NASA CONTRACTOR REPORT 159250

CRYOGENIC COOLING STUDY FOR THE ADVANCED LIMB SCANNER

S. C. RUSSO
HUGHES AIRCRAFT CO.
CULVER CITY, CA 90230

CONTRACT NAS1-16029
MAY 1980

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OBJECTIVES OF THE ALS VM COOLER STUDY PROGRAM

The objectives of the Cryogenic cooling study for the ALS are (1) to determine the feasibility of cooling the ALS instrument to a temperature of 78°K with a VM cycle refrigerator while satisfying system requirements, (2) establish cryogenic requirements, (3) define a preliminary VM refrigerator conceptual design compatible with the ALS instrument and the UARSP spacecraft, and (4) assess development problems and prepare an implementation plan for development of three flight qualified VM coolers.

The results of this study are intended to provide NASA with a better understanding of the suitability of the VM refrigerator for cryogenic cooling of the ALS instrument and provide definitive information needed for the further design and development of cryogenic coolers for the UARSP mission.

This study was conducted by Samuel C. Russo of Hughes Aircraft Co., Culver City California under the direction of Robert D. Averill (technical monitor) of the National Aeronautics and Space Administration, Langely Research Center under Contract NAS1-16029 "Cryogenic Cooling Study For the Advanced Limb Scanner."

ESTABLISH CRYOGENIC REQUIREMENTS

DEFINE PRELIMINARY CONCEPTUAL DESIGN

AND

PREPARE IMPLEMENTATION PLAN

FOR

A VM CYCLE CRYOGENIC REFRIGERATOR (COOLER) TO BE USED IN COOLING THE ALS DETECTOR ARRAY TO 78°K DURING A TWO-YEAR OPERATIONAL SPACE MISSION

SUMMARY

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- DEFINITION OF CRYOGENIC REQUIREMENTS SHOWS VM COOLER MEETS ALS REQUIREMENTS

- THERMAL AND MECHANICAL INTERFACES HAVE BEEN DEFINED

- HUGHES RECOMMENDS A REDUNDANT COOLER SYSTEM

- TOTAL POWER REQUIREMENT CAN BE MAINTAINED BELOW 100 WATTS

REVIEW OF ALS MISSION AND COOLER REQUIREMENTS

The initial phase of this study was devoted to defining the requirements for a VM cryogenic cooler for the ALS mission. The refrigerator requirements, critical interface factors, and the sensor configuration were reviewed by NASA, the instrument manufacturer (Honeywell), and Hughes. Requirements established at the review are summarized in the next three figures. The information provided in these figures forms the refrigerator baseline requirements for the ALS cryogenic cooling system.

Overall Requirements - The ALS instrument must meet the mission requirements of the UARSP as defined in the Conceptual Planning Guide. Although the primary instrument interface will be with the UARSP spacecraft, overall mission requirements of the STS must be satisfied during prelaunch, launch, orbital insertion, and re-entry periods of the mission. UARSP mission requirements must be satisfied in orbit and during orbital transfer periods of the mission. The ALS instrument will be vented to space and shall operate in the space vacuum, thermal, and radiation environment. Instrument temperature will be passively or actively controlled at the spacecraft temperature level and the instrument shall perform to all the specifications of the SOW over the spacecraft maximum temperature range of -10C to +40C with the capability of surviving over the temperature range specified in the UARS Flight Environment Planning Guide.

The mission life of the instrument is a nominal 18 months; however, the cryogenic cooler will be designed for a 20,000 hour life including a 2-year in-orbit mission life (17,500 hours) with 100 start-ups without instrument performance degradation so that an 18-month mission will be assured. The storage life of the instrument shall be a minimum of two years under normal warehouse conditions and utlizing special containers as necessary for environmental protection.

ALS MISSION AND COOLER REQUIREMENTS

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CHARACTERISTIC	REQUIREMENTS
o Refrigeration Temperature, ^{o}K	78 $\pm$2(-195^{o}C $\pm$ 2)
o Refrigeration Temperature Stability, ^{o}K	$\pm$.01 @ 2 min
o Refrigeration Capacity, Watts	0.3
o Input Power	
o Voltage, VDC	28 and 10 (10 VDC reserved for controls)
o Amount, Watts	
Heaters	$\leq$75
Driver Motor	$\leq$15
Electronics	$\leq$10
o Cooldown Time, Hours	$\leq$2.0
o Dynamic Unbalance	$\leq$.07G

CHARACTERISTIC	REQUIREMENTS
o OPERATIONAL MODES	
o OPERATING	NORMAL OPERATION; CAPABLE OF CONTINUOUS OPERATION
o SHUTDOWN	CAPABLE OF DRAWING "0" POWER WITH A RESTART ABILITY OF AT LEAST 100 TIMES
o HEAT REJECTION	
o RATE, WATTS	≤ 100
o METHOD	HEAT PIPE OR LIQUID COOLANT TO SPACE RADIATOR
o TEMPERATURE OF CRANKCASE, ^{o}C	-10 TO 40
o POINT OF MECHANICAL ATTACHMENT	ALS INSTRUMENT
o ENVIRONMENTAL TEMPERATURE RANGE, ^{o}C	-10 TO +40
o OPERATING LIFE, HOURS	$\geq 20,000$
o STORAGE LIFE, YEARS	≥ 2

CHARACTERISTICS	REQUIREMENTS
o DETECTOR DISPLACEMENT TO FOCAL PLANE, M (LATERAL AND AXIAL)	± 0.75
o ENVIRONMENTAL DESIGN REQUIREMENTS	PER UARS FLIGHT ENVIRONMENT PLANNING GUIDE
o EMI DESIGN REQUIREMENTS	PER MIL-STD-461, CLASS 1 (COMMUNICATION AND ELECTRONIC EQUIPMENT)

ALS STUDY TASKS

The ALS cooling study major tasks are summarized below:

Cryogenic requirements - Determine the cryogenic and operating requirements for a VM refrigerator to be used for detector cooling of the ALS instrument. Determination of these requirements include consideration of the mission and environmental constraints of the UARSP mission.

Performance analysis - Conduct a performance analysis for steady state operating conditions resulting in an optimized refrigerator design.

Interface recommendations - Make interface recommendations as to the best approach and most effective concepts which will result in simple and satisfactory interfaces between the VM refrigerator, the ALS instrument, and the UARSP spacecraft.

Conceptual design - As a result of the previous tasks, propose a refrigerator preliminary design concept which establishes the technical feasibility of the proposed application.

Implementation plan - Develop a preliminary implementation plan and schedule describing the general tasks required to design, develop, manufacture, test, and deliver three VM cooler assemblies and one set of ground support equipment (GSE) to support an ALS flight experiment in the mid 1980's time period.

ALS STUDY TASKS

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- DEFINE CRYOGENIC REQUIREMENTS
- ACCOMPLISH PARAMETRIC PERFORMANCE ANALYSIS
- IDENTIFY INTERFACE RECOMMENDATIONS
- CREATE CONCEPTUAL DESIGN
- DEVELOP A PRELIMINARY IMPLEMENTATION PLAN AND SCHEDULE

ALS CRYOGENIC COOLING STUDY SCHEDULE

Work was begun on this program in December 1980. The main objectives, shown on the schedule, are to conduct a study in sufficient depth to define the cryogenic requirements, conduct a parametric performance analysis, and propose a conceptual design. The above tasks have been completed.

ALS CRYOGENIC COOLING STUDY SCHEDULE

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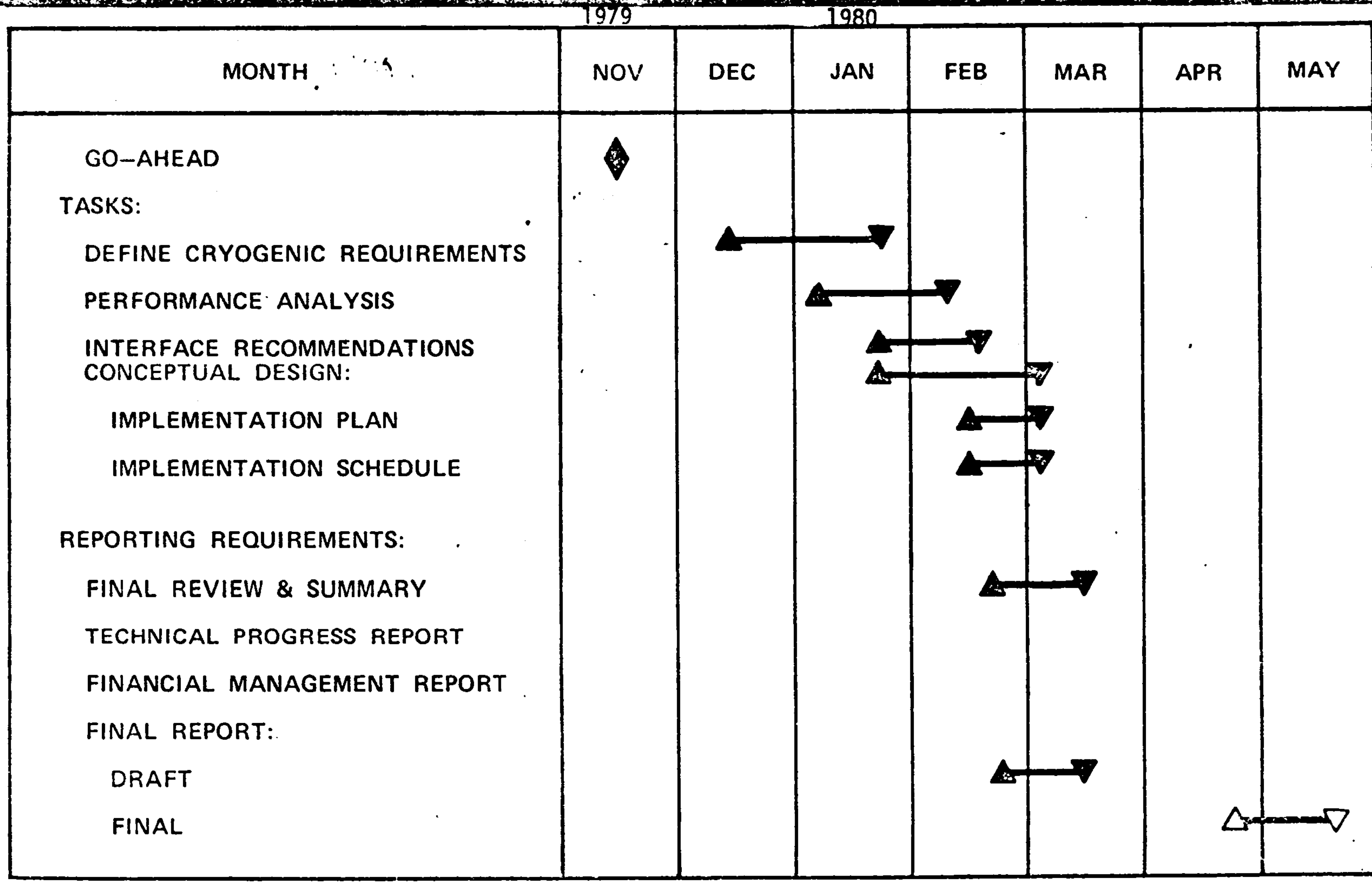

REFRIGERATOR CRYOGENIC REQUIREMENTS

The initial phase of this study was devoted to defining the requirements for a VM cryogenic cooler for the ALS program. Mission requirements and the sensor configuration were reviewed, and an initial design specification was developed

A study of cryogenic cooling requirements for the ALS under conditions identified in the SOW has established the following:

<u>Cooler Selection</u> - Candidate closed cycle coolers for the ALS instrument were the integral VM cooler and the split cycle VM cooler (i.e., a primary power module separated from the refrigeration module).

Hughes has developed and successfully demonstrated the concept of a two-piece (split) VM refrigerator. This design offers a significant advantage when the size or weight constraint at the detector package is critical, or where the vibration input to the instrument package is critical. However, there is a severe power and life penalty associated with the split cycle VM.

The power required by a split VM cooler satisfying ALS mission requirements is approximately 50% higher than a conventional VM cooler, and the estimated maintenance free operating life is only 500 to 1,000 hours. The higher input power requirement results from the velocity and boundary friction heat losses and additional inactive volume associated with a typical split cycle cooler transfer line.

Although the split design offered significant advantages of small envelope dimensions, low weight, and low vibration at the detector, the severe power and life penalty associated with the split cooler could not satisfy the ALS 100 watt cooler input power, and 20,000 hour life requirements. Therefore, the integral VM was selected as the baseline cooler.

The initial concern with the integral VM was the effect of vibration output on the instrument. An evaluation of the integral VM cooler vibration levels indicated that with a balanced unit the typical shaking force is .0014G's at a frequency of 10 Hz. The instrument manufacturer has concluded that the above shaking force magnitude and frequency are within acceptable limits for interface with the ALS instrument.

<u>Vacuum Dewar Design</u> - The detector vacuum dewar must be capable of maintaining vacuum integrity during storage, launch, and throughout mission life. An evacuated glass dewar with the detector mounted within the dewar is the preferred approach. The advantages of this type dewar compared to a vacuum housing is (1) the dewar serves as a support for mounting the detector array off of the cold cylinder eliminating the transmission of shaking forces from the refrigerator to the detector, and (2) there is not a critical mechanical interface requiring conventional sealing techniques which could degrade during storage. The details of the dewar design are addressed in the interface recommendation section of this report.

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REFRIGERATOR CRYOGENIC REQUIREMENTS

LIFE REQUIREMENT - 20,000 HOURS

Life Limiting Component - Operating data on present VM refrigerators shows that the life limiting component to date is the 1st stage or ambient dynamic seal. On the basis of data from a seal development still in progress, present seal designs can ensure no more than 5,000 to 20,000 hours of operation for an integral VM cooler. Hughes is working on development programs with the overall objective of advancing VM cooler technology to satisfy a 40,000 hour minimum life requirement. These programs are comprised of tasks which in part contribute to developing an ambient temperature dynamic seal satisfying the above requirement. Hughes is confident that advances in dynamic seal technology as a result of these programs will ensure operating life in excess of 20,000 hours within the time frame required for development of the ALS cooler.

Start Stop Operation - 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.

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.

Redundant Cooler - As previously stated, the life limiting component on the VM refrigerator is the first stage or ambient dynamic seal. Hughes is confident that, as a result of the seal development effort previously described, the ALS 20,000 hour seal-life requirement will be realized within the time frame required for development of the ALS cooler. However, regardless of the level of reliability of any mechanical component, failures can statistically occur. Therefore, in addition to a single refrigerator cooling system, Hughes is pursuing a system with a redundant VM cycle refrigerator for the ALS instrument. The primary and redundant refrigerator will each be designed to meet the ALS life requirement of 20,000 hours.

LIFE REQUIREMENT - 20,000 HOURS

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- o LIFE LIMITING COMPONENT IS THE AMBIENT DYNAMIC SEAL
  - o BASED ON CONVENTIONAL SEAL MATERIALS AND DESIGNS MAXIMUM SEAL LIFE IS 20,000 HOURS
  - o NEW SEAL MATERIALS AND DESIGNS WITH POTENTIAL FOR LIFE $\geq$ 20,000 HOURS ARE UNDER DEVELOPMENT
- o START STOP OPERATION
  - o OPERATING DATA ON VM REFRIGERATORS INDICATES NO UNUSUAL SEAL WEAR COMPARED TO CONTINUOUS RUNNING WEAR

REFRIGERATOR THERMAL LOADS

The refrigeration capacity required is determined by the following heat loads:

- o Detector thermal load
- o Dewar conduction and radiation load
- o Lead conduction load
- o Heat pipe conduction and radiation load (redundant system)

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 most efficient.

The thermal load due to the detector consists of bias (I^2R) dissipation and aperture thermal shield radiation. The refrigerator must cool the detector loads summarized below.

ALS DETECTOR THERMAL LOAD

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DETECTOR LOAD, WATTS

• DETECTOR BIAS*	0.100 WATT
• APERTURE THERMAL SHIELD RADIATION* WITH 0.38 BY 0.48 CM APERTURE*	0.009 WATT
TOTAL	0.109 WATT

*DETECTOR LOADS DEFINED BY INSTRUMENT MANUFACTURER

REFRIGERATOR THERMAL LOADS (Continued)

Detector/dewar thermal load - The thermal load from the detector dewar is composed of conduction up the glass well and radiation to the glass well. Another source of heat load is gaseous conduction. However, under vacuums of at leat 10^{-4} torr, such conduction is usually so slight that it is ignored.

The major heat load from the dewar is conducted up the glass well. The radiation load consists of radiation from the glass well to the cold finger.

With the single cooler system, the dewar is designed to slip over the refrigerator cold finger with the major heat load, both radiation and conduction, absorbed at the 130°K stage, minimizing the 73°K stage load. The conduction load is absorbed by thermally shorting the dewar to 130°K stage and the radiator load is absorbed by shorting a radiation shield that surrounds the 73°K stage to the 130°K stage.

With a redundant cooler system, it is required to thermally isolate the non-active refrigerator from the detector/dewar. Because of this requirement, it is not practical to thermally short the 130°K stage to the detector dewar. It is, therefore, necessary to absorb the entire dewar thermal load at the 73°K stage. The dewar thermal load has been maintained at acceptable levels by minimizing the physical size of the dewar and, in that manner, minimizing conduction and radiation heat leaks.

The resulting 130°K stage thermal load for the single cooler system is summarized below.

EFFECT OF CRANKCASE TEMPERATURE ON 73°K STAGE CAPACITY AND HEATER POWER

A parametric performance analysis was conducted of the baseline cooler. The available cooling capacity and power input was evaluated as a function of temperature. In this analysis, the speed and fill pressure were maintained at their design values; the calculated performance data represents steady state values.

The figure below shows how refrigeration capacity and required heater power of the cooler varies as a function of crankcase temperature. The crankcase temperature significantly affects the performance of a cryogenic refrigerator. Increasing the 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 cycle pressure.

The higher minimum cycle 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 cylinder decreases. Increasing the hot cylinder temperature increases the refrigeration capacity by increasing the pressure ration of the thermodynamic cycle. Heater power increases slightly with increasing ambient temperature, but can almost be considered constant over the -10°C to 40°C ambient temperature range. The ambient temperature is primarily determined by the heat rejection system of the spacecraft.

EFFECT OF CRANKCASE TEMPERATURE ON
COOLING CAPACITY AND HEATER POWER

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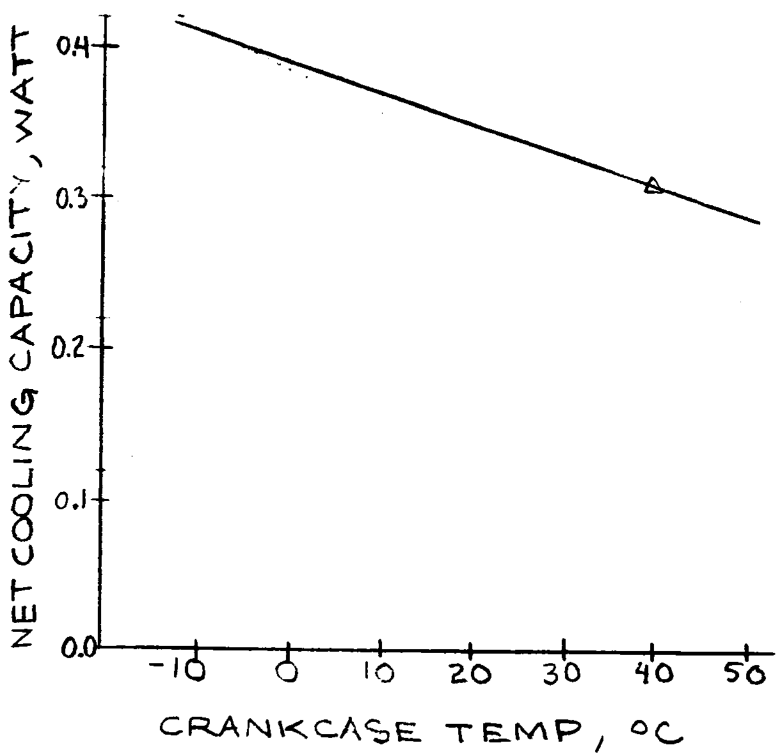
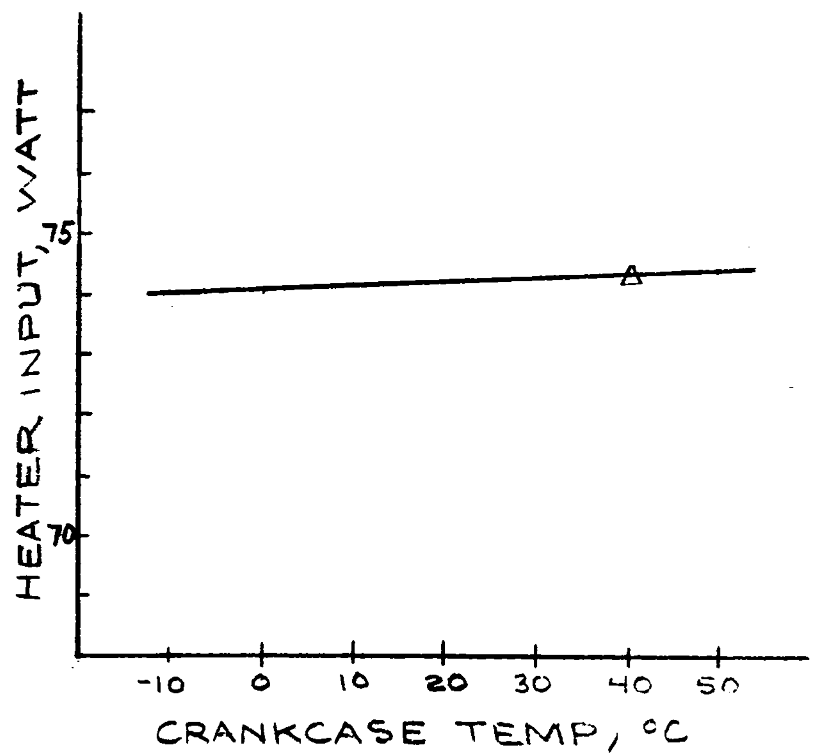
DESIGN POINT: 0.31 WATT AT 73°K

CRANKCASE TEMPERATURE = 40°C
MOTOR SPEED = 250 RPM





EFFECT OF SPEED AND CRANKCASE TEMPERATURE ON PERFORMANCE

A parametric study of the baseline cooler was conducted showing the effect of speed and crankcase temperature on refrigeration load and input power. The refrigeration load is shown as a function of speed in the figures below at crankcase temperatures of 40°C and -10°C.

The refrigeration load and required heater power increase as the speed is increased above the design speed of 250 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 crankcase 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.

EFFECT OF SPEED AND CRANKCASE TEMPERATURE ON PERFORMANCE

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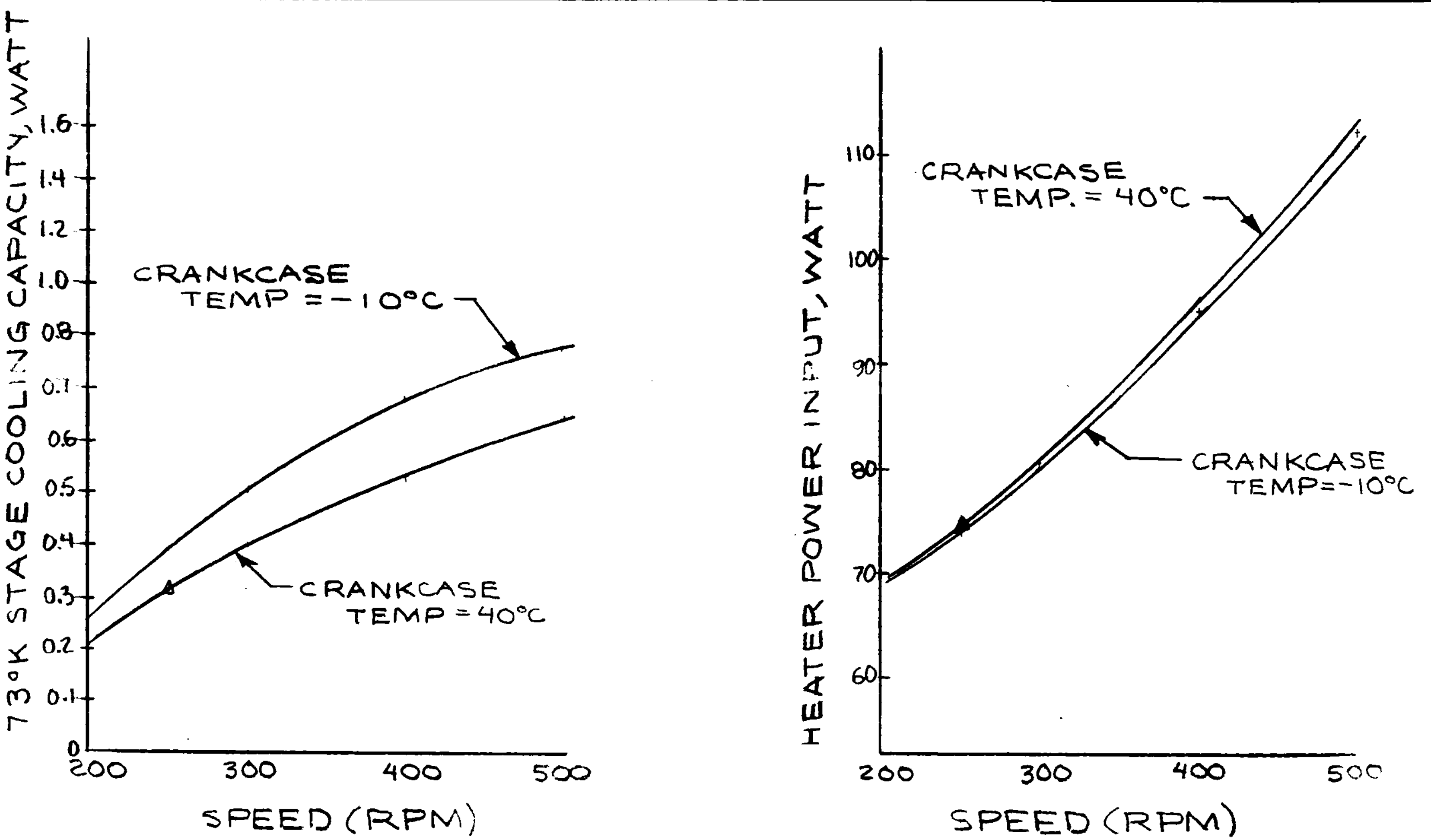

COOLING CAPACITY AND HEATER POWER VS. HOT END TEMPERATURE

When operating at crankcase temperatures lower than $40^{o}C$, power savings can be realized by operating the cooler at reduced hot end temperatures. The curves below indicate that when operating at a crankcase temperature of $-10^{o}C$, the hot end temperature can be reduced from $645^{o}C(920^{o}K)$ to $540^{o}C(815^{o}K)$ with no loss in refrigeration capacity. The corresponding reduction in heater power is from 74 watts to 62 watts (12 watt saving).

COOLING CAPACITY AND HEATER POWER VS.
HOT END TEMPERATURE

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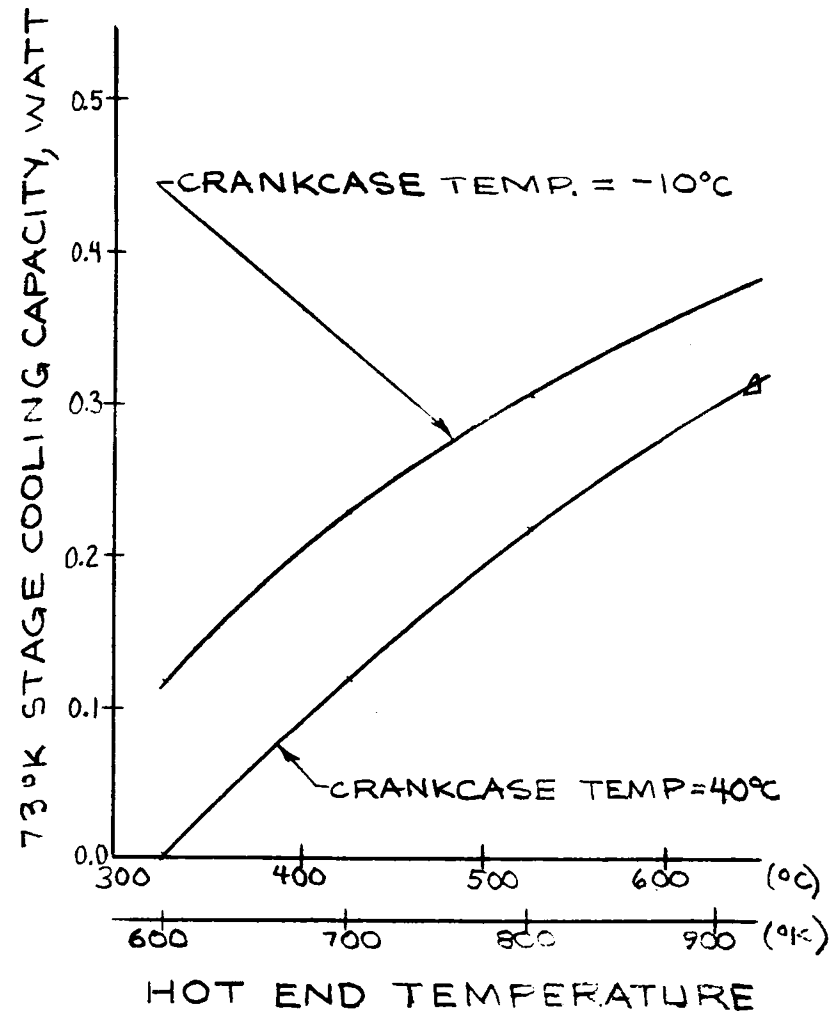

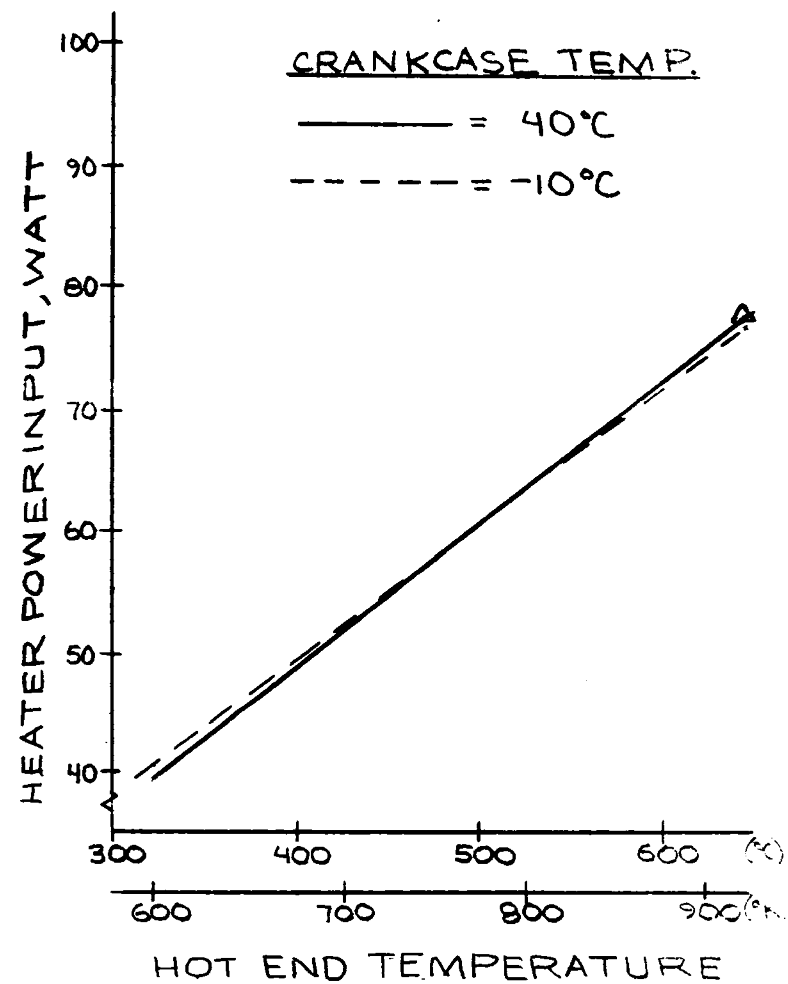

REFRIGERATOR CONCEPTUAL DESIGN

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 veryify that the governing assumptions made during the themal 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.

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REFRIGERATOR CONCEPTUAL DESIGN

MECHANICAL LAYOUT OF BASELINE ALS REFRIGERATOR

The baseline refrigerator conceived to meet the ALS requirement is a two-stage VM cycle cryogenic refrigerator. The figure below is a layout 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. The two cold stages are coaxial. The cold 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. Copper mesh which provides flexible thermal interfaces between the cold stages and the detector/dewar permits the heat loads to be transferred to the refrigerator. 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

MECHANICAL LAYOUT OF BASELINE ALS REFRIGERATOR

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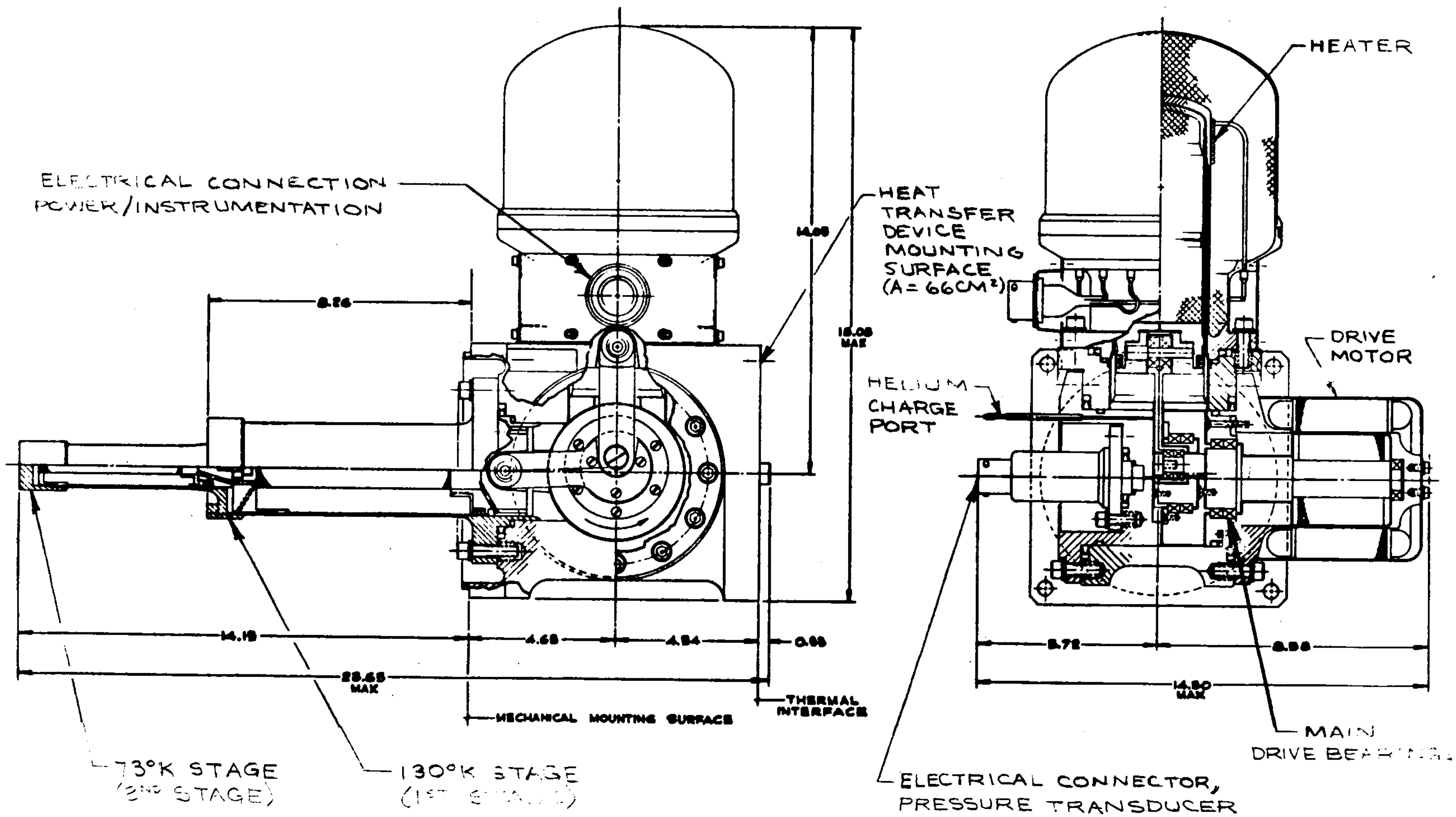

COLD END ASSEMBLY

The regenerators for the first and second stages are built into the center of the cold displacer as shown in the figure below. 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 250 RPM with a 0.3-cm stroke. Rider rings made of teflon loaded polyformaldehyde (Delrin AF) support each section of this assembly. Each displacer can be shimmed axially for optimum heat 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 attachement ends are used.

Two dynamic seals on the cold displacer assembly force the gas to flow through the proper passages between the regenerators. These seals ensure practically no leakage and contribute only slighly to frictional drag. The first stage dynamic seal is a new design developed by Hughes for operation at ambient temperature.

COLD END ASSEMBLY

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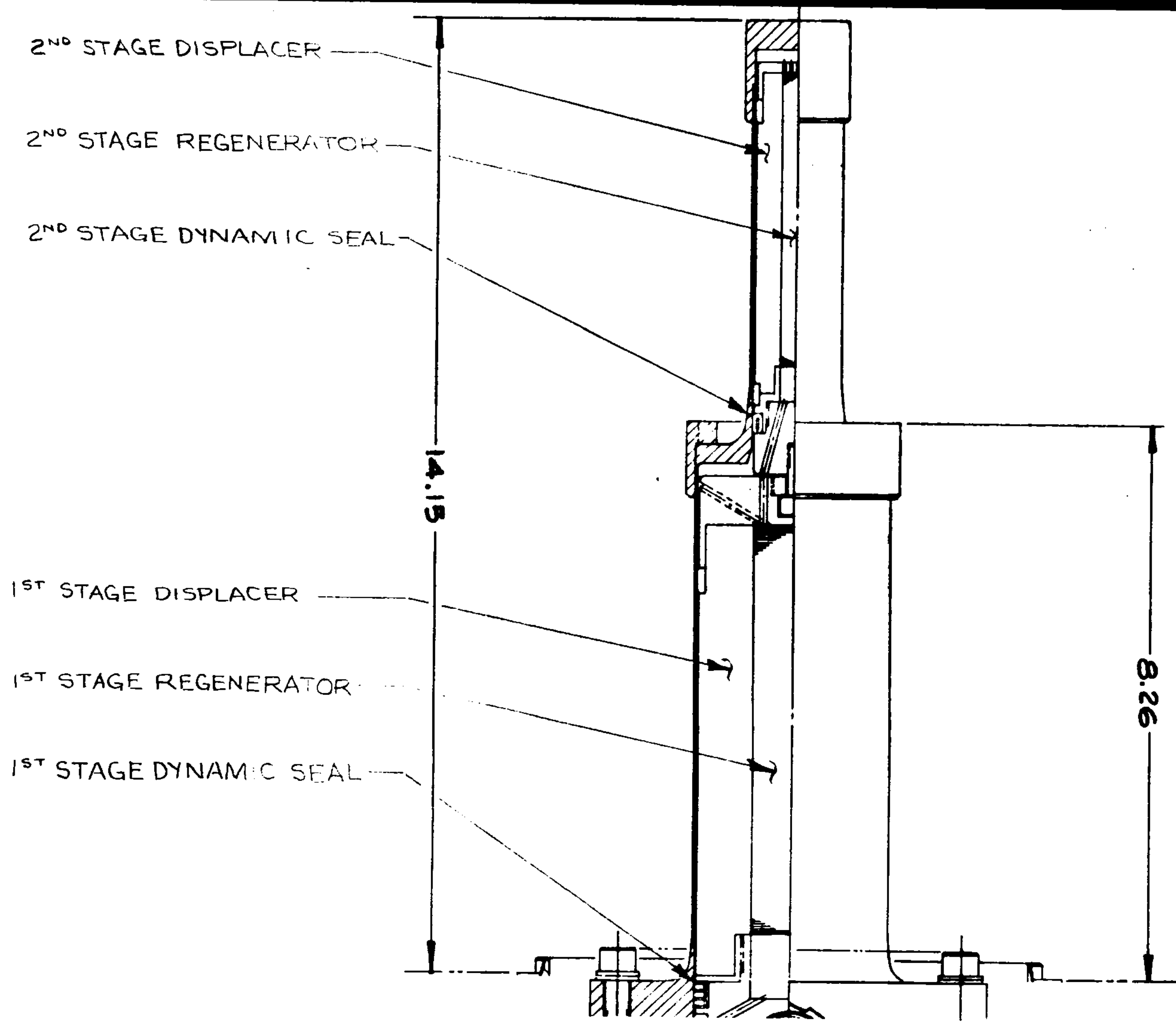


HOT END ASSEMBLY

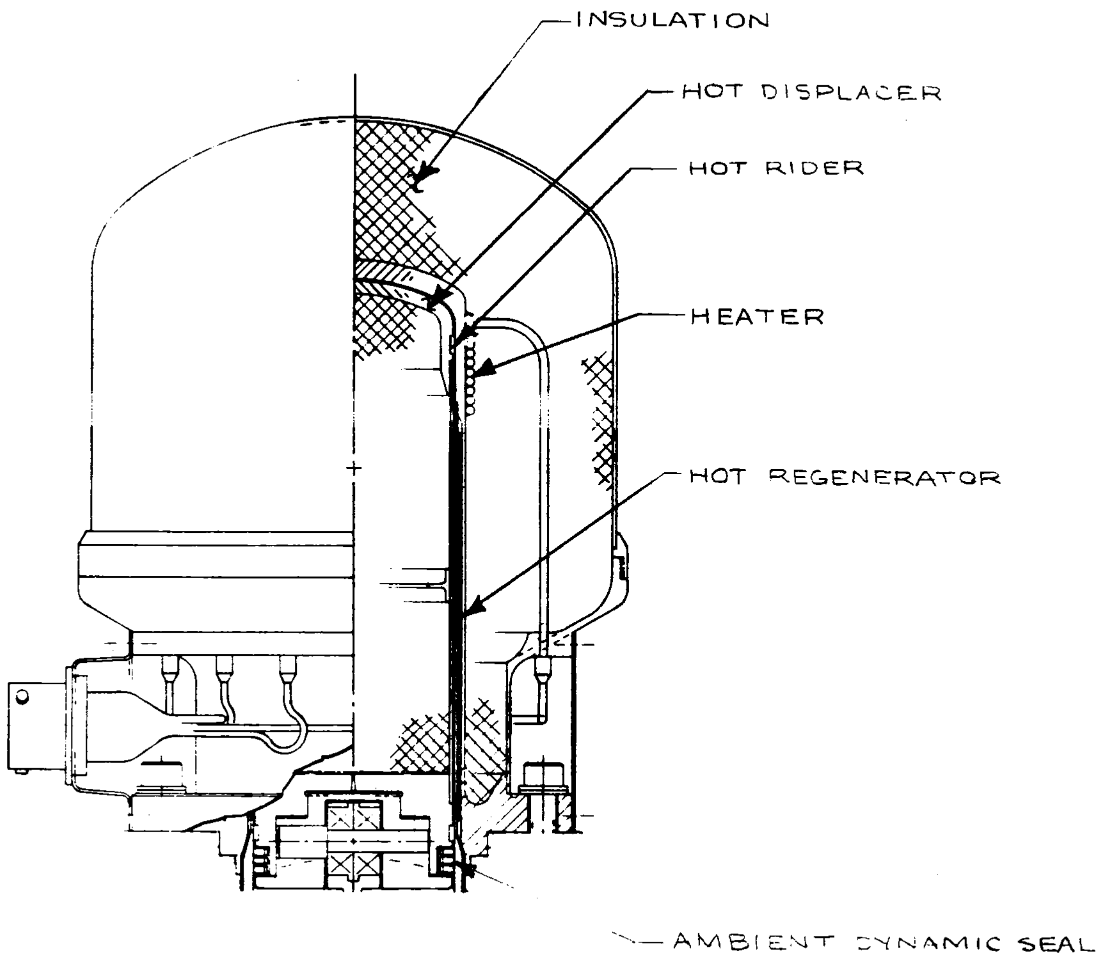
The hot end assembly consists of the hot cylinder with heaters and temperature sensors, a hot regenerator assembly, a hot displacer with rider rings, and an insulation container. The figure below shows the basic configuration of the hot end. The hot cylinder is made of Rene 41, a nickel base alloy whose 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 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 28 vdc 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 resistnace 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 rubs to increase its resistance to collapsing. A spring loaded 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 teflon loaded polyformaldehyde (Delrin AF) rider ring. This material was chosen for the hot end rider based on the findings from the VM seal material development studies.


INSULATION
HOT DISPLACER
HOT RIDER
HEATER
HOT REGENERATOR
AMBIENT DYNAMIC SEAL

DRIVE ASSEMBLY

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.

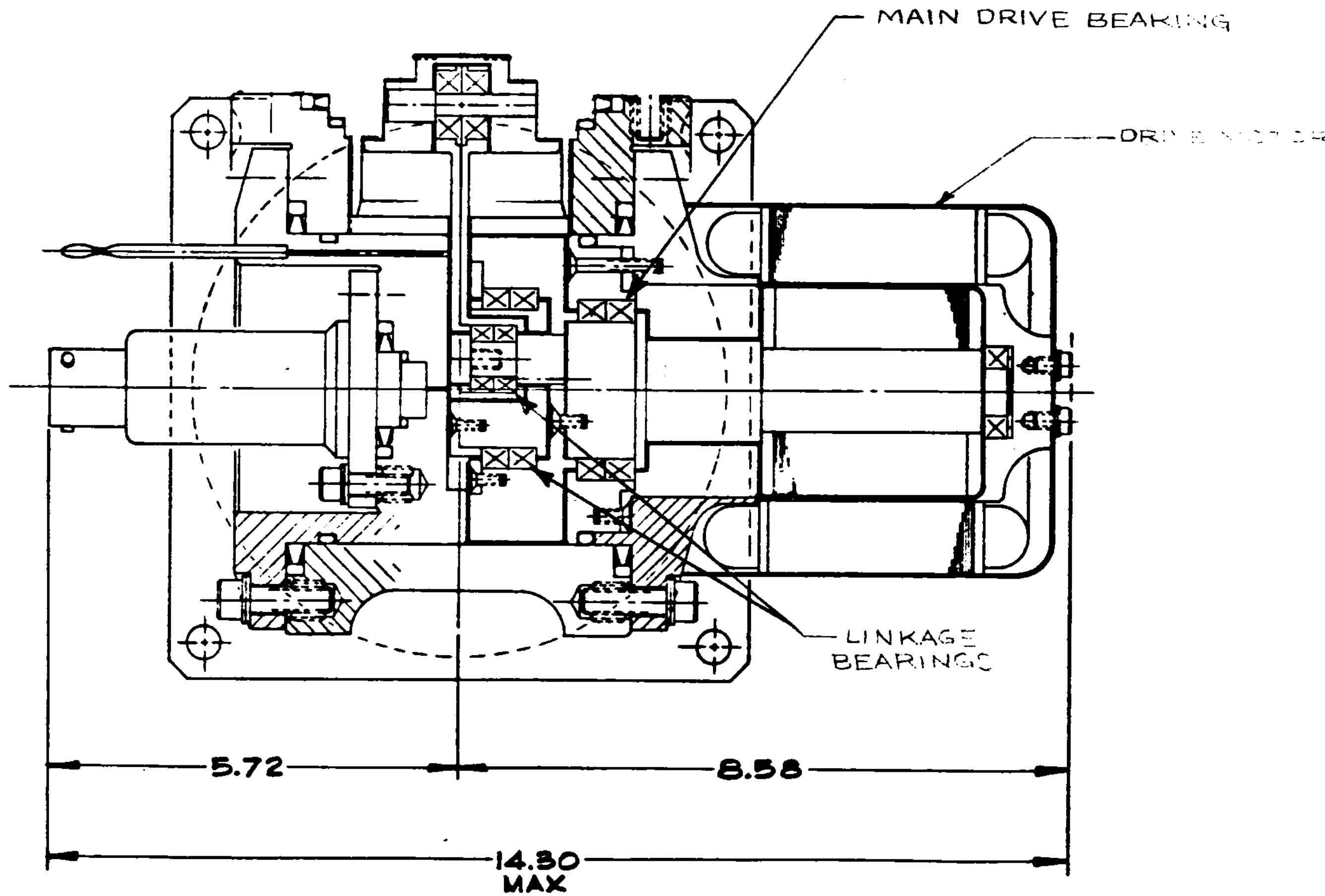
All flanges that serve as helium seals have surfaces for both an elastomeric O'ring and a metal K-seal. The flanges are rigid enough that, after the system is pressurized, the seals can withstand the pulsating pressure and thermal expansion effects at different crankcase temperatures.

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 requirements 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 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.

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 material. The drive housings provide a mounting surface for the main drive bearings. These bearings are the outboard set of duplex pair shown below. 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.


MAIN DRIVE BEARING
LINKAGE BEARINGS
5.72
8.58
14.30 MAX

SOURCES OF REFRIGERATOR DYNAMIC IMBALANCE

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.

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

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

where

F = inertia force

K = a constant

M = mass of reciprocating parts

N = rotational speed

r = radius of crank

ℓ = length of connecting rod

θ = crank angle

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 secondary 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 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. 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.

SOURCES OF REFRIGERATOR DYNAMIC UNBALANCE

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1) LINEAR INERTIA FORCE ACTING ALONG THE AXIS OF EACH CYLINDER: MOTION OF HOT AND COLD DISPLACERS CAUSES THESE FORCES (PRIMARY FORCES CAN BE COMPLETELY BALANCED, MAXIMUM UNBALANCED SECONDARY FORCE = 0.0014G)

2) ROTATING INERTIA EFFECTS DUE TO IMBALANCE OF ROTATING PARTS OF CRANK MECHANISM (CAN BE COMPLETELY BALANCED)

3) COUPLES RESULTING FROM CRANK MECHANISM IMBALANCE FORCES AND COUNTERWEIGHT INERTIA FORCES BEING OUT OF THE PLANE OF THE DISPLACER MOTION - (CAN BE COMPLETELY BALANCED)

PRIMARY INERTIA FORCES ARE BALANCED

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 = mass of hot displacer = 126.7 grm

M_c = mass of cold displacer = 267.4 grm

r_H = radius of hot displacer crank = 0.439 cm

r_c = radius of cold displacer crank = 0.155 cm

l_h = length of hot displacer connecting rod = 3.56 cm

l_c = length of cold displacer connecting rod = 3.3 cm

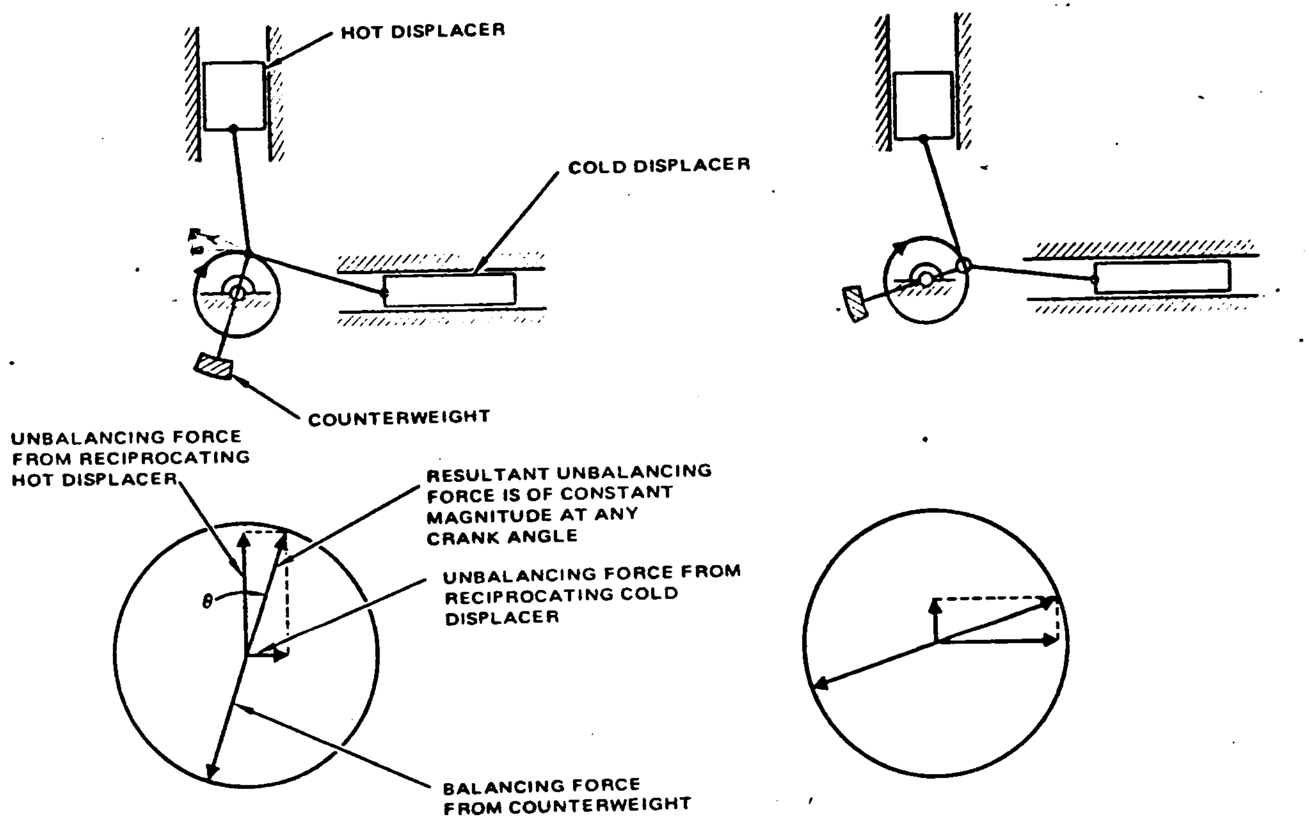
n = rotational speed = 250 rpm

From these values are calculated the following secondary shaking forces. F_h = 0.0470N(.011 lbf) from the hot displacer and F_c = 0.0179N(0.004 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 $\sqrt{F_h^2 + F_c^2}$ = .0503N(0.0113 lbf) or 0.0014g for a 0.038N(8.4 lbf) refrigerator. As the crankshaft rotates, the force varies from maximum to zero and back to maximum with each 90 degrees of rotation.

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 0.0014 g at 5 Hz.

PRIMARY INERTIA FORCES ARE BALANCED

HUGHES



REFRIGERATOR AND EIFU WEIGHT ESTIMATE

Total weight of the refrigerator and EIFU is estimated to be 64.2N(14.43 lb.). The weight breakdown is shown below.

REFRIGERATOR AND EIFU WEIGHT ESTIMATE, NEWTONS

HUGHES

o HOT END ASSEMBLY	6.22	(1.40 LB)
o COLD END ASSEMBLY	8.00	(1.80 LB)
o CRANKCASE ASSEMBLY	11.78	(2.65 LB)
o MOTOR DRIVE ASSEMBLY	11.47	(2.58 LB)
o REFRIGERATOR TOTAL	37.5	(8.43 LB)
o EIFU	26.7	(6.0 LB)
CRYO SYSTEM TOTAL	64.2	(14.43 LB)

MOTOR TORQUE REQUIREMENTS

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.

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. The motor inverter is housed in the EIFU. The motor is designed for a normal operating speed of 250 rpm, but the speed can be varied by ± 10 percent upon command from the EIFU, as required for hot end temperature control.

The motor torque and power requirements are summarized below.

MOTOR TORQUE REQUIREMENTS

HUGHES

AVERAGE TORQUE, CM-N	8.12(11.5 IN-OZ)
PEAK TORQUE, CM-N	10.55(14.95 IN-OZ)
AVERAGE OUTPUT POWER, WATTS	16

CRITICAL INTERFACES

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

CRITICAL INTERFACES TO BE REVIEWED

- COLD CYLINDER/DETECTOR DEWAR
- REFRIGERATOR/ALS INSTRUMENT
- REFRIGERATOR/HEAT REJECTION
- ELECTRONIC INTERFACE UNIT

COLD CYLINDER DETECTOR DEWAR INTERFACE

It is not possible to mount the detector directly on the refrigerator cold cylinder because of the requirement to prevent motion of the detector in both the axial and radial planes. A cold cylinder designed for the required stiffness would have excessive thermal losses.

By mounting the detector on the dewar it is possible to design the cold cylinder for optimum performance and design the dewar for low heat leak. An evacuated glass dewar will be utilized. Alignment of the detector is achieved by precision adjustment of the vacuum dewar.

<u>Single Cooler System</u> - The dewar is designed to slip over the refrigerator cold cylinder and mount to the instrument. (See page 77). Mounting to the instrument will provide for separation of the cooler from the instrument. Cooper mesh provides thermal interfaces between the refrigerator cold cylinder and the detector/dewar at the 130°K and 73°K stages.

<u>Redundant Cooler System</u> - The detector/dewar is designed to mount on the ALS instrument and is thermally coupled to the 73°K cold cylinder by diode heat pipes. The cold cylinder and heat pipe assembly is contained within an evacuated housing.

COLD CYLINDER DETECTOR DEWAR INTERFACE

1. SINGLE COOLER SYSTEM
   - DIRECT MOUNT OF DEWAR TO INSTRUMENT
   - SELF-CONTAINED SLIP-ON DETECTOR/COLD CYLINDER DEWAR
   - DEWAR ADJUSTABLE WITH RESPECT TO OPTICAL SYSTEM WITHOUT DISASSEMBLY
   - DETECTOR DEWAR CONDUCTIVELY COUPLED TO COOLER AT TWO STAGES

2. REDUNDANT COOLER SYSTEM
   - DIRECT MOUNT OF DEWAR TO ALS INSTRUMENT
   - SELF CONTAINED DETECTOR DEWAR AND SEPARATE COLD CYLINDER DEWAR
   - DETECTOR DEWAR ADJUSTABLE WITH RESPECT TO OPTICAL SYSTEM WITHOUT DISASSEMBLY
   - DETECTOR DEWAR THERMALLY COUPLED TO 2ND STAGE OF EACH COOLER BY DIODE HEAT PIPES

COLD CYLINDER/DEWAR INTERFACE
(SINGLE COOLER SYSTEM)

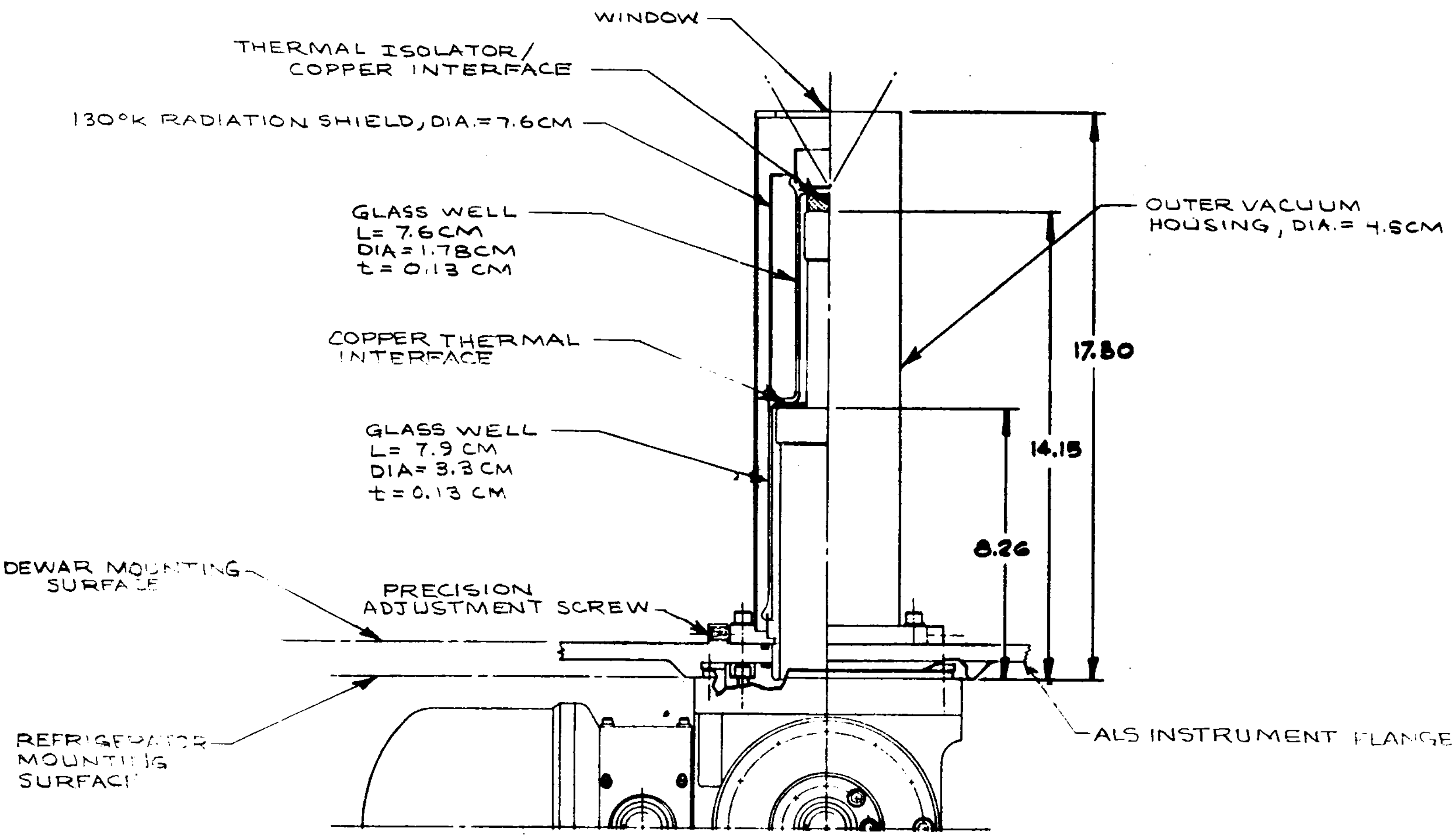
Below is a conceptual view of the detector/dewar mounted to the instrument and mated with the cold cylinder. Braided copper straps or copper mesh provide flexible thermal interfaces between the cold stages and the detector array support. At the 130°K stage, one end of the copper is attached to the cold cylinder and the other end pressed against the dewar. When the cold cylinder is fitted into the glass dewar and the copper 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. At the 73°K stage, a copper mesh is pressed between the cold cylinder and dewar providing a thermal interface between the detector and cold cylinder. The flexible copper interfaces isolate the motion of the refrigerator from the detector and allow the detector to be adjusted independent of the cold cylinder.

The detector/dewar is axially adjusted by using different shims between the dewar and mounting surface. Small radial adjustments are made with locking adjusting machine screws at the base of the dewar.

Mounting the dewar to the instrument, as shown below, provides for separating the cooler from the instrument without disturbing the detector.

COLD CYLINDER/DEWAR INTERFACE
(SINGLE COOLER SYSTEM)

HUGHES



TEMPERATURE DROP ACROSS COLD CYLINDER/DEWAR INTERFACES
(SINGLE COOLER SYSTEM)

With a copper area to length ratio of 0.050 cm and an 0.55 watt first stage heat load, the temperature drop from the dewar wall to the 130°K stage is limited to approximately 2.5°C.

At the 73°K stage a copper pad area to length ratio of 0.05 cm and an 0.19 watt detector thermal load, the temperature drop between the detector and cold cylinder is 0.62°C.

TEMPERATURE DROP ACROSS COOLER/DEWAR INTERFACES (SINGLE COOLER SYSTEM)

1) TEMPERATURE DROP ACROSS FIRST STAGE INTERFACE

$$Q = 0.548 \text{ WATT}$$

$$A/L = 0.050 \text{ cm}$$

$$K = 4.5 \text{ w/cm}^{o}\text{K}$$

$$T = \frac{Q}{\frac{KA}{L}} = 2.44^{o}\text{C}$$

2) TEMPERATURE DROP ACROSS SECOND STAGE INTERFACE

$$Q = 0.185 \text{ WATT}$$

$$A/L = 0.05 \text{ cm}$$

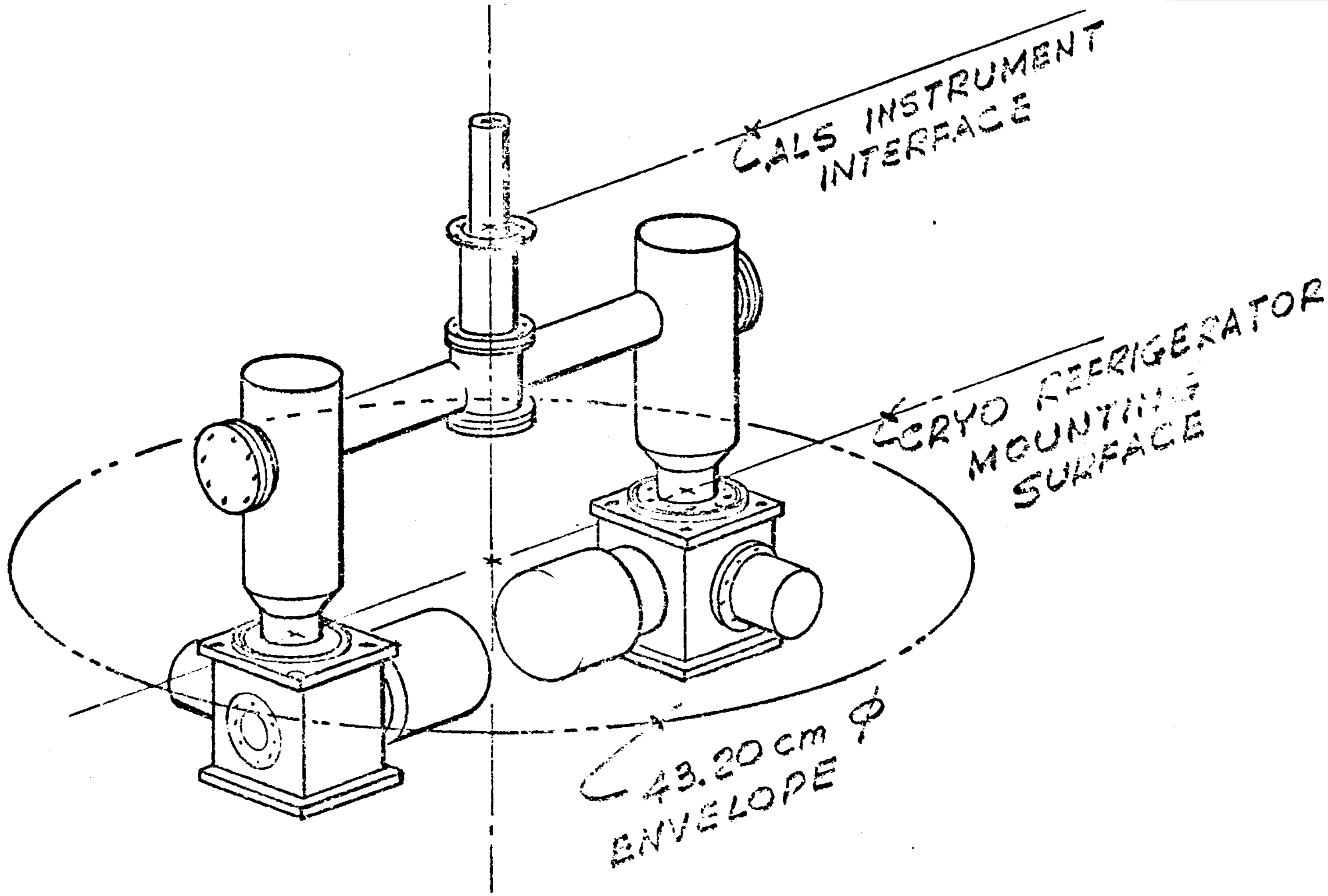
$$K = 6.0 \text{ w/cm}^{o}\text{K}$$

$$\Delta T = \frac{Q}{\frac{KA}{L}} = 0.62^{o}\text{C}$$

3-DIMENSIONAL SCHEMATIC OF

THE REDUNDANT COOLER SYSTEM

HUGHES



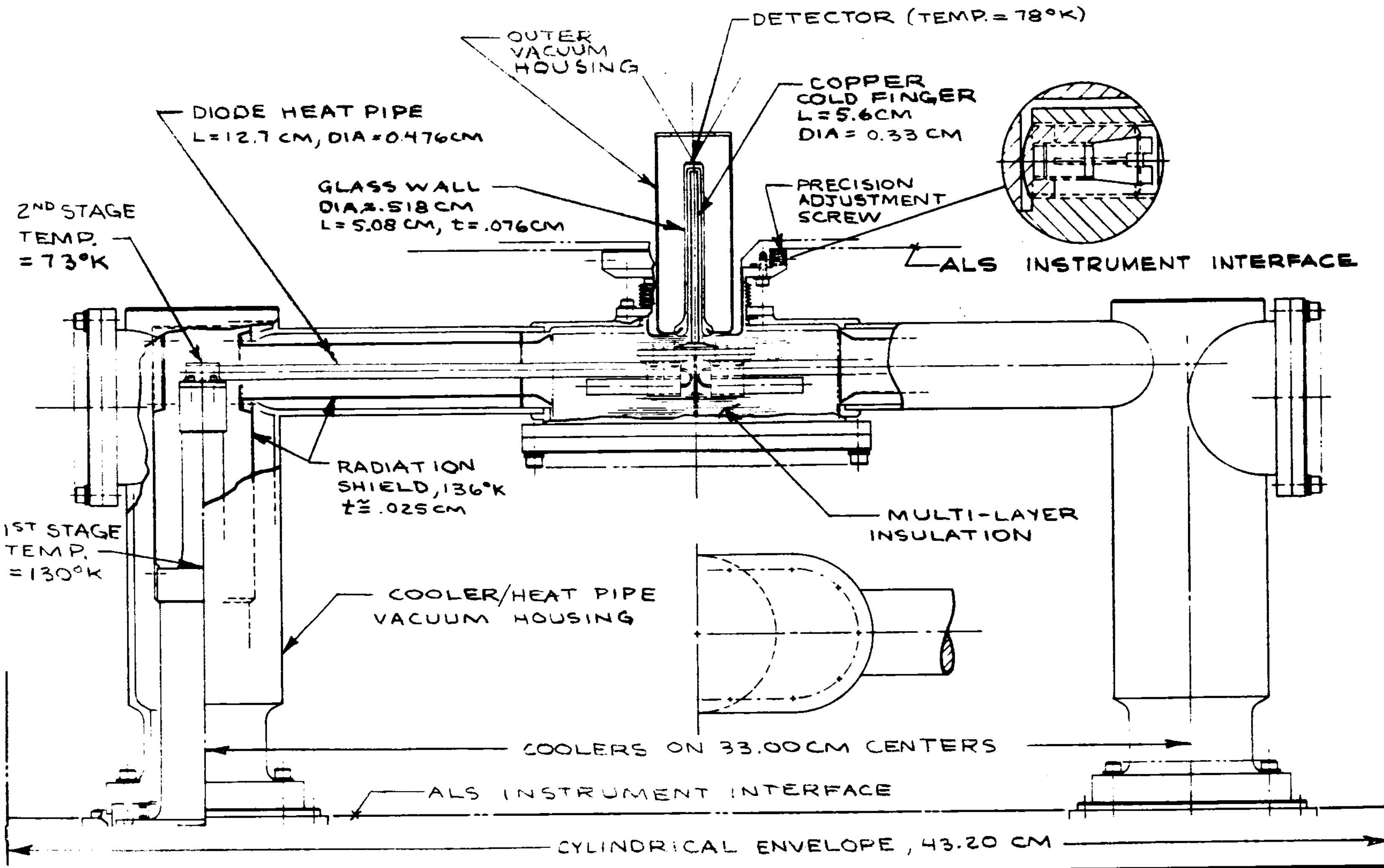
COLD CYLINDER/DEWAR INTERFACE
(REDUNDANT COOLER SYSTEM)

The implementation of a redundant refrigerator requires that the non-active refrigerator be thermally isolated from the detector/dewar. Failure to accomplish adequate thermal isolation will increase the thermal load at the 73°K stage from 0.27 to 0.40 watts; the thermal load at the 130°K stage increase from 0.45 watt to 1.53 watt. The additional loads increase the input power requirement by approximately 20 watts.

The required isolation can be achieved by thermally coupling each refrigerator to the detector dewar with a diode heat pipe. Below is a conceptual layout of a redundant refrigerator system utlizing diode heat pipes to thermally couple the active refrigerator to the detector/dewar and thermally isolate the non-active refrigerator.

COLD CYLINDER/DEWAR INTERFACE (REDUNDANT COOLER SYSTEM)

HUGHES



TEMPERATURE DROP ACROSS COOLER/DETECTOR INTERFACE

(REDUNDANT COOLERS)

The total temperature drop between the detector and cold finger is approximately 5.4°C. The temperature drop results from heat flow across 3 major thermal interfaces as summarized below. Through careful design optimization, the temperature drop could be reduced further.

TEMPERATURE DROP ACROSS COOLER/DETECTOR INTERFACE (REDUNDANT COOLERS)

HUGHES

ΔT ACROSS REFRIGERATOR/DETECTOR INTERFACE = 5.6°C

1) ΔT ACROSS HEAT PIPE

HEAT FLUX = 0.271 WATT

HEAT PIPE CAPACITY = 12°C/WATT

$\Delta T = (0.271 \text{ WATT})(12°C/\text{WATT}) = 3.3°$

2) ΔT ACROSS DEWAR COLD FINGER

$Q = 0.22$ WATT

$A/L = .086 \text{ CM}/5.08 \text{ CM} = 0.17 \text{ CM}$

$K = 6.0 \text{ W/CM} - °C$

$$\Delta T = \frac{Q}{\frac{KA}{L}} = 2.2°C$$

3) ΔT ACROSS EACH HEAT PIPE INTERFACE

$Q = .271$ WATT

$A/L \geq 1.0$ CM

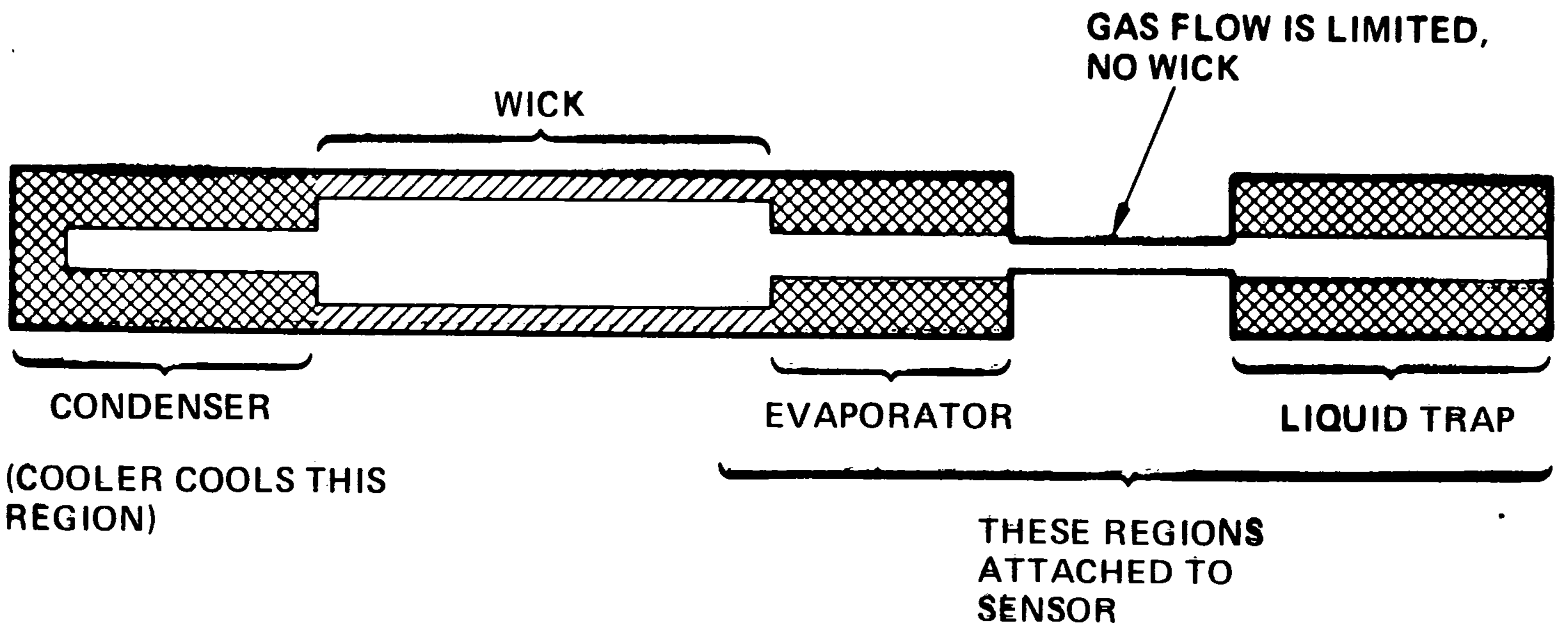
$K = 6.0 \text{ W/CM} -°C$

$$\Delta T = \frac{Q}{KA} = 0.05°C$$

DIODE HEAT PIPE COMPONENTS

IN THE FIGURE BELOW THE BASIC COMPONENTS OF THE DIODE HEAT PIPE ARE ILLUSTRATED.

DIODE HEAT PIPE COMPONENTS



CHARACTERISTICS OF DIODE HEAT PIPE SATISFYING ALS REQUIREMENTS

The design characteristics of a diode heat pipe satisfying ALS system requirements are defined below.

HUGHES

CHARACTERISTICS OF DIODE HEAT PIPE SATISFYING ALS REQUIREMENTS

- OPERATING TEMPERATURE, 75 ^{o}K
- HEAT FLUX CAPACITY, 3 WATTS
- TRANSPORT FLUID, NITROGEN
- WICK, STEEL MESH OR FELT
- GRADIENT LENGTH, 12.7 CM
- SHELL DIAMETER, .187 CM
- SHELL THICKNESS, .025 CM (1027 PSI WITH SAFETY FACTOR OF FOUR)
- THERMAL RESISTANCE 12^{o}C/WATT
- HEAT FLUX AT EVAPORATOR AND CONDENSER, 0.31 W/CM^2
- BURST PRESSURE/4, 1027 PSI
- PEAK GAS PRESSURE, 1000 PSI

HEAT REJECTION INTERFACE

Waste heat is rejected from the VM cooler to a radiator mounted on the spacecraft; this radiator is thermally coupled to a 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 is capable of transferring, by thermal conduction, 90 watts of waste heat from the crankcase heat exchanger to the radiator mounting surface with a maximum temperature drop of approximately 12°C. The temperature gradient around the circumference of the annular heat exchanger is approximately 7°C. Therefore, in order to maintain the crankcase at an effective temperature of 40°C, it will be necessary to maintain the radiator mounting surface on the crankcase at a temperature $\leq$ 28°C.

HEAT REJECTION INTERFACE

- HEAT TRANSFER UNIT (HTU) WILL THERMALLY MOUNT TO REFRIGERATOR CRANKCASE (64 CM AVAILABLE FOR THERMAL INTERFACE)
- (HTU) WILL TRANSPORT WASTE HEAT TO SPACE RADIATOR
- REFRIGERATOR CRANKCASE WILL BE MAINTAINED $\leq 40^{o}C$
- ESTIMATED CRANKCASE TEMPERATURE GRADIENT IS $12^{o}C$

SUMMARY OF HEAT REJECTED

Waste heat is rejected from the VM cooler to a radiator mounted on the vehicle; this radiator or an intermediate heat transfer unit, is mated to a flange on the refrigerator crankcase.

A total of 87 watts must be rejected across the interface. A summary of rejected heat is presented below.

HUGHES

SUMMARY OF HEAT REJECTED

$73^{o}K$ STAGE LOAD	0.42
$130^{o}K$ STAGE LOAD	0.27
HEATER POWER	75
MOTOR DISSIPATION	12
TOTAL HEAT REJECTED	87 WATT

FUNCTIONS OF ELECTRONICS INTERFACE UNIT

The EIFU consists of the motor inverter, temperature control circuits, and all other power conditioning circuits. 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

FUNCTIONS OF ELECTRONIC INTERFACE UNIT (EIFU)

HUGHES

- CONTROLS REFRIGERATOR DRIVE MOTOR
- CONTROLS TEMPERATURE OF COLD AND HOT CYLINDERS
- PROVIDES OVERTEMPERATURE PROTECTION FOR HOT CYLINDER
- PROVIDES SIGNAL CONDITIONING FOR INSTRUMENTATION
- SHUTS OFF POWER SHOULD ANY MALFUNCTION OCCUR (FAULT DETECTION)
- CONTROLS LOGIC FOR PROCESSING COMMANDS FROM GROUND

SECOND STAGE TEMPERATURE CONTROL

The temperature of second stage ($73^{o}K$ 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 an 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 duy 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.

SECOND STAGE TEMPERATURE CONTROL

HUGHES

- o OPTIONS
  1 VARYING HOT CYL TEMP
  3 HEATER ON COLD TIP
  3 HEATER IN DEWAR
  4 BY VARYING MOTOR SPEED
- o RECOMMENDATION

  VARY MOTOR SPEED USING ERROR SIGNAL FROM SECOND STAGE TEMP.

  1) HOT END TEMPERATURE

  2) MOTOR SPEED

EIFU POWER ESTIMATE

Here is given a rough estimate of the power dissipation in the EIFU. The instrumentation output is based on providing scaled analog signals.

The command logic circuit is based on digitally encoded commands and 5-volt logic. The command logic power might be changed by having separate lines for each command, or by using CMOS logic. The total power dissipation is estimated at 11 watts.

EIFU INPUT POWER ESTIMATE

HUGHES

1. ASSUME 10 MA/CHANNEL FOR OUTPUT DATA;
   20 CHANNELS REQUIRE ~ 200 MA
   POWER = 4.8 WATTS

2. ASSUME 20 MA/IC FOR COMMAND LOGIC,
   25 ICs REQUIRE ~ 500 MA; POWER = 1.2 WATTS

3. POWER LOSS IN TEMPERATURE CONTROLLER
   SWITCH = 6 WATTS

4. TOTAL INPUT POWER TO EIFU = 11 WATTS

BLOCK DIAGRAM OF EIFU

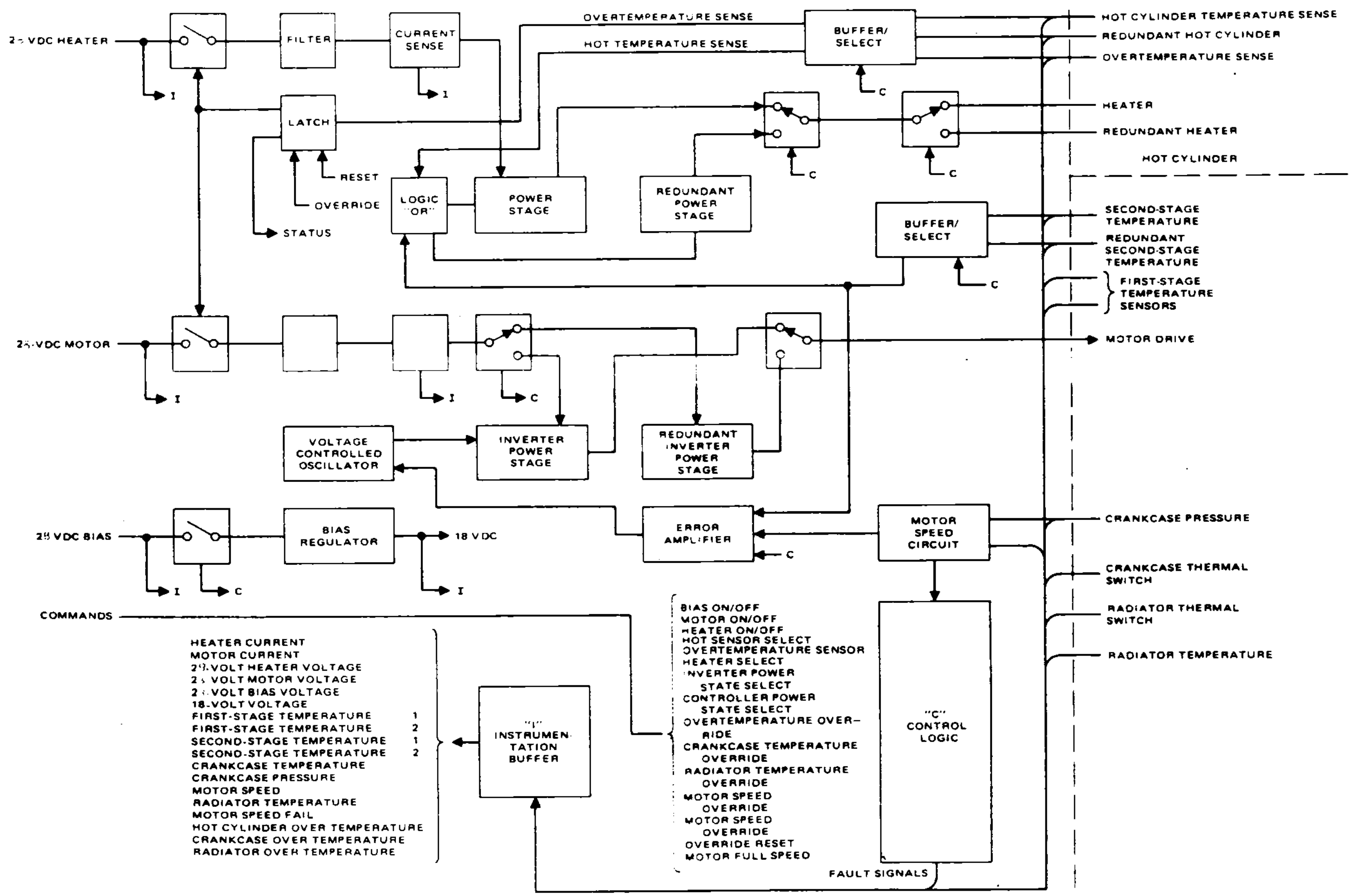
The dashed line at the right shows the interface between the EIFU and the cryogenic cooler. Sensor signals from the hot cylinder, the crankcase, the cold stages, and the radiator are input to the EIFU. Motor drive, heater drive, and transducer excitation voltages are input to the cooler from the EIFU.

Control logic box C accepts command signals from the ground and controls the various refrigerator functions. Since several combinations of sensors may be used to respond to failures in any one or two sensors, it is recommended that the redundancy function be controlled from the ground. The lower left hand portion of the diagram enumerates various instrumentation signals that should be telemetered to the ground in order to monitor refrigerator performance.

The upper part of the diagram depicts the functional signal flow that accomplishes the functions of the EIFU.

HUGHES

BLOCK DIAGRAM OF EIFU



IMPLEMENTATION PLAN AND SCHEDULE

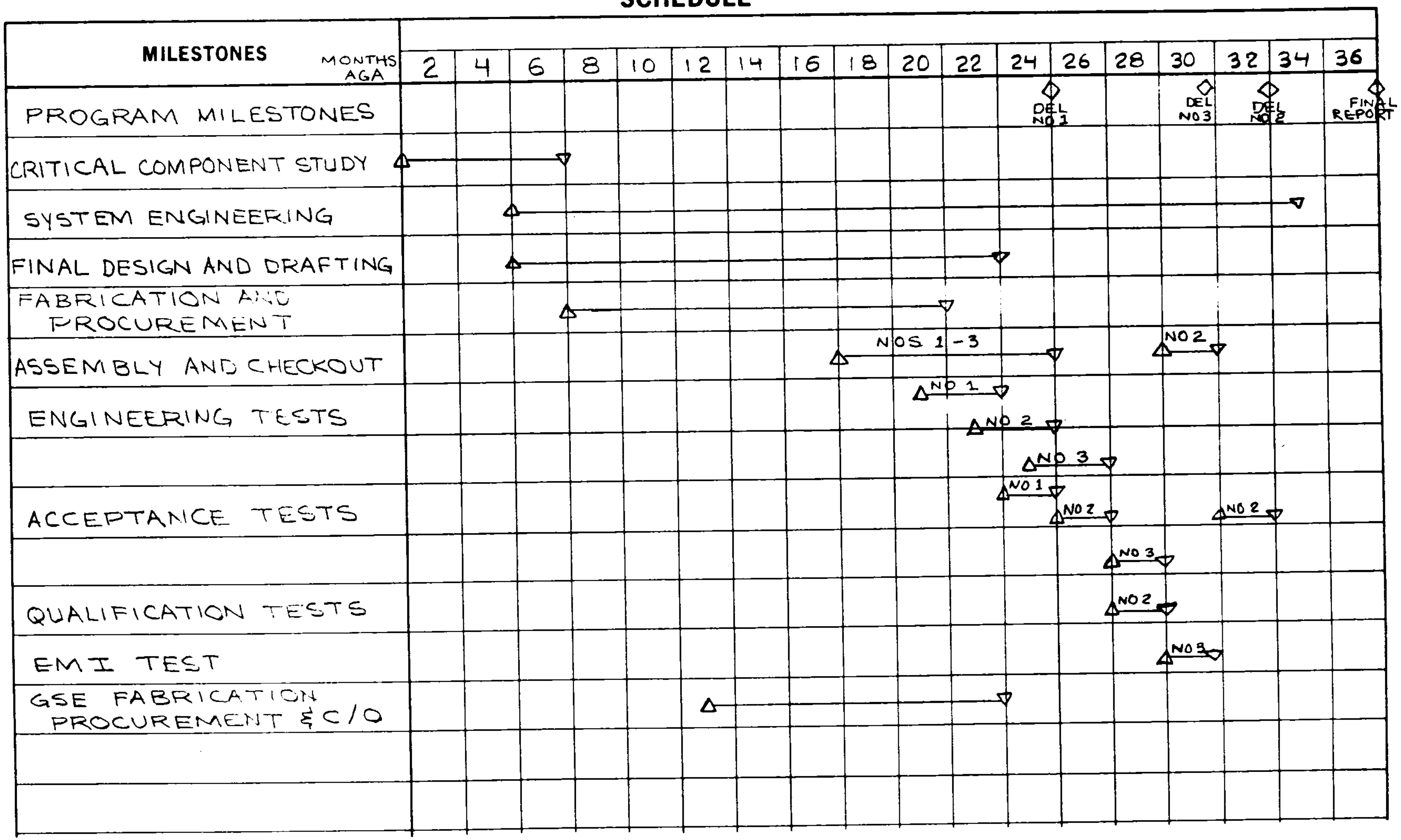
An implementation plan and schedule were developed for a follow on program to consist of critical component study, final design, fabrication, assembly, and test of three ALS VM coolers and one set of ground support equipment (GSE) to support an ALS flight experiment. The schedule below reflects the preliminary plan.

During the interim between the current study and the follow on development program it is recommended that a critical componenent study be performed. This effort would address the design of seals, bearings, and riders and could be completed before beginning principal development program. Such an interim program would facilitate the development effort and result in earlier hardware deliveries.

PROPOSED ALS COOLER DEVELOPMENT SCHEDULE

HUGHES

SCHEDULE



ACTIVE VM TECHNOLOGY PROGRAMS

Hughes Aircraft Company is now engaged in programs related to advancing VM technology state-of-the-art.

The purpose of the 5 year VM component development program is to develop improved designs for life limiting components and to improve refrigerator reliability and efficiency by exploring new design techniques and materials.

The SIRE program will deliver three long life state-of-the-art VM refrigerators for use in spaceborne systems.

The technology currently being developed is expected to advance the state-of-the-art of the life limiting components of VM cryogenic refrigerators to the point where goals of 45,000 hours can be realized.

ACTIVE VM TECHNOLOGY DEVELOPMENT PROGRAMS

HUGHES

1) COMPONENT DEVELOPMENT FOR FIVE YEAR VM COOLER

- o REGENERATOR SEAL DEVELOPMENT

  DEVELOP NEW SEAL DESIGNS

  IMPROVE LIFE CHARACTERISTICS; DEVELOP WEAR DATA

- o COLD REGENERATOR IMPROVEMENT PROGRAM

  IMPROVE RELIABILITY AND EFFICIENCY OF REGENERATORS

- o COOLER LIFE IMPROVEMENT

  DEVELOP REFRIGERATOR WITH LIFE $\geq$ 45,000 HOURS

2) SIRE PROGRAM (CRYOGENIC PORTION)

- o IMPROVE VM TECHNOLOGY; DEVELOP THREE LONG LIFE, SPACEBORNE VM REGRIGERATORS

CONCLUSIONS

The technical requirements for a cryogenic cooler capable of cooling the ALS detector are met with a VM cycle refrigerator utilizing technology developed during the CMP, Hi Cap, VM wear rate, SIRE, 5 Year Component Development 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 ALS program objectives with low developmental risk.

CONCLUSIONS

HUGHES

- TECHNOLOGY DEVELOPED ON PREVIOUS AND CURRENT PROGRAMS IS EXTENDING THE STATE OF THE ART OF VM TECHNOLOGY TO THE POINT WHERE LIFE IN EXCESS OF 20,000 HOURS CAN BE REALIZED
- TECHNICAL REQUIREMENTS FOR ALS COOLING ARE MET WITH A BASELINE VM COOLER UTILIZING EXISTING TECHNOLOGY DEVELOPED ON PREVIOUS AND CURRENT DEVELOPMENT PROGRAMS
- THE VM REFRIGERATOR IS IN AN ADVANCED STATE OF DEVELOPMENT FOR SPACE APPLICATIONS AND CAN BE EXPECTED TO MEET THE ALS OBJECTIVES WITH LOW DEVELOPMENTAL RISK
- THE BASELINE ALS VM REFRIGERATOR AS DEFINED BY THIS STUDY MEETS ALS REQUIREMENTS WITH REASONABLE WEIGHT AND POWER BUDGETS
- IT IS RECOMMENDED THAT INSTRUMENT RELIABILITY BE IMPROVED BY UTILIZING A REDUNDANT VM CYCLE REFRIGERATOR

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA CR-159250		

4. Title and Subtitle	5. Report Date
CRYOGENIC COOLING STUDY FOR THE ADVANCED LIMB SCANNER	May 1980
	6. Performing Organization Code
7. Author(s)	8. Performing Organization Report No.
S. C. Russo	
9. Performing Organization Name and Address	10. Work Unit No.
Hughes Aircraft Company Culver City, CA 90230	11. Contract or Grant No. NAS1-16029
12. Sponsoring Agency Name and Address	13. Type of Report and Period Covered
National Aeronautics and Space Administration Langley Research Center Hampton, Virginia 23665	Contractor Report
	14. Sponsoring Agency Code

15. Supplementary Notes

Technical Monitor: Robert D. Averill, NASA Langely Research Center

16. Abstract

A study was conducted to define the cryogenic operating requirements for VM refrigerator to be used for detector cooling of the ALS instrument. Determination of these requirements included consideration of mission and environmental constraints of the UARSP mission. Based on the operating requirements a detailed refrigerator design and performance analysis was conducted resulting in an optimized VM refrigerator conceptual design. The cooler-instrument-spacecraft interfaces were evaluated resulting in recommendations as to the best approach and most effective concepts resulting in simple and satisfactory interfaces. A preliminary implementation plan and schedule describing the general tasks required to develope, design, manufacture, test, and deliver three VM coolers and required support equipment to support an ALS flight experiment was prepared. The result of this effort show that existing VM cooler technology is sufficient to satisfy ALS mission requirements and schedules.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Advanced Limb Scanner Cryogenic Cooler VM Cycle	Unclassified-Unlimited

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	107	

N-305

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