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NASA CR-145078

STUDY OF A VUILLEUMIER CYCLE CRYOGENIC  
REFRIGERATOR FOR DETECTOR COOLING ON  
THE LIMB SCANNING INFRARED RADIOMETER

by

Samuel C. Russo

JULY 1976



Prepared under Contract NAS 1-14337 by  
HUGHES AIRCRAFT COMPANY  
Culver City, California

for



National Aeronautics and  
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*Sixty Page*

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## SECTION I.

### INTRODUCTION AND SUMMARY

The National Aeronautics and Space Administration (NASA) is pursuing a program to detect and monitor the presence of trace constituents in the earth's atmosphere by using the Limb Scanning Infrared Radiometer (LSIR). The LSIR, which makes radiometric measurements of the earth's limb radiance profile from a space platform, contains a detector assembly that must be cooled to a temperature of  $65 \pm 2^\circ \text{K}$ . Previously, open cycle coolers using liquid or solid cryogenes were used for this cooling, but they were bulky and had limited life.

Recent information indicates that a Vuilleumier (VM) closed cycle cryogenic refrigeration system could replace open cycle systems and would be lighter, smaller, and have a long reliable operational life. However, such things as the use of spacecraft power, waste heat rejection, and instrument and control interfaces must be investigated and their effect on the overall mission established.

The purpose of this study program is to investigate the feasibility of cooling the NASA-type detector package with a VM cryogenic refrigerator and to develop a preliminary conceptual design of a VM refrigerator that is compatible with a flight-type LSIR instrument.

The scope of the LSIR program consists of analytical and design work to establish the size, weight, power consumption, interface requirements, and other important characteristics of a cryogenic cooler that would meet the requirements of the LSIR as summarized in Table 1. The first effort of the study was to define the cryogenic cooling requirements under the conditions that NASA specified. Following this, a parametric performance analysis was performed to define the interrelationships between refrigeration characteristics and mission requirements. This effort led to the selection of an optimum refrigerator design for the LSIR mission.

The third and last part of the program was to generate a conceptual design of the previously identified VM cryogenic cooler system, needed for power conversion, signal conditioning, control, fault detection, safety interlocks, and cryogenic command interfaces. Table 2 summarizes the characteristics of the recommended cryogenic cooler for the LSIR. The table shows that the cooler provides the detector assembly with the necessary 0.3 watt of net refrigeration and requires 77.1 watts of total input power, which is well below the 90-watt source limit. With 89 watt of input power, the cooler can provide 0.5 watt of net refrigeration. The VM cooler design has a thermal interface between the crankcase heat exchanger and the LSIR thermal rejection system and a mechanical interface with the LSIR housing.

This report describes the preliminary design of a cryogenic refrigeration system for a satellite based infrared detector system. The baseline system that evolved is an electrically powered VM refrigerator on which the infrared

TABLE 1. LSIR MISSION AND REFRIGERATOR REQUIREMENTS

Characteristic	Requirements
<ul style="list-style-type: none"> <li>● Refrigeration temperature, °K</li> </ul>	65 ± 2
<ul style="list-style-type: none"> <li>● Refrigeration temperature stability, °K</li> </ul>	±0.5
<ul style="list-style-type: none"> <li>● Refrigeration capacity, watts</li> </ul>	0.2, 0.3, 0.5
<ul style="list-style-type: none"> <li>● Input power</li> </ul>	
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● Voltage, vdc</li> </ul> </li> </ul>	24 and 10 (10 vdc reserved for controls)
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● Amount, watts</li> </ul> </li> </ul>	
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>Heaters.</li> </ul> </li> </ul>	≤80
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>Drive motor</li> </ul> </li> </ul>	≤10
<ul style="list-style-type: none"> <li>● Cooldown time, hours</li> </ul>	
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● From standby mode</li> </ul> </li> </ul>	≤0.5
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● From shutdown mode</li> </ul> </li> </ul>	≤2.0
<ul style="list-style-type: none"> <li>● Operational modes</li> </ul>	
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● Operating</li> </ul> </li> </ul>	Normal operation: capable of continuous operation
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● Standby</li> </ul> </li> </ul>	Standby operation (i.e., no performance requirements, minimum power input) for periods of from 12 to 500 hours
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● Shutdown</li> </ul> </li> </ul>	Capable of drawing "0" power for periods of up to 500 hours with a restart ability of at least 100 times
<ul style="list-style-type: none"> <li>● Heat rejection</li> </ul>	
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● Rate, watts</li> </ul> </li> </ul>	≤90
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● Method</li> </ul> </li> </ul>	Direct radiation to space
<ul style="list-style-type: none"> <li> <ul style="list-style-type: none"> <li>●● Temperature of crankcase</li> </ul> </li> </ul>	Adjustable to within ±5°C within the range of 0°C to 40°C

TABLE 1. LSIR MISSION AND REFRIGERATOR REQUIREMENTS  
(Continued)

Characteristics	Requirements
<ul style="list-style-type: none"> <li>● Interface and mounting requirements</li> </ul>	Per Figures A-1 and A-2 of Appendix A
<ul style="list-style-type: none"> <li>● Environmental temperature range, °C</li> </ul>	-5 to 45
<ul style="list-style-type: none"> <li>● Operating life, hours</li> </ul>	≥17,500
<ul style="list-style-type: none"> <li>● Design pressure levels</li> </ul>	2 times maximum operating pressure
<ul style="list-style-type: none"> <li>● Proof test pressure</li> </ul>	1.5 times maximum operating pressure
<ul style="list-style-type: none"> <li>● Detector assembly/coldfinger assembly alignment tolerance, μm</li> </ul>	20 (DG)*
<ul style="list-style-type: none"> <li>● Axial alignment to focal plane, μm</li> </ul>	±2.5 (DG)
<ul style="list-style-type: none"> <li>● Dynamic variation in detector position (axial and/or radial) μm</li> </ul>	±0.25 (DG)
<ul style="list-style-type: none"> <li>● Environmental design requirements</li> </ul>	Per Table A-1 of Appendix A
<ul style="list-style-type: none"> <li>● EMI design requirements</li> </ul>	Per MIL-STD-461, Class I (communication and electronic equipment)
<ul style="list-style-type: none"> <li>● Qualification test program</li> </ul>	Per Tables A-1 and A-2 of Appendix A
*DG = Design goal	

TABLE 2. CHARACTERISTICS OF RECOMMENDED LSIR CRYOGENIC COOLER

Refrigerator	Two-Stage, electrically powered, VM cycle
Weight, Newtons (N)	
Refrigerator	35.6 (8 lbf)
IFU	25.7 (6 lbf)
Capacity, watts	0.3
Refrigeration temperature, °K	65 ± 2
Input power, watts	
Heater, 24 vdc	50.1
Motor, 24 vdc	16
EIFU, 24 vdc	11
Total	77.1
Cooldown time, hr	
From standby mode	0.43
From shutdown mode	1.25
Heat rejection, watts (Heat rejection temperature = 45°C)	77.1

detector assembly is mounted. The refrigerator and the electronic interface unit (EIFU) designs make maximum use of the technology developed on previous programs.

This program also includes a formulation of plans, schedules, and cost estimates for a possible follow-on program consisting of final design, fabrication, assembly, and test of three space-qualified refrigeration systems (and necessary GSE) based on the preliminary design described in this document. With a mid 1977 start date, the follow-on program would last approximately 26 months.

This study program was conducted under the direction of Robert D. Averill (technical monitor) and Wilfred D. Hesketh (alternate technical monitor)

## SECTION II

### VM CYCLE THEORY OF OPERATION

Figure 1 shows that the VM refrigerator is a constant-volume device, i. e., the geometrical working volume does not vary when any of the three displacers move. However, because of changes in the average temperature of the gas in the working volume of the refrigerator, the pressure varies as the displacers move. Motion of the hot volume displacer toward the center of the crankshaft forces gas to move from the crankcase, which is approximately at ambient temperature, to the hot end of the hot cylinder. This movement increases the average temperature of the gas in the working volume, and hence the pressure in the fixed volume increases. Likewise, motion of the first- and second-stage displacers toward the center of the crankshaft forces some gas to leave the crankcase and flow to the cold ends of the first- and second-stage cylinders. In this way, the working gas is cooled, and its pressure therefore is decreased. To summarize: Motion of the hot volume displacer toward the center compresses the gas, and motion of the cold volume displacer toward the center expands the gas. Reversing either of the motions produces the opposite effect. The operation of the VM refrigerator will be explained by discussing these basic principles:

When the crankshaft is in the North position, the pressure in the working volume assumes some high value  $P_h$  (see Figure 2). As the crankshaft moves toward West, the hot displacer forces gas to flow into the hot cylinder volume, where the gas compresses further; the first- and second-stage displacers force gas into the cold cylinders where the gas expands to result in refrigeration. In a well designed VM refrigerator, these actions are essentially balanced, and little change in pressure occurs. The net result is that a certain amount of the working gas is transferred to the cold end of the cooling cylinders with little change in pressure. As the crankshaft moves from West to South, all three displacers allow the gas to expand, and the pressure falls to a low value  $P_l$ . With further motion to the East, the first- and second-stage displacers compress the gas and the power displacer allows it to expand with little change in pressure. Motion to the North causes the three displacers to compress the gas, and the pressure returns to the value  $P_h$ . Figure 2 shows indicator diagrams of the cold volumes in the first- and second-stage cylinders. In these diagrams, the enclosed areas represent the gross refrigeration produced in the first and second stages during one cycle.

Figure 3a is an indicator diagram for the hot end of the hot cylinder. The area enclosed represents the minimum amount of energy that must be applied to the hot cylinder during each cycle to drive the system. Figure 3b is an indicator diagram of the crankcase volume; the enclosed area represents the amount of energy that must be dissipated from the crankcase. This energy

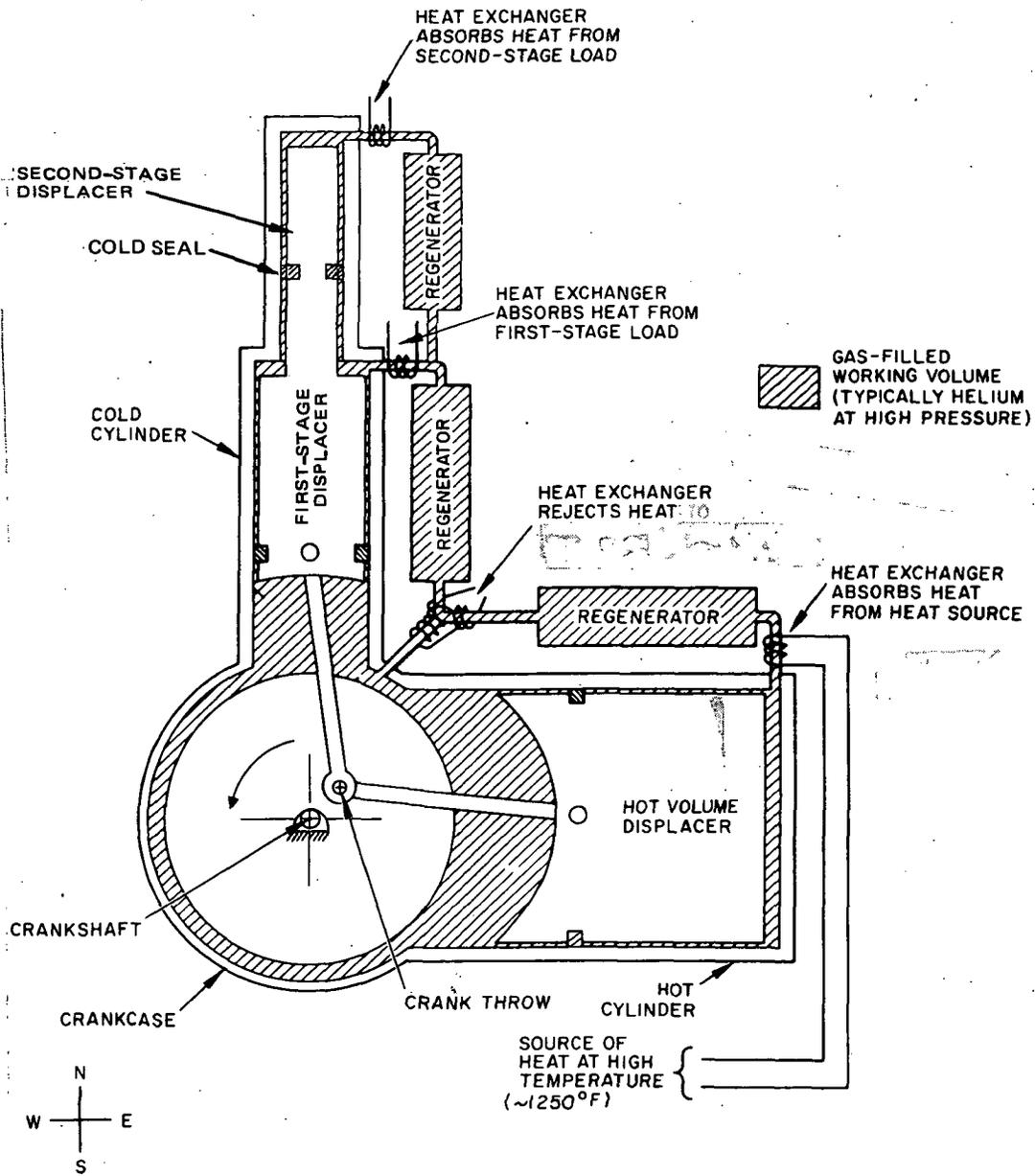


Figure 1. Schematic of basic two-stage VM cryogenic refrigerator.

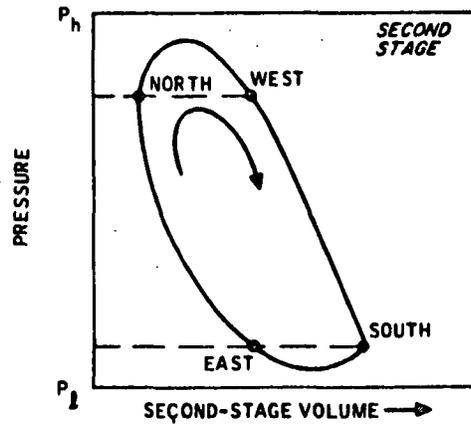
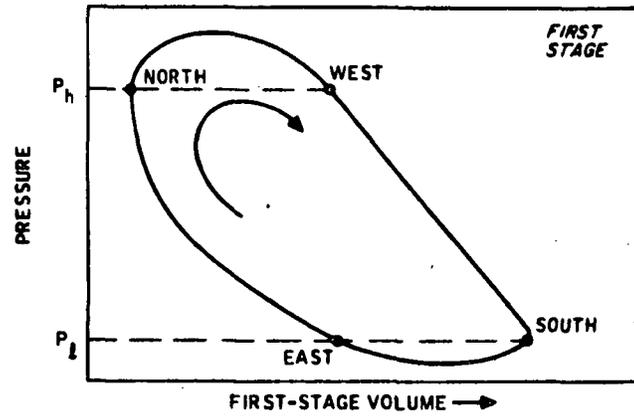
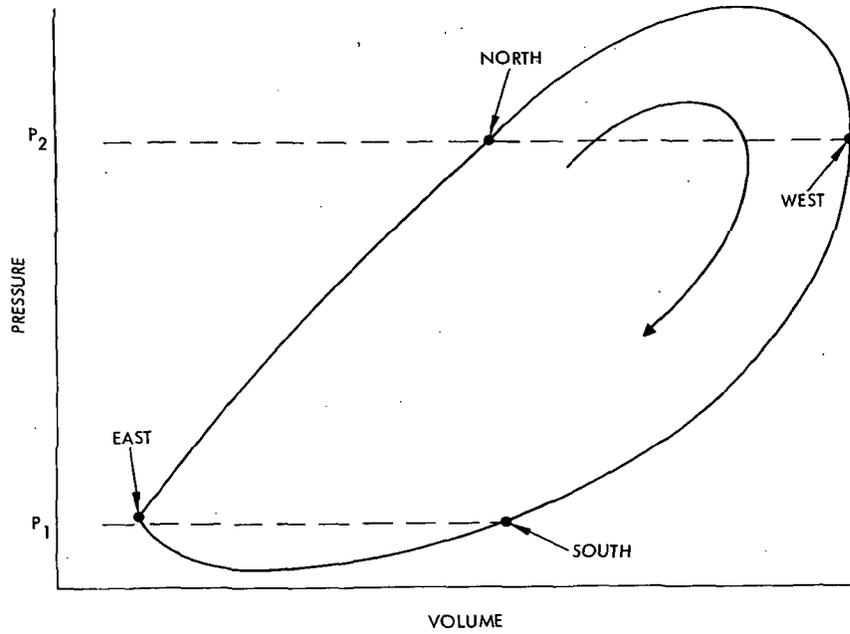
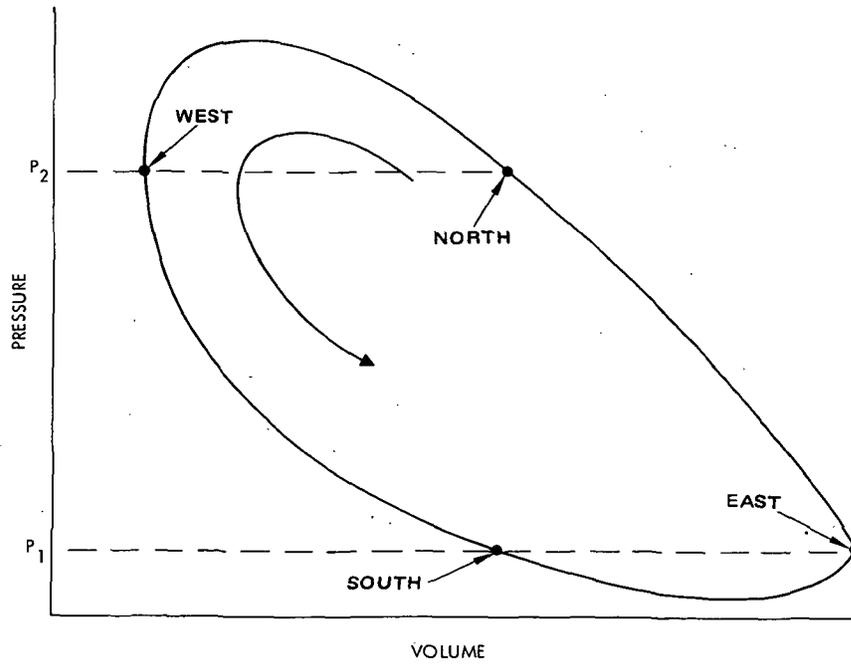


Figure 2. Indicator diagrams of first- and second-stage expansion volumes.



a. Hot expansion volume



b. Crankcase volume

Figure 3. Indicator diagrams of hot expansion volume and crankcase volume.

is equal to the sum of the heat that is added to the hot cylinder and the refrigeration capacities of the first- and second-stage cylinders.

The regenerators shown in Figure 1 store and release heat. Each regenerator is a matrix having a high ratio of surface area to volume, such as that of an array of fine mesh screens. When the gas travels from the ambient temperature volume to the first-stage cold volume, it relinquishes heat to the first-stage matrix element and thereby produces a thermal gradient in the first-stage regenerator. The process is repeated at the second stage as gas flows from the first-stage cold volume to the second-stage cold volume. When the gas flows from the cold volumes, it absorbs heat from each regenerator and emerges at the temperature of the crankcase. A regenerator is also required between the hot volume and the crankcase. Heat enters the regenerator when the gas flows from the hot cylinder to the crankcase, and heat is absorbed from the regenerator when the gas flows from the crankcase to the hot cylinder.

In a practical application of a VM refrigerator, power must be supplied to provide the hot cylinder with thermal energy and to ensure that the motor that moves the displacers is driven at the correct speed. The amount of thermal energy required is considerable, but in small machines the drive motor needs only a few watts of power. The temperature in a hot cylinder approaches  $920^{\circ}\text{K}$ , and the fill pressure when the entire refrigerator is stabilized at  $300^{\circ}\text{K}$  is about 40 atm. The maximum pressure of the gas while a refrigerator is operating is about 55 atm. Hughes has built VM refrigerators that operate at speeds between 200 and 1000 rpm, depending on the application.

Figure 4 shows the general configuration of a small air cooled VM refrigerator; this unit has a capacity of approximately one watt at  $80^{\circ}\text{K}$ . The hot and cold cylinders are usually located 90 degrees apart with the drive motor located at right angles to the axes of the cylinders. A simple slider crank mechanism utilizing a single eccentric drive can then impart the motion to the two displacers. It is also possible to have the hot and cold cylinders collinear if this configuration is more applicable to a specific system. Although this refrigerator could provide the necessary refrigeration for the LSIR, it was not designed for such environment or life requirements. The LSIR refrigerator was designed to satisfy the constraints imposed by the LSIR application.

### SECTION III

#### DETERMINATION OF CRYOGENIC COOLING REQUIREMENTS

The initial phase of this study was devoted to defining the requirements for a VM cryogenic cooler for the LSIR program. Mission requirements and the sensor configuration were reviewed, and an initial design specification was developed (see Appendix A). A study of cryogenic cooling requirements

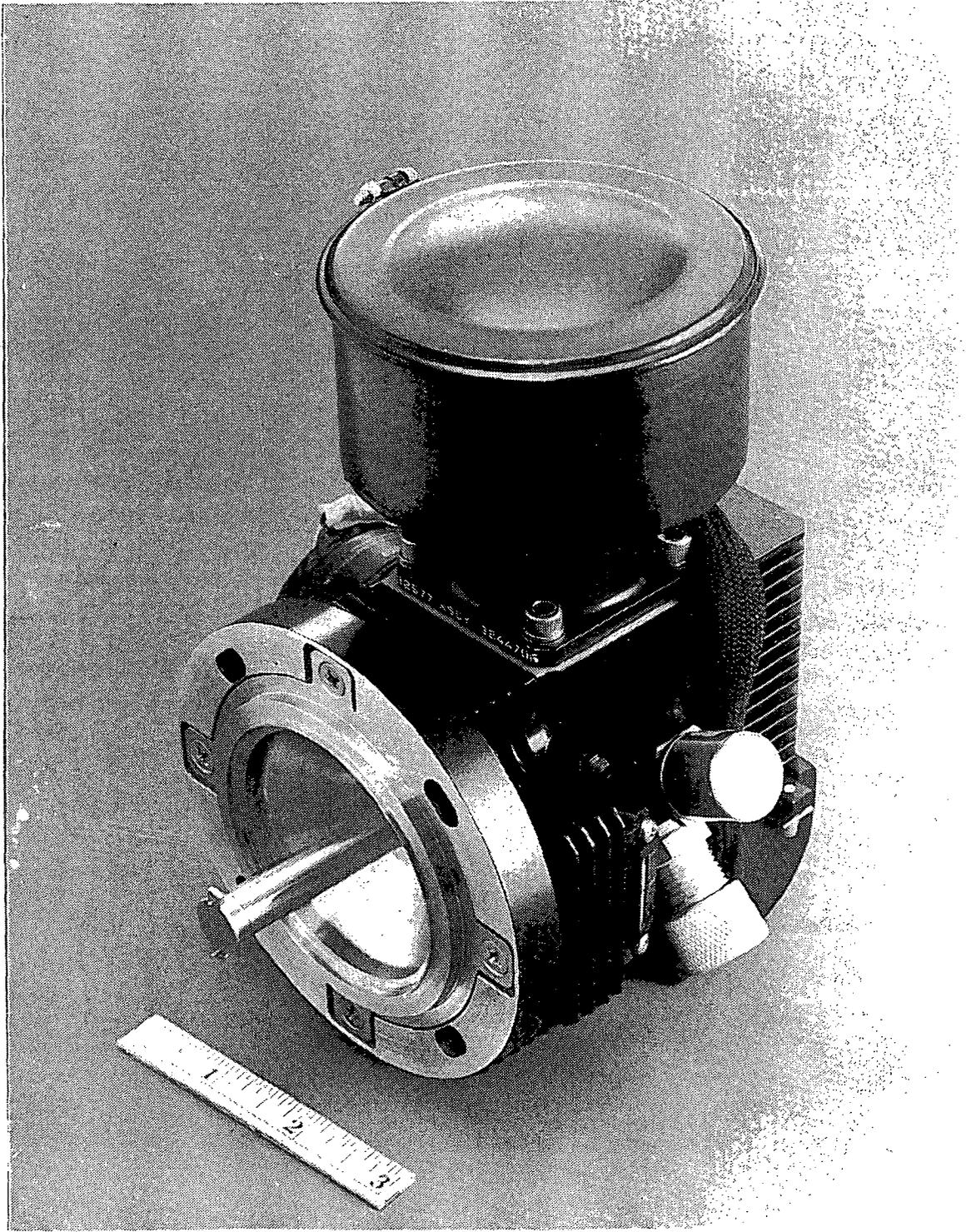


Figure 4. Typical small VM cooler (71-7244).

for the LSIR under conditions identified in the SOW has established the following:

1. The dynamic variation in axial position of the end of the refrigerator cold cylinder due to pressure variation is approximately  $10\ \mu\text{m}$ , which is about 20 times greater than allowed. This value is based on an optimized thermodynamic design with an Inconel cold cylinder that is 7.62 cm long, has a 1.52 cm bore, and whose wall is 0.0254 cm thick. The pressure variations within the refrigerator are between 40 and 52 atm.

It was concluded that reconfiguring the cold cylinder to reduce axial motion to the required  $\pm 0.25\ \mu\text{m}$  limit would result in an exceptionally inefficient machine requiring a total input power of more than 90 watts. It was further concluded that it is not feasible to mount the detector package directly on the cold cylinder. Therefore, a separate low-heat-leak support for the detector array must be designed as part of the refrigerator. This support must have provisions for accurately positioning the detector array. The heat leak through the support structure is included as a parasitic loss to the refrigerator.

A preliminary design study has shown that a conical detector support made of glass approximately 12.7 cm long can satisfy the  $\pm 0.25\ \mu\text{m}$  movement requirement in the axial and radial directions. This is based on the LSIR operational environment as presented in Table II of the statement of work, on the anticipated refrigerator imbalance, and on an 8-gram detector mass. The additional heat load at the  $65^\circ\text{K}$  stage conducted up the support is approximately 0.06 watt.

2. The refrigeration capacity required is determined by the following heat loads.
  - Support thermal load
  - Lead conduction load
  - Detector thermal load

The results of the thermodynamic design and parametric performance analyses, which defined an optimized cooler based on temperature and thermal load requirements, indicate that a two-stage refrigerator is needed. It must cool the following parasitic loads.

Item	First Stage (120°K)	Second Stage (65°K)
Support thermal load, watts	0.48	0.096
Lead conduction load, watts	0.004	0.001
Detector thermal load, watts	-	0.30
Total, watts	0.484	0.397

Support thermal load – The thermal load from the detector support is composed of conduction up this support and radiation to the support structure; see Figure 5. If this support is part of a vacuum dewar, another source of heat load is gaseous conduction. However, under vacuums of at least  $10^{-4}$  torr, such conduction is usually so slight that it is ignored.

The major heat load from the support is conducted up the support and intercepted at the 120°K stage. Radiation to the support at the 120°K and 65°K stages is 0.06 watt and 0.04 watt, respectively. The radiation to the support at the 65°K stage is almost half of the total support thermal load to that stage. A realistic emissivity value of 0.05 was assumed for the support structure and its enclosure. If special care is taken, it may be possible to achieve an emissivity of 0.03; the radiation load at 65°K due to the support would then be 0.016 watt or less.

Lead conduction load – Eight instrumentation leads are required (four leads for each of the cold stages); they are made of manganin and are 0.0127 cm in diameter. As shown above, the losses from these leads are so small at both stages that they do not add significantly to the total loss.

Detector thermal load – The thermal load due to the detector array consists of bias ( $I^2R$ ) dissipation, detector lead conduction, and aperture radiation. It is estimated that there will be 22 leads (including those from temperature sensors and spares) and that the resistance of these leads will be 0.5 ohm. Leads made of either manganin or platinum are used because of the good electrical conductance, low thermal conductivity, and good welding characteristics of these materials.

The use of manganin leads that are 0.5 cm long and 0.01 cm in diameter is assumed. These leads will have a resistance of 0.50 ohm and a heat leak of 0.08 watt. Because of the similar relationship of the thermal conductance and electrical resistance to length and cross sectional area, all 0.5-ohm leads made of

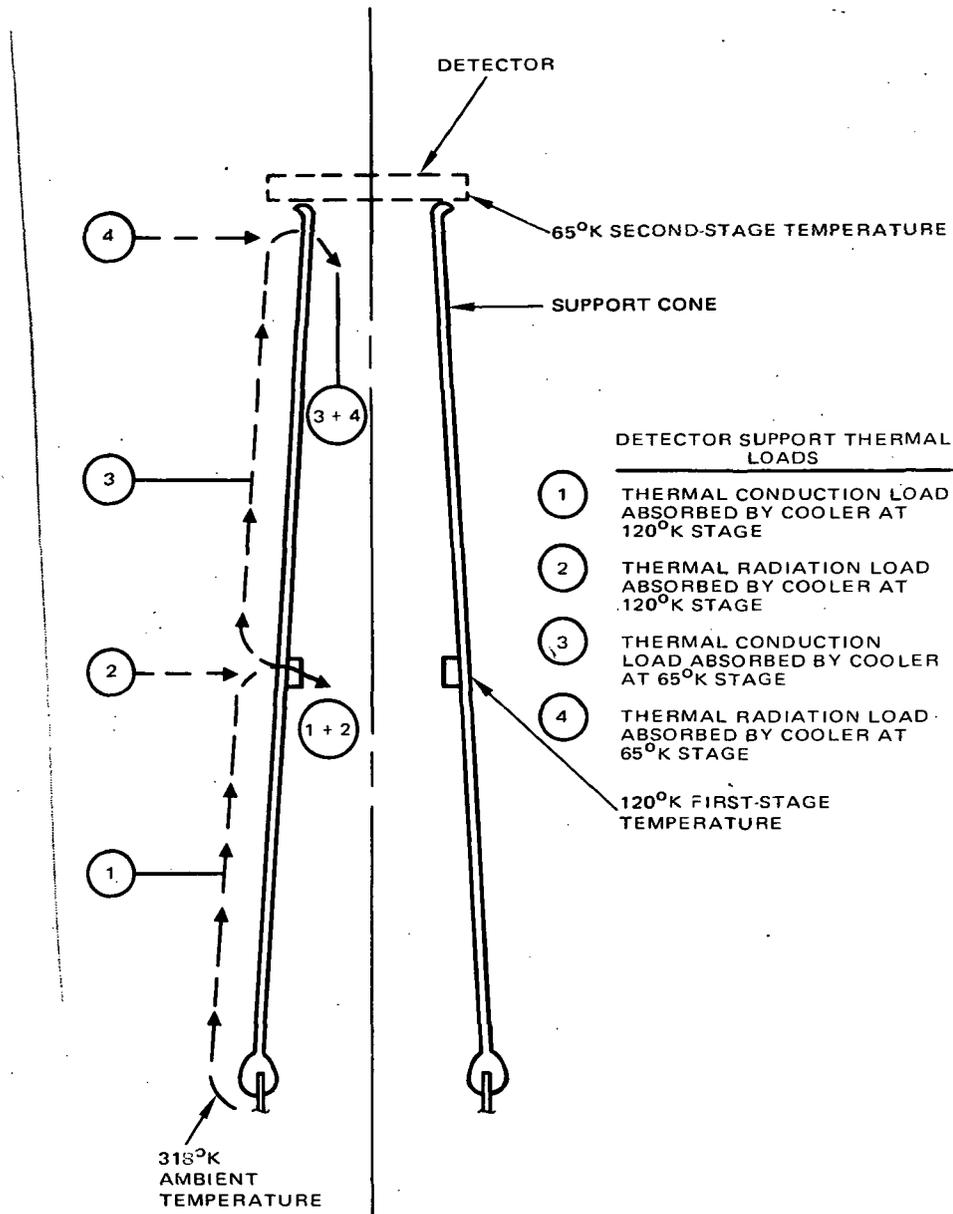


Figure 5. Thermal loads from detector support.

manganin will have the same heat leak. Higher resistance or a different material must be used if the lead loss is to be reduced. It is possible that 1-ohm resistance could be tolerated; this would reduce the lead conduction by 50 percent.

Radiation loss was estimated by using a 3-cm-diameter blackbody at 65°K looking at a gray surface ( $\epsilon = 0.8$ ) at 45°C. This set-up approximates the cold shield on the detector array as it faces the infrared window. The radiation heat transferred to the detector array is then 0.18 watt.

Detector lead conduction and aperture radiation total 0.28 watt; therefore, only 0.02 watt of the total 0.3 watt detector load is available for detector bias. However, should the detector need more than 0.02 watt for bias, the cryogenic cooler recommended has an additional reserve net refrigeration capacity of 0.5 watt with a 79-watt input power requirement.

3. On the basis of previously demonstrated technology, a cylinder temperature of 975°K will allow a rupture life of at least 20,000 hours. By allowing for a temperature drop across the heat exchanger and through the walls, this corresponds to a hot volume temperature of approximate 920°K. This figure is based on stress rupture and creep data for the candidate hot cylinder material (Rene 41) and on low development risk in producing a system that can meet the operating life goals and performance requirements.
4. Operating data on present VM refrigerators shows that the life limiting component is the hot rider ring and that the wear rate of this component varies as a function of how the weight of the hot displacer is supported.

On the basis of data from a test program still in progress, the use of present hot rider materials can ensure more than 20,000 hours of operation in a static environment. Operation on the ground for periods up to 2200 hours under worst case conditions can be tolerated before launch.

The present 1-g wear rate data shown in Figure 6 is based upon tests that have lasted about 4000 hours. These tests were conducted in a 1-g environment with the hot displacer in a horizontal plane, this orientation results in the highest wear rate. By using an expression relating displacer forces to wear rate and employing the 1-g wear data described above, it was possible to analytically determine the effect of a 0-g environment on wear rate (see the wear rate curves in Figure 6\*). It appears from this figure that it will be feasible to operate the refrigerator on the ground for 2200 hours before launch and then operate it in the 0-g environment in space for the rest of the 20,000 hours.

\*In this figure the 0-g curve was superimposed on the 1-g curve to show allowable ground operation time.

HOT RIDER RING IS MAJOR WEAR COMPONENT. IT COULD HAVE LONGER LIFE DURING GROUND OPERATION IF REFRIGERATOR IS ORIENTED SO THAT WEIGHT OF DISPLACER IS NOT ON RIDER

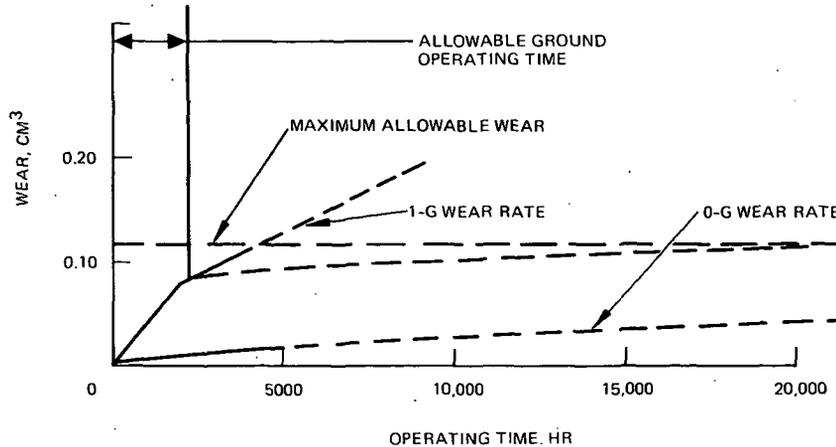


Figure 6. Effect of ground operation on wear rate of hot rider ring.

Wear rate tests are being continued and data for longer test periods is becoming available.

5. The long term effect of stop-start operation on wear rate is not known. However, operating data on VM refrigerators used on airborne and ground based applications where many stop-start cycles are experienced does not indicate unusual wear as compared to continuous running wear. Wear rate data on the components of VM refrigerators is now being generated with the objective of proving at least 20,000 hours of life. Initial results, based on test times of about 4000 hours, have not indicated serious problems.

One effect of start-up operation can be to reduce the capacity of the refrigerator. If the working fluid (helium gas) is contaminated, it will eventually cause the refrigerator to warm up. But, the difference in volumetric flow in and out of the cold regenerator tends to keep the regenerator clean and the refrigerator operating. If the refrigerator is turned off for a short time, the contaminants will rapidly diffuse to the cold regions and become trapped there until the entire refrigerator warms up. If a partially warmed up refrigerator is turned on, the contaminants are concentrated in the regenerator and foul it. If the refrigerator is warmed up to 250°K before it is turned on (it is estimated that this will take 2 hours), the contaminants are swept out of the regenerator and the refrigerator operates normally.

When the gas is not contaminated, start-stop operation does not affect cold end performance.

## SECTION IV

### THERMODYNAMIC DESIGN AND PARAMETRIC PERFORMANCE ANALYSIS

#### Thermodynamic Design Analysis

A thermodynamic analysis was conducted in order to define the configuration and performance of baseline coolers optimized for heat loads of 0.2, 0.3, and 0.5 watt, and three optimized cooler designs were identified. A computer program that Hughes developed was used in designing and evaluating the above coolers. The thermodynamic and heat transfer equations incorporated in the computer program and a description of the computer program as applied to the design of a long life high capacity VM cycle cryogenic refrigerator are presented in references 1 and 2, respectively.\*

The first step in designing each cooler was to define the refrigeration temperatures and thermal loads, the hot volume temperature, and the maximum ambient (or crankcase) temperature. On the basis of size, weight, and operational life requirements, a speed and a cyclic pressure were selected. The type of regenerator matrix and the material used for it were chosen on the basis of the temperature of each stage. The ratio of the regenerator volume to the swept volume for each stage as well as an estimate of the ratio of the total dead volume to the swept volume was determined. A preliminary estimate of the required cold working volumes was also made. The final design parameters to be selected were the maximum allowable clearance between the displacer and the side of the cylinder, the allowable pressure drop in the first stage regenerator and in the hot regenerator, and the thickness of the insulation around the hot cylinder.

On the basis of the values of the parameters selected, a cooler was designed in which the various thermal losses were minimized in order to provide the maximum amount of net refrigeration at each stage. This was done by simultaneously varying stroke, piston side clearances, regenerator lengths, and cylinder and regenerator diameters at each stage until the best combination of the above parameters was reached. This design was then improved by re-evaluating the required swept volume, the ratio of regenerator volume to swept volume, the estimate of the ratio of total dead volume to swept volume, the allowable clearances, and the allowable pressure drops. The cooler was then redesigned. This process was continued until a thermodynamically optimum cooler was defined, i. e., one in which the thermal losses are minimum and the net refrigeration at each stage is maximum. This method was utilized in designing each of the three coolers.

\* 1) R. White, "Vuilleumier Cycle Cryogenic Refrigeration", AFFDL-TR-76-17 (April 1976).

2) R. White, "Computer Program for Optimizing Three-Stage Vuilleumier Cycle Cryogenic Refrigerators", AFFDL-TR-76-27 (February 1976).

In the initial design phase, the following four refrigerator mechanizations were considered:

1. One-stage refrigerator with the same hot and cold displacer strokes.

The one-stage refrigerator is a simple design and is usually accepted for applications above  $65^{\circ}\text{K}$ . The one-stage single-stroke machine (same hot and cold displacer stroke) was not evaluated because, based on preliminary calculations, it was obvious that it could not satisfy the low power requirement of the LSIR.

2. One-stage refrigerator with different hot and cold displacer strokes.

For this one-stage design, the hot displacer stroke and the cold displacer stroke were both optimized for maximum efficiency; the result was that the hot and cold displacers had different strokes. The two-stroke machine is more efficient thermodynamically than a single-stroke machine; however, its mechanical design is more complex.

3. Two-stage refrigerator with the same stroke for the hot and cold displacers.

The two-stage machine is generally more efficient than the one stage at operating temperatures in the range of  $60^{\circ}\text{K}$  to  $70^{\circ}\text{K}$ ; below  $60^{\circ}\text{K}$ , the one-stage machine is not practical. For this design, a displacer stroke that is a compromise between optimum hot and cold displacer strokes was selected. The stroke selected gave the most efficient performance for the single-stroke design.

4. Two-stage refrigerator with different hot and cold displacer strokes.

In this design, the hot and cold displacer strokes were each optimized for maximum efficiency. This design is more efficient than the single-stroke design, but as stated above, the mechanical design, particularly in the crankcase assembly, is more complex.

The three latter mechanizations described above were compared on the basis of refrigerator load and required heater power. Figure 7 shows how the heater power requirement of each design varies as a function of load. It is evident that the two-stage, double-stroke machine is the most efficient and best satisfies the low power requirements of the LSIR program. Therefore the two-stage double-stroke design was selected for the LSIR baseline cooler.

A thermodynamic analysis was conducted to define the best thermal configurations of the baseline cooler for three distinct cases with net refrigeration of 0.2, 0.3, and 0.5 watt\*. Table 3 compared the design variables of the

\*For Case A, net refrigeration = 0.5 watt; for Case B, net refrigeration = 0.3 watt; for Case C, net refrigeration = 0.2 watt.

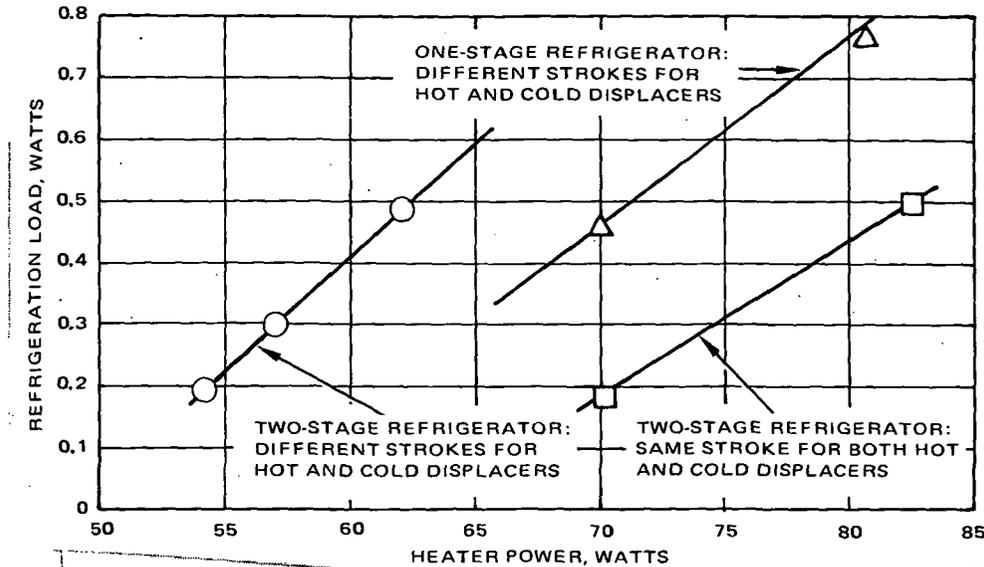


Figure 7. Refrigeration load as a function of heater power (three mechanizations).

three refrigerator designs that resulted from this study; note that the variables for the three machines were almost identical. On the basis of previously demonstrated technology, these values are considered to be consistent with LSIR operating life goals and performance requirements. Based on stress rupture and creep data on the candidate hot cylinder material (Rene 41) and low developmental risk attendant to meeting the operating life goals and performance requirements, a maximum hot cylinder temperature of  $920^{\circ}\text{K}$  was selected.

The coolers were designed to operate at the highest ambient temperature,  $45^{\circ}\text{C}$ . For operation at lower ambient temperatures, the hot cylinder temperature and power consumption can be lowered to maintain the same net refrigeration capacity. This matter is discussed later in this report.

A first-stage temperature was selected on the basis of minimum input power;  $120^{\circ}\text{K}$  was found to be best.

The normal rotational speed of the baseline coolers is 300 rpm; this value is based on design compromises between life, overall size, and contamination. A lighter weight machine operating at higher rpm and using less input power could furnish the same amount of cooling but would have a shorter life. The speed selected is a good compromise among the parameters.

The base cycle pressure at design point operation is 45 atm. The thickness of the various pressure vessel elements was calculated to ensure structural integrity over the cyclic pressure range. The pressure ratio is 1.29.

The pressure drop through the  $120^{\circ}\text{K}$  stage regenerator and hot cylinder regenerator was selected to keep bearing loads and motor torque low in order to extend operating life and minimize required motor power. The pressure drop through the  $65^{\circ}\text{K}$  regenerator was optimized to give the best performance.

TABLE 3. COMPARISON OF DESIGN VARIABLES OF REFRIGERATORS A, B, AND C

DESIGN VARIABLES	REFRIGERATOR A (0.5 WATT)	REFRIGERATOR B (0.3 WATT)	REFRIGERATOR C (0.2 WATT)
Hot volume temperature, °K	920	920	920
Crankcase temperature, °K	318	318	318
First cold stage temperature, °K	120	120	120
Second cold stage temperature, °K	65	65	65
Operating speed, rpm	300	300	300
Low cycle pressure, atm	40.2	40.2	40.2
High cycle pressure, atm	51.9	51.3	51.8
Fill pressure, atm	47	47	47
Pressure drop in first-stage regenerator, atm	0.11	0.11	0.11
Pressure drop in second-stage regenerator, atm	0.21	0.26	0.27
Pressure drop in hot regenerator, atm	0.023	0.025	0.02
Pressure ratio	1.29	1.28	1.29

Table 4 gave the dimensions of the three coolers, which are all approximately the same size; however, the overall dimensions decrease slightly as the design point net refrigeration is reduced from 0.5 to 0.2 watt.

Table 5 summarized the volume parameters. The working volume requirements decrease slightly as the net refrigeration is reduced from 0.5 to 0.2 watt. It is seen that the ratio between reduced dead volume and cold swept volume is the same for each cooler. This indicates that the volumetric relationship for all three configurations is equivalent and that the pressure ratios are identical.

Table 6 summarizes the parasitic heat losses and net refrigeration capacity and shows that the total losses (for a definition of these losses, see Appendix B) increase very little at the cold stages as the net refrigeration is increased from 0.2 to 0.5 watt. This indicates that the penalty for achieving the 65°K temperature is high compared to the additional penalty paid for increasing the net capacity from 0.2 to 0.5 watt. As shown in Table 7, there is only a 13.5-percent increase in required heater power as the net refrigeration is increased from 0.2 to 0.5 watt.

### Parametric Performance Analysis

A parametric performance analysis was conducted of the three coolers that were evolved via the evaluation described previously. The available cooling capacity and power input were evaluated as a function of both ambient temperature and hot cylinder temperature. In this analysis, the speed and fill pressure were maintained at their design values; the calculated performance data represents steady state values.

Three hot cylinder operating temperature were selected for each cooler. The highest hot cylinder temperature is 920°K, the design point value. The lowest hot cylinder temperature is that which just gives 0.2 watt of refrigeration at 65°K and -5°C ambient. The third hot cylinder temperature is midway between the highest and lowest temperature.

Figures 8 through 10 show how refrigeration capacity and required heater power of the three cooler designs vary as a function of ambient temperature for the three selected hot cylinder temperatures. The ambient temperature significantly affects the performance of a cryogenic refrigerator. Increasing the ambient temperature reduces refrigeration capacity for two reasons:

- The larger temperature gradient across the stage increases the thermal losses particularly on the first stage.
- The higher crankcase temperature decreases the pressure ratio by increasing the minimum cyclic pressure.

The higher minimum cyclic pressure significantly affects the performance of the cold regenerators because of the increased mass flow through them. Also, greater heat input is needed not only because the regenerator becomes less efficient but also because a greater amount of P-V work must be generated. The net refrigeration capacity also decreases as the temperature of the hot

TABLE 4. DIMENSION OF REFRIGERATOR

REFRIGERATOR DIMENSIONS	REFRIGERATOR A (0.5 WATT)		REFRIGERATOR B (0.3 WATT)		REFRIGERATOR C (0.2 WATT)	
	120°K STAGE	65°K HOT CY- LINDER	120°K STAGE	65°K HOT CY- LINDER	120°K STAGE	65°K HOT CY- LINDER
Cylinder bore, cm	2.21	1.22	2.46	2.11	1.143	2.34
Regenerator length, cm	5.18	4.80	4.67	5.18	4.39	4.67
Cylinder length, cm	6.35	4.57	5.08	7.62	6.60	6.86
Side clearance, cm	0.008	0.008	0.0254	0.008	0.008	0.036
Regenerator ID, cm	0.881	0.368	2.510	0.836	0.307	2.375
Regenerator OD, cm	-	-	2.565	-	-	2.423
Cylinder thickness, cm	0.0254	0.0254	0.031	0.0254	0.0254	0.031
Stroke, cm	0.300	0.300	0.869	0.300	0.300	0.869
Regenerator matrix	400 mesh	400 mesh	40 mesh	400 mesh	400 mesh	40 mesh
Regenerator location	Inter- nal	Inter- nal	Exter- nal	Inter- nal	Inter- nal	Exter- nal
Liner thickness, cm	-	-	0.015	-	-	0.015
Thickness of hot cylinder insulation, cm	-	-	1.7	-	-	1.7

TABLE 5. VOLUME PARAMETERS OF REFRIGERATORS A, B, AND C

VOLUMETRIC RELATIONSHIPS, CM <sup>3</sup>	REFRIGERATOR A (0.5 WATT)	REFRIGERATOR B (0.3 WATT)	REFRIGERATOR C (0.2 WATT)
Hot active volume	4.12	3.69	3.50
Hot regenerator volume	0.64	0.57	0.54
Active volume of 120°K stage	0.80	0.72	0.69
Volume of 120°K regenerator	2.19	1.97	1.89
Active volume of 65°K stage	0.36	0.32	0.3
Volume of 65°K stage regenerator	0.36	0.23	0.17
Crankcase volume	2.30	2.30	2.30
Additional inactive volume	2.12	1.98	1.90
Ratio between reduced dead volume and cold volume	2.55	2.55	2.55

TABLE 6. PARASITIC LOSSES AND NET REFRIGERATION CAPACITY OF REFRIGERATORS A, B, AND C

	REFRIGERATOR A	REFRIGERATOR B	REFRIGERATOR C
	(0.5 WATT)	(0.3 WATT)	(0.2 WATT)
COLD CYLINDER LOSSES	120°K STAGE	120°K STAGE	120°K STAGE
	65°K STAGE	65°K STAGE	65°K STAGE
Regenerator heat loss	0.39	0.36	0.34
Aerodynamic friction	0.09	0.08	0.08
Cylinder conduction	0.91	0.86	0.84
Displacer conduction	0.44	0.39	0.37
Regenerator conduction	0.05	0.04	0.04
Shuttle	0.59	0.56	0.55
Pumping	0.05	0.03	0.03
Seal	-	-	-
Leads, dewar, and support	0.48	0.37	0.37
Total losses	3.00	2.69	2.62
Gross refrigeration	2.62	2.30	2.24
Interstage heat flow	0.36	0.38	0.44
Net refrigeration	0.02	0.01	0.06
	0.50*	0.32*	0.22*

\*Net refrigeration available for cooling detector.

TABLE 7. HEATER POWER REQUIREMENTS OF REFRIGERATORS A, B,  
AND C

HOT CYLINDER HEATER POWER, WATTS	REFRIGERATOR A (0.5 WATT)	REFRIGERATOR B (0.3 WATT)	REFRIGERATOR C (0.2 WATT)
PV input to gas	13.5	11.8	11.2
Regenerator heat loss	10.3	9.1	8.6
Aerodynamic friction	0.1	0.1	0.10
Cylinder conduction	5.5	5.1	4.8
Displacer conduction	5.1	4.5	4.3
Liner conduction	4.4	4.1	4.0
Regenerator conduction	0.2	0.2	0.2
Piston internal	1.3	1.2	1.1
Shuttle	13.5	9.3	9.0
Pumping	0.1	0.2	0.2
Insulation	7.6	7.4	7.4
Total power to heater	61.9	57.3	55.1

DESIGN POINT: 0.5-WATT COOLING AT 65°K  
HEAT REJECTION TEMPERATURE = 45°C

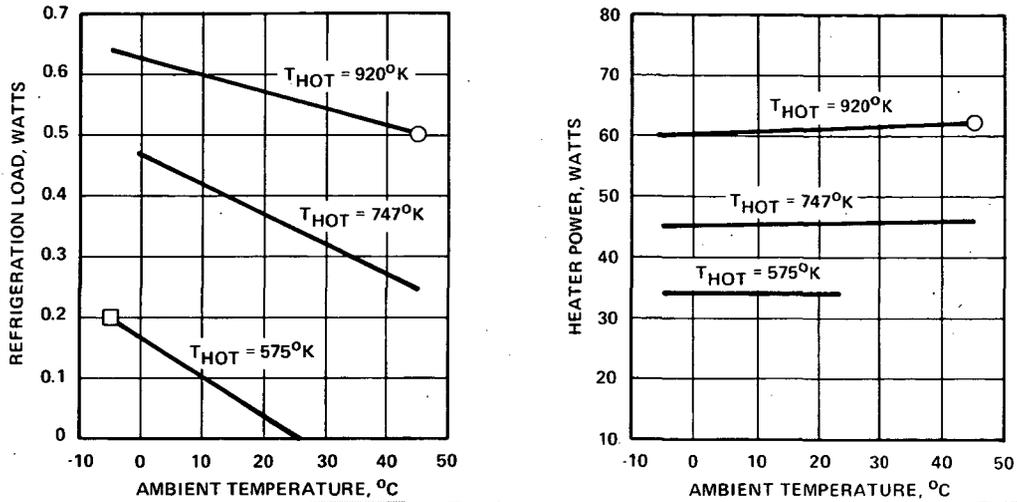


Figure 8. Effect of crankcase and hot cylinder temperature on heater power for refrigerator A.

DESIGN POINT: 0.3-WATT COOLING AT 65°K  
HEAT REJECTION TEMPERATURE = 45°C

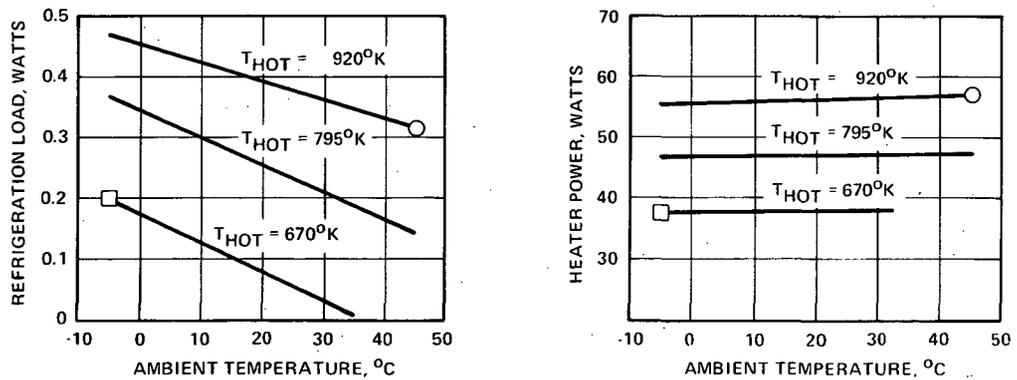


Figure 9. Effect of crankcase and hot cylinder temperature on heater power for refrigerator B.

DESIGN POINT: 0.2 WATT AT 65°K  
REJECTION TEMPERATURE = 45°K

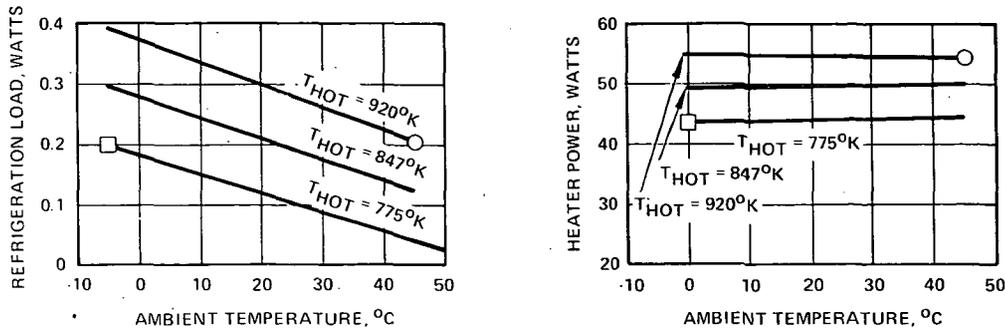


Figure 10. Effect of crankcase and hot cylinder temperature on heater power for refrigerator C.

cylinder decreases. Increasing the hot cylinder temperature increases the refrigeration capacity by increasing the pressure ratio of the thermodynamic cycle. Heater power increases slightly with increasing ambient temperature but can almost be considered constant over the  $-5^{\circ}C$  to  $45^{\circ}C$  ambient temperature range. The ambient temperature is primarily determined by the heat rejection system of the spacecraft.

The results of the parametric study, as presented in Figures 8 through 10, indicate that the coolers designed to operate at 0.2-watt and 0.3-watt capacity require more power than the 0.5-watt cooler when the 0.5-watt cooler is operated at 0.2 watt or 0.3 watt capacity. This result is clearly illustrated in Figure 11 where the heater power requirements for the three

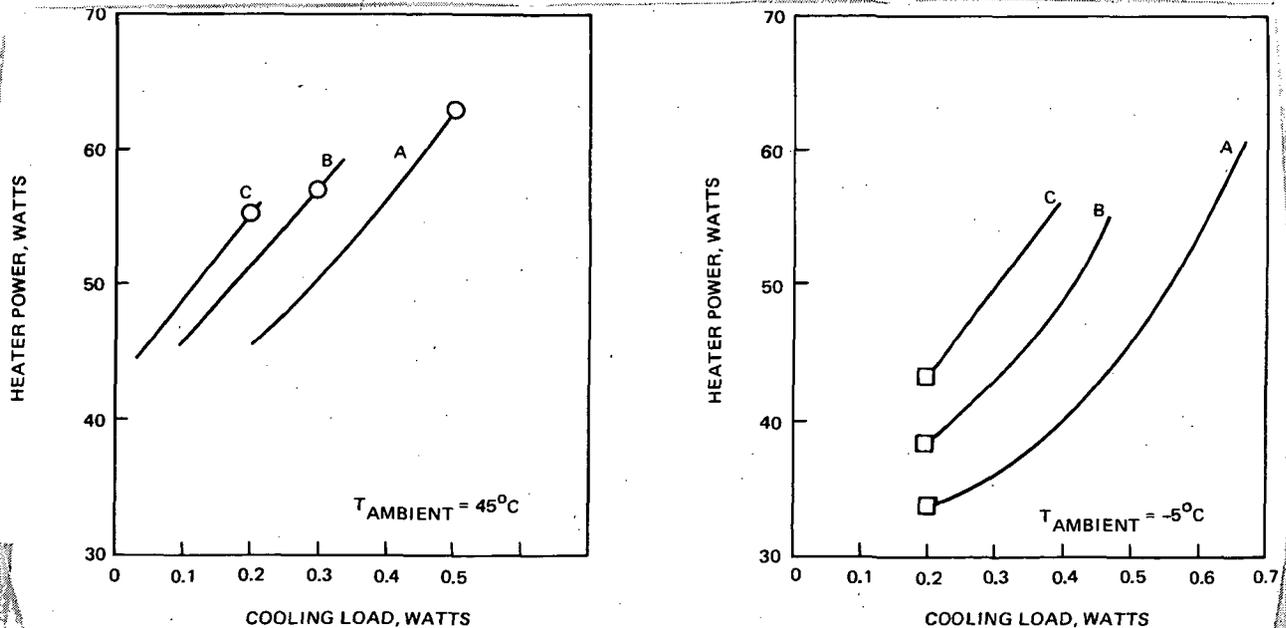


Figure 11. Heater power requirement as a function of cooling load for refrigerators A, B, and C.

cooler designs are presented as a function of load for operation at  $45^{\circ}\text{C}$  and  $-5^{\circ}\text{C}$  ambient temperatures. At  $45^{\circ}\text{C}$  ambient and 0.2-watt net refrigerator load, the heater power requirements for cooler designs A, B, and C are approximately 45 watts, 51 watts, and 55 watts, respectively. At  $-5^{\circ}\text{C}$  ambient and 0.2 watt net refrigeration load, the heater power requirements for designs A, B, and C are approximately 34 watts, 37 watts, and 43 watts, respectively. This result can be better understood by reviewing the hot cylinder losses for each of the three cooler designs as presented in Table 7. It is seen that approximately 80 percent of the heater power required by each of the coolers is needed to compensate for thermal losses from the hot cylinder and that only 20 percent is actually needed in the thermodynamic cycle (P-V work). Hence, when the 0.5-watt cooler is operated at a 0.2-watt or a 0.3-watt load, the reduction in thermal losses realized by operating the large cooler at the lower hot cylinder temperature is much more significant than the accompanying increase in required P-V work. In effect, the ratio of P-V work to thermal losses is increased when the larger cooler is operated at low capacity, but the total power requirement is reduced.

The results of the parametric study given in Figures 8 through 10 show that the 0.5-watt baseline cooler is the most efficient machine and that the penalty for achieving  $65^{\circ}\text{K}$  is high compared to the additional penalty incurred for increasing the design load from 0.2 watt to 0.5 watt. The larger machine provides the potential for 0.2 watt of additional refrigerator capacity beyond the 0.3 watt needed and requires the least amount of heater power and total power at 0.3 watt net refrigeration capacity. Based on the results of the parametric study, the 0.5-watt cooler (refrigerator design A) was selected as the candidate cooler for the LSIR, and a parametric study of this cooler was conducted showing the effect of speed and ambient temperature on refrigeration load and input power. The refrigeration load of refrigerator design A is shown as a function of speed in Figures 12 through 14 at  $45^{\circ}\text{C}$ ,  $20^{\circ}\text{C}$ , and  $-5^{\circ}\text{C}$  ambient temperatures with hot cylinder temperature as a parameter. The hot cylinder temperatures with  $920^{\circ}\text{K}$  (the maximum allowable),  $747^{\circ}\text{K}$  (midway between the highest and lowest temperatures), and  $575^{\circ}\text{K}$  (the lowest temperature that gives 0.2 watt at  $-5^{\circ}\text{C}$  ambient and 300 rpm motor speed. At  $45^{\circ}\text{C}$  ambient and  $575^{\circ}\text{K}$  hot cylinder temperature, refrigerator design A cannot provide cooling.

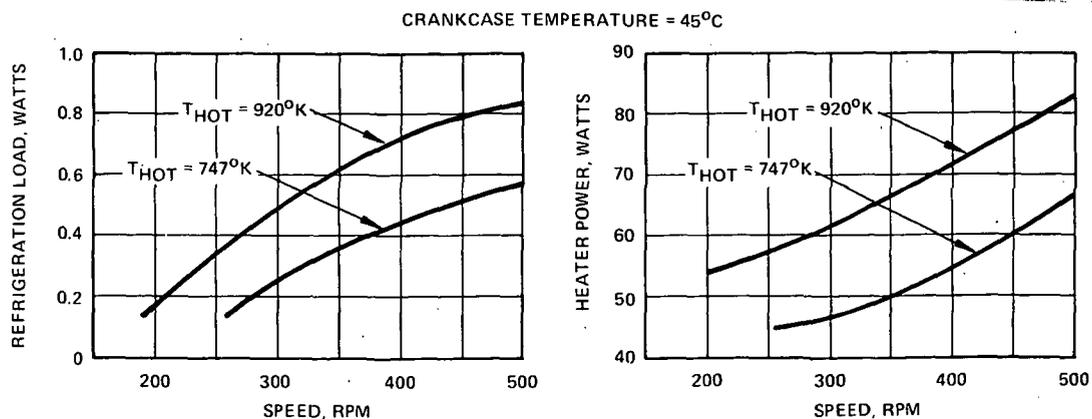


Figure 12. Effect of speed and hot cylinder and crankcase temperature on performance of refrigerator A (0.5 watt) when crankcase temperature =  $45^{\circ}\text{C}$

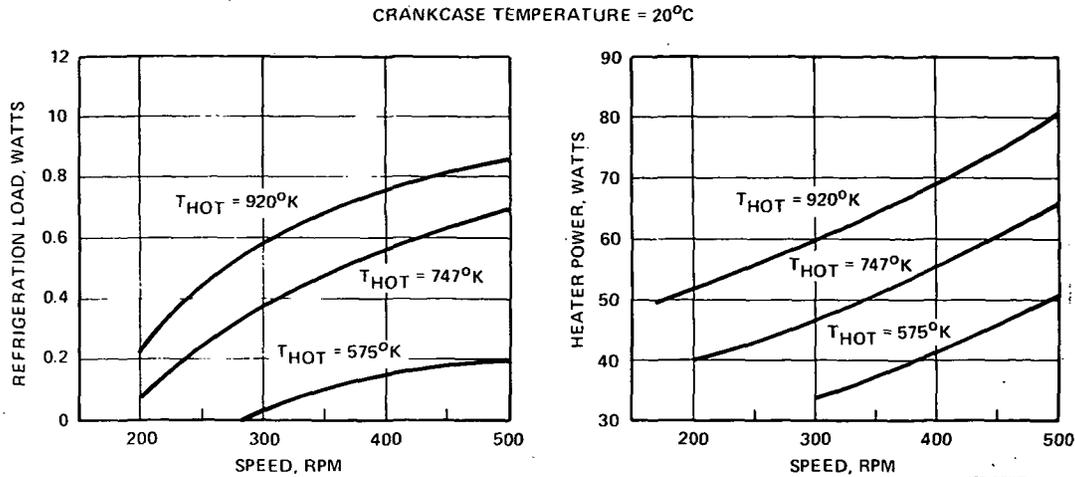


Figure 13. Effect of speed and hot cylinder and crankcase temperature on performance of refrigerator A (0.5 watt) when crankcase temperature = 20°C.

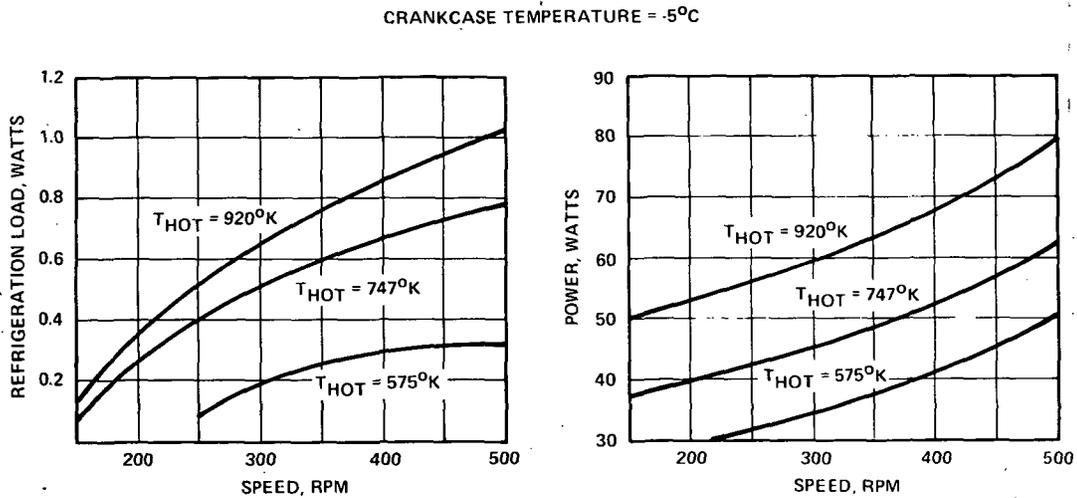


Figure 14. Effect of speed and hot cylinder and crankcase temperature on performance of refrigerator A (0.5 watt) when crankcase temperature = -5°C.

Both refrigeration load and required heater power increase as the speed is increased above the design speed of 300 rpm. Refrigeration load begins leveling off to a maximum value at a speed of approximately 500 rpm. The rpm at which the maximum refrigeration load is realized increases as the ambient temperature is decreased. Although there is a refrigeration load limit, there is no heater power limit, and the power requirement continues to increase with increasing speed.

## Calculations of Transient Performance

Cooldown. — The time required to cool the LSIR detector array to the 65°K operating temperature from initial start-up has been computed. The very accurate detector alignment requirements demanded that the refrigerator cold finger be connected to the heat sink through a compliant heat transfer mechanism, which itself requires additional cooldown time. The first stage of the VM refrigerator must cool the lower dewar and thermal interface structure and intercept the heat load from this lower dewar. The second stage cools the upper dewar, the detector array, and the thermal interface structure and absorbs aperture radiation, detector lead conduction, and thermal loads from the upper dewar. It was assumed that detector bias is zero during cooldown.

The cooldown time was calculated by means of a transient computer program that employs the explicit forward differencing technique and utilizes the temperature dependent properties of the materials. Figure 15 shows the cooldown history. The first stage cools more rapidly because it has the largest capacity. First-stage operating temperature is reached in approximately 1 hour. The detector array reaches the 65°K operating temperature in approximately 1.25 hours.

When operating at the design point (300 rpm; 920°K hot cylinder temperature), the baseline cooler has a 0.5-watt capacity at 65°K and a 0.35-watt capacity at 53°K. Since the maximum load at the second stage is 0.35 watt during cooldown, the detector array is cooled to 53°K when operating at the design point, as illustrated in Figure 15. For operation at 65°K, it will be necessary to lower the hot cylinder temperature once the detector has cooled to 65°K. The EIFU will perform this function; the hot cylinder temperature is controlled in relation to the cold end temperature. Hot end temperature control is discussed later.

The time required to cool the LSIR detector array to operating temperature in less than the 2-hour program requirement. The cooler can be launched at ambient temperature and commanded into operation after the LSIR is in orbit and properly positioned.

Standby mode. — The refrigerator operating parameters for the standby mode have been determined on the basis of the requirement that the refrigerator remain in standby condition with minimum electrical power input and that the refrigerator must cool down to operating temperature from this mode in less than one-half hour. The results of the parametric performance study were utilized in evaluating standby mode requirements. Table 8 shows the effect of reduced speed and reduced hot cylinder temperature on standby mode requirements. Of the two modes, the reduced hot cylinder temperature is recommended since it offers the greatest savings of power for almost equivalent cooldown times. Figure 16 shows cooldown temperature history for the reduced hot cylinder temperature mode.

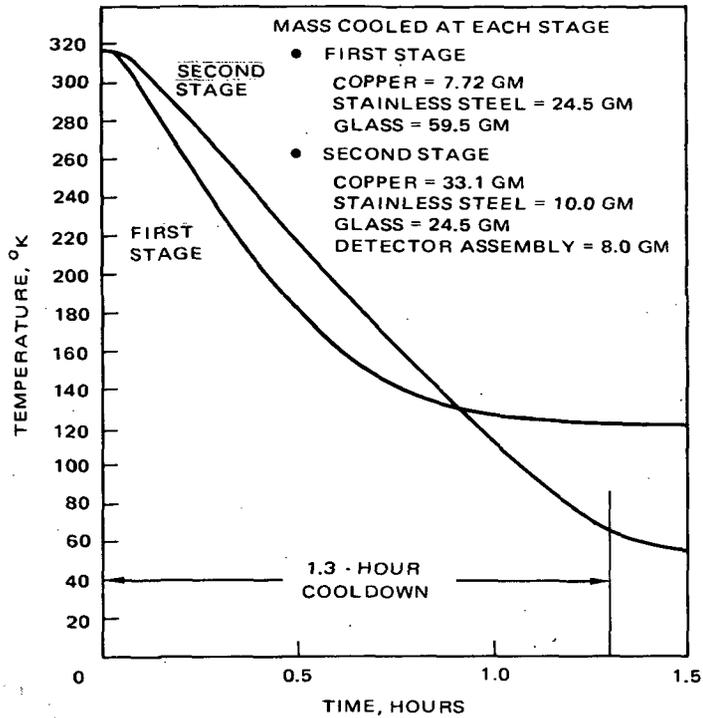


Figure 15. Cooldown history of refrigerator from shutdown.

TABLE 8. POSSIBLE STANDBY MODES

	OPERATING CONDITION		
	NORMAL	REDUCED SPEED	REDUCED HOT TEMPERATURE
COOLDOWN TIME, HOURS	—	0.35	0.43
HOT CYLINDER TEMPERATURE, °K	920	920	570
SPEED, RPM	300	150	300
FIRST-STAGE TEMPERATURE, °K	120	174	166
FIRST-STAGE LOAD, WATTS	0.48	0.35	0.36
SECOND-STAGE TEMPERATURE, °K	65	106	113
SECOND-STAGE LOAD, WATTS	0.6	0.30	0.34
HEATER POWER, WATTS	62	47	31
MOTOR POWER, WATTS	16	13	16
COMBINED POWER, WATTS	78	60	47
POWER SAVED, WATTS	—	18	31

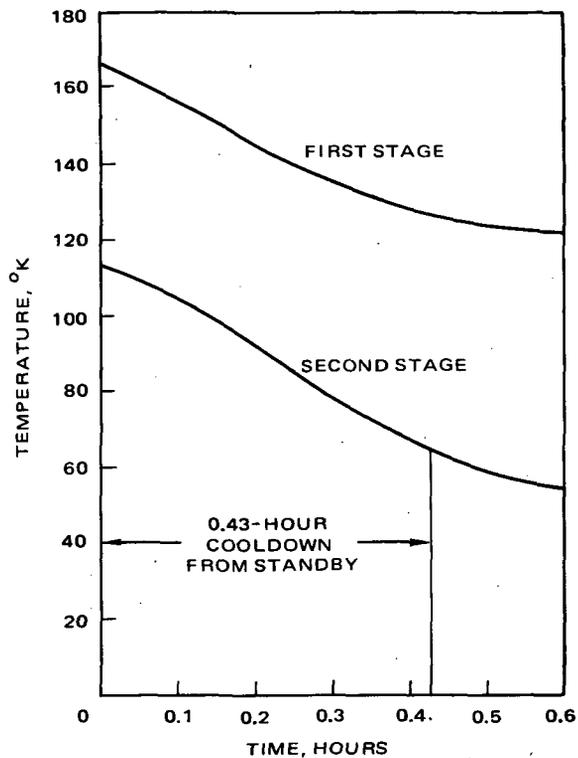


Figure 16. Cooldown history of refrigerator from standby mode.

Cold end temperature stability. — Cold end temperature stability is achieved by varying the hot cylinder temperature and/or the motor speed as a function of cold end temperature; i. e., if the temperature of the cold end falls below the design point, the hot cylinder temperature and the motor speed are reduced in proportion to the magnitude of the cold end temperature drop, and if the temperature rises above the design point, the hot cylinder temperature and the motor speed are increased in proportion to the temperature rise of the cold end. The EIFU performs the necessary control, the details of which will be discussed in a later section. The temperature variation at the cold end is a function of cold stage loading, crankcase temperature, and input voltage. The effectiveness of the cold end temperature control system in maintaining a stable cold end temperature with variations in input voltage was evaluated. A control system consisting of only hot cylinder temperature control and a control system consisting of simultaneous hot cylinder temperature control and motor speed control were evaluated. In the latter system, the motor speed control reduces the effect of the temperature ripple that results from switching the heater on and off slowly and functions as a fine adjustment for maintaining the selected cold temperature. The heater input voltage is 24 vdc; it was assumed that the voltage variation is  $\pm 4$  vdc. In order to simulate a worst case, the voltage was assumed to vary as a step function between the maximum and minimum values (20 vdc to 28 vdc). The cold end temperature

was calculated as a function of the varying input voltage. For the analysis, it was assumed that the detector mass is concentrated on the cold cylinder. The assumed heater input voltage and the calculated cold end temperature are shown as a function of time in Figure 17. The motor speed control is effective in damping out the effects of thermal lag and maintaining a stable cold end temperature variation with a maximum ripple amplitude of approximately  $\pm 0.1^{\circ}\text{K}$ . Without speed control, the amplitude of the temperature ripple is  $\pm 0.25^{\circ}\text{K}$  but the variation appears to be unstable. It is recommended that both the hot end control and speed control be utilized in maintaining cold end temperature stability. Hot end control and the speed control are discussed later.

## SECTION V

### CONCEPTUAL MECHANICAL DESIGN

Following the thermodynamic and parametric performance analyses, which defined the optimum configuration for the baseline refrigerator, a preliminary layout of the conceptual design was made. The layout of the mechanical components is sufficiently detailed to verify that the governing assumptions made during the thermal analysis were valid. For example, the mechanical layout has verified that the effective inactive volume is equivalent to that used in the analytical study; this indicates that the working pressure ratio will be as originally computed. The layout further provides necessary information on the feasibility of meeting operating life requirements. Bearing sizes and load factors as well as other critical bearing surfaces and drive components must be defined before the overall life of the refrigerator can be estimated.

#### Dynamic Balance

Considerable attention was devoted to the dynamic balance of the refrigerator in order to minimize the vibration that the refrigerator would impose on the support for the detector array. The three basic contributors to dynamic imbalance in a VM refrigerator are

- Linear inertia forces acting along the axis of each cylinder; the motion of the hot and cold displacers causes these forces
- Rotating inertia forces due to the imbalance of the rotating parts of the crank mechanism
- Coupling forces resulting when crank mechanism imbalance forces and counterweight inertia forces are not in the plane of the displacer motion.

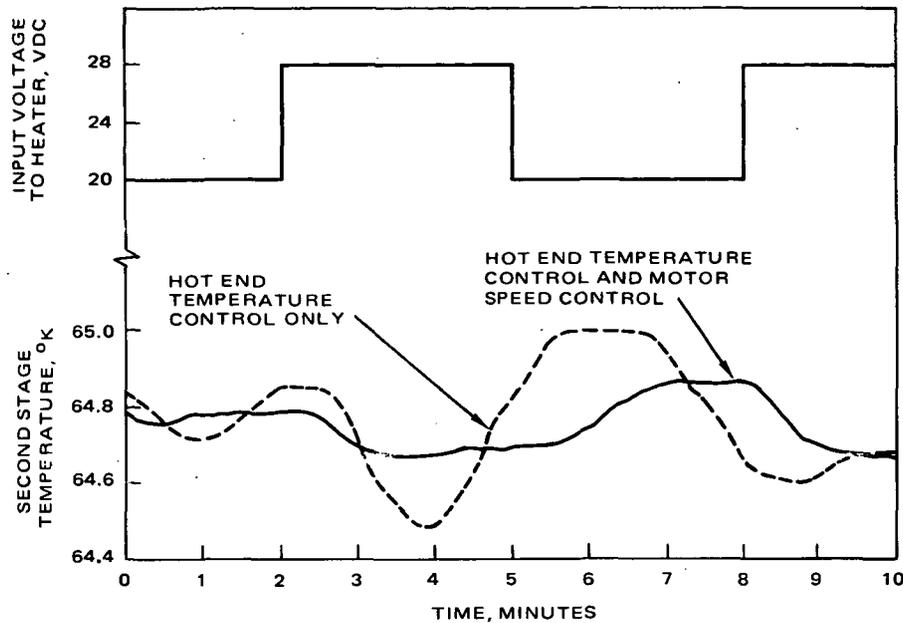


Figure 17. Cold end temperature variation with varying heater input voltage.

The inertia forces that the reciprocating motion of each displacer generates is expressed by the relationship

$$F = KMN^2 r \left( \cos \theta + \frac{r}{l} \cos 2\theta \right)$$

where

F = inertia force

K = a constant

M = mass of reciprocating parts

N = rotational speed

r = radius of crank

l = length of connecting rod

$\theta$  = crank angle

Note that the inertia forces consist of (1) the primary force, which varies sinusoidally at the fundamental frequency (which is equal to the rotational speed) and (2) the secondary force, which varies sinusoidally at twice the rotational speed. By employing relatively short displacer strokes (small crank radius) and long connecting rods, the magnitude of the secondaries forces can be made small with respect to the primary forces. Also, with both displacers in the same plane and arranged 90 degrees apart as they are in the the VM cycle, it is possible to completely balance the primary linear inertia forces. This is accomplished by making the product of the cold displacer mass and crank radius equal the product of the hot displacer mass and crank radius and by adding counterweights. As shown in Figure 18, the rotating force from the counterweight equals the resultant shaking force from the motion of the displacer. This leaves the secondary shaking forces unbalanced, but these are much smaller than the unbalanced primary forces because of the geometry of the connecting rod/crank in the VM refrigerator.

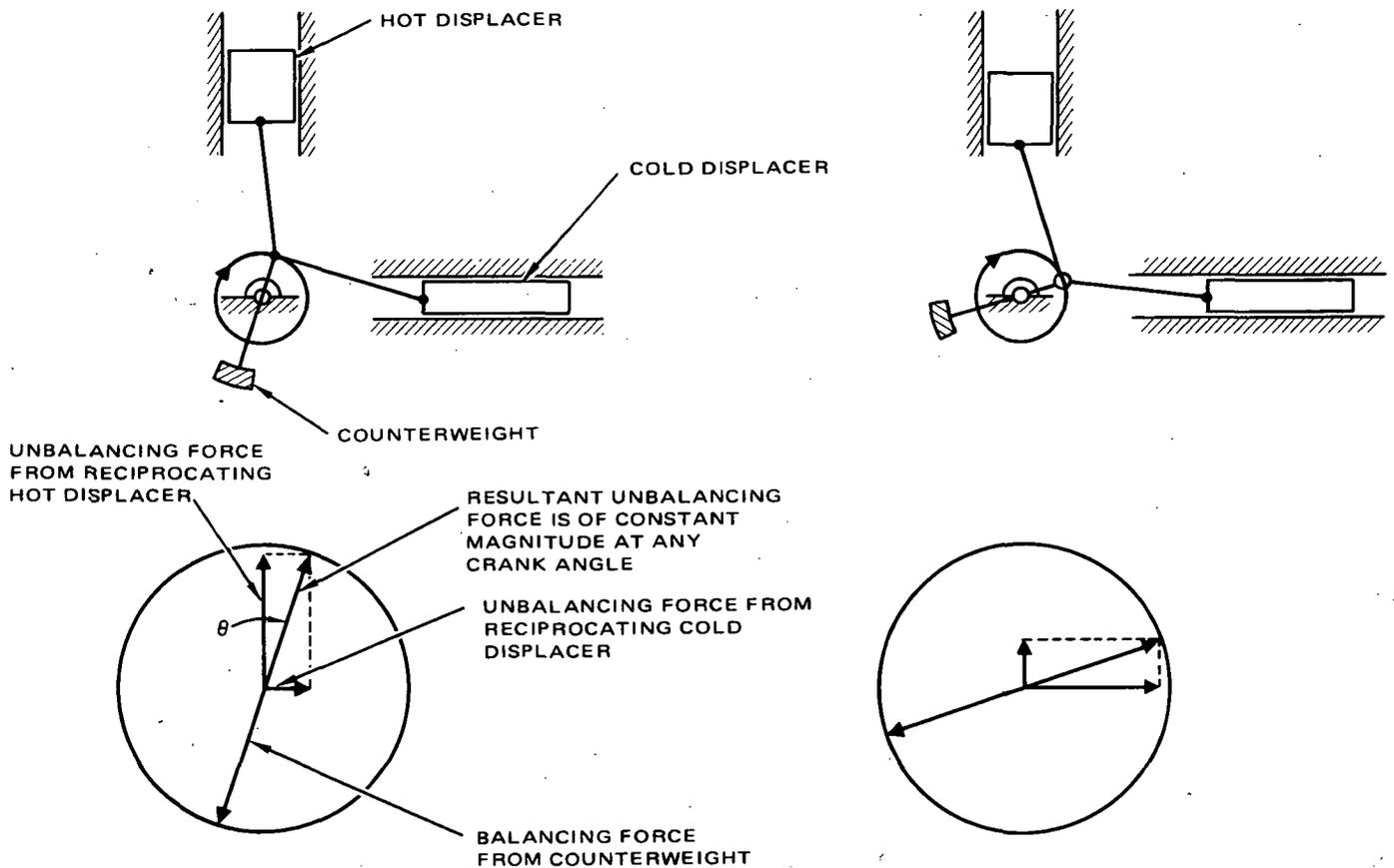


Figure 18. Diagrams for primary inertia forces.

By adding a second counterweight, it is possible to completely balance the rotating inertia forces due to the imbalance of the rotating parts of the crank mechanism. Designing the displacer crank mechanism so that the center of gravity of each is in the same plane eliminates unbalanced coupling forces. The refrigerator parameters affecting dynamic balance are:

$$M_h = \text{mass of hot displacer} = 68.1 \text{ gm}$$

$$M_c = \text{mass of cold displacer} = 196.2 \text{ gm}$$

$$r_H = \text{radius of hot displacer crank} = 0.432 \text{ cm}$$

$$r_c = \text{radius of cold displacer crank} = 0.15 \text{ cm}$$

$$l_h = \text{length of hot displacer connecting rod} = 3.56 \text{ cm}$$

$$l_c = \text{length of cold displacer connecting rod} = 3.3 \text{ cm}$$

$$n = \text{rotational speed} = 300 \text{ rpm}$$

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From these values are calculated the following secondary shaking forces.  $F_h = 0.036 \text{ N}$  (0.008 lbf) from the hot displacer and  $F_c = 0.0133 \text{ N}$  (0.003 lbf) from the cold displacer. Since the maximum secondary forces from both displacers occur simultaneously each 90 degrees of rotation and are in a direction 90 degrees apart, the resultant combined maximum secondary forces is  $\sqrt{F_h^2 + F_c^2} = 0.0378 \text{ N}$  (0.0085 lbf) or 0.001 g for a 0.036 N (8 lbf) refrigerator. As the crankshaft rotates, the force varies from maximum to zero and back to minimum with each 90 degrees of rotation.

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In practice, it may not be possible to completely balance all of the primary forces and to precisely split the rotating counterweights to eliminate all rocking couples. However, the preliminary analysis indicates that the total imbalance forces will be 0.0011 g at 5 Hz, which is negligible compared to the vibration of the space vehicle (0.25 g at 5 to 35 Hz) for which the detector dewar support was designed.

### Description of Refrigerator

The baseline refrigerator conceived to meet the LSIR requirement is a two-stage VM cycle cryogenic refrigerator. Figure 19 is an outline and mounting drawing showing the basic refrigerator, the space radiator mounting surface, and the refrigerator mounting surface. The basic refrigerator consists of a cold end assembly, a hot end assembly, a motor and drive mechanism assembly, and a crankcase housing as shown in the mechanical layout (Figure 20). The detector dewar support was designed to be integrated with the basic refrigerator as illustrated in Figure 20. Also shown in the figure is the cold end assembly, which consists of two basic components: the cold cylinder and the cold displacer. The two cold stages are coaxial. The cold

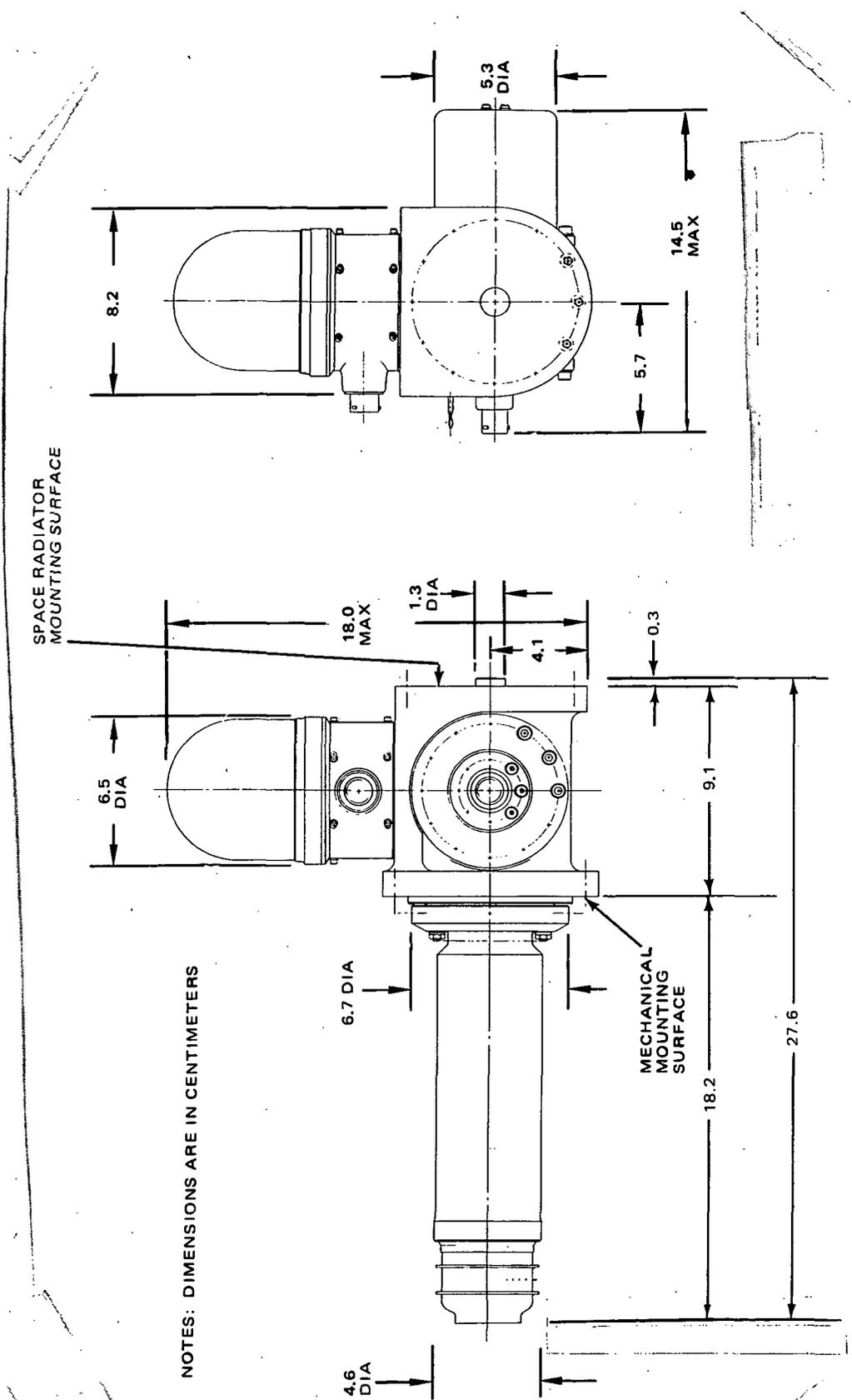


Figure 19. Outline and mounting drawing of LSIR refrigerator.

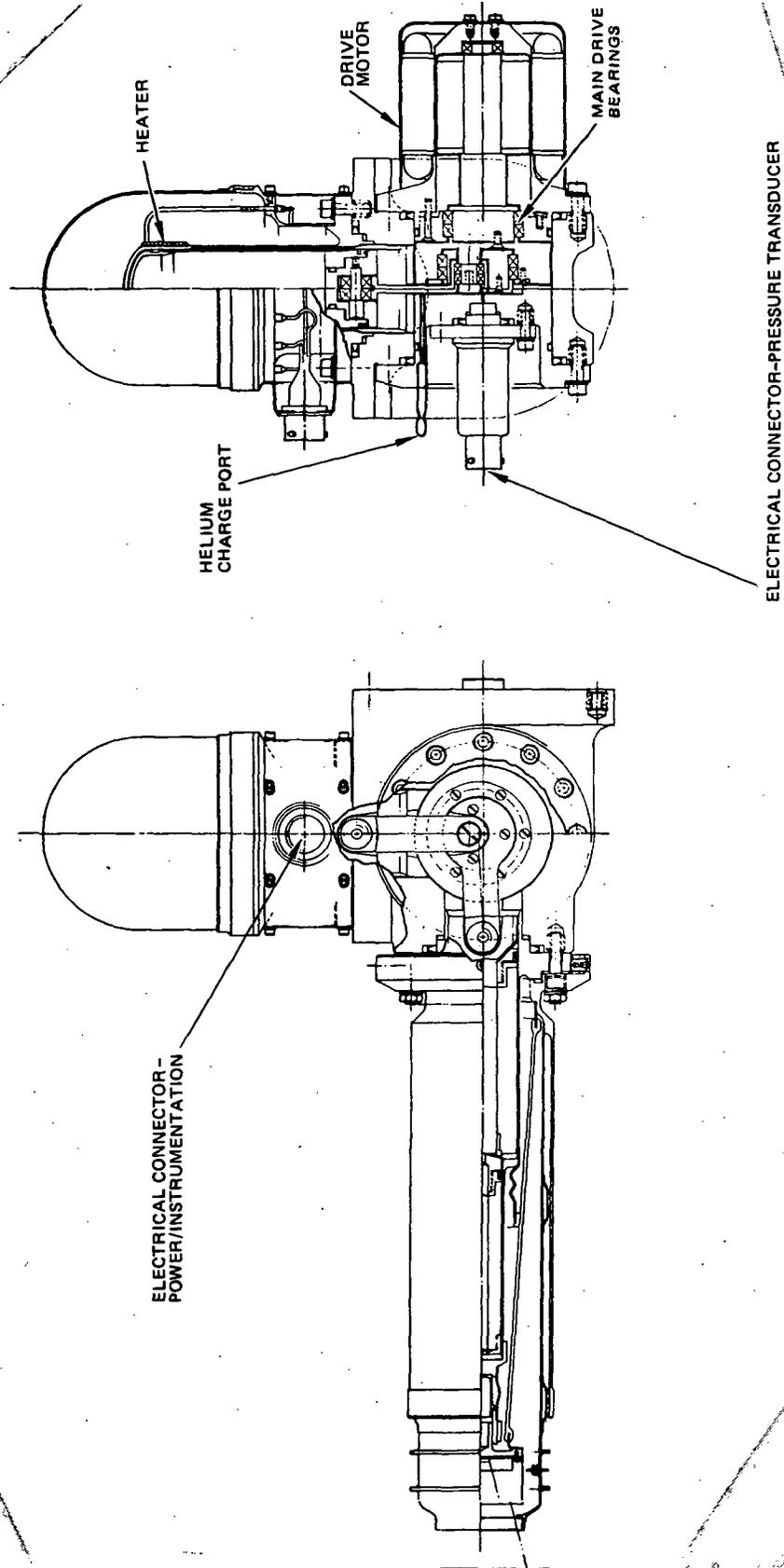


Figure 20. Layout of LSIR refrigerator.

cylinder is a thin walled, stainless steel cylinder, which serves as a helium pressure vessel. The walls of the vessel must be thin in order to reduce the conduction loss between the cold stages, and since they are thin, the dynamic variation in the axial direction due to pressure variations exceeds detector alignment requirements. Therefore, the cold cylinder cannot be used to structurally support the detector array. Braided copper straps, which provide flexible thermal interfaces between the cold stages and the detector support, permit the heat loads to be transferred to the refrigerator. A teflon ring at the 65°K end maintains good mechanical/thermal contact between the straps and the detector mounting surface when the refrigerator cools down. The copper surfaces are gold flashed to prevent oxidation (oxidation would increase the thermal impedance to the cooling load) and to reduce radiation heat load. The integral flange at the base of the cold cylinder provides

- Surface for an O-ring and a metal seal
- A surface for bolting the cold cylinder assembly to the crankcase
- A mounting surface for the detector support dewar.

The regenerators for the first and second stages are built into the center of the cold displacer as shown in Figure 21. The packing material for these regenerators is 400-mesh stainless steel screen.

The cold displacer assembly, consisting of two separate displacers in series, reciprocates at 5 Hz with a 0.3-cm stroke (see Figure 21). Rider rings made of 15-percent glass filled teflon support each section of this assembly. Each displacer can be shimmed axially for optimum head clearance to thereby minimize the dead volume in the cold stage and to keep the piston from hitting the cylinder when the piston is at top dead center. The displacers are made of wound fiberglass in order to minimize thermal conduction losses, thermal distortion, and running clearance changes. Metal attachment ends are used.

Two dynamic lip seals on the cold displacer assembly force the gas to flow through the proper passages between the regenerators. These seals, which are made of 15-percent glass filled teflon and are expanded by continuous garter springs, ensure practically no leakage and contribute only slightly to frictional drag. The seals are of a unique design that is particularly suitable for cryogenic applications.

The hot end assembly consists of the hot cylinder with heaters and temperature sensors, a hot regenerator assembly, a hot displacer with rider rings, and an insulation container. Figure 22 shows the basic configuration of the hot end. The hot cylinder is made of Rene 41, a nickel base alloy whose creep strength will allow 20,000 hours of satisfactory operation. Two sheathed electrical heaters brazed to the cylinder provide the external heat input to the refrigerator.

One of the two heaters on the hot cylinder is the main heater; it heats the hot cylinder and provides temperature control. A temperature controller

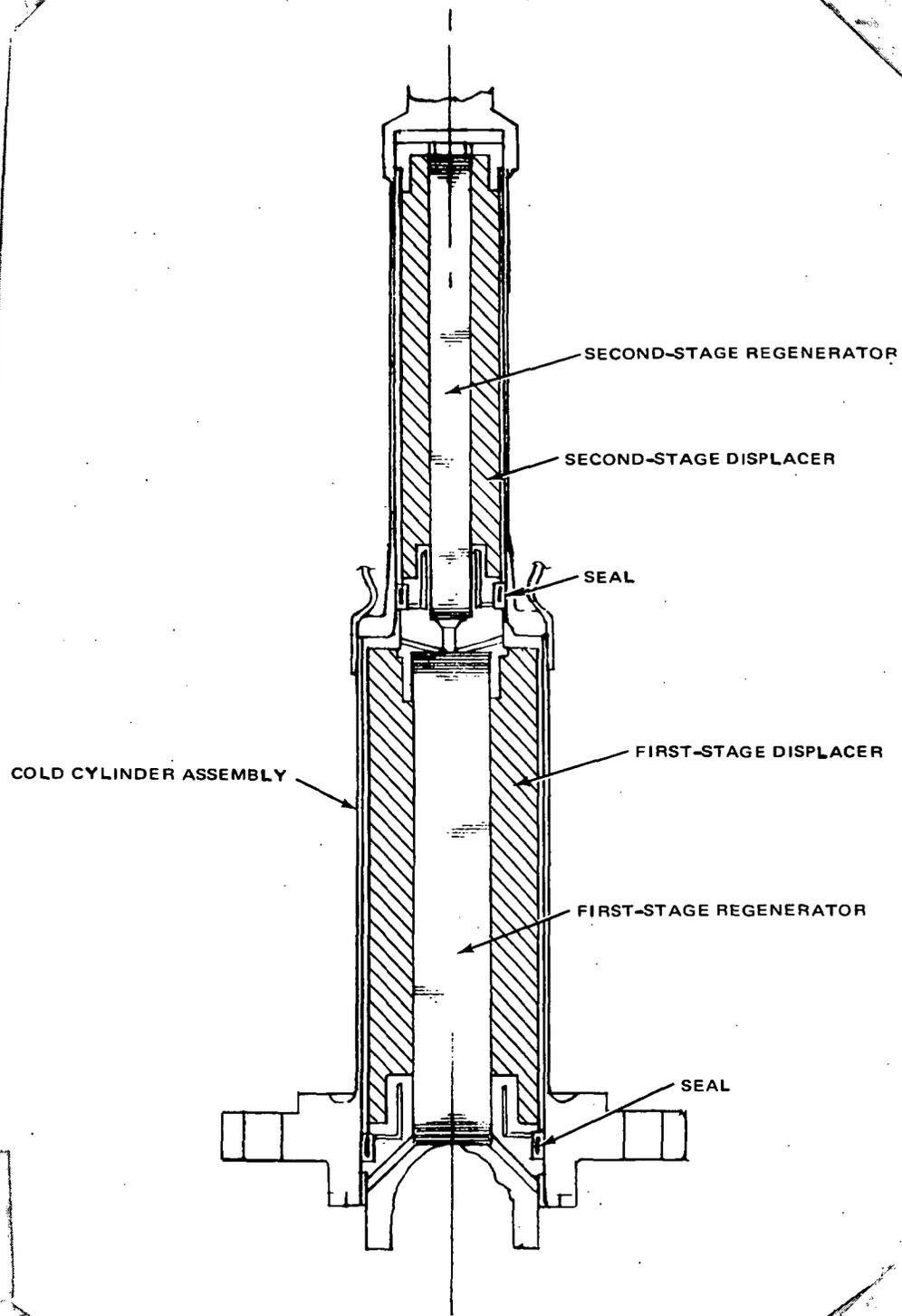


Figure 21. Cold end assembly of LSIR refrigerator.

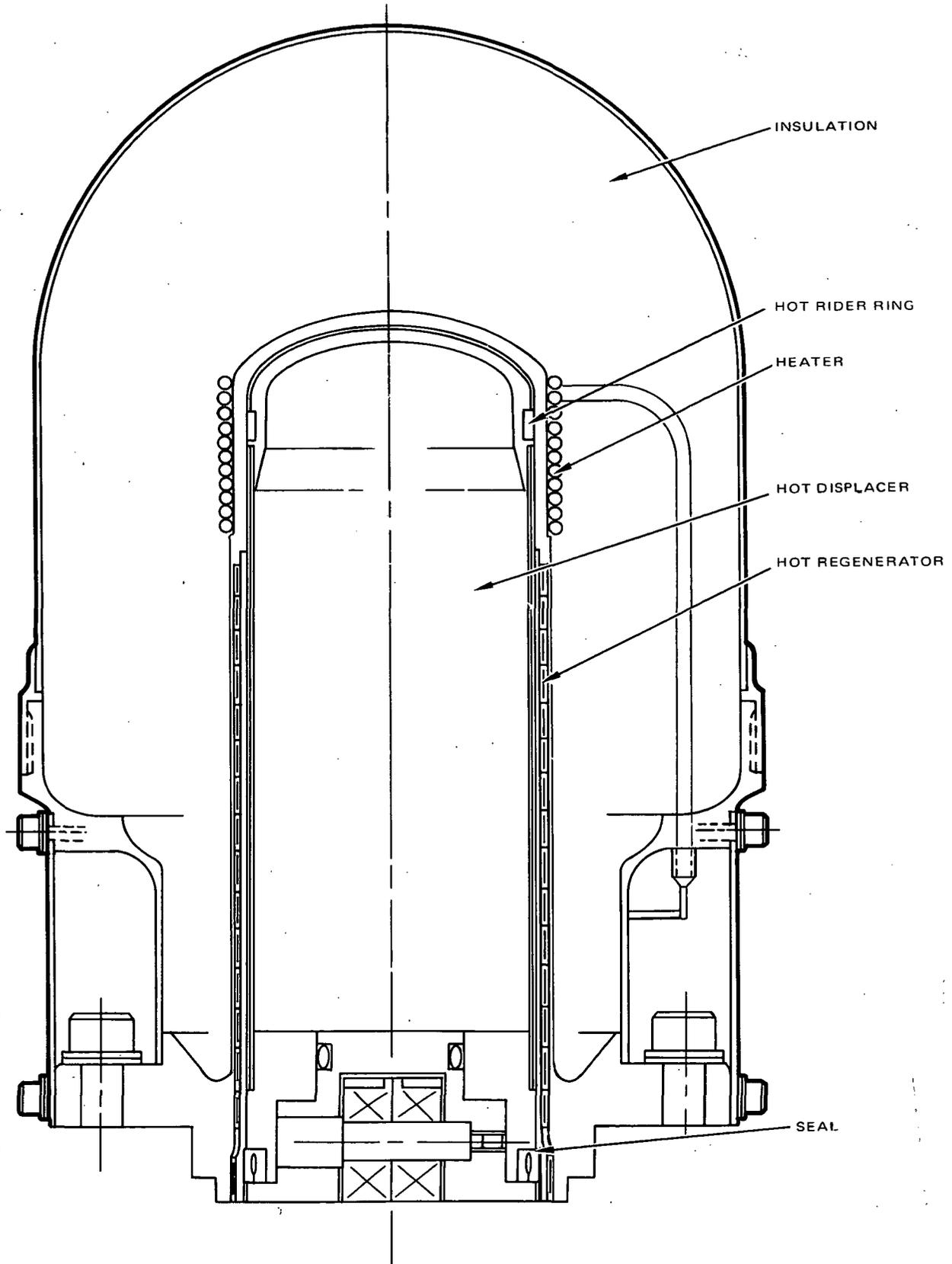


Figure 22. Hot end assembly of LSIR refrigerator.

*PL*

modulates the power to it to thereby maintain the required hot cylinder temperature. The second heater is a redundant heater, which can be energized in case the main heater fails. Both heaters have the same power rating, i.e., 80 watts at the low end of the tolerance on the 24-vac power. The heaters are helically coiled Nichrome V wire in Inconel 600 sheath. Boron nitride insulates the element from the sheath. A 200-series nickel wire is utilized for the termination of the heater elements.

Three identical platinum resistance temperature sensors will be clamped to the hot cylinder. Their attachment clamps are brazed over the outer surfaces of the heater. These sensors are used in maintaining the heater temperature. One of these controls hot cylinder temperature, the second provides independent overtemperature protection, the third is redundant and can, on command, be switched into either the control circuit or the overtemperature circuit should either of the primary sensors fail.

An aluminum enclosure surrounding the hot end assembly contains thermal insulation to reduce the amount of heat loss externally from the hot cylinder. This container is filled with solid Min-K insulation and bolted to the hot cylinder base.

The hot regenerator matrix consists of 40-mesh stainless steel screen packed into the annular space between the cylinder wall and an external regenerator shell. Ports are provided at the top and bottom of the shell, and a seal is used between the shell and the displacer to allow the gas to flow through the regenerator. The hot displacer is an electron beam welded assembly made of Inconel 718. Min-K insulation is packed into the piston before it is welded shut. The assembly is then evacuated and sealed. It has stiffening ribs to increase its resistance to collapsing. A spring loaded lip seal at the ambient end of the hot displacer forces the gas to flow through the external hot regenerator. The hot displacer is supported at the top by a rider ring. The material chosen for the hot end rider will depend on the findings from the VM wear rate program now underway. Preliminary screening of materials for that program indicates that a rider ring of composite refractory metal/molybdenum disulfide (such as Boeing hot compact 6-84-1) sliding on a hard coated Inconel 718 liner may provide the necessary wear life for the LSIR application. The material for the ambient rider ring will also depend on findings from the wear rate program, although riders of 15-percent glass loaded teflon have performed satisfactorily in previous programs.

The housing assembly contains the motor drive assembly, serves as a heat exchange medium between the crankcase heat exchanger and the space radiator, serves as an attachment structure for the space radiator, and serves as the refrigerator mounting surfaces. The housing is made from an aluminum alloy. The use of the space radiator was specified in the SOW. The sizing of this radiator was not a part of this study program.

All flanges that serves as helium seals have surfaces for both an elastomeric O-ring and a metal K-seal. The flanges are rigid enough that, after the system is pressurized, the seals can withstand the pulsating pressure and thermal expansion effects at different coolant temperatures.

A major concern in the design of a long life VM refrigeration system is the leakage of helium through the assembly flange seals. Metal seals with indium plating are used for a hermetic seal. This design will maintain sealing integrity even when the flanges are deflected by as much as 0.002 inch. The rigidity requirement necessitates the large number of bolts used at each closure. Since cyclic pressure loads on the metal K-seal can cause fatigue stresses within its flexible lips, an O-ring is placed between the working gas and each seal in order to reduce these stresses. This ring also seals off the dead volume introduced by the K-seal.

The greater portion of the energy that a VM refrigerator uses is thermal; the drive motor is needed only to overcome frictional forces and pressure drop forces across the hot and cold displacers. The motor driven drive mechanism maintains the proper phase relationship between the displacers. The motor torque and power requirements are summarized below.

	Operating Mode	Standby Mode
Running torque, cm-N	6 (8.5 in-oz)	5.5 (7.8 in-oz)
Starting torque, cm-N	7.7 (10.9 in-oz)	7.3 (10.4 in-oz)
Average output power, watts	16	16

The drive motor is a two-phase induction motor that has the windings separated from the working fluid by a stainless steel shell. This prevents the windings from contaminating the working fluid. However, the added air gap does decrease the efficiency of the motor. Figure 23 shows the configuration of the motor. The motor inverter is housed in the EIFU. The motor is designed for a normal operating speed of 300 rpm, but the speed can be varied by  $\pm 10$  percent upon command from the EIFU, as required for hot end temperature control.

The drive mechanism assembly consists of the linkages for the hot and cold displacers, the crankshaft, the counterweight ring, a crankshaft adapter, the drive housings, the motor stator and rotor, and all of the bearings.

The cold and hot displacers have 0.3-cm and 0.87-cm strokes, respectively. All bearings in this assembly are dry lubricated, duplex, preloaded pairs designed for contact stress levels of less than  $72,936 \text{ N/cm}^2$  under combined loading. The hot displacer bearings have an ID of 0.476 cm and an OD of 0.953 cm. The cold displacer bearings have an ID of 1.9 cm and an OD of 2.54 cm. The bearing retainers are made of rulon A (filled teflon) with a 5 percent  $\text{MoS}_2$ . The crank mechanism is preassembled before it is installed in the crankcase.

In order to minimize the dead volume in the crankcase, the normal clearance between the assembled parts is 0.0127 cm. All components of the drive assembly are made of titanium in order to reduce weight and yet maintain approximately the same coefficient of thermal expansion as that of the bearing

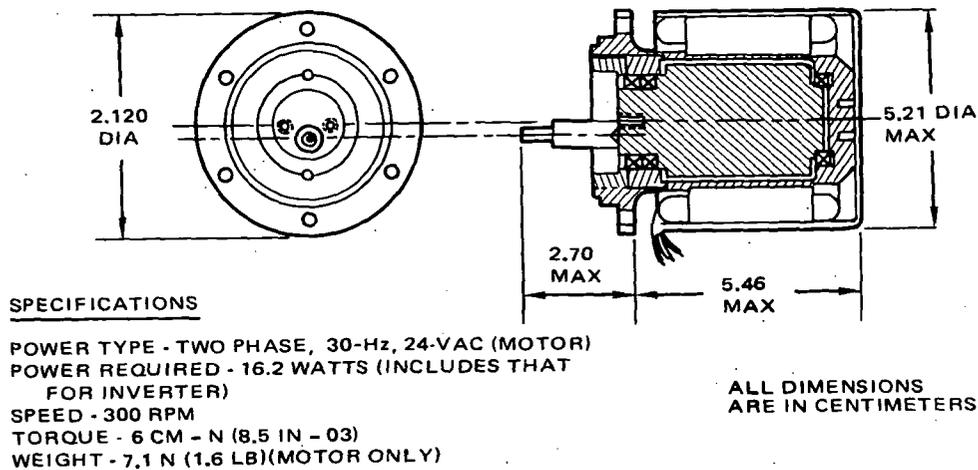


Figure 23. Motor configuration.

material. The drive housings provide a mounting surface for the main drive bearings. These bearings are the outboard set of duplex pair shown in Figure 20. The rotor of the motor turns the crankshaft around the longitudinal axis of the drive housings. The crankshaft contains two eccentrics that drive the cold displacer and the hot displacer linkages. The counterweight ring and crankshaft are connected to the crankshaft and rotate with it. The two internal sets of bearings are the connecting rod bearings, which allow the crank to rotate in relation to the connecting rod of the displacer.

The refrigerator is dynamically balanced to reduce the internally induced vibration that might perturb the infrared sensor. By proportioning the masses of the cold and hot displacer and by utilizing counterweights in the counterweight ring, it is possible to almost completely balance the primary inertia forces. The maximum value of the unbalanced secondary force is calculated to be less than 0.0378N (0.0085 lbf) or less than 0.0011 g.

### Interfaces

The cold cylinder/detector-dewar, crankcase/radiator, and refrigerator/LSIR housing assembly interfaces have been studied. As a result of these studies, interfaces consistent with good system performance and LSIR program requirements have been defined.

Cold cylinder/detector-dewar interface. - It was not possible to mount the detector array directly on the cold cylinder of the refrigerator because of the requirement that the detectors were not to move either axially or radially. A cold cylinder design that would meet this stiffness requirement would have excessive thermal losses.

By using a separate support structure, it is possible to design the cold cylinder for optimum performance and design the support structure for low heat loss and required rigidity. A detector support structure that is part of a permanently evacuated dewar has been demonstrated. This arrangement allows the detector support to be adjusted with respect to the optical system with a minimum of effort. Flexible heat transfer straps to both stages isolate the motion of the refrigerator from the support and allow the position of the detector array to be precisely adjusted.

A glass cone was used as the main support because it has good thermal and mechanical properties and can be easily fabricated and cleaned. Figure 20 is a view of the detector/dewar integrated with the refrigerator. The detector array can be axially adjusted by using different shims between the dewar and the mounting surface. Small radial adjustments are made with four machine screws mounted at the base of the detector/dewar as illustrated in Figure 24.

Braided copper straps provide flexible thermal interfaces between the cold stages and the detector array support. At the 120°K stage, one end of the copper straps is brazed to the cold cylinder and the other end to a copper ring, (see Figure 24). When the cold cylinder is fitted into the glass dewar and the ring is pressed against the dewar wall, there is a thermal interface between the dewar and the cold cylinder. Thermal grease is utilized at this interface. With a copper strap area to length ratio of 0.053 and a 0.48-watt first stage heat load, the temperature drop from the dewar wall to the 120°K cold stage is limited to approximately 2°K.

At the 65°K stage, one end of the copper straps is brazed to the cold cylinder and the other end to a copper ring that is mated to the mounting surface of the detector array; this surface is also copper. A teflon ring slipped around the mating surfaces maintains good mechanical/thermal contact at the interface when the refrigerator cools down. With a copper strap area to length ratio of 0.044 and a 0.04-watt second stage heat load, the temperature drop from the detector array to the 65°K cold cylinder is limited to approximately 1°K.

Figure 25 shows the predicted deflection of the support when subjected to a 1-g load. The specified load from the spacecraft will be 0.25 g. The radial motion of the detectors will be less than  $\pm 0.10 \mu\text{m}$ . The natural frequency of the support is 994 Hz, and the maximum bending stress at the base of the support is  $2.82 \text{ N/cm}^2$ , (4.1 psi).

Refrigerator/Heat Rejection Interface. - Waste heat is rejected from the VM cooler to a radiator mounted in the spacecraft; this radiator is bolted directly to a circular flange in the crankcase region of the refrigerator. The crankcase housing and the radiator mounting surface have been designed to allow the waste heat to be transferred to the radiator without causing large temperature drops in the crankcase. The crankcase and radiator mounting surface configuration shown in Figure 19 is capable of transferring, by thermal

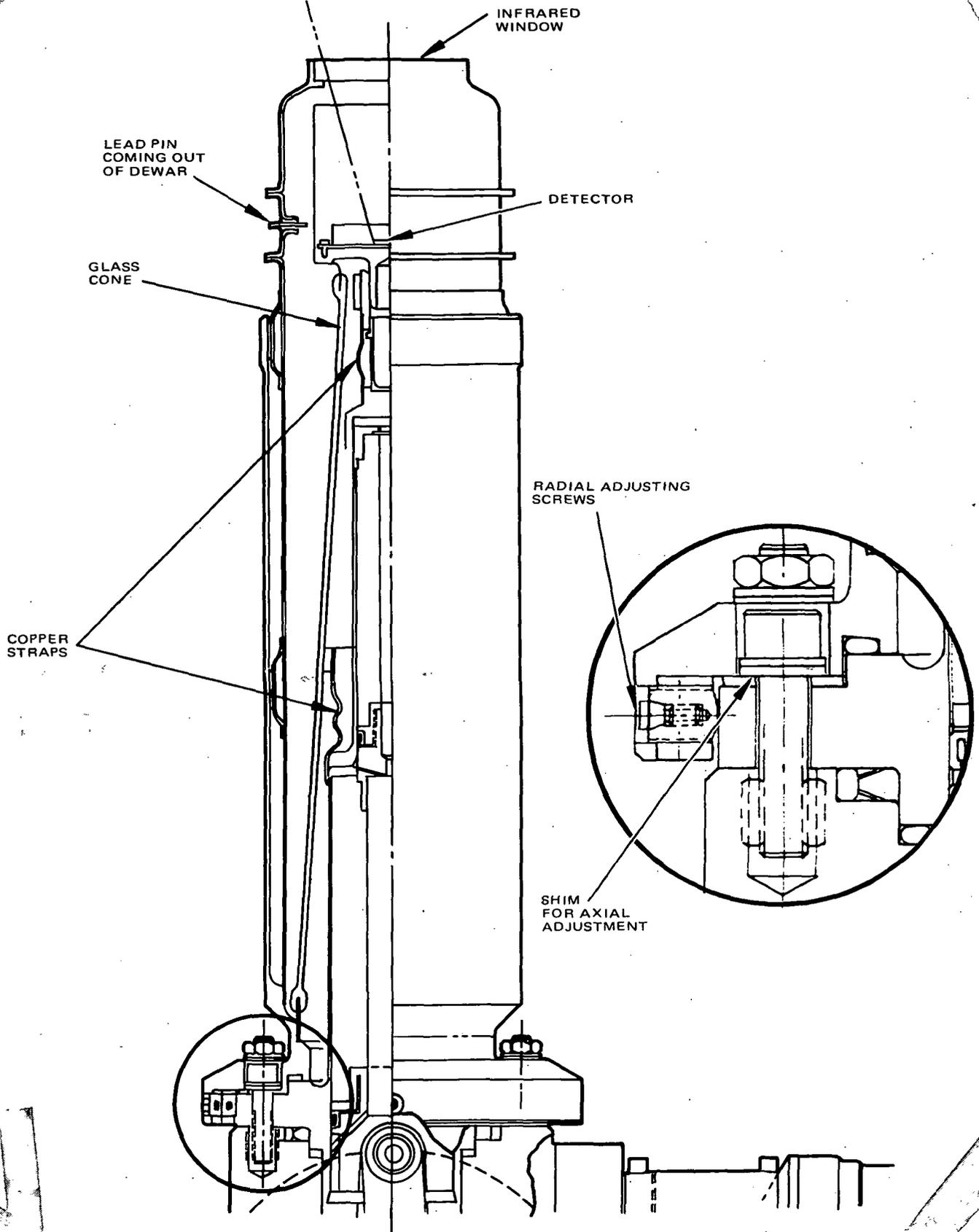


Figure 24. Cutaway of cold cylinder/detector dewar interface.

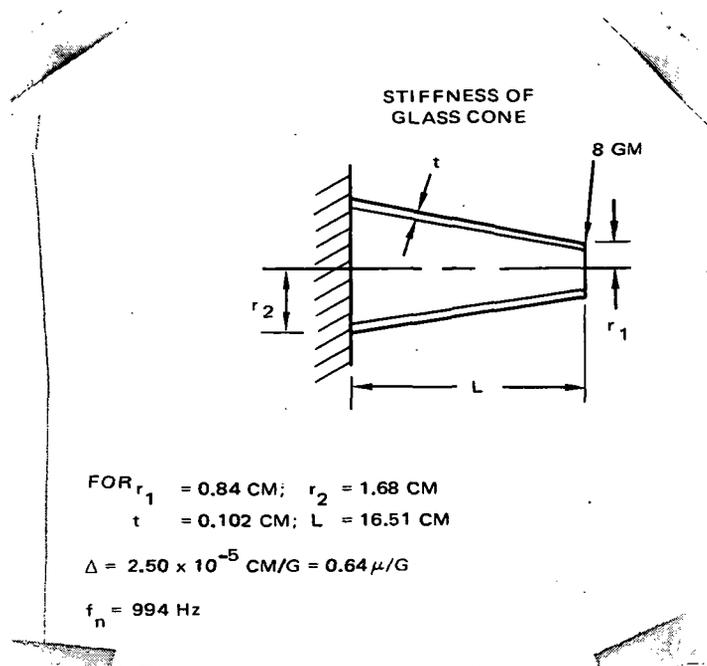


Figure 25. Detector dewar deflection.

conduction, 90 watts of waste heat from the crankcase heat exchanger to the radiator mounting surface with a maximum temperature drop of approximately  $12^\circ\text{K}$ . The temperature gradient around the circumference of the annular heat exchanger is approximately  $7^\circ\text{K}$ .

The temperature gradients in the crankcase were calculated by employing a 19-node mathematical model of the thermal path between the crankcase heat exchanger and the radiator mounting surface. The thermal model was of the nodal network type, which divides the system into discrete thermal nodes that are tied together by conduction.

For operation on the ground, it will be possible to operate the machine by blowing air over the radiator or by replacing the radiator with a liquid cooled heat exchanger.

Refrigerator/LSIR housing assembly interface. — The interface between the refrigerator and the LSIR housing assembly has been defined as well as possible with the information available. Figure 19 gives mechanical interface data on the refrigerator; it shows the space radiator mounting surface, a recommended structural mounting surface for the refrigerator on the housing, and the probable locations for the electrical connector and coolant fitting. The structural mounting surface is a flange with symmetrically located bolt-holes. The type of electrical connector and coolant fitting have not been specified because the requirements are not sufficiently defined. These details are best defined during the design phase when the refrigerator and spacecraft interfaces can be developed together.

## Electronic Interface Unit

The EIFU consists of the motor inverter, temperature control circuits, and all other power conditioning circuits. The overall size of the EIFU is shown in Figure 26, and the block diagram is shown in Figure 27. A detailed installation drawing of the EIFU was not prepared because its final configuration has not been established. A wide variety of EIFU configurations is possible, depending on the electronic packaging concepts used and the size and shape constraints imposed by the spacecraft. The EIFU performs six essential functions; it

- controls the refrigerator drive motor
- controls the second-stage temperature
- controls the hot cylinder temperature
- provides signal conditioning for the instrumentation
- contains the fault detection/interlock circuits
- contains the control logic for processing commands from the ground

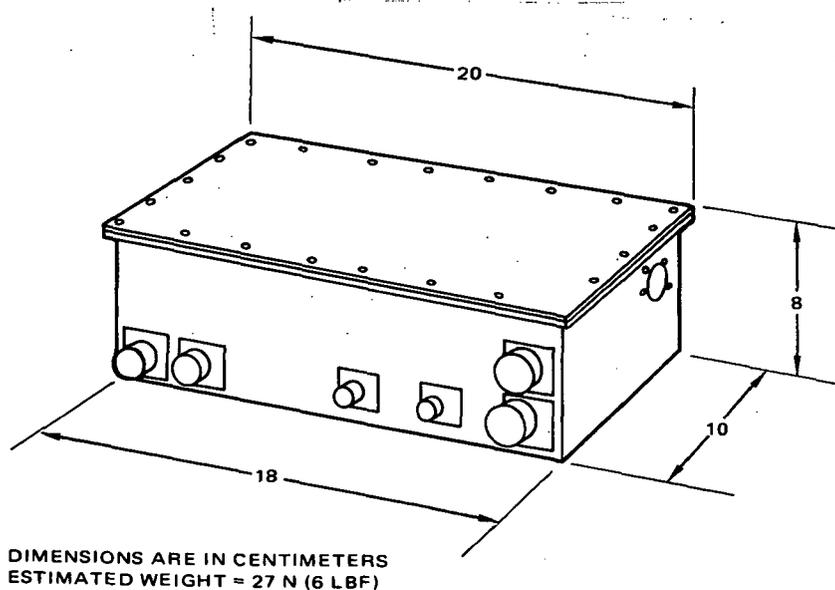


Figure 26. Configuration of typical EIFU.

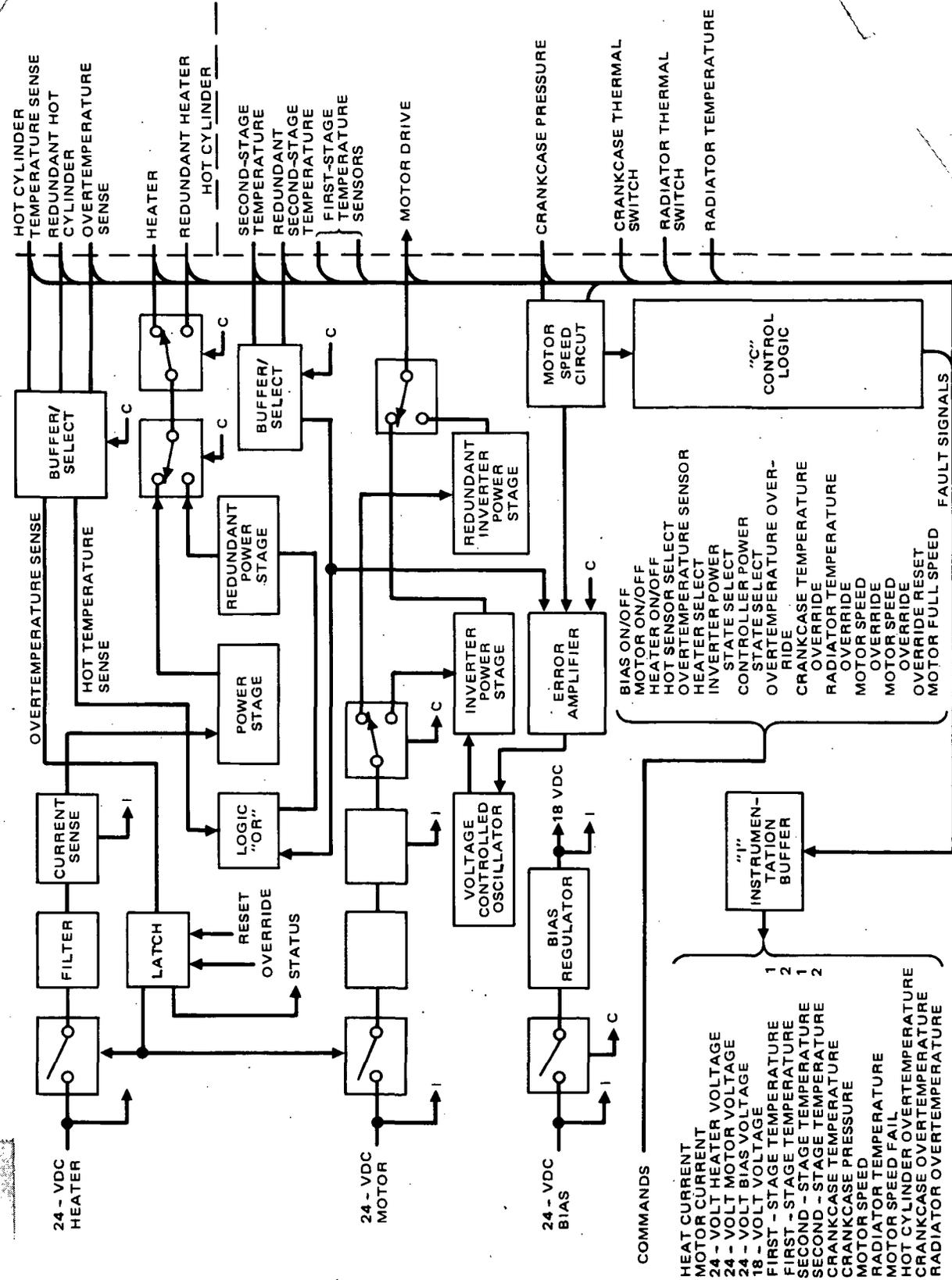


Figure 27. Block diagram of EIFU.

Refrigerator drive motor control. — A 24-vac induction motor was selected to drive the refrigerator. This type of motor was used on previous programs and satisfies the LSIR drive requirements. This motor requires a two-phase a-c drive waveform, with a phase separation of 90 degrees. The motor inverter converts the 24-volt spacecraft power to the two-phase a-c power required by the motor. The speed of the motor depends on the frequency of this a-c power, and this frequency is varied in response to a signal from the second-stage (65°K stage) temperature controller. The frequency signal from the second stage is fed to a logic block that splits the signal into the two phases required by the motor. The logic block also shifts these signals to the level required by the power amplifier. The power amplifier is a power semiconductor switching bridge; one bridge is required for each motor phase. By opening and closing the semiconductor switches in each leg of the bridge, an a-c waveform is generated to drive the motor. Since the power amplifier is a heavily stressed component (from a reliability standpoint) in the motor inverter, a redundant power amplifier has been included to ensure that the motor inverter will meet the long life goal of 20,000 hours.

Second stage temperature control. — Figure 28 is a block diagram of the second-stage (65°K stage) temperature controller. The temperature of this stage is controlled by varying two parameters.

1. Hot end temperature
2. Motor speed

Hot end temperature is varied by varying the amount of power supplied to the hot end heater. Because of the thermal mass at the hot end, this control has a rather slow response. To regulate against more rapidly varying disturbances, the motor speed is varied.

The temperature controller receives temperature sensor inputs from both the cold and hot ends of the refrigerator. The cold end temperature signal is compared to a temperature reference in order to produce an amplifier error signal, which is converted to a 0.10-Hz pulse width modulated drive signal to the hot end heater and a frequency command to the motor inverter. The control circuit processes the error signal from the temperature sensing circuit and generates an output signal that governs the state of the power stage. The power stage is pulsed at a rate of 0.10 Hz. When less power is required, the power to the heater is turned off during part of the 0.10-Hz cycle, i. e., if the temperature of the cold cylinder falls below the set point, the off-portion of the cycle is lengthened (i. e., the duty cycle is reduced), and when the temperature rises above the set point, the off-time becomes shorter. The length of the duty cycle depends on the amount of power that the control heater must provide to maintain the temperature of the cold cylinder at its set point. This power level is a function of cold stage loading, ambient (or crankcase) temperature, motor speed, and input voltage. The 0.10-Hz frequency was chosen to allow long heater switching times to satisfy the EMI requirements of MIL-STD-461.

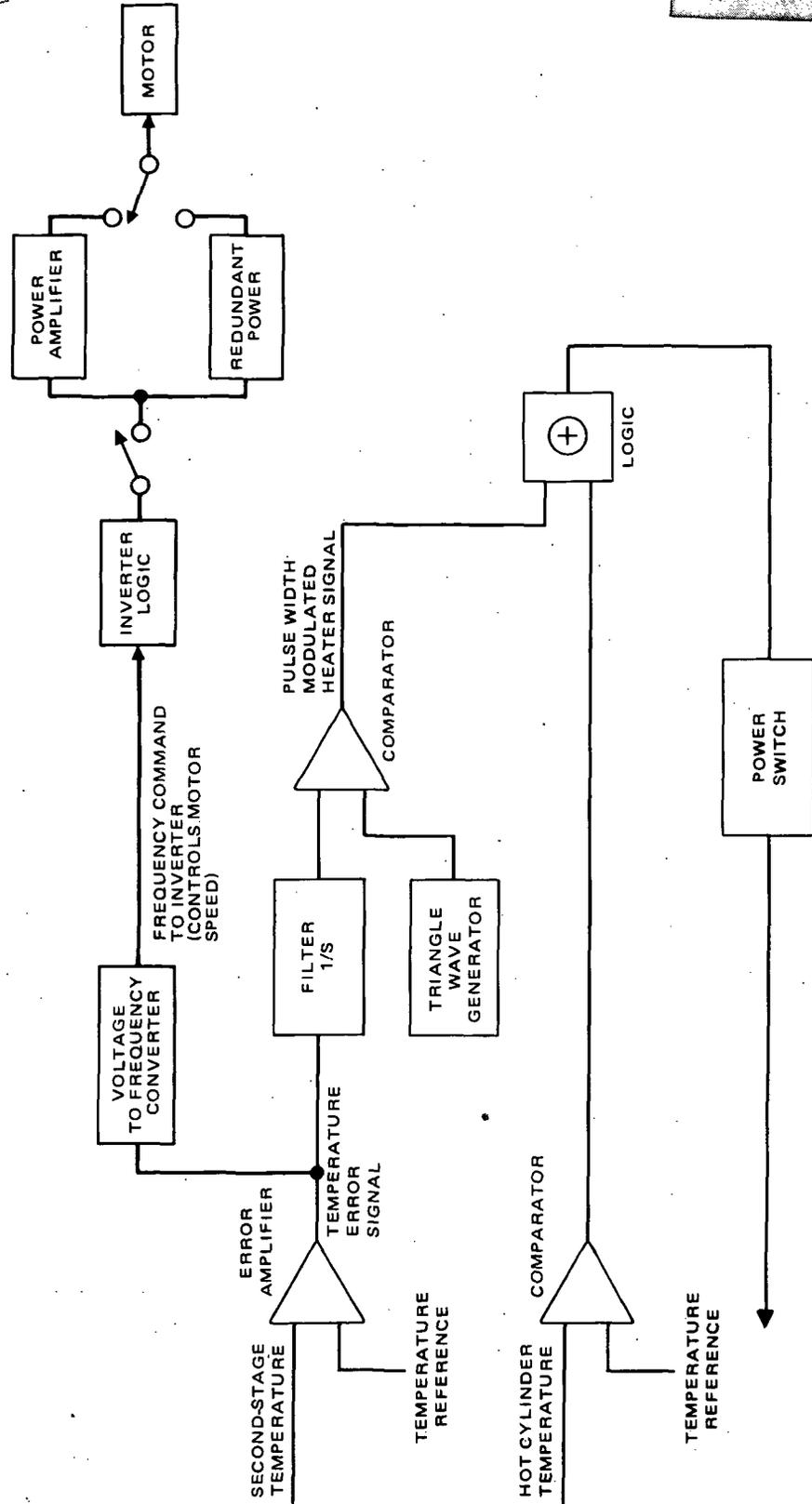


Figure 28. Block diagram of second-stage temperature controller.

The three commonly used types of temperature controllers are proportional d-c, proportional pulse width modulation, and on-off. A proportional d-c controller controls temperature well and has low ripple but is inefficient. Proportional pulse width modulation overcomes the efficiency problem; however, the waveforms involved generate EMI. The on-off controller is efficient, and it is possible to eliminate EMI by slowing switching times; however, there is some ripple in the controlled temperature as a result of the heater's switching on and off. The design selected uses a proportional pulse width modulator with slow switching times. The temperature ripple that results from switching the heater on and off slowly (0.10-Hz, or, once every ten seconds) is reduced by varying the motor speed. In view of its experience with other space systems of this type, Hughes does not expect that the variation in current resulting from the 0.10-Hz heater waveform will cause problems in the power supply or the system. However, this cannot be determined until the power supply and other system components are more fully defined.

Hot end temperature control. - During cooldown, the hot end temperature may overshoot its maximum limit since the control loop will command maximum power until cooldown is achieved. Hence, a second hot end control loop is provided. This loop uses the hot end temperature input and controls the hot end temperature at its maximum value until cooldown occurs; the cold end control loop scales back the hot end temperature to maintain the second stage at its set point.

To provide the fail-safe capability of shutting off power to the heater when an overtemperature condition occurs, the temperature controller contains an overtemperature shutdown circuit. If the primary control sensor fails and if the temperature of the hot cylinder increases to 1005°K, the overtemperature circuit shuts off power to the heater on the hot cylinder. A second resistance sensor (in a bridge) similar to the primary control sensor and mounted on the hot cylinder signals the overtemperature circuit. When the output from this bridge exceeds a predetermined error signal, the overtemperature circuit activates the fault logic.

Instrumentation. - The instrumentation buffer (see Figure 27) generates the required excitation (1) for gallium arsenide diodes that monitor cold stage temperature, (2) for the thermistors that monitor crankcase temperatures, and (3) for the pressure transducer that monitors the refrigerator gas pressure. The instrumentation buffer also includes the electronics for buffering and scaling the instrumentation outputs for telemetry. Instrumentation outputs include first- and second-stage cold end temperatures; hot cylinder temperature; motor speed, crankcase pressure; heater, motor, and bias regulator current; and the four failure signals (hot cylinder overtemperature, motor failure, crankcase overtemperature, and radiator overtemperature).

Fault detection/interlocks. - To protect the refrigerator, the EIFU can detect the following fault conditions:

- Overtemperature of the hot cylinder
- Excessive crankcase temperature

- Overtemperature of space radiator
- Drive motor failure

If any of these malfunctions occurs, power to the motor and to all of the hot cylinder heaters is turned off, but power is still applied to the low power control circuits to enable the EIFU to accept commands and to telemeter data.

Figure 29 is a block diagram of the failure logic. An override is provided for each of the fault conditions; the fault signal must be present and the override signal absent before the fault can cause the power to be shut off. If a failure occurs, a signal is telemetered to indicate which mode of operation is affected in order that the cause of the fault may be determined.

Each of the fault signal relays can be reset. The fault must be removed and the appropriate relay must be reset by proper command (reset) before the refrigerator can resume operation. If an override has been employed, the appropriate relay must be reset by a proper command (override reset) to reactivate the protective circuit.

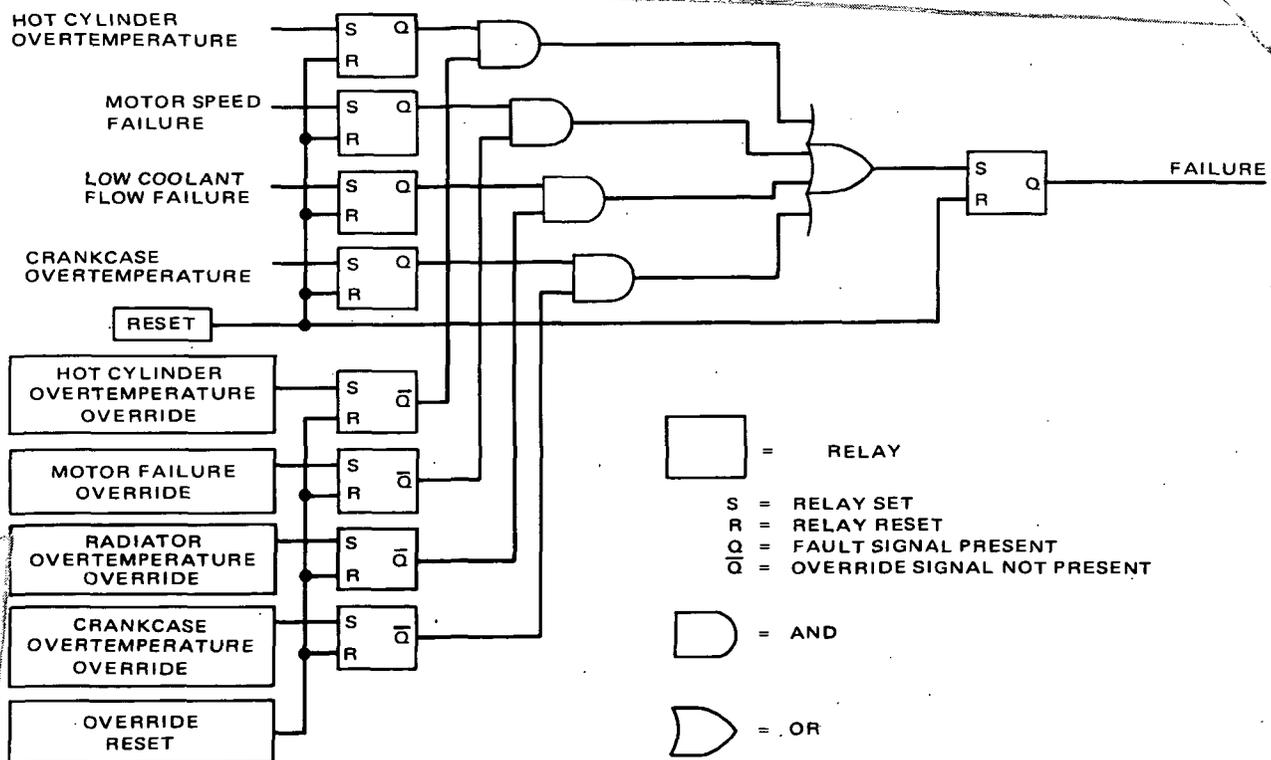


Figure 29. Failure logic.

Command interface. - The EIFU is the command interface between the spacecraft and the refrigerator. All commands are in the form of a short 24-vdc relay signal. Each command latches the appropriate relay in the EIFU; the commands and a description of each are included in appendix C. Redundancy functions will be controlled from the ground.

## SECTION VI IMPLEMENTATION PLAN

An implementation plan and schedule were developed on the basis of a possible follow-on program to consist of final design, fabrication, assembly, and test of three LSIR VM coolers and one set of ground support equipment (GSE) to support an LSIR flight experiment. This is in keeping with the SOW, and the schedule shown in Figure 30 reflects this preliminary plan.

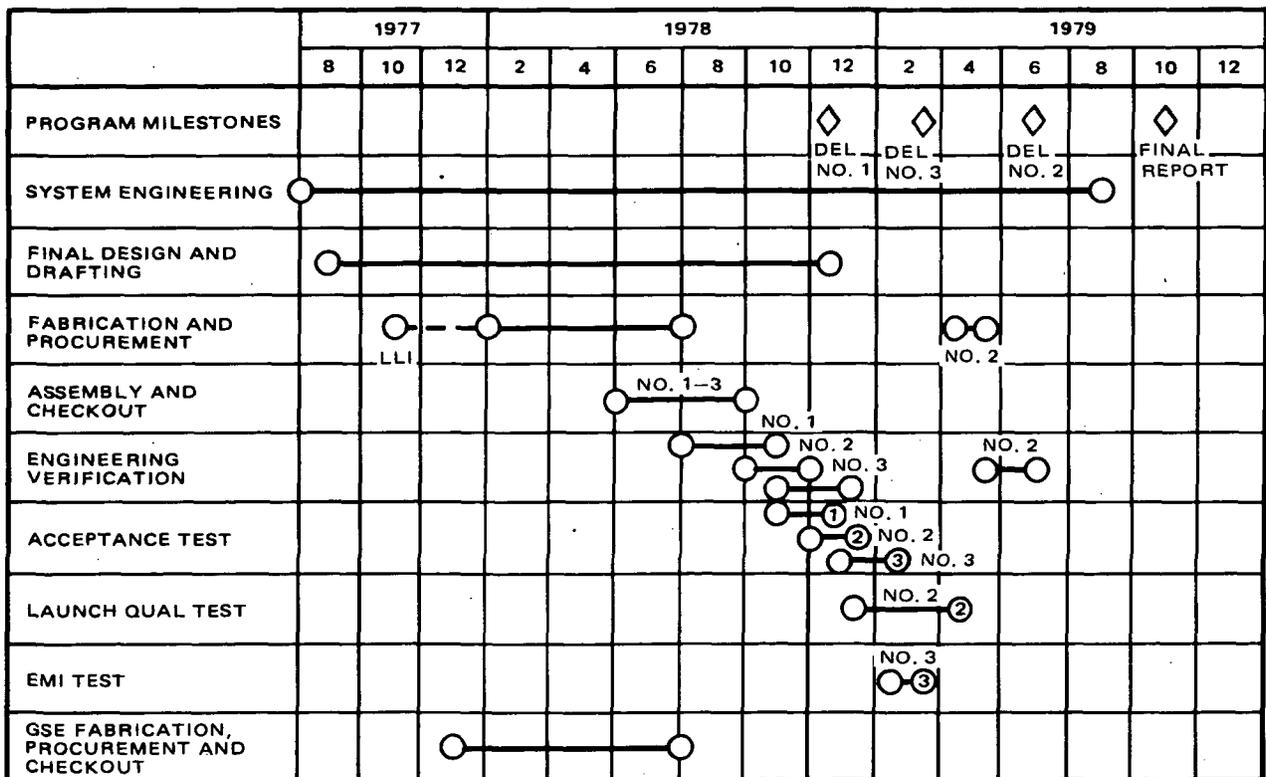


Figure 30. Program schedule (it is assumed that three sets of GSE are available).

## General Tasks

Any follow-on program requiring a flight LSIR cooler will include the general tasks briefly described below:

- System Engineering - Define in detail the specific requirements for applying the refrigeration system to the LSIR spacecraft and infrared sensor. Coordinate cooler requirements with those for a spacecraft heat rejection system, for a power supply system, and for telemetry, command logic, etc.
- Final Detail Design - Prepare a detail design, analysis, and working drawings for the refrigerator and the IFU; these should include procurement specifications, test specifications, etc. Before they are released, all drawings will be checked and approved by materials or components specialists and by the responsible cryogenic system engineer.
- Fabrication - Fabricate and/or procure parts for refrigeration systems. All parts, components, and subassemblies will be subjected to quality control inspection in accordance with approved Hughes Aircraft Company procedures for space equipment.
- Assembly and Checkout - Each refrigeration system will be assembled under quality control surveillance and then checked out to verify that the unit operates correctly in response to all commands. Preliminary leakage tests will verify that there are no coolant leaks and that the helium leak rate is within acceptable limits.
- Engineering Verification and Development Tests - All units will be tested to verify that they meet acceptable performance levels before they undergo formal acceptance tests. For the first unit, some development work may be needed to bring performance within specifications. All following units would be subject to similar modifications.
- Acceptance Test - This is a formal test, conducted under quality control surveillance, to demonstrate that each refrigeration system meets the specified requirements. This test would be defined in more detail in any follow-on effort. After a system passes acceptance tests, it will be considered qualified for sensor integration, launch qualification tests, EMI tests, life tests, or other tests that may be required.
- Launch Qualification Test - One or two units would be subjected to a series of launch qualification tests as described in the current SOW. The number of units used would be a function of the schedule.
- EMI Test - One unit will be subjected to EMI tests. It is expected that this type of test would be repeated later as part of the tests of an integrated system.

- Life Test - Although the SOW does not call for a life test and such a test is not shown in the typical program schedule, Hughes recommends that consideration be given to subjecting one of the refrigeration systems to an extended life test in order that as much operational experience as possible may be gained before an LSIR system is launched.
- Ground Support Equipment - Design, procure, assemble, and check out the necessary ground support equipment. This equipment typically will consist of the following.
  - Cryogenic test console; this comprises electrical power supplies, command input provisions, instrumentation and status outputs, etc.
  - Heat rejection unit to provide liquid cooling for the refrigeration system.
  - Vacuum station to provide the necessary vacuum environment during all refrigeration tests.

Should a life test be conducted, an additional automated data acquisition system will permit the manpower needed for data gathering and interpretation to be substantially reduced.

### Test Plans

Acceptance test. - Following informal engineering development and verification tests to ensure that performance is acceptable, all units will be subjected to a formal acceptance test conducted under quality control surveillance to demonstrate that the cooler meets the specified requirements. Table 9 lists these tests. The operating time and start-stop cycles accumulated in cooldown tests and capacity tests will be credited toward the 100-hour burn-in and ten start-stop cycle requirements. The purpose of these tests is to identify any weak or marginal components that might contribute to infant mortality type failures or that might fail as a result of thermal cycling of the refrigerator hot cylinder. If a unit fails to pass the acceptance tests, it will be reworked and retested as deemed appropriate by the responsible cryo system engineer. Of course, all rework and retests will be conducted and documented according to formal procedures and will be monitored by quality control personnel.

TABLE 9. ACCEPTANCE TESTS

- Verify conformance of system to drawing requirements
- Verify operation of command and safety devices
- Verify that no coolant leakage can be detected
- Determine helium leak rate
- Determine cooldown time
- Determine refrigeration capacity at design temperatures for
  - Normal operation
  - Standby operation
- 100-hour burn-in
- Ten start-stop cycles over the full range of hot cylinder temperatures

Launch qualification test. - Following acceptance tests, one refrigeration system will be subjected to a launch qualification test. The test procedures of MIL-STD-810C or MIL-STD-1540 will serve as a guide for these tests wherever applicable. Table 10 lists tests that make up the launch qualification tests. The explosive atmosphere test will be conducted first to demonstrate that the cooler operates safely. The succeeding tests need not be performed in any particular order, except of course the final acceptance test. The test levels for the various environmental tests will be established after the launch vehicle is selected and the environmental conditions are known.

EMI test. - One refrigeration system will undergo EMI tests. Because of the schedule based on one GSE, this system will likely be the one employed for launch qualification tests. The EMI tests are also listed in Table 10. They will be conducted in accordance with the procedures of MIL-STD-462 and any specific requirements for the LSIR program.

TABLE 10. LAUNCH QUALIFICATION AND EMI TESTS

<ul style="list-style-type: none"><li>● Launch qualification test - one system<ul style="list-style-type: none"><li>● Acceptance</li><li>● Explosive atmosphere</li><li>● High temperature</li><li>● Acceleration</li><li>● Vibration</li><li>● Shock</li><li>● Space simulation</li><li>● Acoustic noise</li><li>● Pyrotechnic shock</li><li>● Repeat of acceptance test</li></ul></li><li>● EMI test - one system<ul style="list-style-type: none"><li>● Radiated interference and susceptibility</li><li>● Conducted interference and susceptibility</li></ul></li></ul>
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Schedule

The schedule shown in Figure 30 for the follow-on program was based on a minimum time delivery of the first cooler to sensor integration in 16 months with test support of all units provided by three sets of GSE. The start date of 1 July 1977 was assumed; hence, hardware tests and deliveries would be completed before 1980 with few months leadtime. Because of the fabrication and procurement lead time needed, the first unit will be delivered after only a minimum of engineering development and before any launch qualification and EMI tests can be conducted. These last tests will be conducted with the second unit while the first unit is being integrated with the sensor. Unit 2 will be refurbished after its tests and will be delivered following acceptance test and delivery of unit 3.

It would be preferable and lower the development risk if the LSIR program would allow unit 1 to be delivered to sensor integration after launch qualification and EMI tests are completed. In addition, although a life test is now shown on the schedule, it is highly recommended that one of the refrigeration system be life tested early in the program with the support of a second set

of GSE and an automated data acquisition system. The program schedule shown can be made more flexible, even with the added life test, if the various tests can proceed simultaneously with the above mentioned second set of GSE.

## SECTION VII

### RELIABILITY AND LIFE STUDIES

Technology developed during the CMP and the Hi Cap programs and technology being developed during the current wear rate program are expected to advance the state of the art of the life limiting components of VM cryogenic refrigerators to the point where life goals of 20,000 hours can be realized. During these programs, the following major refrigerator life limiting components have been studied.

- Dynamic seals
- Bearings
- Rider rings
- Static seals
- Electric heater

The following information obtained during the analytical and experimental studies provides assurance that the refrigerator will meet its performance and service goals.

- Dynamic seals

Seals are required between the cold stages of the cold displacer to force the helium gas to flow through the regenerators and to prevent gas from bypassing the regenerators by flowing in the clearance between cylinder and displacer.

A seal is also required on the hot displacer to prevent gas from bypassing the hot regenerator. This seal operates at the crankcase temperature.

In the VM refrigerator, the pressure drop through the regenerators (and therefore, the pressure differential across the seal) is held to a minimum (less than 0.4 atm)

After extensive tests conducted as part of the Hi Cap program, it was found that glass loaded teflon provides the lowest wear rate of any of the candidate seal materials and can be expected to satisfy the 20,000-hour life requirement.

- Bearings

One of the biggest advantages of the VM refrigerator is that it can operate with relatively small forces transmitted through the drive system. The forces acting on the drive system are small because there is no mechanical compression; forces that must be overcome are friction between rubbing parts, pressure drop across the regenerators, and inertia. In addition, the VM refrigerator can operate at slower speeds and be more perfectly balanced than can refrigerators employing other gas cycles. For these reasons, bearings lubricated with solid type lubricants can be employed. Grease or oil lubricated bearings are not desirable because they contaminate the helium gas and degrade the performance of the refrigerator. Prior to the Hi Cap program, bearings using a retainer made of Duroid 5813 had been used successfully in VM refrigerators that had an operating life of more than 1000 hours. As part of the Hi Cap program, an extensive bearing study was conducted to learn what actually limited the life of the bearing (i. e., lubricant material, ball, race materials, etc.) and also what could be done to increase life expectancy. From the studies, it was concluded that failure generally occurs when a ball in the bearing wears through a retainer between ball pockets. Therefore, to extend the life of these bearings, every effort has been made to decrease the wear rate of the ball retainer. Based on studies conducted during the Hi Cap program, it was concluded that ball bearings made of rulon A with 5 percent MoS<sub>2</sub> would allow life requirements to be met. Because the proper preload is essential to achieving long life, matched pairs preloaded at the factory are used.

- Rider Rings

Riders rings on the cold displacer assist the seals in holding the displacer in the center of the cylinder. If the displacer moves off center, gas may leak past part of the piston seal, and the increased compression load at the other side of the seal will increase the seal lip loading and cause greater wear. The riders absorb these thrust loadings. The shuttle and pumping thermodynamic losses are computed on the basis of the optimum side gap between displacer and cylinder. As this gap changes, the magnitude of these losses can change and hence reduce the refrigeration.

Riders are used on the hot displacer for the same reasons that they are used on the cold displacer. However, a special problem exists because one rider must be located toward the end of the displacer, which operates in a 920°K environment. In the past, split rings made of carbon materials have been used for these riders. However, in extended life applications, the excessive amount of carbon dust wear debris that is generated could block regenerators and various flow passages inside the refrigerator. Accelerated wear rate component tests are now being conducted with the Hi Cap

refrigerator and several small coolers. In addition, a test module is being fabricated and will soon be in service. The purpose of the wear rate program is to test several candidate rider ring materials in order to learn the projected long term operating capabilities of each. The material chosen for the hot displacer rider ring will depend on the findings from that program. Preliminary data indicates that a rider ring of composite refractory metal/MoS<sub>2</sub> (such as Boeing hot compact 6-84-1) sliding on a hard coated Inconel 718 liner may provide the necessary wear life for the LSIR application.

- Static Seals

In general, there are two methods of ensuring long term retention of helium within the refrigerator. One is to use some type of hermetic seal to essentially keep any helium from leaking from the refrigerator (i. e., the leakage rate is so low that it causes an insignificant reduction in pressure within the refrigerator during the 20,000-hour operating life). The other method is to use some type of seal that has a known leakage rate (e. g., O-rings) and then provide an auxiliary volume of gas to replenish the gas lost from the refrigerator in such a way that only a slight reduction in pressure occurs at any interval during the useful life of the refrigerator.

Studies of various types of helium seals have shown that the use of elastic metal shaped hermetic seals is a good means of ensuring long term retention of helium. If a seal having a known leak rate is used, there are penalties associated with the size of the plenum volume and the weight of the gas that must be used. In addition, some type of regulating device must be inserted between the refrigerator and the plenum volume in order to sense a prescribed decrease in pressure and to allow the system to be refilled without the danger of overpressurization. No completely satisfactory method of replenishing the helium gas has yet been found. Therefore, considerable experimental effort has been expended to learn whether metallic seals will perform adequately in the refrigerator.

These static seal tests have indicated that elastic metal seals can be used for hermetic sealing. The refrigerator will utilize one elastomeric O-ring and one metal seal at each joint, and no make-up gas will be required.

The O-ring serves several purposes. First, it absorbs the cyclic pressure changes within the refrigerator so that the metallic seal is not continuously flexing. Second, it limits the leak rate if the metallic seal becomes momentarily unseated because of external loading, such as launch vibration. Thermal cycling tests have demonstrated that a metallic seal will reseal itself. The O-rings will be utilized as gas seals during the development test phase; metallic seals need be utilized during life test.

To summarize: All external joints of the refrigerator will be sealed by both an elastomeric O-ring and an elastic metal seal. The metal seals will be plated with indium, and the flange interfaces will be designed to ensure that the deflections will be less than 0.001 inch. No external gas supply will be required.

- Electric Heaters

Electric resistance heaters are used to supply high temperature energy to the helium gas in a VM refrigerator. At present, materials technology does not allow extended operation of the Inconel 718 hot cylinder above  $1000^{\circ}\text{K}$ . Therefore the heaters must be capable of continued operation for at least 20,000 hours at a temperature that will maintain the cylinder head temperature at  $920^{\circ}\text{K}$ .

The heaters incorporate a helically coiled heating element. This element has several advantages over the straight wire type. For the same resistance, the heating element in the helically coiled design is longer, and therefore the diameter of the wire can be greatly increased. Such a wire will have a longer operating life. In addition, when the heater is swaged, the diameter of the wire in the coil remains constant since swaging only spreads the coil apart. The life of a straight wire heater can be severely reduced if the swaging causes a local reduction in the diameter of the heating element. Also during on and off cycling, the wire spreads rather than stretches when in a coil; hence, it undergoes a much lower stress. Finally, the heating element is close to the heater sheath, and therefore the temperature gradient between them is minimum. The lead wires of the heater are generally high purity nickel. Nickel is used because it has a low electrical resistance and does not heat up significantly when current is applied to it.

Hughes discussed with Watlow Electric Company the design considerations relevant to achieving a 20,000-hour operating life. From this discussion it was concluded that existing heater technology for VM refrigerator applications is consistent with the 20,000-hour life requirements for the refrigerator. A heater at Hughes has accumulated over 30,000 hours of test life without failure.

## SECTION VIII

### CONCLUSIONS

The technical requirements for a cryogenic cooler capable of cooling the LSIR detector array are met with a VM cycle refrigerator utilizing technology developed during the CMP, Hi Cap, and VM wear rate programs. Because of these programs, the VM refrigerator is in an advanced state of development for space applications and can be expected to meet the LSIR

program objectives with low developmental risk. Without development, there is no refrigerator available that could satisfy the long life and low power requirements of the LSIR.

The preliminary design of the LSIR baseline cryogenic cooler, as identified in Tables 3 through 7, meets the LSIR requirements with reasonable weight and power budgets, i. e., the cooler

- Weighs 61.28 N (14 lb)
  - 35.58 N (8 lb) for the instrument mounted refrigerator
  - 25.7 N (6 lb) for the EIFU
- Requires 77.1 watts of power at 24 vdc when providing 0.3 watt of refrigeration at 65°K.

The vibration that the refrigerator induces can be kept well under 0.01 g at the refrigerator mounting surface.

The operating life goal of 20,000 hours will be evaluated in relation to the results of the current VM wear rate program. On the basis of the data available from the wear rate test program after 4000 hours of operation (see Figure 6), the use of present hot rider materials ensures more than 20,000 hours of operation in a static environment. Test results also indicate that operation on the ground for periods up to 2200 hours under worst case conditions can be tolerated before launch. These tests are being continued. The long term effect of stop-start operation on wear rate is not known. However, operational data on VM refrigerators used in airborne and ground based applications where many stop-start cycles are experienced does not indicate unusual wear as compared to continuous running wear. Present technology will allow other elements of the VM refrigerator to meet the operating life goals.

An implementation plan and schedule have been developed for the possible follow-on program. The schedule shown in Figure 30 was based on a minimum time delivery of the first cooler to sensor integration in 16 months with test support of all units provided by three sets of GSE. The start date of 1 July 1977 was assumed; hence, hardware tests and deliveries would be completed before 1980 with few months leadtime. Because of the fabrication and procurement lead time needed, the first unit will be delivered after only a minimum of engineering development and before any launch qualification and EMI tests can be conducted. These last tests will be conducted with the second unit while the first unit is being integrated with the sensor. Unit 2 will be refurbished after its tests and will be delivered following acceptance test and delivery of Unit 3. Although a life test is not shown on the schedule, it is highly recommended that one of the refrigeration systems be life tested early in the program.

APPENDIX A  
DESIGN SPECIFICATION

1.0 SCOPE

1.1 Scope - This specification establishes the design requirements for a closed cycle 65°K VM refrigerator to be used to cool a spaceborne limb scanning infrared radiometer (LSIR).

2.0 APPLICABLE DOCUMENTS

2.1 Applicable Documents - The latest issue of the following documents provides standards and guidelines for this study. In the event of conflict between documents referenced here and other details specified herein, the latter shall be considered a superseding requirement.

SPECIFICATIONS

<u>NUMBER</u>	<u>TITLE</u>
MIL-I-6181D	Interference Control Requirements Aircraft Equipment
MIL-W-6858C	Welding, Resistance: Aluminum
MIL-D-1000	Drawings, Engineering and Associated Lists
MIL-W-22759B	Wire, Electric, Fluorocarbon-Insulated Copper and Copper Alloy
MIL-B-5087B	Bonding, Electrical and Lightning Protection for Aerospace System
MIL-C-5541C	Chemical Films and Chemical Film Material for Aluminum and Aluminum Alloys

## STANDARDS

<u>NUMBER</u>	<u>TITLE</u>
MIL-STD-100A	Engineering Drawing Practices
MIL-STD-461	Electromagnetic Interference Characteristics, Requirement for Equipment
MIL-STD-462	Electromagnetic Interference Characteristics, Measurement of
MIL-STD-463	Definition and System of Units, Electromagnetic Interference Technology
MIL-STD-130C	Identification Marking of U. S. Military Property
MIL-STD-129D	Marking for Shipment and Storage
MIL-STD-143A	Specifications and Standards, Order of Precedence for Selection of
MIL-STD-810B	Environmental Test Methods for Aerospace and Ground Equipment
MIL-STD-21985C	Wire, Electric, Extruded TFE-Fluoro- carbon Insulated, Silver-Coated Copper, 600 volt.
FED STD-123	Domestic Shipping - Marking for

## OTHER DOCUMENTS

<u>NUMBER</u>	<u>TITLE</u>
NHB 5300.4 (1C)	Inspection System Provisions for Aeronautical and Space System Materials, Parts, Components, and Services
NHB 5300.4 (3A)	Requirements for Soldered Electrical Connections
NHB 5300.4 (1A)	Reliability Program Provisions for Aeronautical and Space System Con- tractors
MIL-HDBK-5A	Metallic Materials and Elements for Flight Vehicle Structures
NHB 9501.2A	Procedures for Reporting Cost Infor- mation from Contractors
NASA SP-5002	Soldering Electrical Connectors

### 3.0 REQUIREMENTS

#### 3.1 Temperature Requirements

3.1.1 Cold Cylinder - The cold cylinder will maintain the  
Detector assembly at  $65^{\circ}\text{K} \pm 2^{\circ}\text{K}$ . The temperature shall be stable in this  
range to within  $\pm 5^{\circ}\text{K}$  during nominal thermal load operating conditions.

3.1.2 Crankcase - The crankcase temperature normal operating  
range shall be  $-5^{\circ}\text{C}$  to  $+45^{\circ}\text{C}$ .

3.1.3 Hot Cylinder - Three hot cylinder operating temperatures will be selected. One hot cylinder temperature will be the maximum allowable that gives maximum refrigeration capacity at 65°K and 45°C ambient. The lowest hot cylinder temperature will be that which just gives minimum refrigeration capacity at 65°K and -5°C ambient.

### 3.2 Performance Requirements

3.2.1 Refrigeration Capacity - The required refrigeration is to be determined during the study. This capacity will include the heat loads resulting from detector bias, detector lead conductivity, aperture radiation, dewar heat leak, detector support structure conductivity, and lead conductivity of required instrumentation. Upon defining the combined thermal load a baseline refrigerator will be designed for this load. Two additional refrigerators will be designed, one with a heater power requirement 10% greater than the baseline refrigerator and another with a power requirement 10% less than the baseline refrigerator.

3.2.1.1 Detector thermal load - Cold end loads resulting from detector bias, detector lead conductivity, and aperture radiative load will be  $.25 + .05$  watts.

3.2.2 Operating life - The VM refrigerator shall satisfy thermal requirements during continuous steady state operation for a minimum operating life of 20,000 hours including a minimum of 100 start-ups without performance degradation.

3.2.3 Standby - The refrigerator shall be capable of maintaining a standby condition with minimum electrical power input for periods from 12 hours minimum to 500 hours maximum.

3.2.4 Shutdown - The refrigerator shall be capable of maintaining a shut down condition with zero electrical power input to 500 hours with a restart ability of at least 100 times.

3.2.5 Cooldown Time from Shutdown - At an ambient temperature of +45°C the time required for cooldown to steady-state operating conditions shall not exceed two hours.

3.2.6 Cooldown Time from Standby - At an ambient temperature of +45°C time required for cooldown shall not exceed one-half hour.

3.2.7 Storage Life - The storage life of the cooler shall be a minimum of two years under normal warehouse conditions and utilizing special shipping containers for environmental protection.

3.2.8 Electromagnetic Interference - The VM cooler shall be designed to meet the minimum electromagnetic interference for Class No. 1 (Communication and Electronic Equipment) of MIL-STD-461.

3.2.9 Performance Degradation - The refrigerator shall be designed such that motor speed or power input can be adjusted over a limited range to compensate for performance degradation which may occur over the life of the refrigerator.

### 3.4 Power Requirements

3.4.1 Heater Power - Heater power to the hot cylinder shall be minimum and as a design goal shall not exceed 80 watts.

3.4.2 Drive Motor Power - Drive motor power shall be a minimum and as a design goal shall not exceed 10 watts.

3.4.3 Power Source - Electrical power shall be provided from the spacecraft bus through the LSIR Power system at a nominal 24 VDC or 10 VDC.

3.4.4 Heat Rejection - All heat generated by the refrigerator will be rejected to space by means of a space radiator mounted to the refrigerator crankcase. Requirements for transferring heat from the crankcase to the radiator are to be determined. This system will be capable of rejecting the heat input to the hot cylinder, the refrigeration load, and the heat dissipated by the drive motor. As a design goal maximum heat rejection will be 90 watts.

3.4.4.1 Crankcase Heat Exchanger - The crankcase heat exchanger shall be defined for either a heat pipe or a circulating coolant LSIR thermal rejection system. The VM cooler design shall include a description of the heat exchanger parameters and of the interface for mating the crankcase heat exchanger with the LSIR thermal rejection system.

### 3.5 Structural Requirements

3.5.1 Alignment Tolerances - As a design goal the alignment of the Detector Assembly with respect to the housing assembly center line shall be maintained within a 20  $\mu\text{m}$  diameter of true position, and the axial position of the Detector Assembly shall be maintained within  $\pm 2.5\mu\text{m}$  of the specified focal plane under steady state operating conditions.

3.5.2 Dynamic Variation in Detector Position - The dynamic variations in true position of the Detector Assembly resulting from VM cooler mechanical vibration and transient temperature and pressure variations shall be less than  $+0.25 \mu\text{m}$  in both radial and axial directions.

3.5.3 Pressure Levels - The refrigerator shall be designed to withstand, at operating temperatures, pressures two times the maximum operating pressure and shall be proof tested at room temperature to 1.5 times the maximum expected service pressure.

3.5.4 Mechanical Interface - The LSIR Housing Assembly will be the point of attachment and primary mechanical interface and reference datum plane with the spacecraft. The refrigerator shall be mounted entirely on the Housing assembly with no secondary mechanical interface with the spacecraft (see Figures A-1 and A-2).

3.5.4.1 Mechanical Mounting - The VM refrigerator will be mechanically attached to a circular flange on the Housing Assembly by means of symmetrically located bolts. The interface shall be defined in detail as part of the design study.

3.5.5 Detector-Cooler Interface - The interface between the Detector Assembly and cold cylinder is to be determined. It will be established if the resulting dynamic variation in detector position is acceptable with direct mounting of the detectors to the cold cylinder. In the event that detector motion is too large a separate support for the detector array will be designed as part of the refrigerator and a flexible interface such as braided copper wire will be used to thermally couple the detector and cold cylinder.

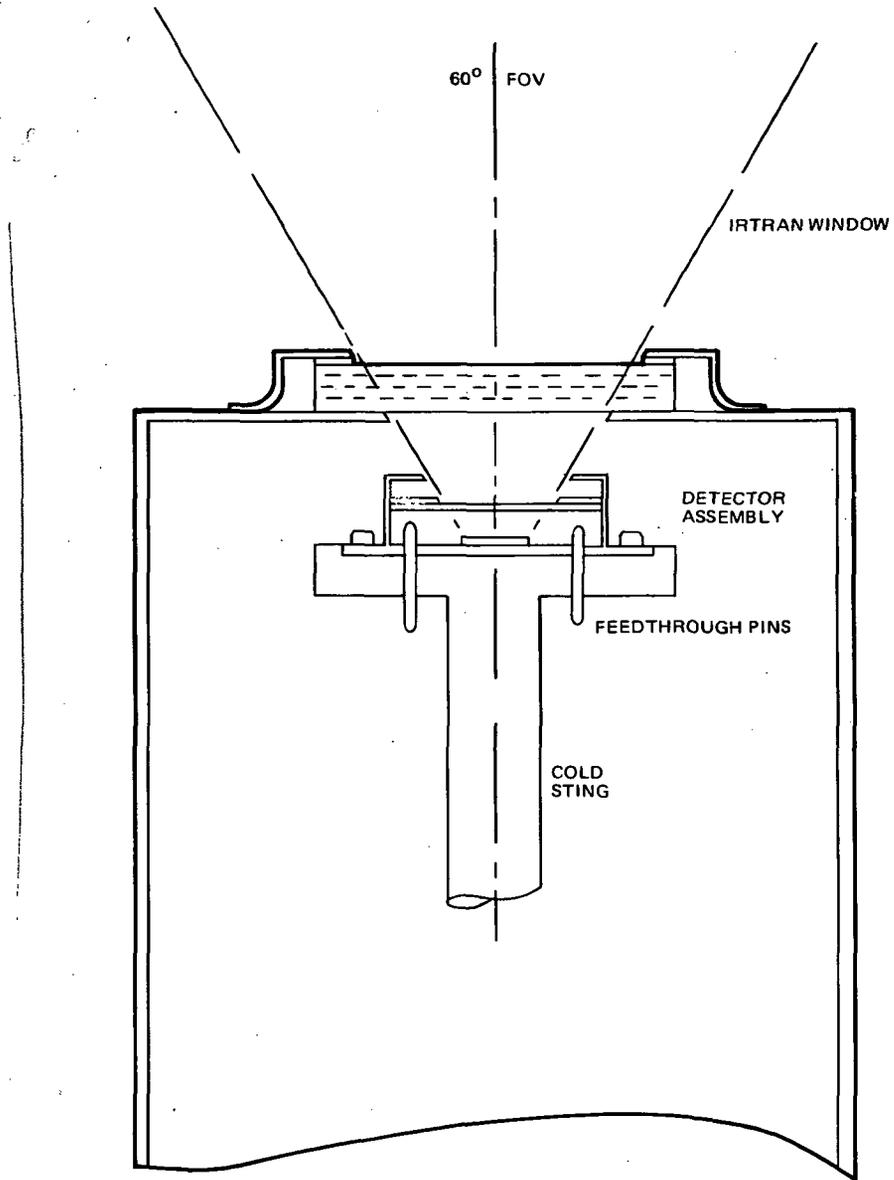


Figure A-1. Mounting schematic for detector assembly.

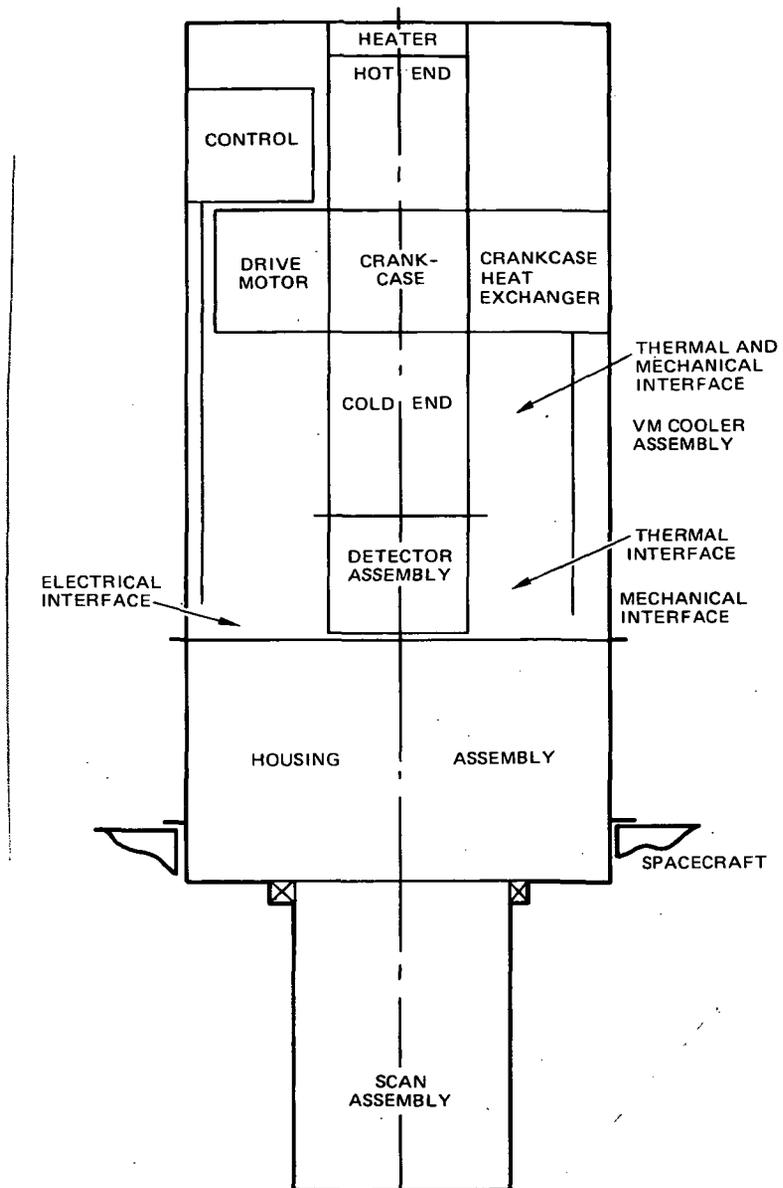


Figure A-2. LSIR nomenclature.

3.5.6 Environmental Design - The VM refrigerator and all components shall withstand ultimate design loads and maintain the required alignment following to any of the space vehicle environmental design requirements specified in Table A-1.

3.5.7 Bench Handling - The cooler shall be capable of withstanding shocks encountered in servicing as specified in MIL-STD-810 Method 516.2, Procedure V.

3.5.8 Dimensional Configuration - The dimensions of the refrigerator are to be determined.

3.5.9 Weight - The total weight of the refrigerator and electronics is to be determined.

### 3.6 Electrical Requirements

3.6.1 Power Receptacle - A power receptacle shall be provided for the following power signals.

24 VDC	Supply to heaters
24 VDC	Return
24 VDC	Supply to motor
24 VDC	Return
10 VDC	Supply to controls
10 VDC	Return
	Ground

TABLE A-1

LSIR ENVIRONMENTAL DESIGN REQUIREMENTS

ENVIRONMENTAL QUALIFICATION

Vibration - Sinusoidal - 2 octaves/minute

<u>Frequency Range</u> (Hz)	<u>Amplitude - Thrust Axis</u> ("g" - 0 to Peak - 0.5" max D.A.)
5 - 25	7.0
25 - 100	15.0
100 - 200	7.0
200 - 2000	5.0
	<u>Transverse Axes</u>
5 - 100	10.0
100 - 2000	5.0

Vibration - Random - Three Axes

<u>Frequency Range</u> (Hz)	<u>Power Spectral Density</u> (g <sup>2</sup> /Hz)	<u>g - RMS</u>	<u>Duration</u>
20 - 300	.023 to .090 at 4 db/oct		
300 - 2000	.090	12.8	2 min/axis

Acceleration

<u>Thrust Axis</u>	<u>Lateral Axis</u>	<u>Duration</u>
18 g	3 g	1 min/axis

Thermal-Vacuum Cycle

Pressure	10 <sup>-5</sup> mm Hg or less
Thermal Cycle	-5°C to 45°C
Duration	3 full cycles lasting a total of approximately 196 hours

3.6.2 Command Receptacle - A receptacle for the following command signals shall be provided.

VM cooler power on

VM cooler power off

VM cooler standby

VM cooler operate

3.6.3 Instrumentation - The VM refrigerator assembly shall include housekeeping and monitoring instrumentation and a receptacle for instrument signals identified as monitoring instruments and specified in Paragraph

3.6.4 Housekeeping - The housekeeping instrumentation necessary to maintain control and regulation of the VM refrigerator shall be defined.

3.6.5 Monitoring - Sensors necessary for sensing the following parameters and relaying the signals to the LSIR Data System shall be included in the VM refrigerator:

- A. Detector temperature
- B. Crankcase temperature
- C. Hot end temperature
- D. Crankcase Pressure
- E. Motor Speed
- F. Motor current
- G. Heater current
- H. Cooler operating status

#### 4.0 QUALITY ASSURANCE PROVISIONS

4.1 System Engineering - Define in detail the specific requirements for applying the refrigeration system to the LSIR spacecraft and infrared sensor. Coordinate cooler requirements with spacecraft and sensor engineers to develop a spacecraft heat rejection system, a power supply systems, telemetry, command logic, etc.

4.2 Final Detail Design - Detail design, analysis, and working drawings for the refrigerator and the interface unit (IFU); these include procurement specifications, test specifications, etc. Before they are released, all drawings will be checked and approved by materials or components specialists and by the responsible cryogenic system engineer.

4.3 Fabrication - Fabricate and/or procure parts for refrigeration systems. All parts, components, and subassemblies will be subjected to quality control inspection in accordance with approved Hughes Aircraft Company procedures for space equipment.

4.4 Assembly and Checkout - Each refrigeration system will be assembled under quality control surveillance and then checked out to verify that the unit operates correctly in response to all commands. Preliminary leakage tests will verify that there are no coolant leaks and that the helium leak rate is within acceptable limits.

4.5 Engineering Verification and Development Tests - All units will be tested to verify that they meet acceptable performance levels before they undergo formal acceptance tests. For the first unit, some development work may be needed to bring performance within specifications. All following units would be subject to similar modifications.

4.6 Acceptance Test - A formal test shall be conducted under quality control surveillance to demonstrate that each refrigeration system meets specified requirements.

4.6.1 Test Methods and Conditions

4.6.1.1 Ambient Test Conditions - All tests will be performed under laboratory ambient conditions,  $25 \pm 5^{\circ}\text{C}$  with the capability of selecting operating crankcase temperature anywhere within the range specified in Paragraph 3.1.2.

4.6.1.2 Temperature Stability - Unless otherwise specified, all temperature measurements will be made after the refrigerator has stabilized.

4.6.1.3 Refrigeration Capacity - A test dewar duplicating the detector interface and thermal heat load of the detector assembly shall be mounted to the refrigerator. The resulting temperature shall meet the requirements of Section 3.1.1.

4.6.1.4 Cooldown Time - The refrigerator shall cool the test dewar described in Paragraph 4.7.3. within the time and temperature requirements of Paragraph 3.2.6.

4.6.1.5 Alignment Tolerances - Alignment tolerances shall meet the requirements specified in Paragraph 3.5.1 during normal steady state operation.

4.7 Launch Qualification Test - One or two units will be subjected to a series of launch qualification tests to demonstrate environmental capability to meet the requirements specified in Table A-1 and A-2. The unit must satisfactorily pass the acceptance tests of Paragraph 4.7 prior to and subsequent performance of the qualification tests.

4.7.1 Qualification Test Procedure - The procedure for conducting the tests shall be as defined in MIL-STD-810B. In the event of conflict between MIL-STD-810B and this specification, the latter shall govern.

TABLE A-2

LSIR OPERATIONAL ENVIRONMENT

Temperature - Spacecraft

NOMINAL	0°C to 40°C
MAXIMUM	-5°C to 45°C

Pressure - Ambient

GROUND TEST - CHECKOUT	Ambient
GROUND TEST - THERMAL VACUUM	10 <sup>-5</sup> mm Hg or less
SPACE	10 <sup>-9</sup> mm Hg or less

Spacecraft Vibration - Random

(Hz)	<u>g-RMS</u>
0.25 - 10	.125

Space Vehicle Vibration - Sinusoidal - 2 octaves/minute

<u>Frequency Range</u>	<u>Amplitude</u>
(Hz)	("g" - 0 to peak)
5 - 35	0.25

Space Vehicle Acoustic Environment

Continuous white noise with a frequency spectrum of 30-10,000 Hz at an overall sound pressure level history as follows

0 - 15 secs	145 db
15 - 90 secs	130 db

4.8 Electromagnetic Interference Test - One cooler will undergo EMI tests. They will be conducted in accordance with the procedures of MIL-STD-461 and any specific requirements for the LSIR program.

APPENDIX B

## LIST OF SYMBOLS USED IN APPENDIX B

- A = cross sectional area
- C = circumferential length or wetted perimeter of a cylinder
- $C_p$  = specific heat of a gas at a constant pressure
- $D_e$  = equivalent hydraulic diameter
- d = diameter
- F = force
- f = friction factor
- G = velocity of mass, i. e., mass flow rate divided by corresponding flow area
- $g_c$  = gravitational conversion factor
- $h, h', h''$  = heat transfer coefficient (primes indicate direction of flow)
- J = conversion factor
- k = thermal conductivity
- L = length
- m = mass
- $\dot{m}$  = mass flow rate
- N = cyclic speed
- n = number of moles of working fluid
- $n_n$  = number of nodes formed along circumference during collapses
- P = pressure
- Q = thermal power
- R = gas constant for working fluid
- r = radius
- $r_h$  = hydraulic radius
- S = radial clearance between piston or displacer and cylinder bore

s = stroke of displacer  
T = temperature  
t = time  
V = volume  
v = average velocity of displacer  
W = wall thickness or work  
X = distance  
Y = stroke  
Z = compressibility factor for working fluid  
 $\Delta P$  = pressure drop  
 $\eta$  = efficiency  
 $\theta$  = crank angle  
 $\lambda$  = number of thermal units associated with material  
 $\rho$  = density of a gas  
 $\tau$  = time

#### Applicable Subscripts

c = cold or cold cylinder  
cc = crankcase  
ci =  $i^{\text{th}}$  cold volume  
cy, w = cylinder wall  
d, p = displacer or piston  
env = environmental or ambient  
f = friction  
g = gas  
h = high, hot, or hot cylinder  
i = inner or 1, 2, 3 . . . . etc.  
l = low

M = mechanical  
max = maximum  
min = minimum  
net = net  
pu = pumping  
rf = frictional forces at regenerator  
rh = efficiency losses at regenerator  
r, reg = regenerator  
s = related to spherical cap  
sh = shuttle  
sr = related to stiffening rings  
t = total  
ut = total of heat transfer load  
v = void, or inactive volumes  
 $\Delta P$  = pressure drop

## APPENDIX B

### INHERENT THERMODYNAMIC AND HEAT TRANSFER LOSSES OF VM REFRIGERATOR

Since it is impossible to build perfect regenerators and to eliminate heat transfer processes in the refrigerator, a number of equations are presented that account for regenerator and heat transfer losses.

The undesirable or inherent thermodynamic and heat transfer loads can be summarized by the expression

$$Q_{ut} = Q_{sh} + Q_{pu} + Q_p + Q_w + Q_f + Q_{rf} + Q_{rh}. \quad (B-1)$$

In general, these are the losses that must be cooled at  $V_{c1}$  and at  $V_{c2}$  or that must be supplied to  $V_h$ . The separate expressions for the loads in Eq. (B-1) are

#### Shuttle Loss

$$Q_{sh} = 0.188 Y_p^2 C \frac{k_g}{S} \frac{(T_h - T_c)}{L_{cy}}. \quad (B-2)$$

Shuttle heat transfer is a transfer of heat caused by the oscillation of the displacer within its cylinder if the thermal gradients of the displacer and of the cylinder are identical at the midpoint of the stroke. This transfer occurs when the displacer absorbs heat at the hotter half of its travel because at this position there is a difference between the temperature of the cylinder (cold end) wall and that of the displacer, and heat flows from the cylinder wall to the displacer. During the other portion of the stroke, the difference between the temperature of the displacer and that of the cylinder causes this heat to be deposited in the vicinity of the load. If the motion of the displacer is approximately harmonic and if the thermal time lag of the cylinder and displacer materials is small compared with the reciprocating time, the shuttle heat transfer can be computed from Eq. (B-2), which is derived as follows:

$$Q = \frac{Ck}{S} X\Delta T, \quad (B-3)$$

where

$$k/S = h$$

$$\Delta T = \frac{(T_h - T_c) X}{L_{cy}}$$

$$X = \frac{Y_p}{2} (1 - \sin \omega \tau)$$

The average thermal power that is deposited in the refrigerating volume when the piston moves a distance  $X$  that is measured from mid-stroke position ( $X = 0$ ) is  $Q/2$ . Thus, Eq. (B-3) becomes

$$Q = \frac{Ck (T_h - T_c) Y_p^2}{8SL_{cy}} (1 - \sin \omega \tau)^2. \quad (B-4)$$

The average thermal energy  $E_a$  that is deposited in the refrigerating volume during a time increment  $\Delta \tau$  is  $Q \cdot \Delta \tau$ , and for one cycle, Eq. (B-4) gives

$$E_c = \frac{Ck (T_h - T_c) Y_p^2}{8SL_{cy} \omega} \int_{\pi/2}^{3\pi/2} (1 - \sin \theta)^2 d\theta \quad (B-5)$$

since  $d\tau = d\theta/\omega$ . Integrating Eq. (B-5) gives

$$E_c = \frac{3 Ck (T_h - T_c) Y_p^2 \pi}{16SL_{cy} \omega}. \quad (B-6)$$

The average thermal power input to the refrigerating volume during one cycle is obtained by dividing Eq. (B-6) by the time the piston spends in the refrigerating volume ( $\tau = \pi/\omega$ ). Thus,

$$Q_{sh} = 0.188 Y_p^2 C \frac{k_g (T_h - T_c)}{S L_{cy}}. \quad (B-7)$$

### Pumping Loss

$$Q_{pu} = \frac{2(\pi d_c)^{0.6} L_{cy}^{1.0} (P_{max} - P_{min})^{1.6} N^{1.6} C_p^{1.6} (T_h - T_c) k_g^{-0.6}}{1.5 Z R^{1.6} \left( \frac{T_h + T_c}{2} \right)^{1.6}} S^{2.6} \quad (B-8)$$

Pumping heat transfer occurs as the expansion chamber undergoes pressure changes, and the annular space around the expansion displacer therefore alternately becomes pressurized and depressurized. This variation in pressure follows the system pressure, and the gas that enters and leaves the annular space transports heat to or from the gas in expansion volumes  $V_c$  or  $V_h$ .

Because of the small annular space, it is assumed that the flow is laminar, and

$$\frac{hD_e}{k_g} = 1.5 \left( \frac{mC_p}{k_g L_{cy}} \right)^{0.4} \quad (B-9)$$

is used in deriving Eq. (B-8). Eq. (B-9) is presented in Reference 1, and the derivation of Eq. (B-8) is shown in Reference 2.

### Heat Transfer Through Displacer

The axial heat transfer through the displacer wall is due to the difference in the temperatures of the ends of the displacer; it is expressed as

$$Q_p = \frac{k_p \pi (d_p^2) (T_h - T_c)}{4L_p} \quad (B-10)$$

### Heat Transfer Through Cylinder Wall

This heat transfer is due to the difference in the temperatures of the ends of the cylinder; it is expressed as

$$Q_w = \frac{\pi k_{cy} P_{max} d_c^2}{2L_{cy} \sigma' \left( 1 - \frac{P_{max}}{2\sigma'} \right)} (T_h - T_c). \quad (B-11)$$

Heat Generated by Friction Between Displacer and Cylinder Wall (Seal)

This refrigeration load is due to friction between the displacer and the cylinder wall, i. e.,

$$Q_f = \oint F_f dL. \quad (B-12)$$

Heat Load Due to Aerodynamic Friction in Regenerator

$$Q_{rf} = \frac{\Delta P_{reg} (P_{max} + P_{min}) V_{co} N}{\rho J Z R T_c}, \quad (B-13)$$

where

$$\Delta P_{reg} = \frac{(P_{max} + P_{min})^2 V_{co}^2 N^2 f L_r}{2 Z^2 R^2 T_c^2 g_c^2 A_{reg}^2 \rho D_e}$$

$$Q_{rf} = \frac{(P_{max} + P_{min})^3 V_{co}^3 N^3 f L_r}{2 Z^3 R^3 T_c^3 g_c^2 A_{reg}^2 \rho D_e J}. \quad (B-14)$$

This refrigeration load is caused by aerodynamic heating when the working medium flows through the regenerator.

Heat Load Due to Limiting Value of Film Coefficient in Regenerator

$$Q_{rh} = (1 - \eta_r) C_p (T_h - T_c) \frac{(P_{max} + P_{min}) V_{co} N}{2 R T_c Z_c}, \quad (B-15)$$

where

$$\eta_r = \frac{h' h'' L_r^2}{h' h'' L_r^2 + h'' L_r C_p Gr_h} \Bigg|_{in} \Bigg|_{out}$$

The value of  $\eta_r$  is  $0.995 \leq \eta_r \leq 1.000$  and  $0.9 \leq \eta_r \leq 1.000$  for the cold and hot cylinders, respectively.

The limiting value of the film coefficient in the regenerator prevents sufficient cooling of the working fluid as it flows from the hot to the cold end of the regenerator. The amount of energy that remains in the working fluid because of this limiting value is computed from Eq. (B-9).

Another method of studying the thermodynamics and heat transfer of the VM refrigerator consists of dividing the refrigerator into several volumes, each volume representing a thermodynamic system, and then deriving an energy balance for each system. The result of applying this method is a number of unsteady state energy equations of the form

$$\begin{aligned} \Sigma \text{ energy} &= mC \frac{dT}{d\tau} \\ \dots\dots\dots & \\ \dots\dots\dots & , \end{aligned} \tag{B-16}$$

a number of mass flow rate equations of the form

$$\begin{aligned} \dot{m}_{in} - \dot{m}_{out} &= \frac{d(\dot{m})}{d\tau} \\ \dots\dots\dots & \\ \dots\dots\dots & , \end{aligned} \tag{B-17}$$

and a number of pressure drop equations of the form

$$\begin{aligned} \Delta P_{ij} &= \frac{m_{ij}}{k_{ij}} \\ \dots\dots\dots & \\ \dots\dots\dots & , \end{aligned} \tag{B-18}$$

where i and j refer to separate elements.

By solving these equations simultaneously and by applying the equation of state for the working medium, it is possible to determine the temperature, the mass of working fluid, and the pressure in any one of the thermodynamic systems at any instant that the refrigerator is operating. Thus, a history of the unsteady state and steady state thermodynamic behavior of the refrigerator is obtained for each particular design. The design (and hence the inputs to the equations) can be varied so that the last solutions of the equations give an optimum design for the refrigerator. This method is complicated and requires a computer since there are generally a large number of energy equations, and they have to be solved by means of the finite difference technique. However, both the transient and steady state performance of the refrigerator can be studied by this method.

## REFERENCES FOR APPENDIX B

- 1) R. White, "Vuilleumier Cycle Cryogenic Refrigeration," AFOL-TR-76-17 (April 1976)
- 2) R. White, "Computer Program for Optimizing Three-Stage Vuilleumier Cycle Cryogenic Refrigerators," AFOL-TR-76-27 (February 1976).

## APPENDIX C.

### DESCRIPTION OF INPUT COMMANDS

IFU ON - All instrumentation and fault detection circuits are energized and allow motor and heater commands to be recognized.

IFU OFF - All power to the IFU is removed.

MOTOR ON - A command is recognized only when the IFU is on. The motor inverter is powered with 100 vdc.

MOTOR OFF - Motor inverter and heater power is removed; normal and redundant inverter commands can be recognized.

HEATER ON - A command is recognized only when the IFU is on; heater is powered with 24 vdc.

HEATER OFF - Heater power is removed. Normal and standby operation and hot cylinder redundant/normal heater connection commands can be recognized.

STANDBY OPERATION - A command is recognized only when the heaters are off. The return path of the main heater is disconnected, and the hot cylinder temperature controller is set for standby mode.

SELECT NORMAL INVERTER POWER STAGE - A command is recognized only when the motor is off; the normal inverter power stage is connected to the motor; the redundant inverter power stage is disconnected.

SELECT REDUNDANT INVERTER POWER STAGE - A command is recognized only when the motor is off. The redundant inverter power stage is connected to the motor, and the normal inverter power stage is disconnected.

SWITCH REDUNDANT HEATER TO REPLACE MAIN HEATER - A command is recognized only when the heaters are off. The main heater is disconnected, and the redundant heater is connected to function as the main heater.

OVERRIDE OVERTEMPERATURE SHUTDOWN FOR HOT CYLINDER - A hot cylinder overtemperature condition will be ignored, and the refrigerator will continue to operate.

OVERRIDE OVERTEMPERATURE SHUTDOWN FOR CRANKCASE - A crankcase overtemperature conditions will be ignored, and the refrigerator will continue to operate.

OVERRIDE HIGH RADIATOR TEMPERATURE SHUTDOWN - A high radiator temperature will be ignored, and the refrigerator will continue to operate.

OVERRIDE MOTOR FAILURE SHUTDOWN - A low motor speed condition will be ignored, and power will continue to be applied to the refrigerator.

OVERRIDE RESET - All override conditions are cleared, and the IFU will respond to a fault condition.

RESET - Fault latches are cleared if fault conditions no longer exist, and the failure shutdown latch is cleared when the fault latches are cleared or overridden.

SELECT NORMAL TEMPERATURE SENSOR CONNECTION FOR HOT CYLINDER - Sensor 1 is connected to the control loop; sensor 2 is connected to the overtemperature fault detector; the redundant sensor is disconnected.

SELECT REDUNDANT SENSOR TO REPLACE SENSOR 1 - Sensor 1 is disconnected, and the redundant sensor is connected to the control loop.

SELECT REDUNDANT SENSOR TO REPLACE SENSOR 2 - Sensor 2 is disconnected, and the redundant sensor is connected to the overtemperature fault detector.

SELECT SENSOR 2 TO REPLACE SENSOR 1 - Sensor 1 is disconnected, and sensor 2 (redundant sensor if sensor 2 has been previously replaced by redundant sensor) is connected to the control loop and to the overtemperature fault detector.

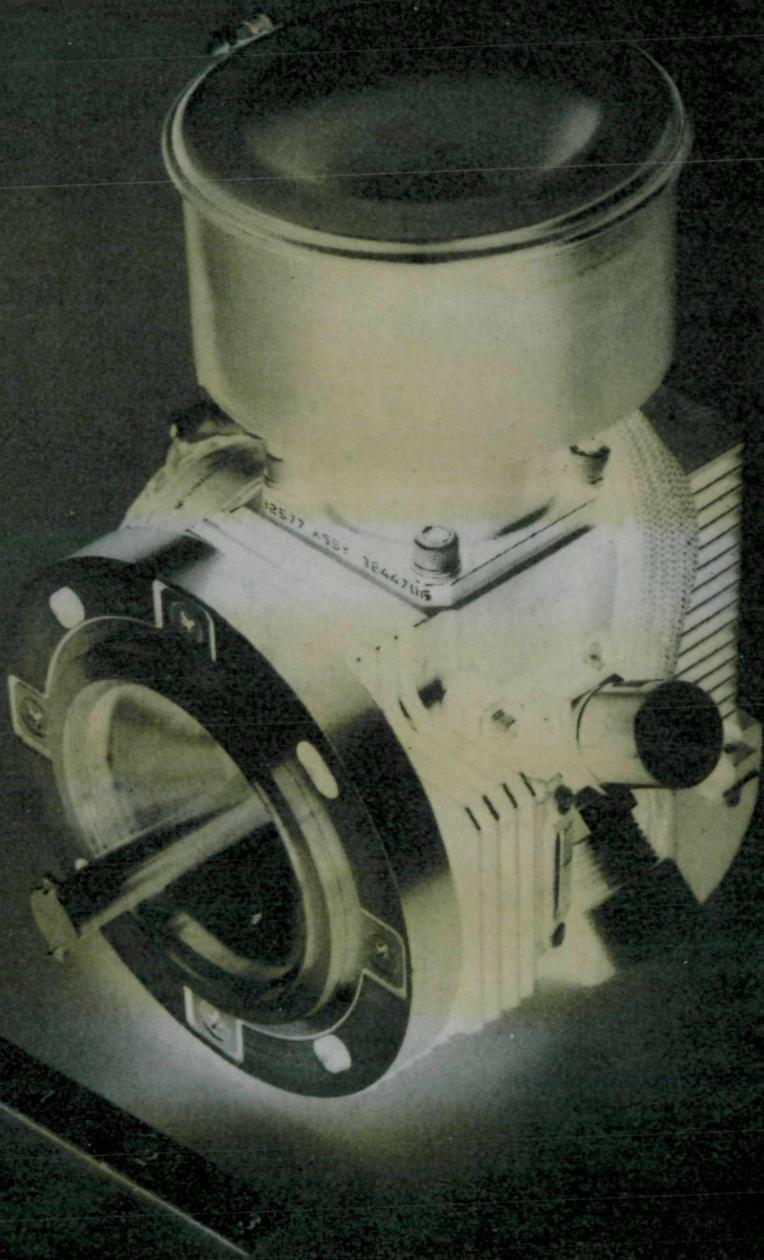


Figure 4. Typical small VM cooler (71-7244).