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COOLING SYSTEMS FOR SATELLITE REMOTE SENSING INSTRUMENTATION

BY

R J COPELAND AND J A OREN

NASA CR-132517

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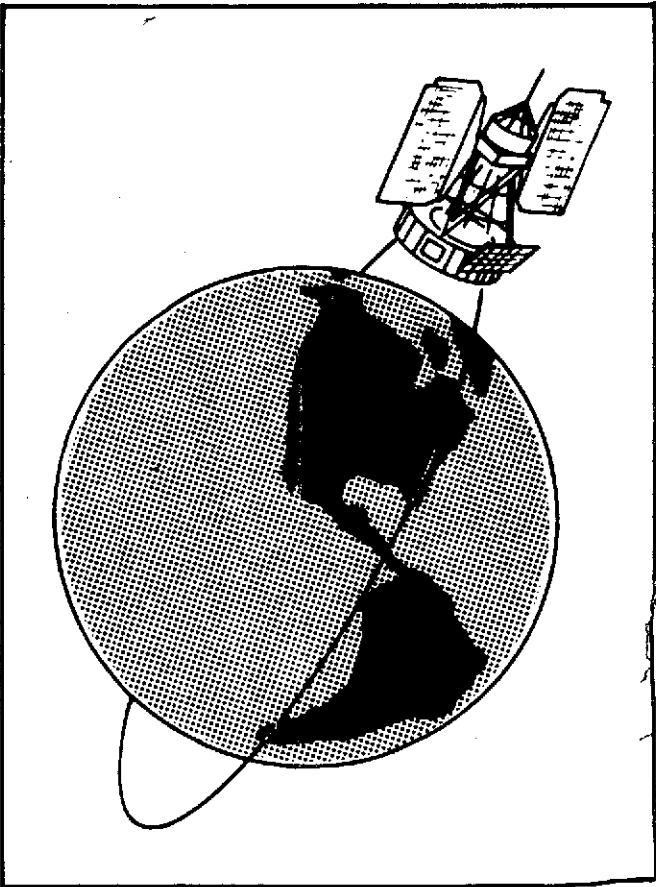
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16 SEPTEMBER 1974

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SUBMITTED BY

VOUGHT SYSTEMS DIVISION
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P.O. BOX 5907

Dallas, Texas

TO

NASA LANGLEY RESEARCH CENTER
Hampton, Virginia

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VOUGHT SYSTEMS DIVISION
LTV AEROSPACE CORPORATION

PERFORMED UNDER NASA-LRC CONTRACTS

NAS1-10900
and
NAS1-13500

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SATELLITE REMOTE SENSING INSTRUMENTATION

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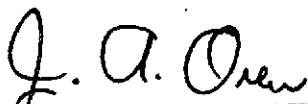NASA CR-132517

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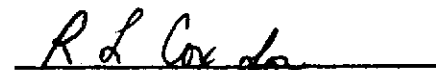P. O. Box 5907
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FOREWORD

This report documents the results of a study of coolers for
remote sensors on a satellite. The study concentrates on accumulating
characteristics of cooler for use in the preliminary design of a
Pollution Monitoring Satellite. The study was performed by the
Vought Systems Division of LTV Aerospace Corporation during the
period of January to September, 1974, as Support Item 127 of Contracts
NAS1-10900 and NAS1-13500.

Mr. Jim Pleasants and Dr. Bob Greene of NASA-LRC were the
technical monitors for this study, and their office supplied the
requirements of typical sensors employed in the study.

TABLE OF CONTENTS

APPENDICES

LIST OF FIGURES

LIST OF TABLES

1.0   SUMMARY

This report describes a study performed by the Vought Systems Division (VSD) of LTV Aerospace Corporation under NASA Langley Research Center Contracts NAS1-10900 and NAS1-13500, during the period of January thru June 1974. The purposes of the study were:

(a) determine the characteristics and use of proven and state-of-the-art cryogenic cooling systems for six (6) specified ranges of performance

(b) determine the systems most applicable for each of the six cooling categories

(c) perform conceptual designs for a candidate system for each of these six representative cooling categories

The state-of-the-art systems to be considered were those that will be available by approximately 1976 for a 1979-1980 flight of a Pollution Monitoring Satellite (PMS).

The six cooling categories or ranges of interest were:

I/III   50 milliwatt, 300 milliwatt, and 1000 milliwatt cooling load at 77°K

IV,V   50 and 500 milliwatts cooling load at 195°K

VI   1000 milliwatts cooling load at 300°K

The desired mission life for each of the categories is 2 years with some of the categories requiring intermittent operation and others continuous operation.

An extensive literature and industry search was performed to obtain characteristics of systems applicable as cryogenic coolers in the range of interest. The search was concentrated on systems applicable to long life in the environment of space. Applicable systems were identified for each category and their characteristics were tabulated.

A preliminary study identified those systems most applicable to each of the six cooling categories. The systems were categorized by cooling temperature as follows:

I/III   77°K   :   Vuilleumier, Joule-Thomson, Solid Cryogen

IV,V   195°K   :   Thermoelectric, Passive Radiator

VI   300°K   :   Semi-passive Radiator, Active Radiator

A more detailed study was then performed on the above systems to identify the most effective or promising cooler for each category. Conceptual designs were subsequently performed on the most promising system for each of the 6 categories along with three additional variations. Information of

1

primary interest was weight, volume and power requirement. Table 1-1 summarizes the conceptual designs.

A study objective was to accumulate data on all promising coolers for the PMS. Thus some cooler types were selected for conceptual designs for categories for which other cooler were lighter weight. For example, the VM's for either Category II and III could be sized for the smaller capacity of Category I, and the resultant weight and volume would be smaller than any of the Category I designs.

The loads and temperature for the six categories were defined by NASA for anticipated requirements. As development continues, conditions or constraints may change and may dictate the choice of another type of cooler. To assist in making selections scaling data are included. Weight, volume and power for different cooling requirements are presented as a function of life expectancy, capacity, and heat load. In addition the operating characteristics of each conceptual design are briefly described. An in-depth discussion of coolers is presented in Reference 1.

The following additional conclusions can be made as the result of this study:

 (1) Cryogenic cooler technology is available or will be available by 1976 to satisfy requirements for satellite remote sensing instrumentation. However, limited development of this technology may be required.

 (2) Systems using expendables are most applicable to missions with duration of two requires or longer.

 (3) Additional development should yield significant reductions in weight and volume, or increased reliability.

CATEGORY	COOLER TYPE	WEIGHT* LB	VOLUME	POWER* WATTS
I. 77°K, 50 MW **	JOULE-THOMSON MODIFIED JOULE-THOMSON SOLID CRYOGEN	300 100 100/140	27" D SPHERE 17" D SPHERE 21" D X 35"	0 2 0
II. 77°K, 1000 MW **	VM	84	6½" D X 25" + 35" X 35" RADIATOR	88
III. 77°K, 300 MW	VM DUAL VM	66 89	5" D X 23.5" + 31" X 31" RADIATOR 5" D X 23.5" + 5" D X 23.5" + 31" X 31" RADIATOR	65 65
IV. 195°K, 50 MW **	THERMOELECTRIC	1.87	.75" X .67" X .4" + 5.5" X 5.5" RADIATOR	2.06
V. 195°K, 500 MW	SHIELD PASSIVE RADIATOR OSR UNSHIELDED RADIATOR	3.5 3.9	6.12" X 11.88" D 8.48" X 8.48" RADIATOR	≈0 2.3
VI. 300°K, 1000 MW	SEMIPASSIVE RADIATOR	2.63	2.65" X 2.65" RADIATOR	2.7

*POWER EQUIVALENT WEIGHT @ 0.55 LB/WATT IS INCLUDED IN WEIGHTS WHERE POWER IS REQUIRED

** INTERMITTENT DUTY CYCLES; ALL OTHERS ARE CONTINUOUS FOR TWO YEARS

TABLE 1-1   CONCEPTUAL DESIGN SUMMARY

2.0   INTRODUCTION

A new series of satellites are currently being planned by LRC to remotely sense conditions in the environment of the earth. Of particular interest are pollution of the land, water and air. As envisioned each spacecraft could contain from 4 to 10 sensors. For the spectral bands of interest, cooling of the detectors to cryogenic temperatures will be necessary for most sensors. The spacecraft design will depend upon the number and type of sensing instruments and their associated support equipment. This study presents weight, volume, shape, power requirements and operating characteristics for several representative cryogenic coolers, as well as conceptual designs of several of the more promising types. This information is provided for use in the preliminary design of remote sensing instruments and spacecraft.

The study consisted of three general tasks. The first task was to perform a literature and technology search to determine the characteristics and use of proven flight systems and state-of-the-art systems that will be available by approximately 1976 for a 1979/1980 flight. This search was limited to systems in the 60 to 300°K refrigeration temperature range and included expendables, closed cycle refrigerators, thermoelectric coolers, and radiator coolers.

The second task was a preliminary screening of feasible cooling system for each of six cooling system categories with requirements which were determined by NASA to be representative. The categories are defined in Table 2-1. The objective of the preliminary screening was to identify promising systems for each category. The items evaluated in the trade-off included weight, volume, radiator area, life expectancy, power required, reliability and cost. Figure 2-1 illustrates the heat load definition.

The third task was to perform a conceptual design for a promising cooler for each of the six categories. A candidate system for each instrument category was selected for design following completion of Task II. The design included component identification, dimensions, volume, power, and a narrative description.

CATEGORY	DETECTOR TEMPERATURE $^\circ$K	DETECTOR COOLING LOAD MW	LIFETIME YEARS	DUTY CYCLE
I	77	50	2	A + B
II	77	1000	2	A + C
III	77	300	2	D
IV	195	50	2	A + B
V	195	500	2	D
VI	300	50−1000	2	A + B

DUTY CYCLE DEFINITION

    A — CONTINUOUS OPERATION FOR 1 MONTH

    B — 5 TO 8 SELECTED DAYS OPERATION PER MONTH

    C — 10 DAY CONTINUOUS OPERATION PER 6 WEEK PERIOD

    D — CONTINUOUS OPERATION

TABLE 2-1   INSTRUMENT COOLING CATEGORIES
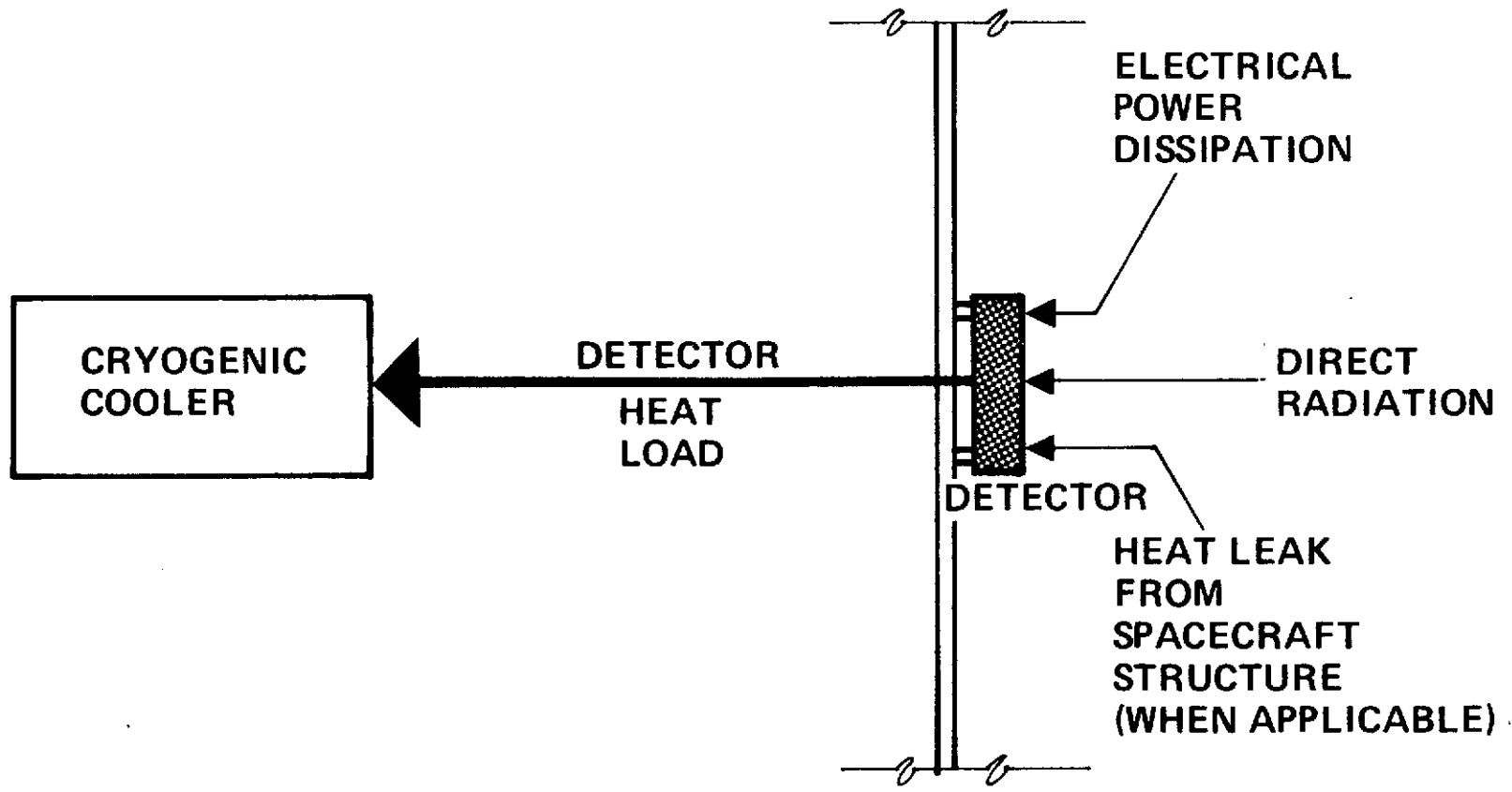
FIGURE 2-1   DEFINITION OF DETECTOR HEAT LOAD ON CRYOGENIC COOLER

The results of Task I are described in Sections 3.0 and 4.0. The results of Task II are given in Section 5.1. The results of Task III are given in Sections 5.0 and 6.0. Some advanced concepts are identified in Section 7.0. Conclusions and Recommendations are given in Sections 8.0 and 9.0.

3.0   SUMMARY DISCUSSION OF CRYOGENIC REFRIGERATION SYSTEMS

The following description of cryogenic refrigeration systems is taken from Reference (1).

"The refrigeration systems available for cryogenic cooling may be categorized into four fundamental types: closed-cycle mechanical refrigerators, open-cycle expendable systems, passive radiative coolers, and thermoelectric coolers.

The simplest and potentially most reliable method of providing cryogenic cooling is to utilize the low temperature sink of space directly by using a radiator. The concept is attractive since the system is passive, requires little or no power and is capable of high reliability for extended periods. The low temperature radiator must be shielded against direct sunlight and the parent spacecraft and, in the case of near-earth orbits, heat inputs from direct thermal emission and reflected sunlight from the earth and its atmosphere. Primary limitations of this approach are the rapid increase in radiator size with decreasing temperature and the parasitic heat leak into the radiator.

Thermoelectric coolers are appropriate to provide cryogenic temperatures for small wattage heat sources. Based on the Peltier cooling effect arising from passing a current through a junction of different materials, thermoelectric coolers provide a simple, lightweight, reliable method of cooling. These systems are limited primarily by the low efficiencies (in the order of one percent) and maximum operating temperature difference between the hot and cold junctions.

For operation at lower temperatures and/or higher cooling capacities, open-cycle expendable systems are appropriate. The simplest, least expensive approach is to use a Joule-Thomson (J-T) cooler wherein a high pressure gas (in the range of 1000 to 6000 psia) combined with a J-T expansion valve (consisting of an orifice, small diameter tube heat exchanger, shield, etc.) results in cooling of the gas and ultimately provides a source of liquid at the point to be cooled. The utilization of helium, hydrogen, argon or nitrogen enables developed units to provide cooling from approximately 4.2 to 77°K at capacities in the range of 0.50 to 10 W. The primary limitation is the high weight penalty for storage of high-pressure gas (approximately 4 lb per pound of gas for N_2

and 15 for H_2). One advantage of this approach over cryogenic storage is the ability to provide intermittent operation over a long period of time.

Cryogenic fluids stored as liquids in equilibrium with their vapors (subcritical storage) can provide a convenient constant temperature control system for ground-based or advanced aircraft and spacecraft applications. A variety of fluids are available which provide temperatures ranging from 4.2 (helium) to 240 K (ammonia). The primary limitation of this approach is the complex tank design required to minimize boiloff and the direct relation of weight and volume requirements with elapsed time.

The same fluids can be stored at pressures above their critical pressures (supercritical storage) as homogeneous fluids, thus eliminating phase separation problems encountered during weightless conditions in space. A weight penalty as compared with subcritical storage normally accrues as a result of the higher operating pressures required. The added flexibility of a single phase homogeneous fluid makes this approach competitive, in some cases, with closed-cycle mechanical refrigerators for missions of 60 to 90 days.

The sublimation of a solidfied cryogen can provide reliable refrigeration for small wattage heat sources for periods measured in months to a year or longer. This approach utilizes a solidified cryogen in conjunction with an insulated container, an evaporation path to space, and a conduction path from the coolant to the device being cooled. Advantages over the use of cryogenic liquids include a higher heat content per pound of coolant, and a higher density storage resulting in less storage volume, and the lower temperature solid phase can permit a gain in sensitivity in certain infrared detectors. Temperatures ranging from 10K (using hydrogen) to 90K (using methane) and 125K (using CO_2) or higher are achievable. Limitations involve restrictions on detector mounting, specialized filling procedures, and temperature control requirements.

A limited number of fluids exist which can be stored at ambient temperature (thus eliminating the boiloff problem of cryogenic fluids) and thermodynamically manipulated to provide cooling at about 100°K. However, development is required to advance this concept to the hardware stage.

To provide cooling in the range of approximately 4 to 100 K or higher in capacities ranging from a fraction of a watt to 100 W for periods of months to years, closed-cycle mechanical refrigerator systems may be required. The

most significant components of mechanical refrigerator systems involve a power supply, power conditioning equipment, the refrigerator itself, and the heat rejection system which normally includes a remote radiator and a heat transport loop. A number of different refrigerator cycles are in various stages of development for space applications, but none currently have demonstrated the capability to operate maintenance-free in excess of 2000 hr. The high penalty for spaceborne power places a premium on refrigerator efficiency. Reliability also is of primary importance.

Stirling and Vuilleumier (VM) cycle systems possess several of the primary requirements for spaceborne refrigeration systems. The primary area of application of these systems is at temperatures above about 10 K and for capacities below 50 to 100 W. Both of these cycles can be classed as intermittent flow systems and utilize thermal regenerators to store and release energy during the completion of each cycle. As the specific heat of all materials becomes very small near absolute zero, the effectiveness of storing thermal energy, and thus the efficiency of both these cycles, becomes very poor at temperatures below about 10 K.

Gas-bearing supported turbomachinery utilizing reversed Brayton and Claude cycles is being developed for use with very low temperatures and/or high capacity systems. Turbomachinery units appear to have the best potential for long life. However, the high power requirements (due primarily to low efficiencies of turbocompressors and expanders) make these systems competitive only at temperatures below about 20 K at higher capacities where the component efficiencies improve.

Rotary-reciprocating machinery utilizing the Brayton or Claude cycle (in which portions are rotated as well as reciprocated) also shows promise for long life and potentially provides a minimum power system at temperatures below about 20 K. Both turbomachinery and rotary-reciprocating units are being developed."

4.0   SURVEY RESULTS

Literature and industry surveys were performed during the early portion of the cooler study to determine the characteristics of a) proven flight cooler systems and b) state-of-the-art cooler systems which could be available by 1976 for a 1979/1980 flight. The survey included cooling systems with a cooling capability in the range of 0.05 watts to 5 watts at a cooling load temperature of 60°K to 300°K. Primary emphasis during the survey was placed on systems with potential for long life (1 to 2 years) operation in the environment of outer space.

The literature search resulted in 70 reports which were considered pertinent to the program. A bibliography of the reports presented in Appendix A is categorized by cooler type. Reference $(1)^*$ is considered directly applicable to this study.

The industry survey resulted in a number of systems being identified in the ranges of interest. These systems were categorized by type and temperature level. The results are tabulated in Appendix B.

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References are included in Section 10.0

5.0   SELECTION OF COOLER TYPES FOR CONCEPTUAL DESIGN

The trade-off and rationale items for the selection of cryogenic coolers for preliminary design are presented in this section. Initially the available coolers were screened for each category to eliminate obvious non-contenders from further consideration. Next a trade-off was conducted to identify the most promising concept for each category. The categories are only representative, and different conditions (e.g., duration, capacity, restrictions) may dictate the selection of another cooler. Recognizing this fact, at least one conceptual design was selected for each promising cooler.

Section 5.1 presents the guidelines for the selection of promising coolers employed in this study. This data is also useful for temperature, durations, and capacities outside the range of interest of this study. Section 5.2 presents the coolers selected for conceptual design and the rationale for each.

5.1   Cryogenic Systems Selection Guidelines

To aid in selecting the type of cryogenic refrigeration system for use in a given application, two system selection maps have been prepared (Figures 5-1 and 5-2). Three primary variables (temperature, refrigeration capacity, and mission duration) are required in most cases to properly identify the most desirable system. Figure 5-1 shows refrigeration capacity in watts versus mission duration in days, while Figure 5-2 shows refrigeration capacity versus refrigeration temperature ($^\circ$K). It should be clearly noted that these merely represent guidelines and in many cases, especially where regimes are in close proximity or even overlap, additional criteria such as booster payload capability, geometry limitations, type of orbit, the reliability required, the development cost and time available, etc., will determine which system is most appropriate. These charts are based on the technology that either exists or is under development to the point at which a given system can be applied in the next two to three years. Figure 5-1 establishes the fundamental concepts that can be considered and Figure 5-2, along with various charts in the report, defines the specific system.

The upper bounds shown in Figure 5-1 for open-cycle systems are based on weight limitations associated with a 3000- to 4000-lb spacecraft and a cryogenic storage limitation of 90 days. However, tankage currently under

12
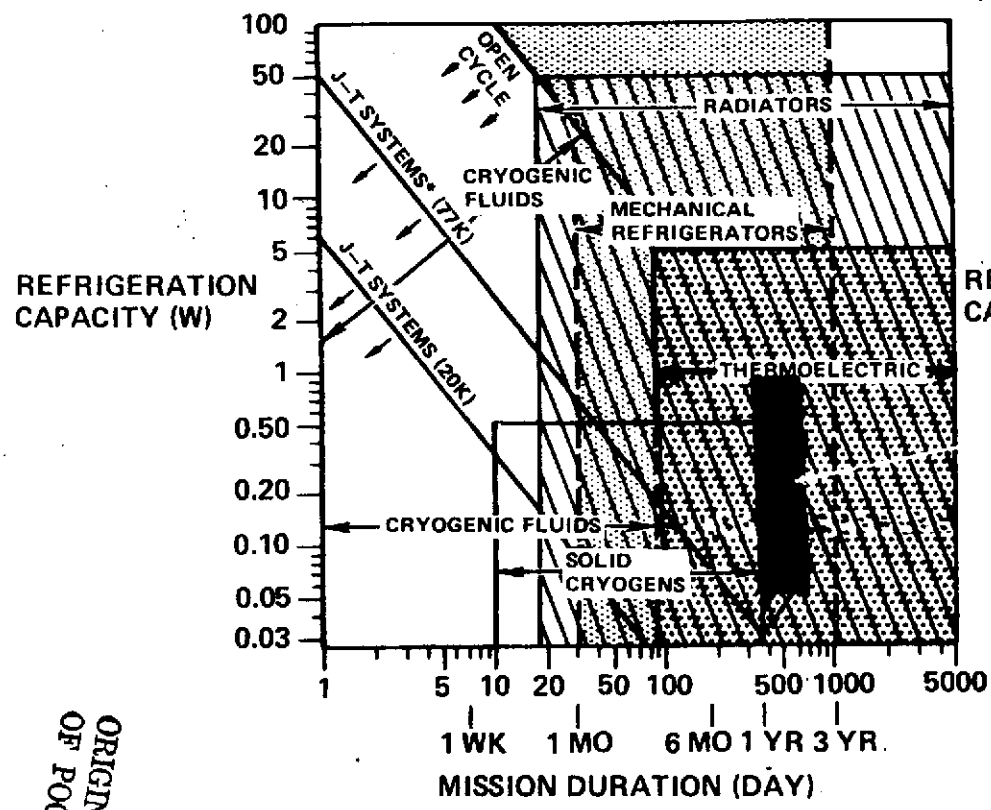
*HIGH PRESSURE GAS STORAGE WITH J—T EXPANSION

FIGURE 3-1

REFRIGERATION CAPACITY VS DURATION
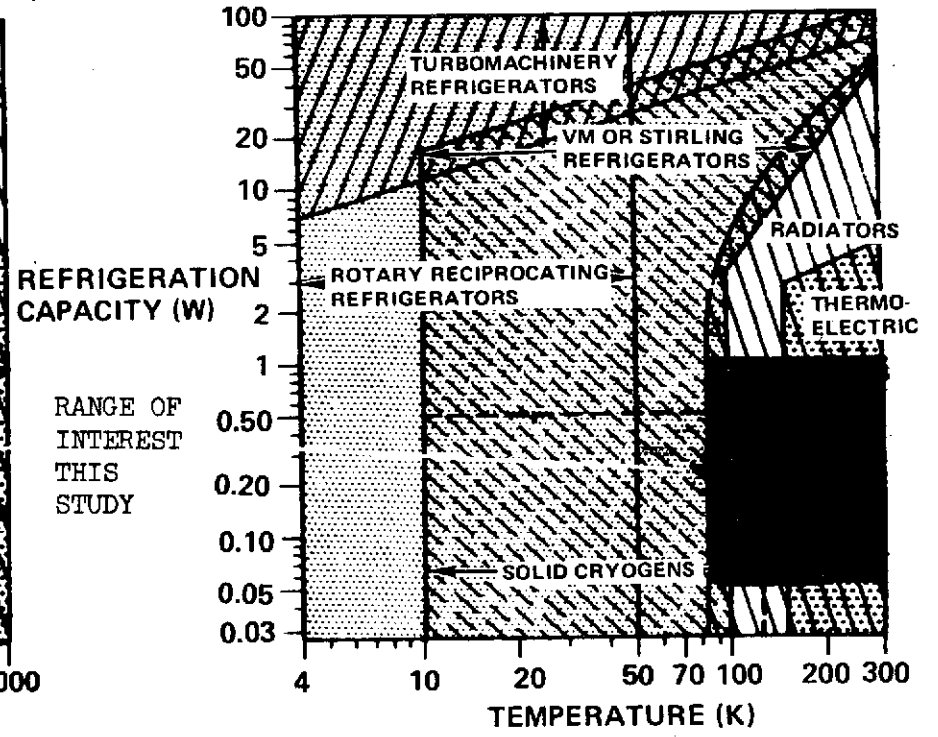
NOTE: OPEN-CYCLE FLUID SYSTEMS ARE
APPLICABLE AT ANY POINT ON THIS
CHART BECAUSE THEY ARE LIMITED
ONLY BY DURATION

FIGURE 3-2

REFRIGERATION CAPACITY VS TEMPERATURE

PRELIMINARY SYSTEM SELECTION

development indicate durations of six months to a year and longer may be feasible. Open-cycle systems utilizing cryogenic fluids are applicable at essentially any point within the capacity and temperature boundaries of Figure 5-2 and are limited primarily only by weight (as determined by mission duration and refrigeration capacity). For very short durations and capacities high-pressure gas systems coupled with J-T expansion valves provides the simplest, most economical means of refrigeration. However, the weight penalties for high-pressure gas storage limit the practical duration. Solid cryogenic systems are most attractive for capacities of less than about 0.50 W for temperatures ranging from about 10K (hydrogen) to about 125K (CO_2) but are feasible for substantially larger loads.

For temperatures above about 100 K passive systems utilizing radiators become attractive, especially where long mission durations are involved. This may preclude the use of mechanical refrigerators. As the temperature increases, the radiator becomes more effective due to the fourth power radiation factor and is competitive at larger refrigeration capacities. Beyond about 150 K, and for loads less than about 2 to 5 W, thermoelectric systems become competitive, although the most attractive area of application is in the fractional wattage load range.

Mechanical refrigerators are applicable over a wide range of conditions from about 4 K to near 100 K at moderate capacities and possibly to higher temperatures at relatively higher capacities. Many specific applications may exist at capacities from a fraction of a watt up to 100 W for mission durations ranging from a month to possibly three years and beyond as limited by refrigerator capabilities.

Stirling cycle systems have been developed and are operational in aircraft (and one spacecraft). These machines are the most efficient in terms of power requirements down to about 10K (as limited by regenerator efficiencies) at moderate capacities. Maintenance-free operational life of 1000 hours has been demonstrated. Vuilleumier cycle systems recently developed have inherent capability for longer life than Stirling cycle units and have the potential advantage of being powered directly by a heat source. A VM unit was recently successfully operated in space, while a number of units are in various stages of development for aircraft and spacecraft uses.

Turbomachinery or rotary-reciprocating machinery utilizing Brayton or Claude cycles are applicable from temperatures of around 4 K up to 40 or 50 K. The lower efficiencies associated with the turbomachinery systems will generally restrict applications to the larger capacities. Both of these systems may have potential for extremely long life. Components have been developed and fabricated; however, no significant overall system performance data or operating experience are available.

The primary area of interest in this study is shown on Figures 5-1 and 5-2. The coolers of interest for each temperature region and a one to two year duration are as follows:

77°K	–	Vuilleumier
		Stirling
		Joule-Thomson
		Solid Cryogens
195°K	–	Thermoelectrics
		Passive Radiators
300°K	–	Active Radiators
		Semi-Passive Radiators

Based on data obtained in the literature Stirling cycles were eliminated from conceptual designs since they have little promise of meeting a 2 year life without maintenance. For applications on the order of 6 months, the Stirling cycles are excellent candidates.

5.2   <u>Comparison of Coolers</u>

Trade-off data were generated to select the best coolers for each category. In Task II potential suppliers were contacted and sizing data obtained for each type of cooler. Using that data equivalent weights were calculated for each promising cooler. Where power penalties were applicable, the following factor was applied:

$$W_e = 0.55 \frac{LBS}{WATT(electrical)} = (\frac{0.25 \ Kgm}{WATT(electrical)})$$

That factor is typical of a solar cell array and is employed in all studies requiring an estimate of equivalent weight.

All promising coolers were compared solely on the basis of weight. Volume and cost are also extremely important to the selection of a cooler. However, equivalent weight is good indication of volume requirements and equipment

costs are many times secondary to the launching costs associated with weight. No attempt is being made to make "final" selections. Rather, the objective is to identify the type of cooler generally most suited to a given cooling requirement. The trade-offs for each temperature range are presented in the following paragraphs.

5.2.1   77°K Trade-Offs

Table 5-1 presents approximate weights for three coolers applied to three categories of applications. Joule-Thomson (J-T) is reasonable only at the smallest integrated heat load and too heavy for categories II and III (order of 6000 lbs). The Vuilleumier (VM) coolers have the lowest weight for all three categories for a two year system. However, for one year life the solid cryogens have the lowest weight in all categories. J-T does not become weight attractive until the integrated heat loads are very small.

The J-T cooler weight was estimated from data in Reference 5-1 and property data for N_2. The cooler unit itself is very small (typically 1.0 pound). Nearly all of the weight is for storage of the expendable gas (N_2) at high pressure and room-temperature. Typical tankage penalty factors average around 3 to 4 lb (tank + N_2) per lb of N_2. Since typical cooling effects for N_2 are 1 to 3 watt-hours of useful heat load per lb of N_2, large quantities are required; and thus a large system weight. These units however have two very important advantages; low cost and high reliability.

The solid cryogen weight was estimated from parametric data supplied by Aerojet General (Reference 2). A 2 year duration and CH_4 expendables were employed for the weight estimates. No power is required for this cooler. However, these designs are expensive and have not been demonstrated for a 2 year application. Solid cryogens have been used in space, however for very limited durations. Problems with controlling and predicting the parasitic heat leak have caused actual durations to fall short of projections by 30% or more.

Vuilleumier (VM) systems require a cooler unit, power for the cycle, and heat rejection by radiators. Cooler unit weights were obtained from data in Ref. 1. Power and heat rejection requirements were estimated from data in Ref. 3. Based on the above power penalty and radiators at 300°F, the total equivalent weights were summed. The VM has the advantage of low weight and small volume.

16

TABLE 5-1

77°K TRADE-OFFS

	APPROXIMATE WEIGHTS*		
	JOULE-THOMSON	SOLID CRYOGEN	VM

CATEGORY I
  — 2 YEAR LIFE
  — 50 MW DETECTOR LOAD
  — 174.5 TO 257.3 WATT-HOUR
    INTEGRATED HEAT LOAD
	300 TO 400 LBS	80 LBS	50 LBS

CATEGORY II
  — 2 YEAR LIFE
  — 1000 MW DETECTOR LOAD
  — 4731 WATT-HOUR
    INTEGRATED HEAT LOAD
	—	150 LBS	130 LBS

CATEGORY III
  — 2 YEAR LIFE
  — 300 MW DETECTOR HEAT
    LOAD, CONTINUOUS
  — 5260 WATT-HOUR
    INTEGRATED HEAT LOAD
	—	150 LBS	90 LBS

*BASED ON PRELIMINARY SIZING DATA OBTAINED FROM SUPPLIERS AND INCLUDING RADIATOR, POWER PENALTIES AND PARASITIC HEAT LEAK AS APPLICABLE

However, development is still in an early stage; and life expectancies are still projections based on data on a limited number of components rather than being observed from a full system. Improvements are still being made and current projections indicate that a 2 to 5 year life is feasible, even without maintenance.

Both the J-T and solid cryogens require venting of the expendables; the highest rate being with the J-T. Some form of thrust nullification will be needed on a satellite to minimize the effect on the orbit. Since thrust nullifiers are imperfect, additional expendables for the reaction control system may also be needed. Expendables may also contaminate the environment of the spacecraft. If there are one or more experiments requiring cooling to a temperature which is lower than the condensation temperature of the expendables, the vented expendable may accumulate on the detector. The condensed film may absorb the radiation from the item(s) being monitored thus eliminating any useful data. That effect is strongly affected by the temperatures, the nature of the expendable, and the location and direction of the exhaust. Due to the latter effect, venting normally causes no problems to cryogenicly cooled detectors.

5.2.2   195°K and 300°K Trade-Offs

Table 5-2 presents approximate weights for 195°K and 300°K coolers for three categories of applications. Passive radiators offer the lowest weight option over all heat load ranges, durations and either 195°K or 300°K applications.

Weights for 195°K passive radiator were scaled based on data in Reference 4. These radiators impose an orientation constraint on the satellite since they must have a clear view of deep space to provide the necessary heat rejection. Shields are employed to increase the number of orbits and locations on the satellite. The shields are normally either conical or parabolic. Although the weight impact for the shields is not large, the volume is increased by an order of magnitude. Thus low temperature radiators are suited only to application where the orbit, location on the satellite and available volume allow

TABLE 5-2

195°K AND 300°K TRADE-OFFS

	APPROXIMATE WEIGHTS*			
	SOLID CRYOGEN	THERMO-ELECTRIC	PASSIVE RADIATOR	ACTIVE RADIATOR
CATEGORY IV — 2 YEAR LIFE — 50 MW HEAT LOAD — 174.5 TO 257.3 WATT-HOUR INTEGRATED HEAT LOAD — 195°K	18.4 LBS	1.2 LBS	1 LBS	—
CATEGORY V — 2 YEAR LIFE — 500 MW HEAT LOAD CONTINUOUS — 8766 WATT-HOUR INTEGRATED HEAT LOAD — 195°K	—	12.3 LBS	2 TO 3 LBS	—
CATEGORY VI — 2 YEAR LIFE — 1000 MW HEAT LOAD CONTINUOUS — 300°K	—	—	¼ LB	¼ LB BUT MORE COMPLEX AND EXPENSIVE THAN PASSIVE

*BASED ON PRELIMINARY SIZING DATA OBTAINED FROM SUPPLIERS AND INCLUDING RADIATORS, POWER PENALTY AND PARASITIC HEAT LEAK AS APPLICABLE

300°K radiators reject heat well above the equivalent sink temperature at all locations and orbits of the satellite. The radiator should be coated with a material having a high emittance and low solar absorptance, such as white paint. Weight in Table 5-2 were estimated assuming a full view of deep space. The area requirements were then calculated from Stefan-Boltzman's law as follows:

$$A = \frac{Q}{\epsilon \, \sigma \, T^4}$$

where: A   =   area

Q   =   detector heat load

ϵ =   emittance of the radiator surface (0.9 is typical)

σ =   Stefan-Boltzman constant

T   =   absolute temperature of detector

The radiator area weight was then determined assuming a typical value of 1-2 lb/ft^2.

Passive radiators have only direct conduction paths to distribute heat in the radiator. Active radiators employ either pumped fluid loops or heat pipes in addition to direct conduction paths to distribute heat through the radiator. In large sizes the passive radiators need very thick solid metal conduction paths; thus very heavy devices result when conduction distances are greater than 5 to 10 inches (12.7 to 25.4 cm). For longer distances the active system divides the radiator into an array of small passive radiators with heat delivered by a pumped fluid or heat pipe. The required conduction distances are very small for the detector heat load in this study (0.1 inches, 2.5 cm) and passive radiators are best at 300°K. For dissipating the large heat load from an active refrigerator such as the VM large radiators (approximately 30 x 30 inches, or 75 x 75 cm) are required, and active radiators provide significant weight savings.

Thermoelectric coolers (T/E) weights were scaled from data in Reference 5. The largest factor (over 3/4 of the total) is the power penalty to run the thermoelectric device. At very small sizes the penalties are small, but as size increases very large penalties result (compare the 50 mw and 500 mw, 195°K coolers versus passive radiators). T/E rejects heat at high temperatures (300 to 325°K). At that temperature the radiators impose no constraints on the

orbit of the satellite or location on the spacecraft. Thus T/E may be applied when passive radiators can not.

Solid cryogen weights were scaled from data supplied by Aerojet-General, Reference 2. NH_3 was selected for the expendable since it has the highest cooling capacity and minimum volume at 195°K. A rather large weight results due to the large impact of the parasitic heat leak for a two year application. Both passive radiators and thermoelectric coolers have lower cost and smaller volumes and do not require venting of expendables. Hence solid cryogens are not good choices for these categories.

5.2.3   110° – 140°K and 243°K Coolers

Two other categories of coolers were considered in this study; 110°-140°K range and a 243°K range, both with 50 to 1000 mw heat load and 1 to 2 year duration. No preliminary designs were prepared for either but promising candidates are identified in the following paragraphs.

110°-140°K Coolers

Table 5-3 presents the weights of promising coolers. A heat load of 76 mw at a temperature of 125°K was arbitrarily selected since data on solid cryogens were directly available. Data for both one and two year duration are presented to show the relative impact of lifetime.

Shielded, passive radiators are the first choice for this range. Orientation constraints and locations on the satellite are manditory.

Vuilleumier refrigerators are very attractive for the 110° to 140°K range. The power penalties, the radiator and VM cooler sizes are all small. The weight is larger than with the shielded radiators but there are no orientation constraints.

Thermoelectric devices are not useable in this range. Potentially new materials can be developed which can provide a cooling effect. Even then the power penalty is likely to be excessive.

Solid cryogens are very attractive for one year or less. The penalties for expendables for two years become quite large.

Certain liquids can be stored at room temperature and thermodynamically manipulated to provide cooling at cryogenic temperatures. Ethane is one of these fluids. For continuous use the expendable requirement makes that approach heavy even for one year use. The concept has promise for intermittent use. Then the expendables are required only for the useful heat load and the expendables may be stored for long periods of time without loss.

21

TABLE 5-3

110-140°K RANGE COOLER TRADE-OFFS

	APPROXIMATE WEIGHTS[*]	
COOLER	1 year duration	2 year duration
Solid Cryogen, CO_2 @ 125°K	20 lbs.	50 lbs.
Liquid Ethane (stored at 300°K, evaporated at 125°K)	50 lbs.	90 lbs.
Shielded Radiator (125°K)	15 lbs.	15 lbs.
VM	30 lbs.	30 lbs.
T/E	Outside Usable Range	

[*]Based on 76 MW continuous heat load 666 watt-hours integrated heat load for one year, 1332 watt-hours for 2 years.

22

<u>243°K Coolers</u>

At this temperature level some type of radiator is the best choice for all heat load and durations. Available coatings provide low solar absorptance and high emittance (i.e., low sink temperatures). The following equivalent sink temperatures for typical coatings are obtained for the conditions indicated:

CONDITION	OPTICAL SOLAR REFLECTORS	SILVER BACKED TEFLON	WHITE PAINT
Sun synchrous orbit, earth oriented satellite, radiator facing deep space	4°K	4°K	4°K
Any orbit, earth oriented satellite, radiator facing away from earth	197.5°K	235°K	272°K
		[Equilibrium temperature in full sun]	
Any orbit, earth oriented satellite, radiator facing earth	249°K[1] 255°K[2]	249°K[1] 258°K[2]	249°K[1] 270°K[2]

(1) Earth in darkness
(2) Above sub-solar point

Either optical solar reflectors [silver backed, thin fused silica tiles] or silver backed Teflon can be employed where the radiator may face full sun. If a full view of the earth is necessary, no radiator can reach the desired temperature. An active cooler, most probably a Thermoelectric device would be required. For conditions between the two extremes, radiators may or may not need an active cooler. Data are presented in Appendix C to determine the incident thermal radiation and thus the equivalent sink temperature as a function of orbital parameters and location on the spacecraft.

5.3   <u>Selections for Conceptual Design</u>

Table 5-3 presents the coolers which were selected for conceptual designs. Not all selections are optimum for each category as defined. Since the categories are only representative, at least one conceptual design was chosen for all promising coolers. These designs illustrate the operation of the system and present representative weight and volume data.

TABLE 5-3

SELECTIONS FOR CONCEPTUAL DESIGNS

CATEGORY	COOLER	RATIONALE
I. 77°K, 50 MW	JOULE-THOMSON	● J-T BECOMES BEST CHOICE AT VERY LOW HEAT LOADS AND IS LOWEST COST SYSTEM
	SOLID CRYOGENS	● SOLID CRYOGEN BECOMES THE BEST CHOICE AT LIFE OF A YEAR OR LESS
II. 77°K, 1000 MW	VM*	● LOWEST PENALTY SYSTEM
III. 77°K, 300 MW	VM*	● LOWEST PENALTY SYSTEM
IV. 195°K, 50 MW	THERMO-ELECTRICS	● NO WEIGHT OR COST ADVANTAGES BUT ALLOWS FREEDOM OF LOCATION OF COOLER
V. 195°K, 500 MW	PASSIVE RADIATOR	● LOWEST PENALTY SYSTEM HIGH RELIABILITY, BUT REQUIRES SPECIFIC LOCATION ON THE SPACECRAFT
VI. 300°K, 50 TO 1000 MV	PASSIVE RADIATOR	● ACTIVE RADIATOR HAS HIGHER COST AND WEIGHT

*ACTIVE RADIATOR SELECTED FOR HEAT REJECTION DUE TO NEED FOR REMOTE LOCATION AND WEIGHT SAVINGS OVER PASSIVE RADIATOR

The Joule-Thomson and solid cryogen coolers were selected for category one, 77°K. Joule-Thomson is very heavy; but has the advantage of being a very low cost system. At very small heat loads the weight penalties are small enough for this cooler to be favored over others. Solid cryogens are relatively expensive and are not well suited to life on the order of two years. For application with less than one year of life, solid cryogens can be the lightest weight system.

Vuilleumier (VM) coolers were selected for both category II and III. The expendable systems require large quantities at these heat loads and are thus not very attractive. Stirling cycle machines can offer even lower penalties. However, those coolers are limited to approximately 6 months of life without maintenance. For one to two year durations, the VM's are much more promising.

Passive radiators provide lower penalties than active coolers at 195°K. The devices are highly reliable but require both specific orbits and locations on the spacecraft. When orbital parameters do not allow the use of radiators, T/E are best. When the heat loads are small, the penalties are very small for T/E; and in some designs T/E may be chosen over passive radiators to maintain close proximity of the detector and cooler.

At 300°K passive radiators are not limited in either orbit or location on the spacecraft. For the small heat load of category VI, the very simple passive radiators offer both cost and weight advantages over active radiators. Active radiators were selected to reject the heat from the VM coolers. The quantity of heat and size of the radiators are sufficiently large to warrant the active system.

Conceptual designs were generated for the specific conditions of all 6 categories. The designs are representative of similar heat loads and durations. No conceptual designs were generated for the 110-140°K and the 243°K ranges coolers for these last two conditions are similar to one of the other conceptual designs.

6.0   CONCEPTUAL DESIGNS

The conceptual designs for all 6 categories are presented in this
section. The operating fundamentals on each cooler are presented first. Then
the equipment is described. Weight, power and envelope data are presented
for the specific conditions. Parametric scaling data for other heat loads and
durations are included to extrapolate the design to other conditions.

6.1   Category I

Category I requires 50 mw of cooling on an intermittent duty cycle
250 watt-hour of useful heat load in a 2 year period. Two expendable coolers
were selected, Joule-Thomson (J-T) and solid cryogens.

The intended application does allow for additional development. Advanced
concepts for both types are considered to insure that the data are representative
of the best hardware which could be obtained.

6.1.1   Joule-Thomson (J-T) Cooling

The Joule-Thomson devices employ the cooling effect from the expan-
sion of a gas from high pressure to low pressure without heat transfer or work
being done by the gas. The process occurs at constant enthalpy and is called
throttling or Joule-Thomson expansion. Once the gas has been cooled (with or
without liquification of part of the gas), the useful heat load is transfered
to the gas from the detector.

The magnitude of the cooling effect is a function of the temperature
and pressure of the gas. The most effective use of the expendables occurs
by cooling the gas before expansion. With all gases the J-T cooling does not
occur until the gas is below the inversion temperature [40°K, Helium 204°K
Hydrogen and 250°K Neon; the other gases of interest have inversion temperatures
in excess of room temperature]. High pressure in general increases the
magnitude of the J-T effect. However, above certain pressures and at
temperatures less than the inversion point, the cooling effect is reduced
as pressure increases.

26

Figure 6-1 presents the schematic of a typical J-T cooler and property data for Nitrogen (N$_2$). The gas is normally stored at room temperature and at high pressure. A control valve is employed to prevent loss of expendables when the detector is not in use. A regenerative heat exchanger is employed to cool the gas prior to expansion. The expansion valve may be one of two types in common use: 1) a simple orifice which causes a fixed consumption rate regardless of the detector heat load and 2) a thermostaticly controlled valve. The latter provides a much lower consumption rate and is baselined for these studies. After being expanded the gas removes heat from the detector. The gas is then warmed in the regenerative heat exchanger and then vented overboard near room temperature conditions.

Property data for Nitrogen is included in Figure 6-1. The storage tank is initially at very high pressure (order of 3000 to 7500 psia, 200 to 500 atm, and 300°K). The cooling in the regenerative heat exchanger lowers the N$_2$ to 150° to 165°K at essentially the same pressure. The gas is then expanded to 1.0 ATM and 77°K, forming some liquid. The detector heat load evaporates this liquid. The gas is then warmed to near the initial room temperature. As the pressure in the tank is lowered, eventually the expansion process produces only N$_2$ gas at 77°K as indicated by the line marked "final" on Figure 6-1. No useful heat load can then be removed. Typically this is a pressure on the order of 1000 psia (70 ATM). The remaining N$_2$ in the tank is rather significant but unusable in the system. The average useful cooling effect was calculated as 2.8 watt-hr/pound of N$_2$, based on a 7500 psia initial pressure and all of the N$_2$ in the tank. Air and O$_2$ were also considered. Both have a higher cooling effect, but the difference is minimal and the gains are not normally worth the safety hazards.

The cooling effect is much less than that for liquid N$_2$ storage. If the N$_2$ could be sufficiently cooled prior to expansion, a factor of 9 reduction in expendables can be obtained. With only regenerative cooling available no significant increases can be obtained, (the heat exchanges is already 95% efficient). The addition of a second cooler can remove this limitation.

Figure 6-2 presents data on a modified Joule-Thomson cooler. A thermoelectric (T/E) cooler removes part of the heat from the high pressure

CONTROL VALVE

REGENERATIVE HEAT EXCHANGER

EXPANSION VALVE

Q, DETECTOR HEAT LOAD

EXHAUST

SCHEMATIC

ROOM TEMPERATURE HIGH PRESSURE GAS STORAGE TANK

NITROGEN TEMPERATURE-ENTROPY CHART

N$_2$ TANK CONDITIONS

INITIAL

FINAL

EXHAUST

REGENERATIVE COOLING

REGENERATIVE HEATING

JOULE-THOMSON EXPANSION

Q = USEFUL HEAT LOAD

ENTROPY, S — (Kg Cal /Mole)/°K

$$(\overline{\Delta h}) = \int (\Delta h)\ \frac{dm}{mi}$$

$$\overline{(\Delta h)} = 2.8\ \frac{\text{WATT-HR}}{\text{LB}}$$

@ 7500 PSIA INITIAL PRESSURE
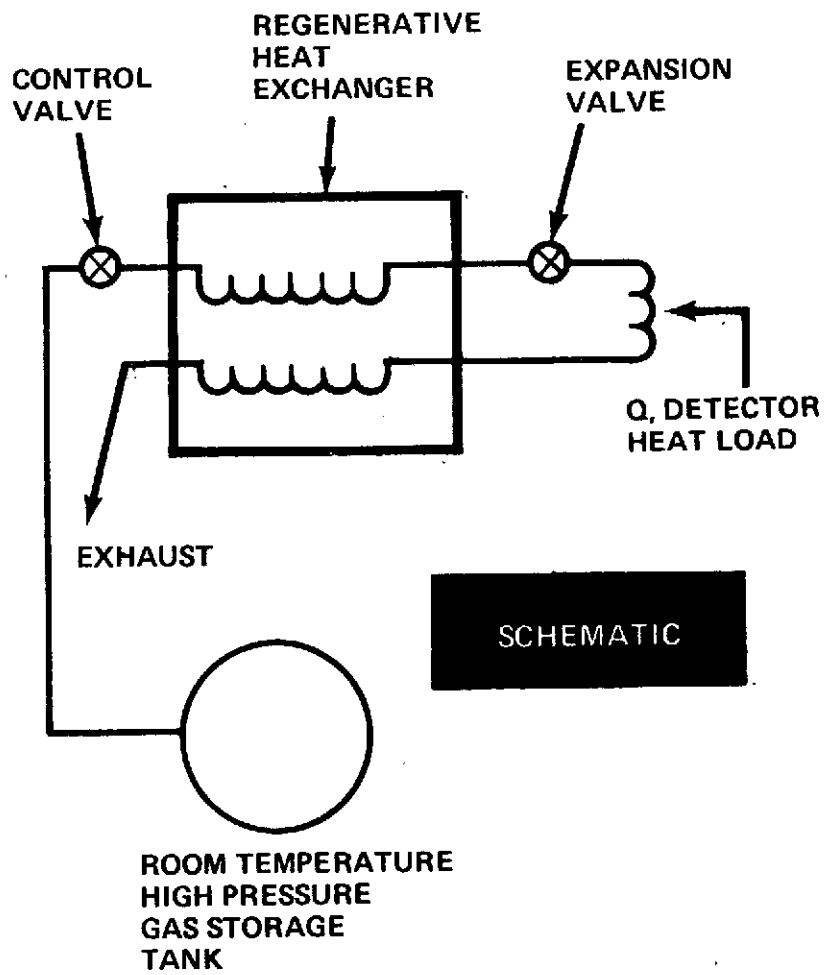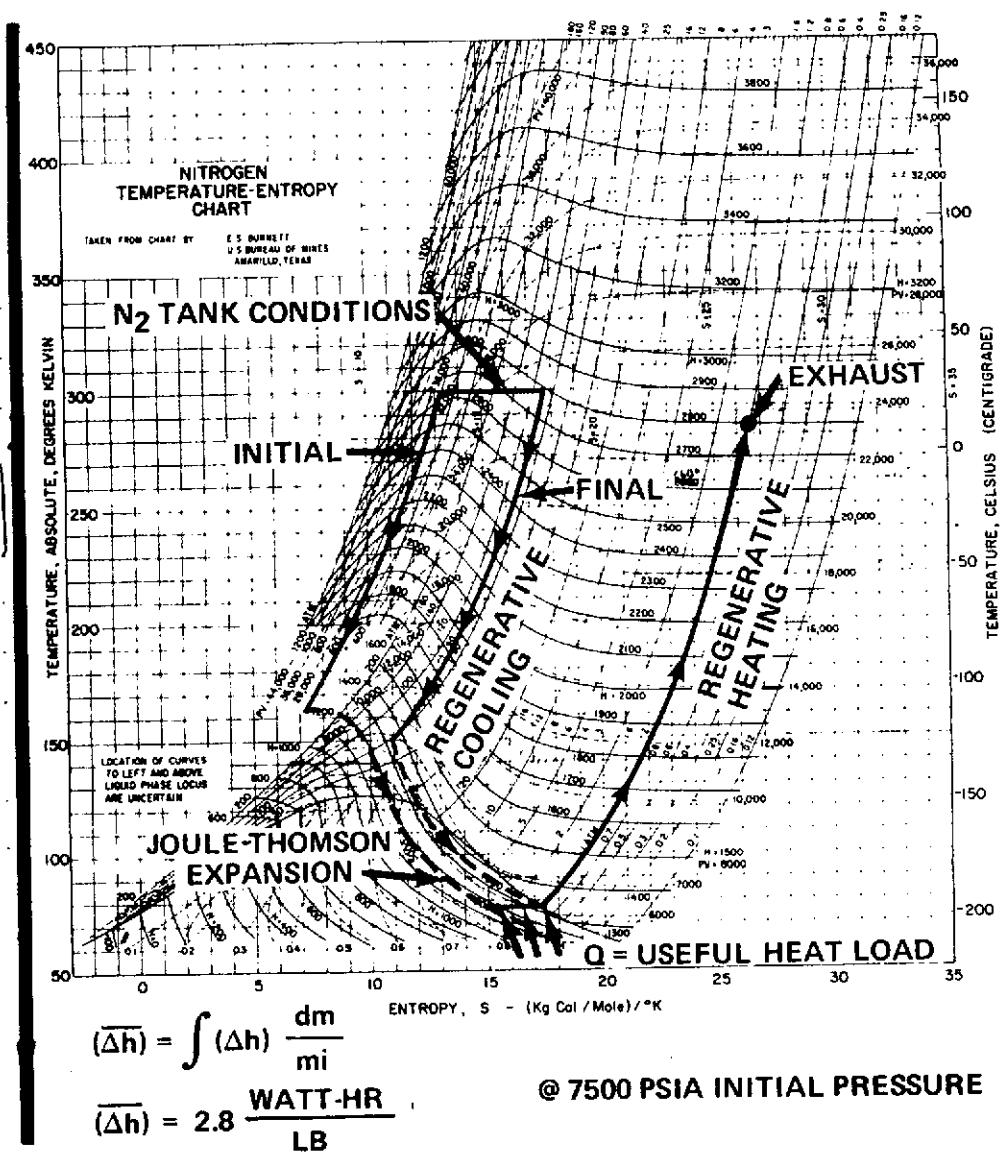
FIGURE 6-1

JOULE-THOMSON COOLING

28

29

**NITROGEN
TEMPERATURE-ENTROPY
CHART**

TAKEN FROM CHART BY   C S BURNETT
U S BUREAU OF MINES
AMARILLO, TEXAS

N₂ TANK CONDITIONS

INITIAL

1ST STAGE
REGENERATIVE
COOLING

FINAL

EXHAUST

HEAT
REMOVED
BY T/E
COOLER

1ST STAGE
REGENERATIVE
HEATING

2ND STAGE
REGENERATIVE
COOLING

LOCATION OF CURVES
TO LEFT AND ABOVE
LIQUID PHASE LOCUS
ARE UNCERTAIN

2ND STAGE
REGENERATIVE
HEATING

JOULE-THOMSON
EXPANSION

Q, USEFUL HEAT LOAD

ENTROPY, S - (Kg. Cal./Mole)/°K

TEMPERATURE, ABSOLUTE, DEGREES KELVIN

TEMPERATURE, CELSIUS (CENTIGRADE)

**COOLING CAPACITY OF
MODIFIED JOULE-THOMSON**

LIMITED BY
GROUND
SERVICING
EQUIPMENT

Δh, ENTHALPY CHANGE AT 77°K
PER UNIT MASS OF N₂
K CAL/KG-MOLE

7500

TANK PRESSURE: PSIA IN THOUSANDS

$$(\overline{\Delta h}) = \int (\Delta h) \ \frac{dm}{mi}$$

$$(\overline{\Delta h}) = 6.65 \ \frac{\text{WATT-HR}}{\text{LB}} \text{, BASED ON ALL N}_2 \text{ IN TANK}$$

@ 7500 PSI INITIAL TANK PRESSURE

FIGURE 6-2

MODIFIED JOULE-THOMSON COOLING
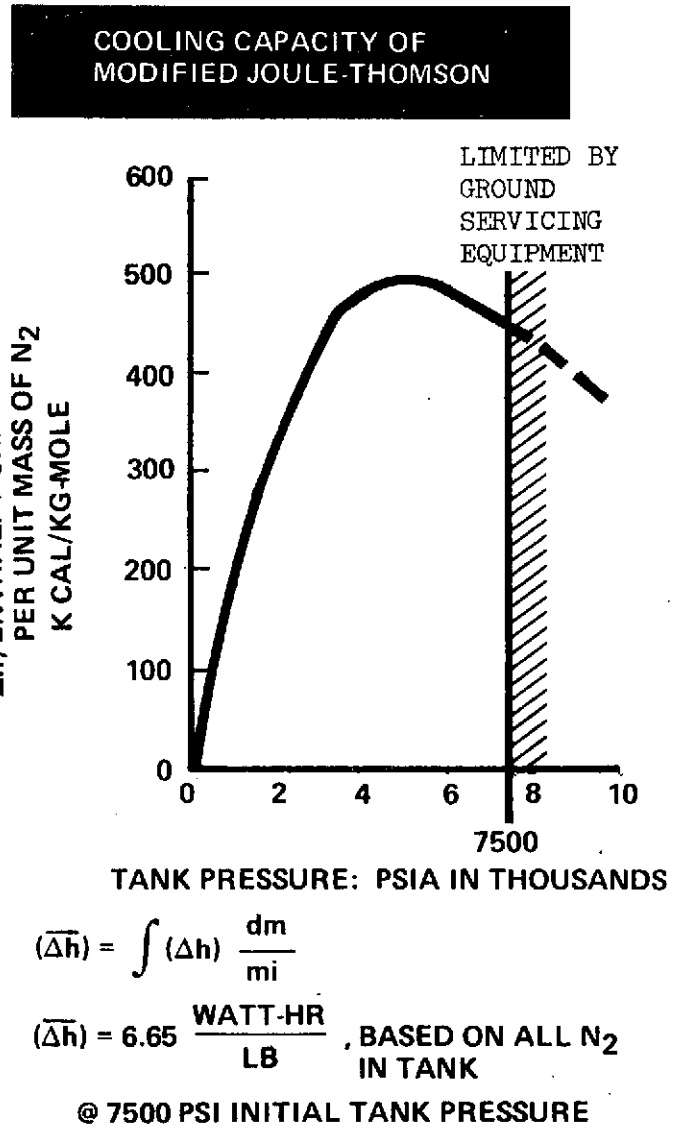
gas (the shaded area of Figure 6-2). The T/E cold junction is at 195°K, cooling the high pressure gas to 200°K with an integral heat exchanger on the cooler. Two regenerative heat exchangers are employed, one upstream of the T/E cooler (1st stage which reduces the requirements on T/E cooling), and one downstream of the T/E cooler (2nd stage). The remainder of the system is unchanged; and the low pressure gas is exhausted at the same temperature as in the simple J-T cooler.

The right hand side of Figure 6-2 presents the cooling effect per unit mass of N_2 as a function of the tank pressure. The maximum cooling effect occurs at 5000 psia (340 ATM). However, the maximum average cooling effect (based on all N_2 initially in the tank) occurs with an initial pressure of approximately 9000 psia (600 ATM) for both simple and modified J-T. However, the maximum pressure which can be provided to a satellite prior to launch will be limited to about 7500 psia (500 ATM) by ground servicing equipment.

The average cooling effect is 6.65 watt-hr/lb at 300°K, and 7500 psia initial tank conditions, based on all N_2 in the tank. This value is 2.4 times the 2.8 watt-hr/lb for the simple J-T cooler. This difference is due to the increased cooling effect per unit mass and useful range of pressure [cooling is obtained with pressures as low as 150 psia versus 1000 psia minimum with simple J-T].

The modified J-T was baselined for this study, since the savings over simple J-T are very significant. The modified J-T system has not yet been demonstrated with hardware. However, demonstration of the system can be accomplished within the time constraints and the technical risks are small due to the high development status of J-T and T/E hardware.

Figure 6-3 presents a conceptual design of modified J-T systems. The weight of each item and the diameter of the high pressure tank are included. For purposes of comparison the conventional J-T system weight and tank diameter are also included. With both systems the N_2 gas and storage tank are the most important penalty factors.

This conceptual design assumes a high technology N_2 storage tank material. There are several currently available; typical examples are:

1) High strength, aged and cryogenic form stainless steel tanks
2) Glass filament wound tanks
3) Carbon filament wound tanks
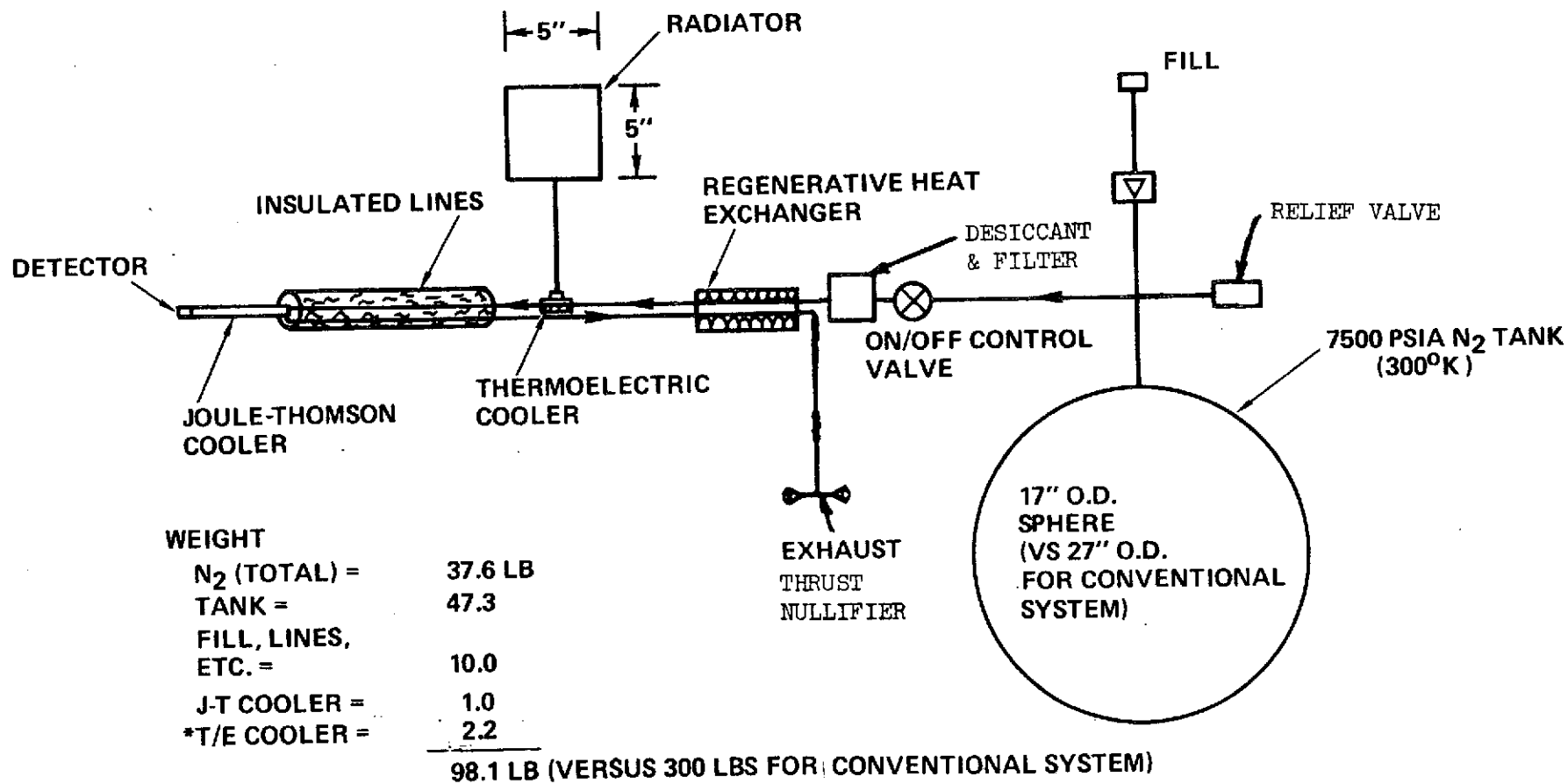
FIGURE 6-3

MODIFIED JOULE-THOMSON COOLER
77°K, 250 WATT-HOURS IN A 2 YEAR PERIOD

All of the above incur a penalty of 2 to 2-1/2 (lbs of N_2 + tank) per (lb of N_2). Other tank materials are available with a lower cost but a higher weight penalty. Typical materials are:

1) Titanium Alloys
   (e.g.; Ti - 6 AL - 4V)

2) Steels
   (e.g., SAE 4340)

The weight penalty factors are 3 to 4 (lbs of N_2 + tank) per (lb of N_2). The volume is not significantly affected by the choice of tank material.

The fill provisions include a check valve, relief valve and lines (typically 1/8" dia. tubing). An on/off control valve is employed to save gas when cooling is not required. A desiccant bed and filter are included in the system to provide a very low dew point and to prevent the freezing out of ice in the cryostat. The exhaust N_2 is expelled overboard through a thrust nullifier to prevent disturbing the orbit and attitude of the satellite. The condition of the exhaust are near one atmosphere pressure (14.7 psia) and at room temperature. All of these items have an estimated weight of about 10 lbs.

The T/E cooler is the same unit as Category IV cooler described in Section 6.3. The unit in the modified J-T incorporates a high pressure heat exchanger on the cold junction and is slightly heavier than the unit shown in Figure 6-26.

The Joule-Thomson cooler is a standard unit with integral 2nd stage regenerative heat exchanger. The 1st stage regenerative heat exchanger is upstream of the T/C cooler; the construction of both regenerative heat exchangers is similar. J-T coolers are typically on the order of 0.2 to 0.5 inches in diameter and 1 to 2 inches long exclusive of any insulation. With a dewar, the dimensions increase to 0.5 inches to 1 inch diameter and 1-1/2 to 2-1/2 inches long. Mounting provisions and tubing connections are in addition to these dimensions. J-T cooler (with 2nd stage) and the 1st stage regenerative heat exchanger are similar in size and mounting. Both units will weigh about 1.0 lb

Sizing data for both the conventional J-T and modified J-T systems are presented in Figures 6-4 and 6-5 respectively. The penalties are basically a function only of the integrated heat load. The life of the system is not normally a factor, for even greater than two years of use. Cool down times are typically 1 to 5 minutes and represent only a small portion of one day (1440 minutes) of operation. However, when calculating the integrated heat load, the mass, specific heat, and temperature change of the detector and all

INTEGRATED HEAT LOAD : WATT-HOURS

FIGURE 6-4

JOULE-THOMSON SCALING DATA

CONDITIONS: 1) 7500 PSIA N$_2$ TANK @ 300°K
                 2) INDEPENDENT OF LIFE
                 3) 50 MW HEAT LOAD

INTEGRATED HEAT LOAD : WATT-HOURS
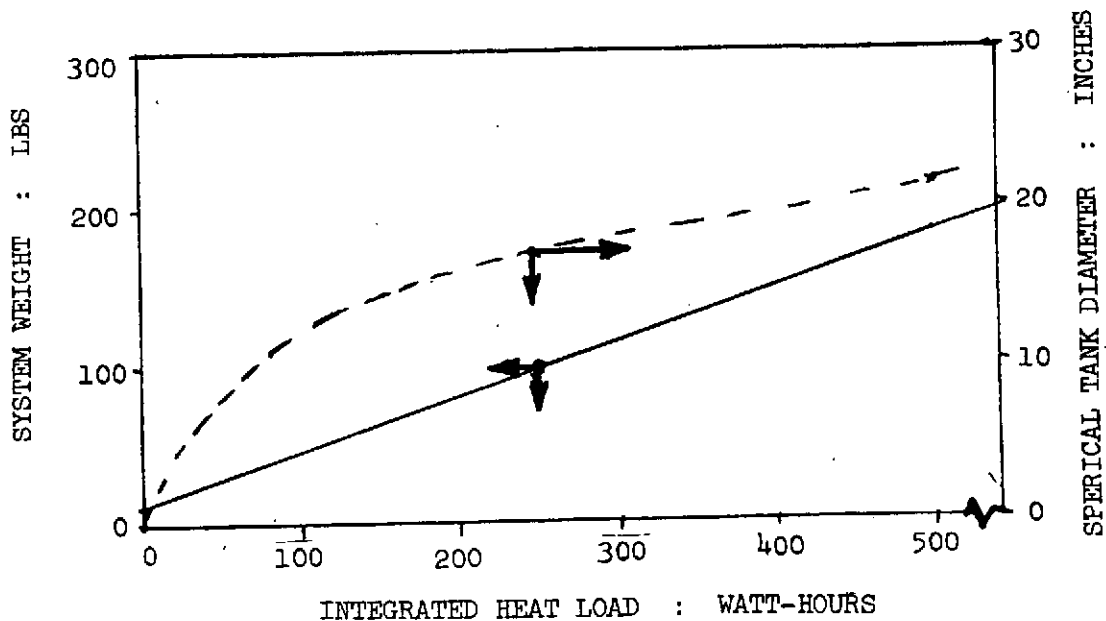
FIGURE 6-5

MODIFIED JOULE-THOMSON SCALING DATA

equipment cooled by the J-T unit must be added to the detector heat load for every cool down cycle.

The data on Figures 6-4 and 6-5 have been calculated for a 50 mw heat load. The conventional J-T will be slightly affected by heat loads up to 1 to 2 watts but not greatly. Modified J-T are more sensitive to heat load due to the added T/E cooler requirements; that factor may be determined by lineraly scaling the T/E cooler with heat loads and adding the increase to the weight data.

The weight data for both the conventional and modified J-T includes the supporting hardware indicated on Figure 6-3. None of these items change for increasing the expendables and only the tank is increased. The tank diameter is included in Figure 6-5 for spherical tanks. Due to the rapid increase in weight and volume with increased cooling requirements, the J-T systems are normally best suited to the very small heat loads with short durations. The next section describes a lighter weight expendable system for larger capacities.

6.1.2   Solid Cryogen Cooling

Solid cryogen cooling employs the heat of sublimation of a solid to achieve a cooling effect. The basic features of the device are shown on Figure 6-6. The detector heat load is conducted into the cryogen by a heat transfer rod. From this fins extend out into the volume and transfer heat to the colder cryogen. The cryogen sublimates forming a gas which then leaves via the exhaust port. Temperature is controlled to the desired level by adjusting the vapor pressure. The mass is insulated from the environment to minimize the loss of cryogen to parasitic heat leak.

Solid cryogens have important advantages over J-T and liquid cryogens. Higher cooling effect per unit mass and per unit volume of expendable can be obtained as illustrated below:

VAPOR PRESSURE
CONTROL

SUPER
INSULATION

HEAT TRANSFER
ROD

DETECTOR

GAS SPACE
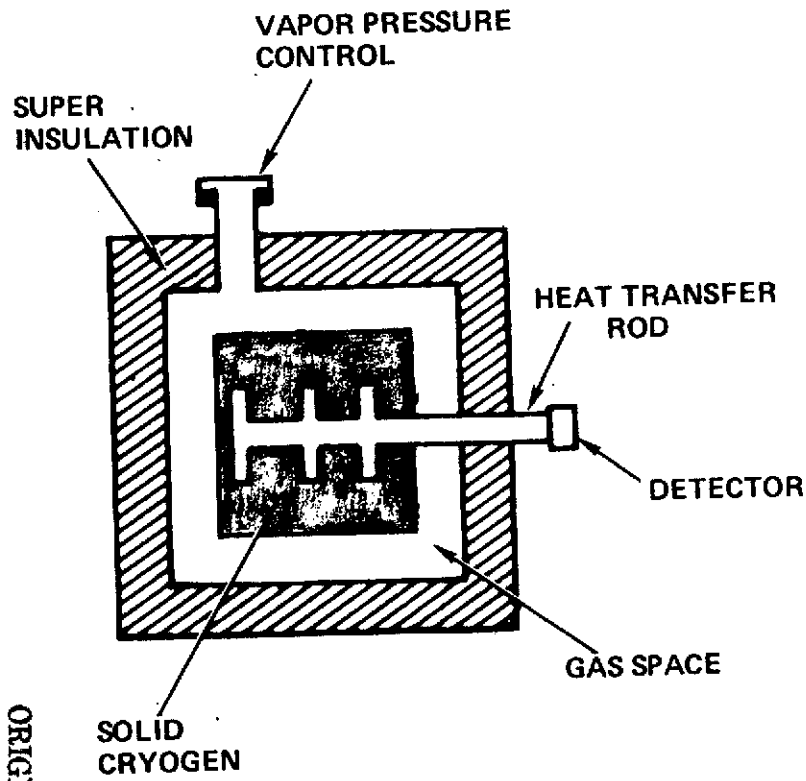
SOLID
CRYOGEN

FIGURE 6-6
BASIC FEATURES

PRESSURE –
PSIA

LIQUID

LIQUID/
VAPOR

VAPOR

SOLID

HEAT OF
SUBLIMA-
TION

77°K

SOLID/
VAPOR

ENTHALPY – BTU/LB
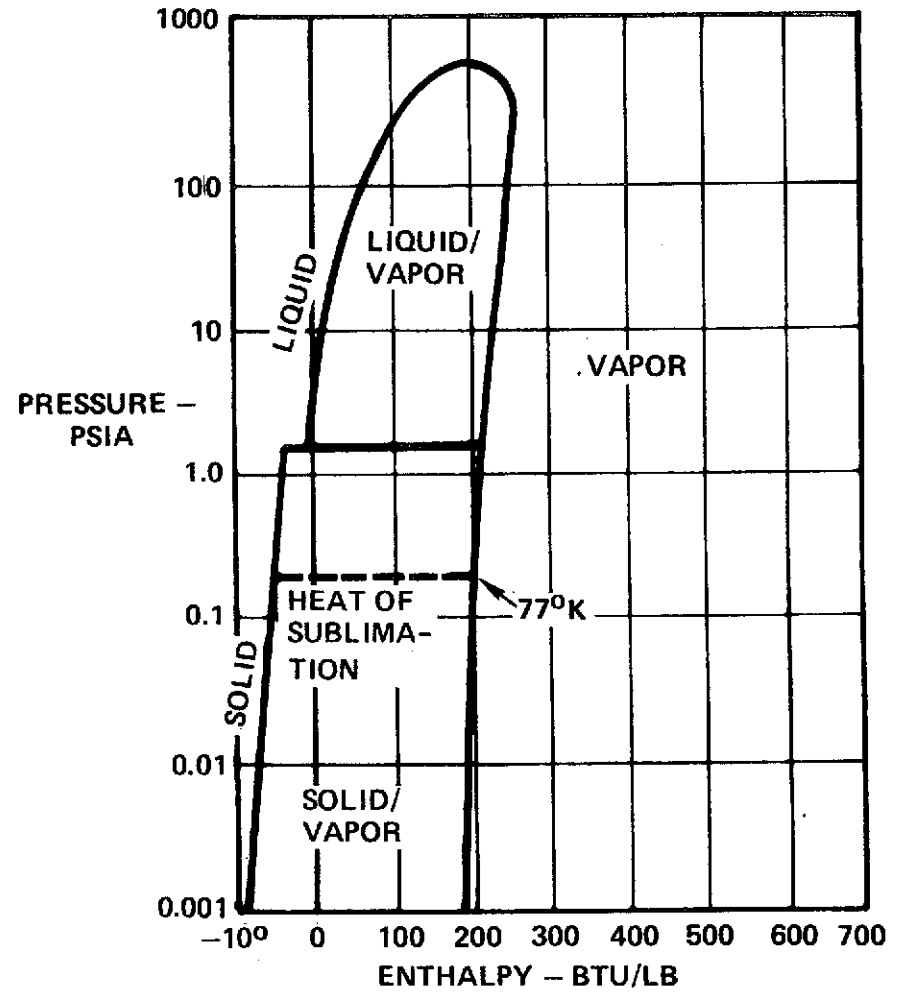
FIGURE 6-7
PROPERTIES OF CH_4 (METHANE)

SOLID CRYOGENIC COOLERS

	COOLING EFFECT @ 77°K	
	---	---
CRYOGEN	Δh * Watt-Hr/Lb	$\rho \cdot \Delta h$ * Watt-Hr/ft^3
Solid Methane	77.3	2505
Solid Argon	25.8	2737
Liquid Nitrogen	25.1	1265
Modified J-T (7500 psia)	6.65	148
J-T (7500 psia)	2.8	62

*Exclusive of storage tank, insulation, and other system components

The solid cryogens (both methane and Argon) have a much larger cooling effect than either the J-T or liquid coolers. Figure 6-7 illustrates the reason. The heat of sublimation includes the heat of vaporization and the heat of fusion. In addition the density of solids is normally higher than the liquid. Thus for a given useful heat load the quantity of solid will weigh much less and take less volume. In addition the liquid systems have zero "g" phase separation and pressure control problems which are not applicable to solids. [The solids stay in place at all times.] Solid cryogens do have a continuous parasitic heat leak from the environment into the storage container; this effect causes a continuous loss of cryogen which does not remove any of the detector heat low. At very small heat loads the parasitic heat leak causes the largest loss of the expendables. Thus the effective cooling effect is greatly reduced and J-T can become both lighter weight and smaller.

Single stage and two stage systems have been developed for 77°K coolers. Single stage coolers employ only one expendable cryogen; two stage coolers employ one cryogen at 77°K and a second cryogen at a higher temperature (100 to 200°K) to remove part of the parasitic heat leak. The latter approach is generally better because more effective cryogens are available. Two stage coolers are baselined for this study.

Figure 6-8 presents a comparison of expendable cryogens. The coolers are compared for a typical application, 55 mw continuous detector heat load at 77°K and a one year duration. Based on data for an Argon/CO$_2$ cooler, Ref. 6, the heat leak at 77°K was estimated to be equal to the detector heat load and at the upper stage to be a little more than twice the detector heat load. The total heat loads on the cryogens are thus 110 mw at 77°K and 120 mw net at the

ONE YEAR DURATION

55 MW DETECTOR HEAT LOAD

Q_{LEAK} ($77^{\circ}K$) = 55 MW

Q_{LEAK} (UPPER STAGE) = 120 MW

Q_{LEAK}

$77^{\circ}K$

DIA

DIA

UPPER STAGE

INSULATION

$Q_{DETECTOR}$ (55 MW)

SHIELD
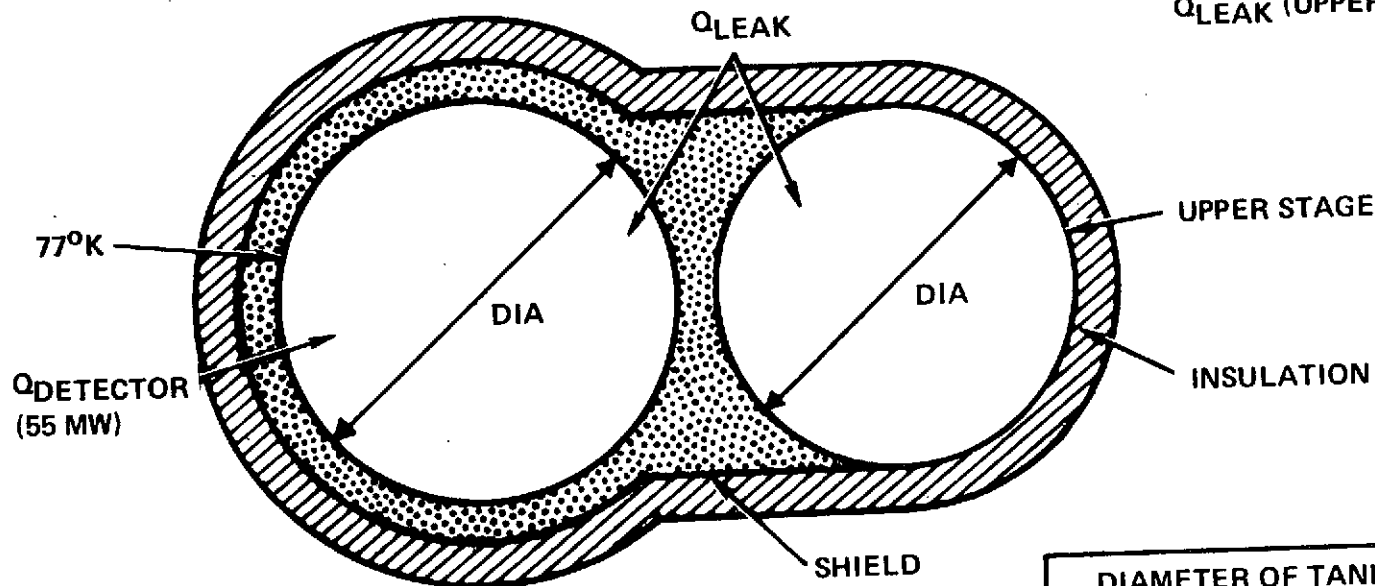
CONCEPT	WEIGHT LBS	VOLUME, FT^3	DIAMETER OF TANK AS SPHERES	
			$77^{\circ}K$	UPPER
			1.95 FT	
H_2*	20 TO 30	3.9	1.40	0.88
$CH_4 + NH_3$	43	1.8	1.33	0.96
$O_2 + CO_2$	75	1.7	1.37	0.88
ARGON + NH_3	67	1.7	1.37	0.96
ARGON + CO_2	85	1.8		

*AT $10^{\circ}K$ CPΔT OF VAPOR USED FOR HEAT LOAD

FIGURE 6-8

COMPARISON OF EXPENDABLE CRYOGENS

37

upper stage.

The 77°K cryogen is surrounded by an insulation blanket. A thermal shield is outside this insulation blanket and another insulation blanket is outside the shield. The shield is maintained at the same temperature as the upper stage by a conductive path to the upper stage storage tank. Part of the heat leak from the environment is thus conducted in the upper stage. For the assumed conditions 175 mw of heat leak reach the shield, of which 120 mw are transfered to the upper stage and 55 mw are leaked to the colder, 77°K cryogen. The magnitude of these heat leaks will be affected by the temperature of the upper stage cryogen; the upper stage heat leak will be smaller for higher temperature cryogens. However the lower stage heat leak will be larger for higher temperature upper stage cryogens. In the sum little difference in total weight and volume is expected for differences in upper stage cryogens.

The lightest weight system is solid hydrogen. That concept employs the heat of sublimation to remove the heat leak. The heat capacity of the vapor from 10°K to 77°K is employed for the detector heat load. The upper stage cooler is provided by returning the vapor to cool the shield with the heat capacity from 77°K to 150°K. Although the solid H_2 system is the lightest system, it is also the largest, by more than a factor of 2 over alternative approaches. The concept has not been developed and a large amount of work remains to be done. One should recognize the potential of this concept; some applications require cooling for more than one detector, sometimes at different temperatures. Other applications are very weight critical, for both conditions solid H_2 is very attractive. However, due to the volume penalty and the need for additional development, solid H_2 was eliminated from this study.

Methane/Ammonia was selected for the conceptual design. This concept has the lowest weight (other than H_2) and there is very little difference in volume in the other concepts. There are safety hazards with these cryogens but the problems can be solved.

Figure 6-9 presents a conceptual design of a solid cryogen, CH_4/NH_3 cooler. The sizes for both an intermittent and a continuous heat load are included on Figure 6-9; intermittent on the left and continuous on the right. For the conditions specified for Category I, either the intermittent or continuous size may be applicable, depending upon the nature of the detector heat load. If the electrical dissipation is the main factor, then the power dissipation can be terminated and the intermittent heat load design is applicable. If most of the detector heat load is due to conduction and radiation
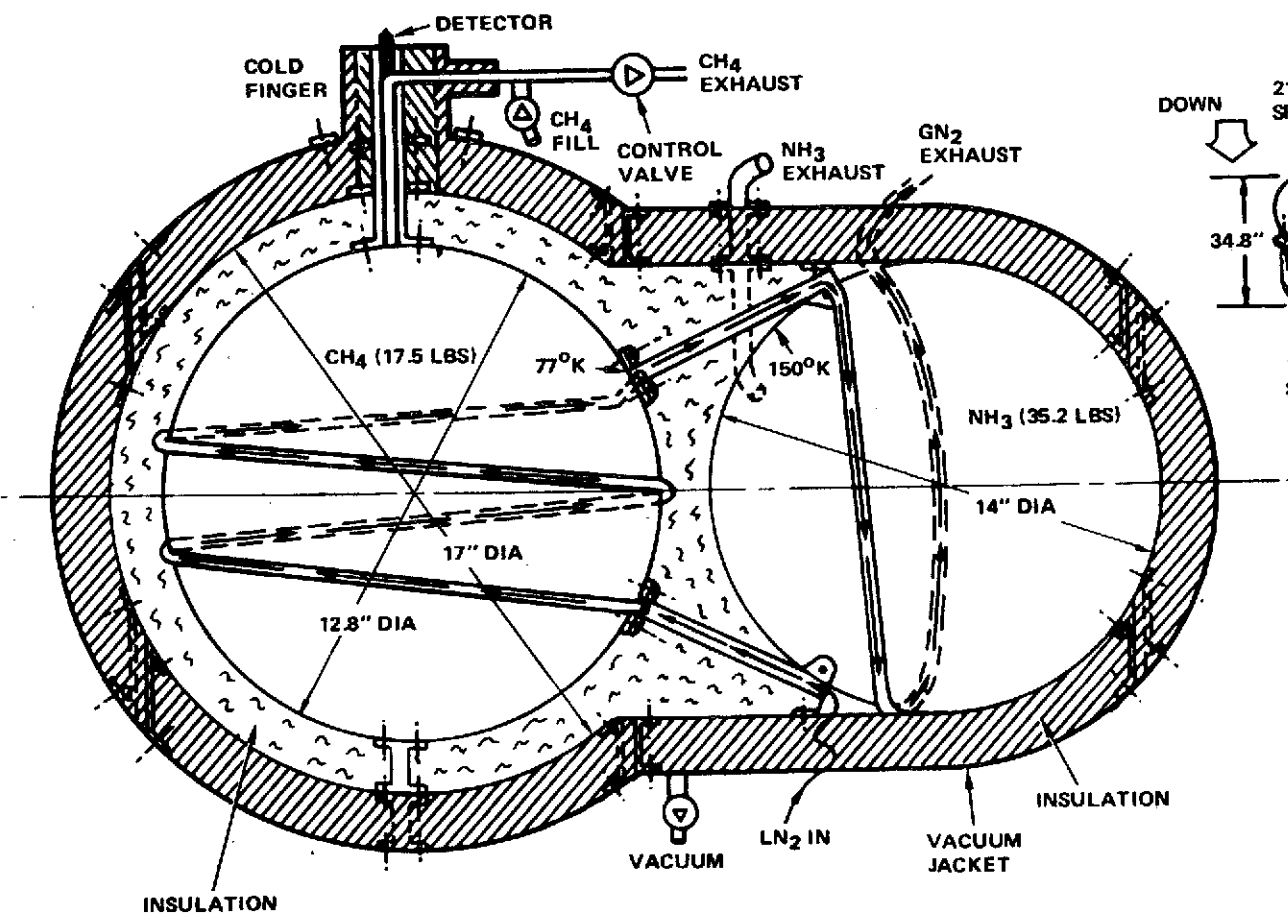
CH4/NH3 SOLID CRYOGEN COOLER
INTERMITTENT 50 MW HEAT LOAD
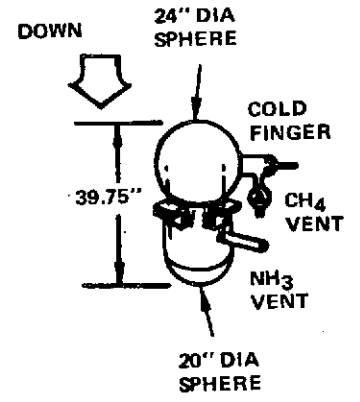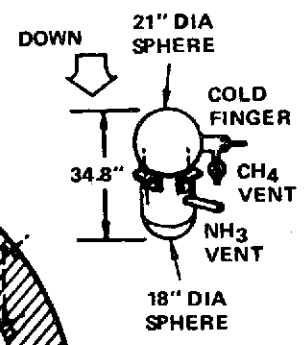
CH4/NH3 SOLID CRYOGEN
CONTINUOUS
50 MW HEAT LOAD

FIGURE 6-9

SOLID CRYOGEN CONCEPTUAL DESIGN

heat loads from the environment, the heat load continues and the large
unit must be used.

The design of both the continuous and intermittent are shown on the left
side of Figure 6-9 and are identical except for size. The detector is mounted to
a cold finger, either directly or by a flexible connection. Heat is conducted
from the detector down a solid copper or aluminum rod into the CH_4 tank at
77°K. The cold finger is attached to the metal walls of the tank which
distributes the heat. Attached to the walls are fins in the form of a very
low density (2-3 lb/ft^3) aluminum foam. This foam extends throughout the
interior of the tank to distribute heat to all points and to provide a large
surface area. The heat sublimes the methane and the gas leaves the volume
through a hollow tube. The pressure is controlled by an active valve in the
tube and hence the temperature [Aerojet has developed such a valve]. An alter-
native approach to temperature control is a long length of small diameter
tubing. That approach is good for continuous heat loads, but for intermittent
duty temperature tolerances may be exceeded (a 3.5°K shift in temperature level
may result due to intermittent duty). The same tube is employed to fill the
tank with CH_4 and a check valve is employed to isolate the fill lines.

The tank is filled by supplying room temperature gas to the fill line.
Liquid nitrogen is also supplied and passes through the insulation to a heat
exchanger on the outside of the CH_4 tank. The tank is cooled to 77°K by the
LN_2, and freezes the methane out on the foam. The LN_2 then passes over the
ammonia tank where NH_3 is also frozen out. The now gaseous N_2 again passes
through the insulation and is vented.

The NH_3 tank construction is similar to the CH_4 tank. No cold finger
is required since there is no detector heat load. Only the heat which leaks
through the outermost insulation blanket is transferred to the NH_3 from both
the area around the NH_3 tank and the shield surrounding the CH_4 tank. The
exhaust line has no control valve, since the NH_3 temperature is not critical.

The insulation between the exterior wall and the NH_3 tank + shield
and that between the shield and the CH_4 tank are multilayer insulation
(50 layers/inch). The heat leaks per unit area were assumed to be the same
as the measured values of the Lockheed Solid Argon - CO_2 cooler, described
in Reference 1. That design requires very complex techniques in the construc-
tion and also includes a high risk that the objectives may not be reached.
As an example the Lockheed system was designed for a total heat leak to the

CO_2 of 76 mW (0.26 BTU/hr and 40 mW (0.135 BTU/hr) to the argon including the 25 Mw IR detector load. The initial tests of the unit resulted in heat loads approximately three times the predicted values. After considerable redesign, improved insulation and reconstruction, the final thermal tests were considerably improved but still short of the design goals. The final measured heat leak was 74 mW (0.25 BTU/hr) to the CO_2 and 28.6 mW (0.098 BTU/hr) to the argon not including the IR detector load. In order to permit the one-year operation, the IR detector load had to be reduced to 17.6 mW. The CH_4/NH_3 system in Figure 6-9 is based on the redesigned measured heat leaks.

The shape chosen for both tanks was spherical. This choice reduces the parasitic heat leak by 13% from that of the best cyclinder (length/diameter ratio = 1). The resultant savings in the system weight and volume is approximately 20%. Construction of a spherical shape is more difficult, but the savings are probably worth the extra cost, although cost will already be high.

Table 6-1 presents a comparison of the weight of intermittent and continuous duty solid cryogen coolers. The intermittent duty cycle can provide 250 watt-hours of useful detector cooling at 77°K in a 2 year period or about a 30% duty cycle. The continuous duty cooler can provide 877 watt-hours at 77°K in 2 years. The difference in weight is mainly due to two effects. One is the larger heat load; the other is more parasitic heat leak due to the increased area of the larger tank. For the latter reason the quantity of NH_3 must also increase. The structure weight increases due to the larger sizes. The total weight increase for continuous duty is only 40% for more than a factor of three change in useful heat load. This effect illustrate the importance of the parasitic heat leak. Most of the CH_4 in the intermittent design is employed to remove that heat (14.3 lbs out of the total of 17.5 lbs, only 3.2 lbs goes to useful heat load).

Figures 6-10 and 6-11 present the effect of duration and equivalent heat load. These data are based on the conceptual design previously presented. The effect of duration is even more pronounced than heat load. For example a 50 mw continuous heat load requires 42 lbs for one year but 140 lbs for 2 years; much more than a factor of 2 increase in weight.

Figure 6-11 presents the effect of equivalent heat load on cooler weight. Equivalent heat load is defined as the integral of the intermittent heat load divided by the total duration. This integral includes the heat capacity of the detector and all equipment which may have changed temperature (mcp ΔT) and the number of cycles. If those factors are small, the intermittent

TABLE 6-10

CATEGORY I

SOLID CRYOGEN WEIGHT ANALYSIS

INTERMITTENT DUTY (30% ON; 50 MW)		CONTINUOUS DUTY* (50 MW)	
• CH_4 (CRYOGEN ONLY) =	17.5 LB	CH_4 (CRYOGEN ONLY) =	35.0 LB
• NH_3 (CRYOGEN ONLY) =	35.2 LB	NH_3 (CRYOGEN ONLY) =	46.5 LB
• STRUCTURAL (INCLUDING TANKS) =	43.0 LB	STRUCTURAL (INCLUDING TANKS) =	56.0 LB
TOTAL	100 LBS	TOTAL	140 LBS

*IF ALL OR MOST OF THE DETECTOR POWER DISSIPATION AND HEAT LEAK CAN NOT BE STOPPED.

NOTES: 1) 77°K Heat load
          2) NH$_3$/CH$_4$ 2 Stage Device
          3) 300°K Environment

EFFECT OF LIFE ON SOLID
CRYOGEN SIZE

FIGURE 6-10

EFFECT OF HEAT LOAD ON SOLID
CRYOGEN WEIGHT
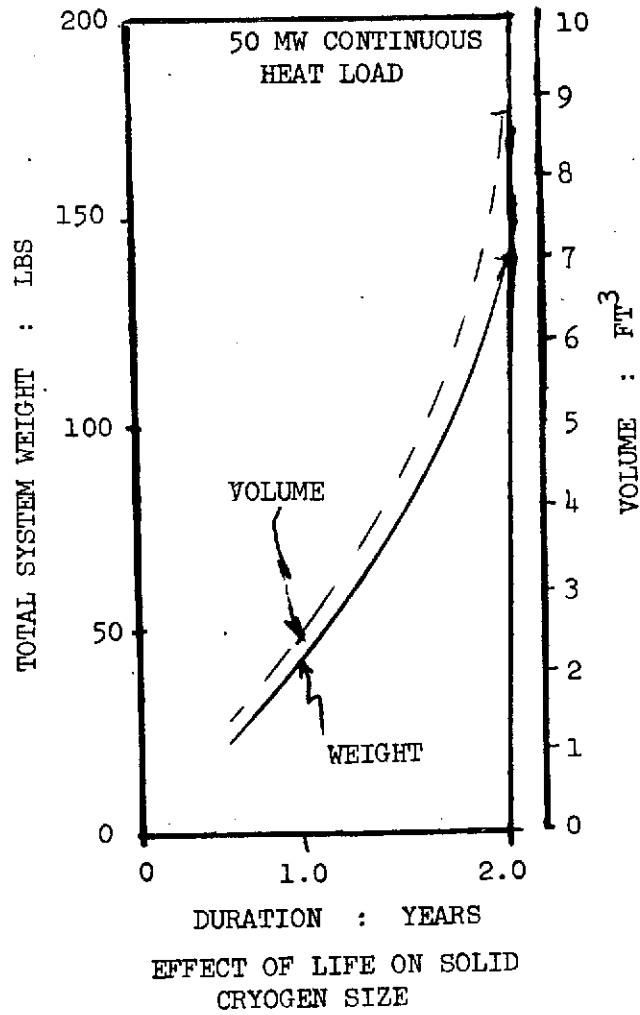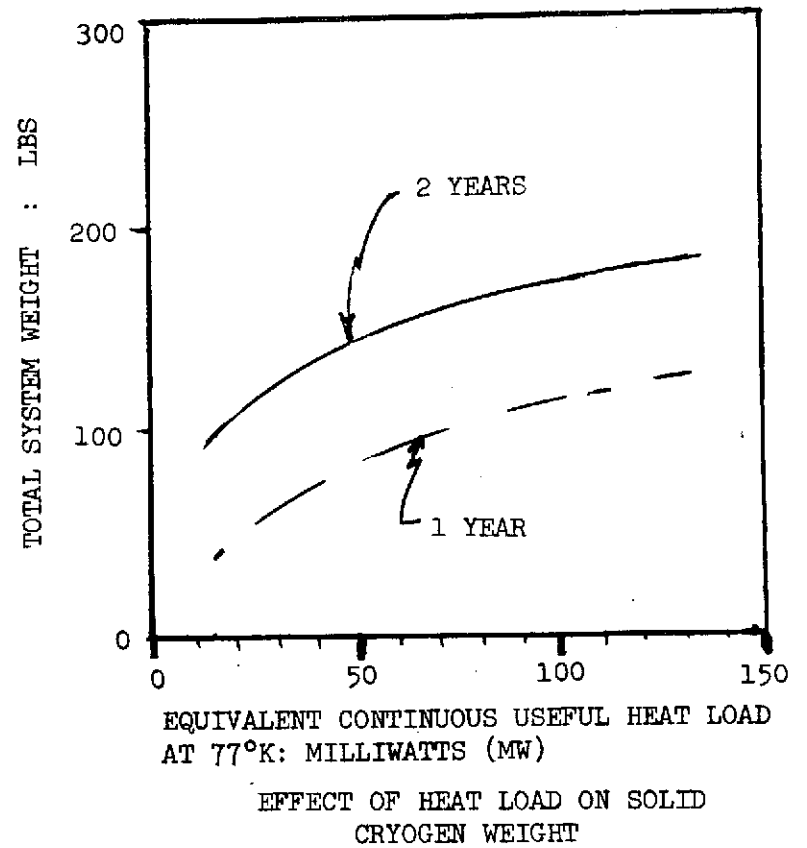
FIGURE 6-11

duty of Category I,(250 watt-hours total, at 50 mw) has an equivalent
heat load of only 14 mw. This unit would be the same size and weight as
one designed for continuous duty at 14 mw. The lack of dependence
upon peak heat load is true at all heat loads with solid cryogens. The
large surface areas for heat transfer result in little dependence upon
heat transfer rates causing these units to be capacity limited rather than
rate limited.

6.1.3   Comparison of J-T and Solid Cryogen Coolers

Joule-Thomson and solid cryogen coolers employ expendables. As
such both coolers are best suited to small heat loads and modest to short
durations. The rate of expendable consumption is highest with J-T and hence
only the very small integrated heat loads are applicable (less than 100 watt-
hours). The solid cryogens are best suited to modest or high heat loads but to
durations on the order of one year. As shown earlier in Section 5.2, Vuillemeir
coolers become most attractive for 2 years or more of life and high detector
heat loads.

Table 6-2 presents a comparison of the features of the J-T and solid
cryogens. J-T has the advantage of being an off-the-shelf system. Flight
hardware can be obtained with minimal cost and delivery delays. The convention-
al J-T system is quite heavy and large due to the requirements for expendable.
The modified J-T can greatly reduce the penalties (by a factor of 3 or more).
Additional development work is needed but sufficient development time is available
before a PMS flight. The system is a straight forward application to two
developed systems and minimal problems are expected. The J-T cooler expendable
consumption can readily be stopped when cooling is not required. The life time
of system components is very high even without maintenance. Thus the life is
virtually independent of the duration of the mission, and the J-T coolers are
sized almost totally by the integrated heat load.

Many cryogens are available for a 77°K application; due to high cooling
capacity per unit weight and volume. NH_3/CH_4 are considered best. If weight
is extremely important and volume less critical, a H_2 system may be preferred.
All solid cryogens require complex, high cost designs to minimize the effect
of parasitic heat leak. This parameteric is very strong in determining system
size; and prototype units have frequently underestimated it by factors of 3.
Even with redesign heat leaks have been a factor of 2 larger than predicted,
resulting in reduced detector heat load or system life or both. Although

TABLE 6-2

COMPARISON OF J-T AND SOLID CRYOGEN COOLERS

JOULE THOMSON	SOLID CRYOGENS
• CONVENTIONAL J-T 　– LOW COST & SIMPLE 　– LARGE WEIGHT & VOLUME PENALTY	• NH_3/CH_4 ARE BEST CRYOGENS • COMPLEX DESIGN WITH HIGH COST • HIGHLY RELIABLE (NO MOVING PARTS • LOW WEIGHT BUT HIGH VOLUME
• MODIFIED J-T 　– REQUIRES ANOTHER COOLER 　– A MINIMINAL AMOUNT OF 　　DEVELOPMENT 　– LOW WEIGHT AND VOLUME PENALTY	• EXPENDABLE REQUIREMENTS 　– STRONGLY AFFECTED BY THE TOTAL 　　DURATION 　– LESS STRONGLY AFFECTED BY THE 　　INTEGRATED HEAT LOAD (WATT-HOURS)
• EXPENDABLE REQUIREMENTS 　– DETERMINED ONLY BY THE INTEGRATED 　　HEAT LOAD (WATT-HOURS) 　– INDEPENDENT OF STORAGE DURATION	• MOST APPLICABLE TO DURATIONS OF A 　YEAR OR LESS
• MOST APPLICABLE TO SMALL HEAT LOADS	

there is a risk of premature exhaustion of expendables, the units have been reliable. No moving parts are employed resulting in a minimal number of failures. The expendable is stored in a high density solid; however, because the cryogen must be well insulated, the volume becomes quite large. For example the modified J-T cooler has a weight approximately equal to the solid cryogen with the same heat load, however the modified J-T requires only a 17" diameter sphere versus a 21" diameter sphere + a 18" dia x 14" cylinder for the solid cryogen. The expendable requirements for solid cryogens are strongly affected by duration in the range of 1 to 2 years; the effect of capacity is significant but not as strong. Due to the strong effect of heat leak, the solid cryogens are best suited for durations of a year or less. The alternative for long durations, the VM, is described in the next section.

6.2   Categories II and III

The Vuilleumier cycle cooler was selected for conceptual design for categories II and III (77°K cooling load and detector cooling loads of 300 and 1000 milliwatts) as discussed in Section 5.0. From the survey results described in Section 4.0 the AiResearch Fractional Watt Cooler[3,7]* (Figure 6-12) was chosen as the basis for conceptual designs. Reasons for this choice were:

a)   The Fractional Watt Unit is designed for long life operation in the space environment. Design life is 2 years continuous operation with a 5 year design goal.

b)   The design performance of the Fractional Watt Unit is in the range of that required for categories II and III. The design point was 250 milliwatts cooling load at 65°K.

c)   A significant amount of scaling information is available from reference 8.

A description of the basic operation of the VM cooler which is taken directly from Reference 9 is given in Section 6.2.1. This is followed by a discussion on scaling of the VM cooler in Section 6.2.2 and conceptual designs in Section 6.2.3 thru 6.2.5.

6.2.1   Basic VM Operation[9]

The VM cooler (Figure 6-13) is essentially a heat engine driving a refrigerator. Thus the input energy to the device is supplied as heat, which

_____

* Superscripts indicate References in Section 10.0

46
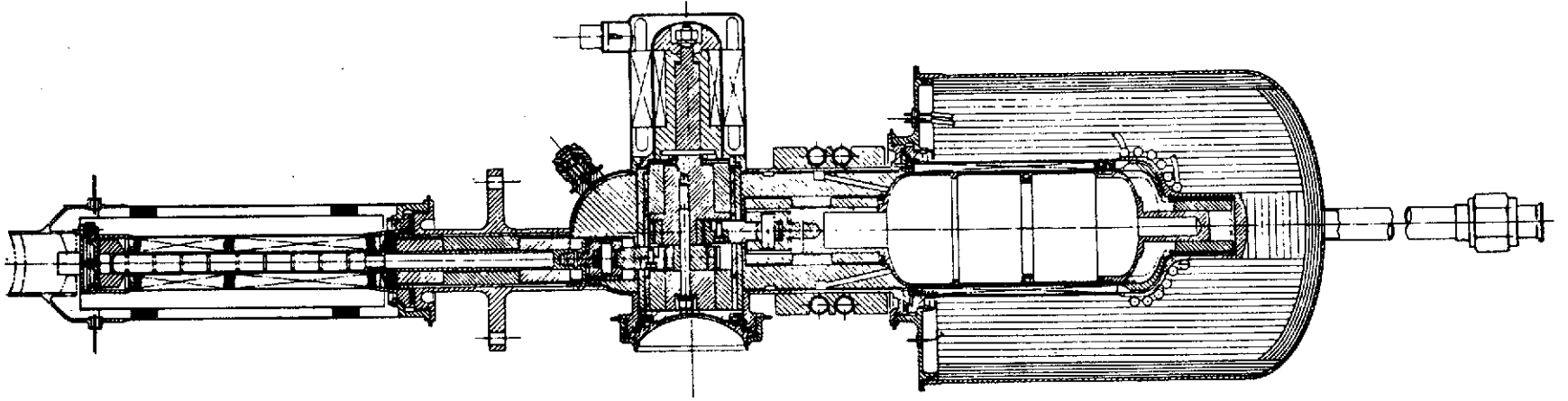
FIGURE 6-12

AIRESEARCH FRACTIONAL WATT VM COOLER

COOLING LOAD

COLD SPACE

DISPLACER -
REGENERATORS

HOT
SPACE

HEAT
SOURCE

HEAT
REJECTION

(a)

CRANKSHAFT DRIVEN BY
ELECTRIC MOTOR
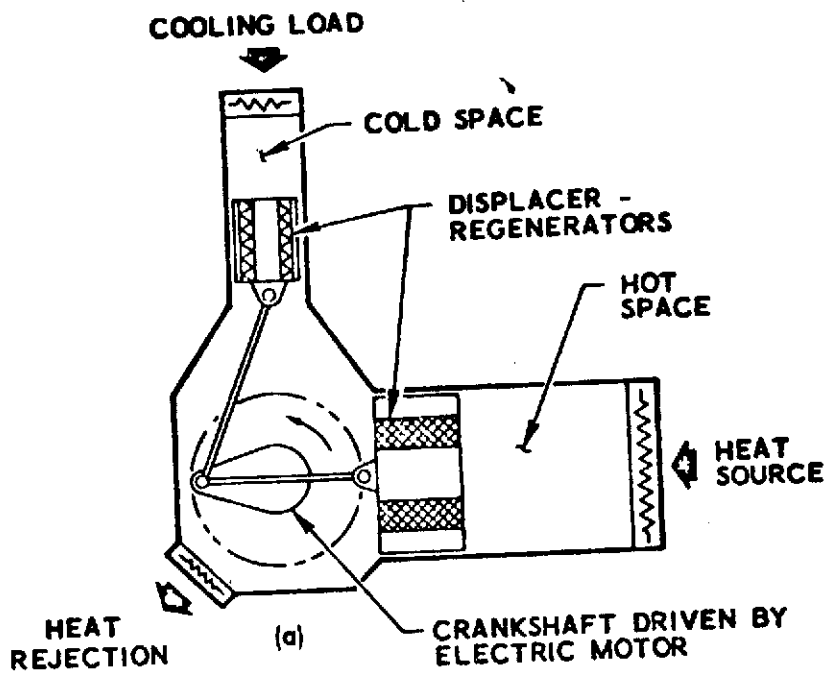
FIGURE 6-13

VM REFRIGERATOR

minimizes the required drive-motor size, and allows the direct use of a long lifetime heat source such as a radioactive isotope. The displacers move the gas from one cylinder of the refrigerator to the other, but do not directly compress or expand the gas.

The steady state operation of the VM refrigeration cycle will now be explained. Reference will be made to Figure 6-14 (a schematic of four crank positions encountered during operation and the pressure-volume diagrams for the cold cylinder, hot cylinder, and total gas volmme for the VM cycle). In this explanation we assume that the pressure drops across the regenerators is zero, and thus for the design depicted the pressures in the three sections are always equal. (In the actual case, the regenerator pressure drops are comparatively small.) Furthermore, we note that the shapes of the pressure-volume diagrams in Figure 6-14b are close to those resulting from the crank type of design shown in Figure 6-13. Finally, it should be noted that the following is a simplified view of the operation of the VM engine, in that only the predominant processes for a given crank position are pointed out. The actual operation of the refrigerator is much more complicated.

We begin with the crank in the South position (Figure 6-14(a). At this point the cold displacer is at its maximum displacement position, and the hot displacer is only at its half-maximum position. The mean gas temperature in the VM refrigerator is relatively low, and consequently the gas pressure is low. This is shown in the hot cylinder and cold cylinder P-V diagrams in Figure 6-14(b). Since this low pressure is the result of the West-South expansion process (as will be shown), heat is absorbed from the refrigeration load and heat source.

As the crank turns to the East position, both the hot and cold cylinder volumes decrease. Part of the cold gas is forced through the cold regenerator, which is at some mean temperature, $\bar{T}_{CRG}(T_C < \bar{T}_{CRG} < T_A)$, where it is heated to nearly T_A before entering the ambient section. Similarly, most of the hot regenerator which is at some mean temperature, $\bar{T}_{HRG}(T_A < \bar{T}_{HRG} < T_H)$, where it is cooled to nearly T_A before entering the ambient section. Heat, which is to be used later in the cycle, is thus stored in the hot regenerator. Finally, since both the hot cylinder volume and cold cylinder volume have decreased, the mean gas temperature, and consequently the gas pressure, change very little during this process. Nevertheless, some gas expansion (caused by the small

a) VM CYCLE

b) PRESSURE VOLUME DIAGRAMS FOR VM CYCLE
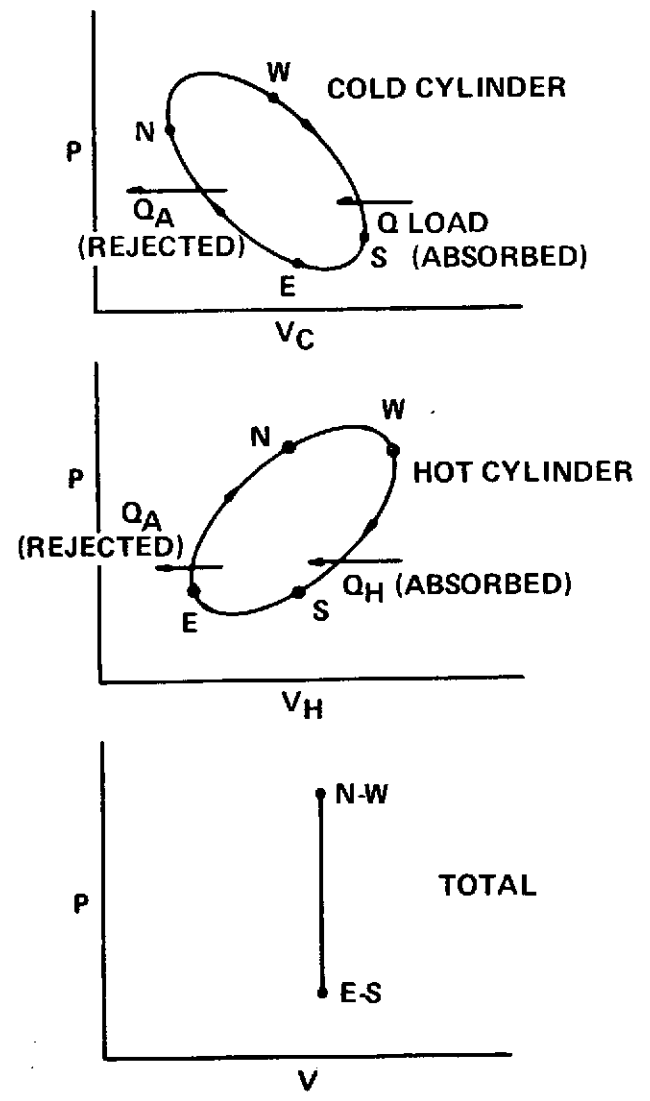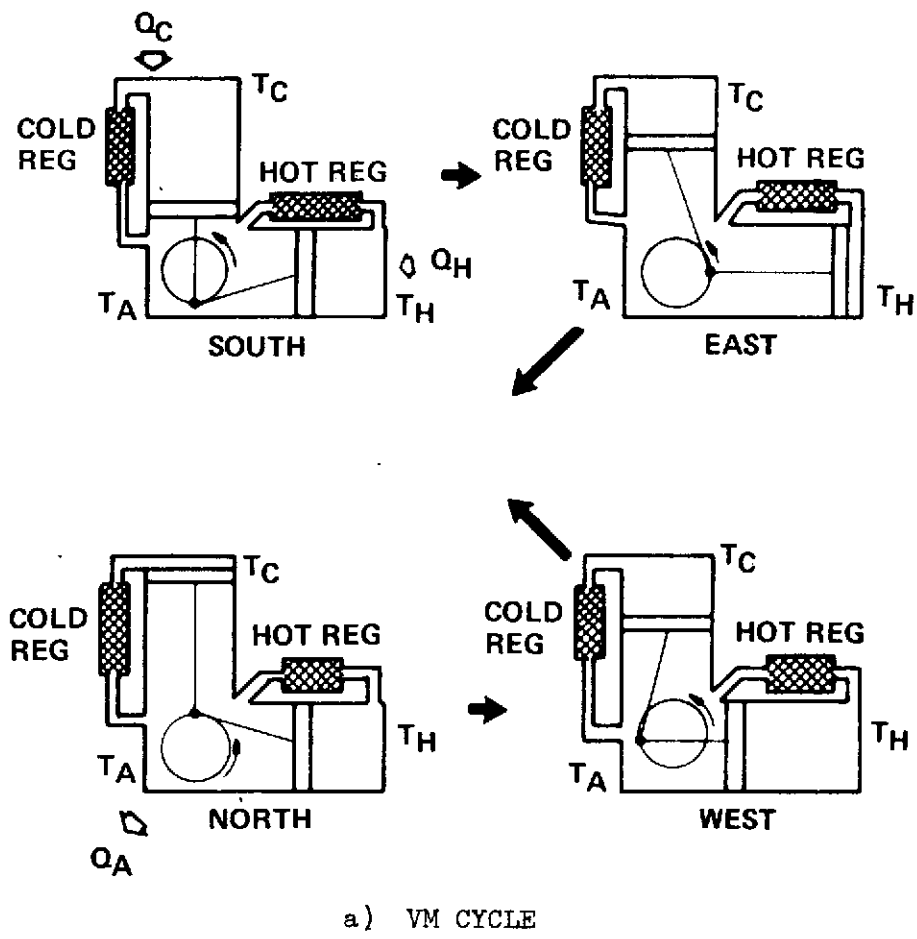
FIGURE 6-14 : VM REFRIGERATOR OPERATION

pressure decrease that does occur), with the resulting heat absorption from the load, does take place.

As the crank moves from East to North, the hot cylinder volume increases and the cold cylinder volume decreases. The cold gas which is forced through the cold regenerator is heated to nearly T_A, while part of the ambient gas flows through the hot regenerator, is heated (from stored energy) to nearly T_H, and enters the hot cylinder. The net effect of the hot cylinder volumetric increase and the cold cylinder decrease is an increase in the mean gas temperature and gas pressure. Hence, this process is one of gas compression, and for the temperatures to remain constant, heat must be rejected. Therefore, heat is rejected at the ambient section.

As the crank turns from North to West, the volumes of both the hot cylinder and cold cylinder increase. Part of the ambient gas moves through the cold regenerator, releases heat to it, and enters the cold volume at nearly T_C. On the hot side, part of the ambient gas moves through the hot regenerator, absorbs heat, and enters the hot volume at nearly T_C. Since both the cold volume and hot volume increase, the system pressure does not greatly change. Nevertheless, there is some compression with a corresponding heat rejection at the ambient section.

The crank now turns from West to South, decreasing the hot cylinder volume while increasing the cold cylinder volume. Part of the hot cylinder gas is forced through the hot regenerator where it is cooled to nearly T_A, while part of the ambient gas is forced through the cold regenerator where it is cooled to nearly T_C. The mean gas temperature, and consequently the gas pressure, decreases, resulting in gas expansion. This causes heat absorption at the cold and hot ends.

The net affect of the described processes is that heat is absorbed at the hot and cold cylinders, and rejected at the ambient section.

6.2.2   Scaling of the VM Cooler

The items of primary interest in the conceptual designs are weight, volume, power requirement, life and radiator area. The weight consists of VM cooler weight, radiator weight, and the equivalent weight of the power at 0.55 lb/watt. The VM cooler weight and volume are relative constant for the cooling load range of interest (0.3 to 1.0 watts) as shown in Figure 6-15. The equivalent power weight is a function of the COP for a given cooling load. COP can be approximated as a function of the cycle temperatures by the relation[10]
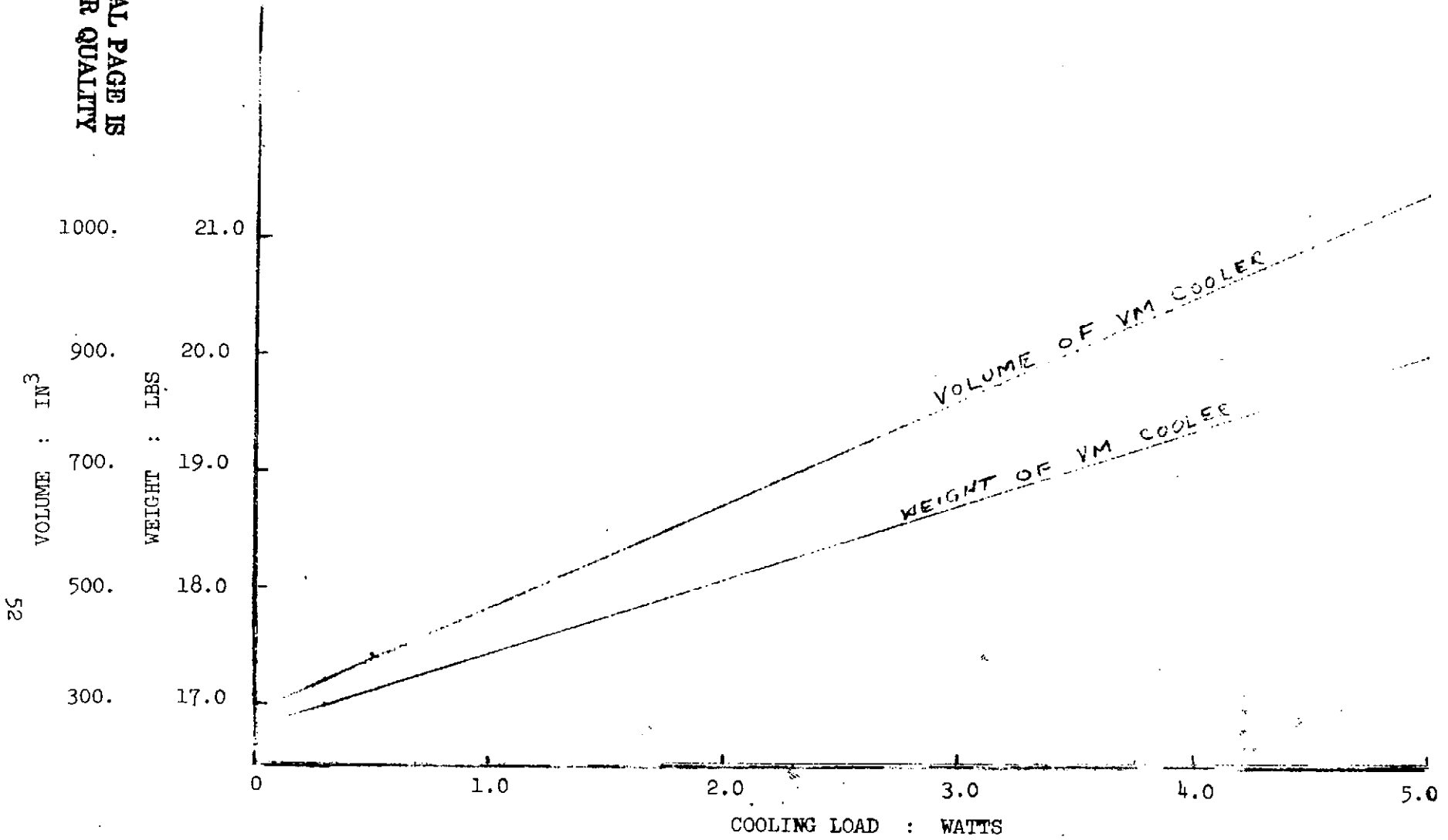
FIGURE 6-15 : FRACTIONAL WATT VM VOLUME AND WEIGHT

$$COP = K_{VM} \left(\frac{Tc}{Th}\right) \frac{(Th - Tsw)}{(Tsw - Tc)} \qquad (6\text{-}1)$$

where:

Tc = cold temperature

Tsw = sump temperature

Th = hot cylinder temperature

$K_{VM} = \dfrac{COP}{COP_{ideal}}$

K_{VM} is a strong function of Q_c at constant temperatures as shown in Figure 6-16. While Tc and Th are fairly well fixed, Tsw is free to be optimized. Since,

$$Q_H = \frac{Q_c}{COP} = \frac{1}{K_{VM}} \left(\frac{Th}{Tc}\right)\left(\frac{Tsw - Tc}{Th - Tsw}\right) \qquad (6\text{-}2)$$

Q_H increases as Tsw increases and thus, a low Tsw is desired. Radiator area is also a function of Tsw and Q_H. The area can be expressed as

$$A = \frac{Q_H + Q_L}{\sigma\epsilon\eta[(T_{sw} - \Delta T_{sr})^4 - T_s^4]} \qquad (6\text{-}3)$$

where:

σ = Stefan Boltzman constant

ϵ = emissivity

η = fin effectiveness

ΔT_{sr} = temperature drop from sump to radiator

T_s = external sink temperature

It can be seen that radiator area increases with decreasing T_{sw}. Thus, an optimum T_{sw} can be determined.

The total cooling system weight was calculated versus T_{sw} and T_s as follows:

$$\dot{w}_T = P_p \cdot Q_H + P_r \left[\frac{Q_H}{\sigma\epsilon\eta[(T_{sw} - \Delta T_{sr})^4 - T_s^4]}\right] + P_F \qquad (6\text{-}4)$$

where:

P_p = power weight penalty
= 0.55 lb/watt for this study

P_r = radiator weight penalty
= 1.5 lb/ft^2 for this study

P_F = fixed weight penalties

Q_H is determined from equation (6-2)

K_m
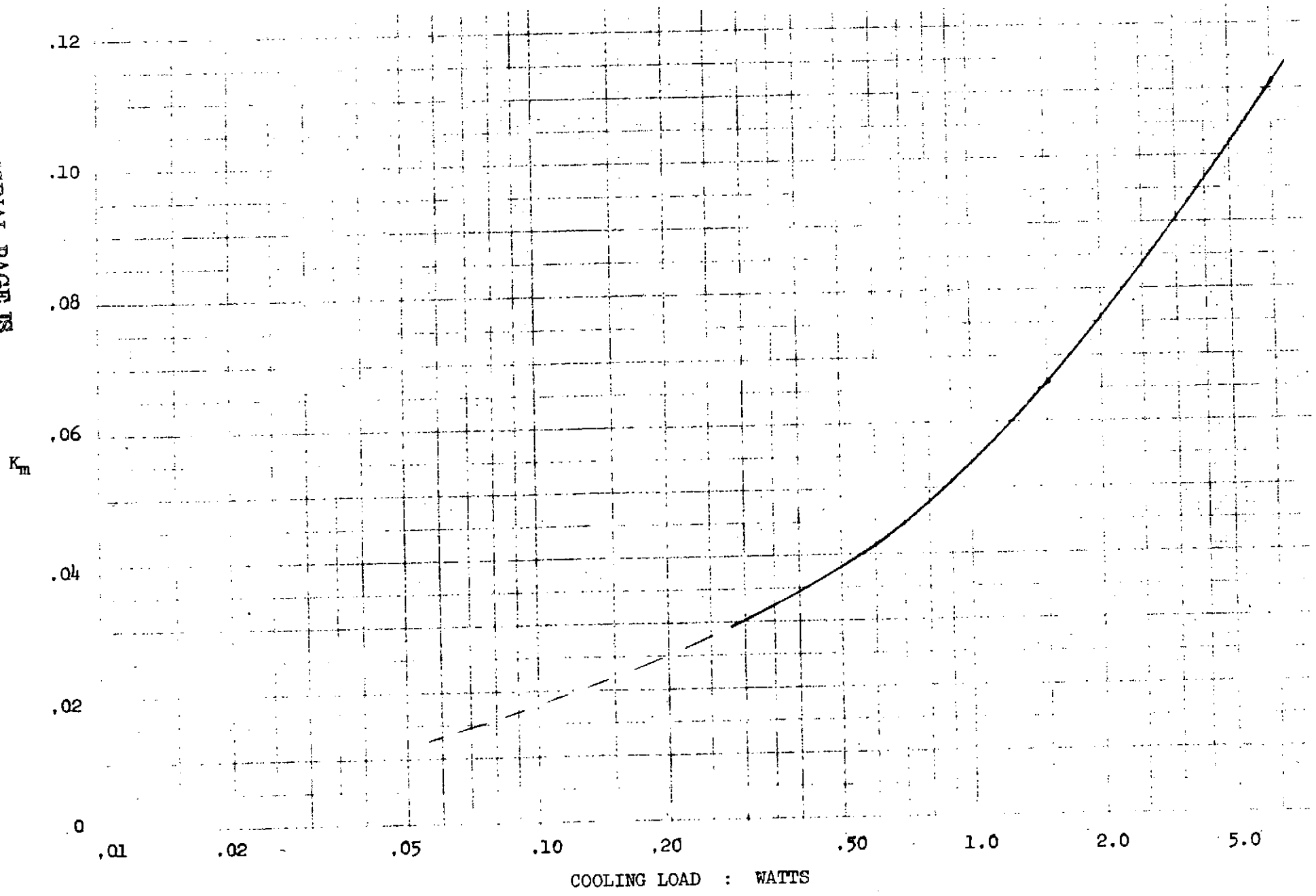
COOLING LOAD : WATTS

FIGURE 6-16 : VM COP RATIO VS COOLING LOAD

The weight versus T_{sw} and T_s is shown in Figure 6-17. Three sink temperatures are shown. Each corresponds to a different coating assuming the sun shines directly on the panels. A coating of silver-backed Teflon was assumed which resulted in an optimum sump temperature of approximately 295°K. The white paint coating optimum T_{sw} was 325°K compared to the 333°K used for the Fractional Watt Cooler. Based upon this study, a sump temperature of 300°K was used for conceptual designs.

An estimate was made of total VM system weight vs cooling load. This curve is shown in Figure 6-18. The system weight shown in this curve includes the VM unit weight, the heat rejection system and the equivalent weight of power.

6.2.3   Category II Conceptual Design

The design requirements for Category II were as follows:

. Cooling Temperature : 77°K

. Cooling Load   : 1000 Milliwatts

. Mission Life   : 2 years

. Duty Cycle   : Continuous operation for 1 month
                    followed by 10 days continuous
                    operation per 6 week period

A sketch of the conceptual design is shown in Figure 6-19. The design consists of a scaled fractional watt VM cooler and a heat pipe radiator attached for heat rejection. The VM cooler is approximately 25" in length, 6-1/2" diameter on the hot cylinder end and 2" diameter at the cold cylinder end. The cooler estimated weight is 20 pounds. The heat rejection system consists of a 35.1" square panel coated with silver-backed Teflon (coating, α = .08, ε = 0.8) with four aluminum/ammonia heat pipes attached. The heat pipes are 3/8 inch O.D. with a condenser length of approximately 35" and an adiabatic length of approximately 3 feet. The estimated total heat rejection system weight including the adiabatic heat pipe sections is 16 pounds.

The power required for the cooler is estimated at 88 watts (78 watts for the hot cylinder heater and 10 watts for the crank motor). The weight equivalent of the power at 0.55 lb/watt is 48 pounds. Total weight of the system is 84 pounds.

A summary of the Category II conceptual design is given in Table 6-3.
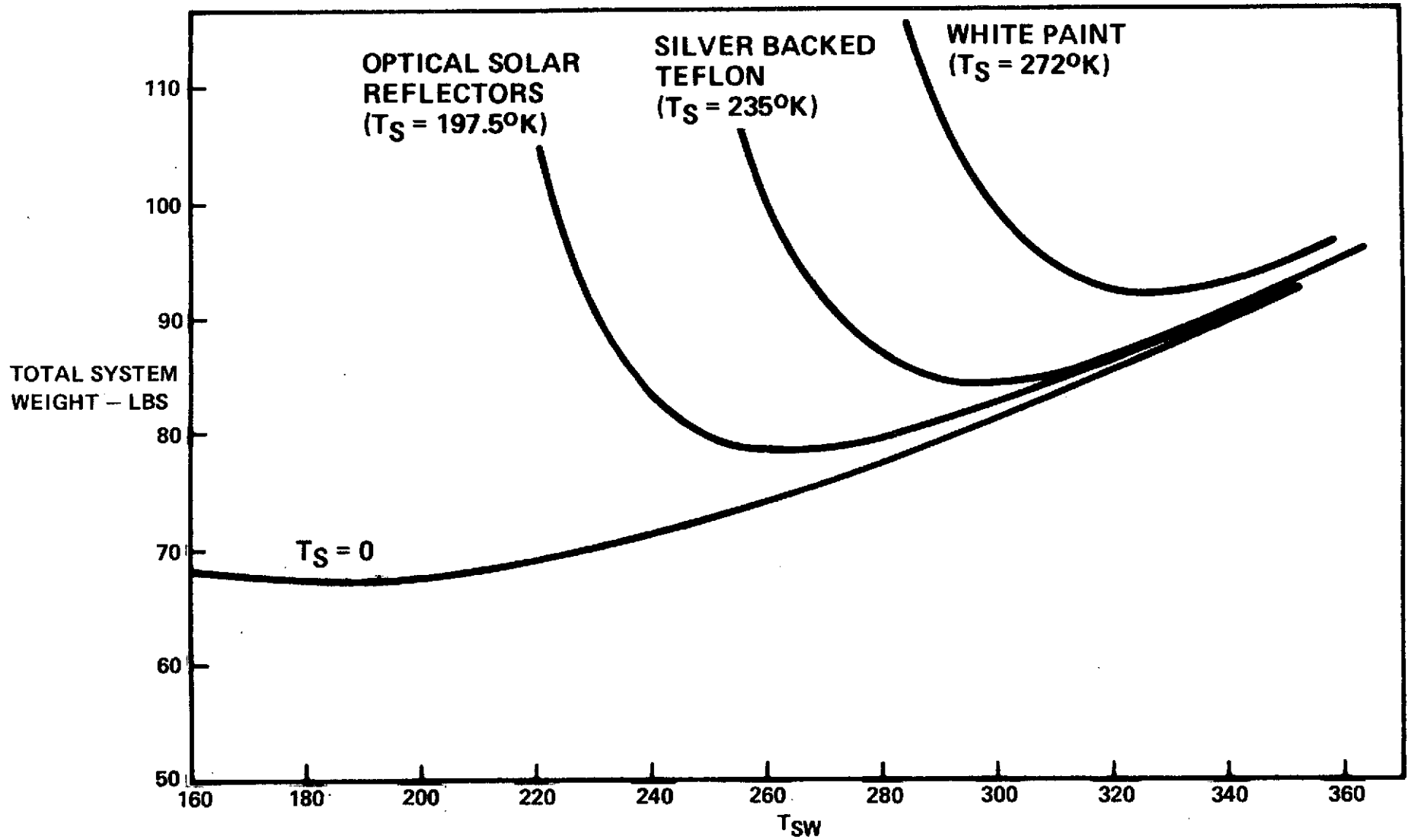
FIGURE 6-17

DETERMINATION OF OPTIMUM VM SUMP
TEMPERATURE
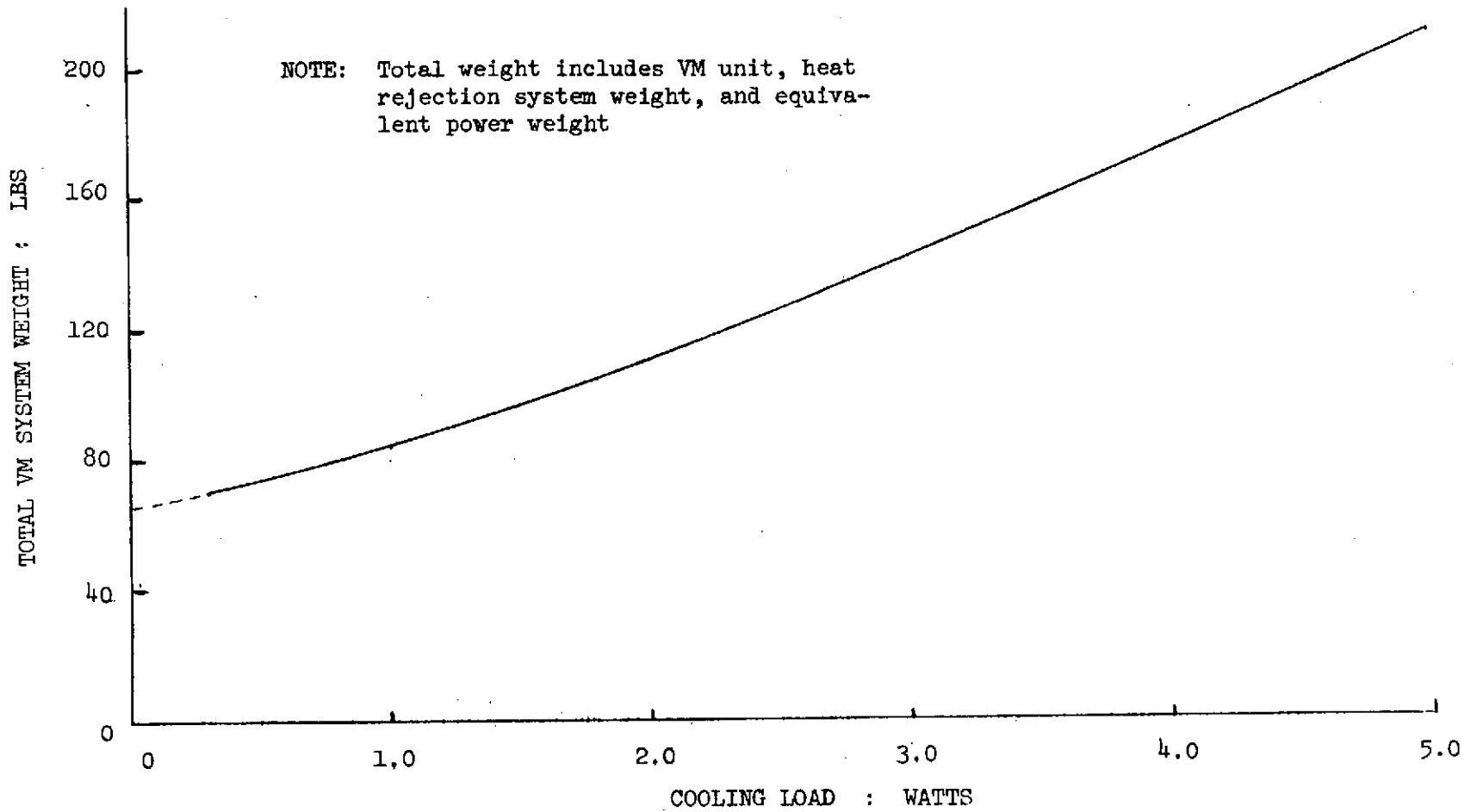
FIGURE 6-18 : TOTAL VM SYSTEM WEIGHT VS COOLING LOAD

NOTE: Total weight includes VM unit, heat rejection system weight, and equivalent power weight

TOTAL VM SYSTEM WEIGHT : LBS

COOLING LOAD : WATTS

57

6½"D

VUILLEUMIER
COOLER

35.1"

4.39"

.375" D
TYP

8.78" SPACING (TYP)

25"

2"

7"

78 W
POWER

2" D

8" D

35.1"

4 ALUMINUM AMMONIA HEAT PIPES

1000 MW
COOLING LOAD
@ 77°K

RADIATOR WITH
SILVER BACKED TEFLON
COATING
($\alpha = 0.08, \epsilon = 0.80$)

FIGURE 6-19

CATEGORY II CONCEPTUAL DESIGN

TABLE 6-3

CATEGORY II CONCEPTUAL DESIGN DESCRIPTION

- VM COOLER DESCRIPTION
  - TYPE: VM (AIRESEARCH BASELINED)
  - LUBRICATION: DRY
  - WORKING FLUID: HELIUM
  - VOLUME: 25" X 7" X 6½"
  - WEIGHT: 20 LBS
  - PRESSURE VARIATION: 1000/885 PSIA
  - HIGH TEMPERATURE: 853°K
  - LOW TEMPERATURE: 77°K
  - SUMP TEMPERATURE: 300°K
  - COOL DOWN TIME: 15 MINUTES
  - POWER: 88 WATTS

- HEAT REJECTION SYSTEM
  - TYPE: HEAT PIPE RADIATOR
  - WEIGHT: 16 LBS (INCLUDING HEADERS)
  - VOLUME: 35" X 35" X 1"
  - HEAT PIPES: .375" OD AMMONIA/ALUMINUM (4)
  - COATING: SILVER BACKED TEFLON ($\alpha_S = 0.08$; $\epsilon = 0.8$)

- WEIGHT SUMMARY
  - VM MACHINE   20 LB
  - ADIABATIC HEAT PIPE SECTION   3
  - RADIATOR   13
  - POWER @ .55 LB/WATT   48
  - TOTAL   84 LB

6.2.4   Category III Conceptual Design

The design requirements for Category III were as follows:

- . Cooling Temperature : 77°K
- . Cooling Load   : 300 Milliwatts
- . Mission Life   : 2 years
- . Duty Cycle   : continuous

A sketch of the conceptual design is shown in Figure 6-20. The design consists of a scaled fractional watt VM cooler and a heat pipe radiator attached for heat rejection. The VM cooler is approximately 23.5" in length, 5" diameter on the hot cylinder end and 2" diameter at the cold cylinder end. The cooler estimated weight is 17 pounds. The heat rejection system consist of a 31.0" square panel coated with silver-backed Teflon (coating, α = .08, ϵ = 0.8) with four aluminum/ammonia heat pipes attached. The heat pipes are 3/8 inch O.D. with a condenser length of approximately 31" and an adiabatic length of approximately 3 feet. The estimated total heat rejection system weight including the adiabatic heat pipe sections is 13 pounds.

The power required for the cooler is estimated at 65 watts (62 watts for the hot cylinder heater and 3 watts for the crank motor). The weight equivalent of the power at 0.55 lb/watt is 36 pounds. Total weight of the system is 66 pounds.

A summary of the Category III conceptual design is shown in Table 6-4.

6.2.5   Alternate Conceptual Design for Category III

The total active life requirement for Category III (2 years continuous operation) is a relatively severe requirement for a single VM unit with current demonstrated technology. (An AiResearch 5 watt engineering model was tested for 5000 hours and a minimum 2 year life was projected but has not been demonstrated[11].) Because of this uncertainty in meeting the long life with a single VM unit, an alternate conceptual design is proposed for Category III identical to that of Section 6.2.4 except a standby VM unit is included. A schematic of the alternate conceptual design is shown in Figure 6-21. The backup VM, which weighs 17 pounds, requires a means of thermal isolation/engagement such as the thermal switch which has an estimated weight of 5 pounds. These two additional items plus a pound for additional heat pipe header for the heat rejection system results in 23 additional pounds for the redundant VM unit.

A summary of the alternate Category III conceptual design is given in Table 6-5.

60

5" D

VUILLEUMIER
COOLER

31"

3.875"

0.375" D
TYP

7.75" SPACING (TYP)

2" D

6"

4 ALUMINUM — AMMONIA HEAT PIPES

31"

23.5"

62W
POWER

2" D

8"

300 MW
COOLING LOAD
@ 77°K

RADIATOR WITH
SILVER BACKED TEFLON
COATING
($\alpha = 0.08,\ \epsilon = 0.8$)
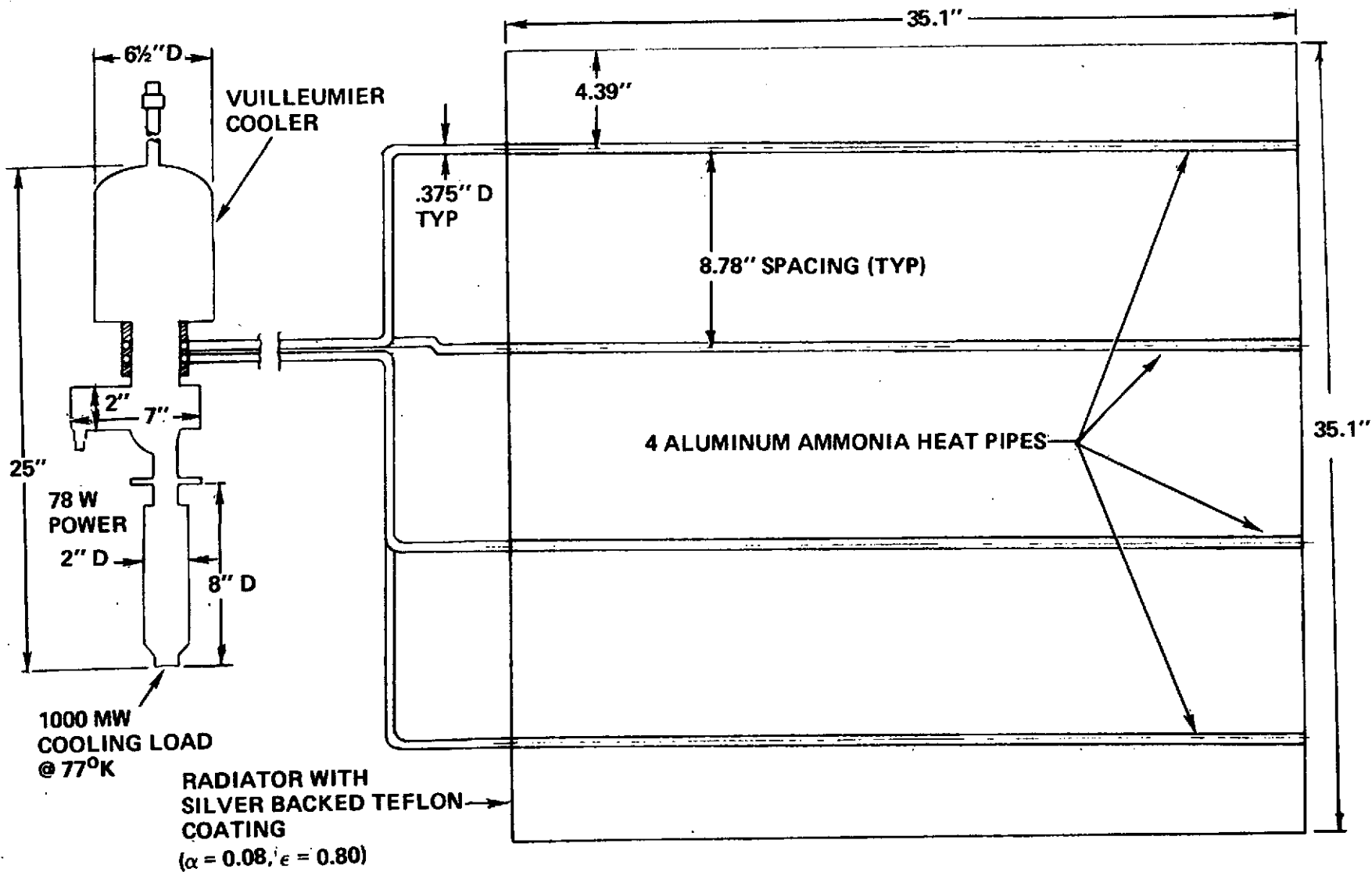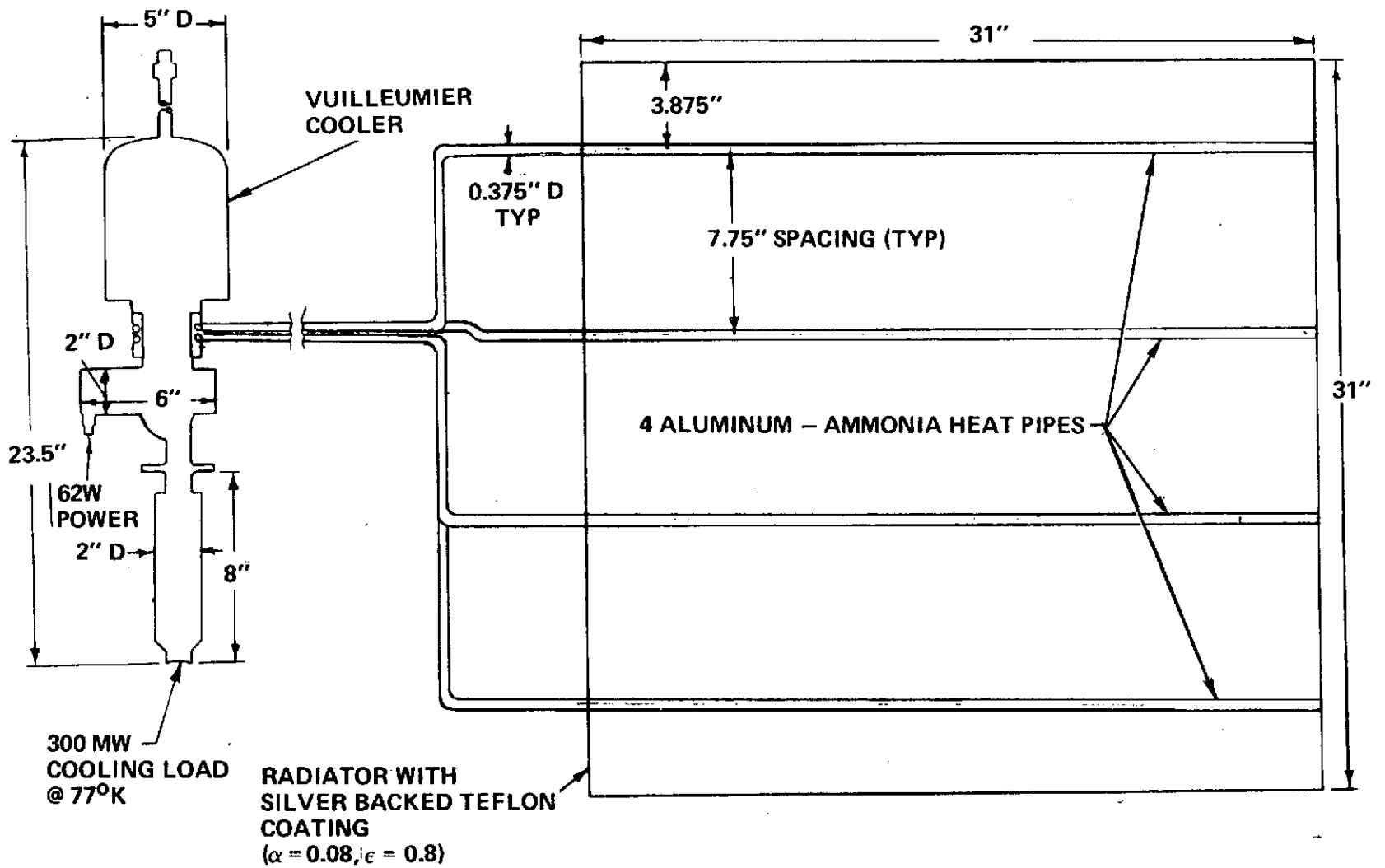
FIGURE 6-20

CATEGORY III CONCEPTUAL DESIGN

TABLE 6-4

CATEGORY III CONCEPTUAL DESIGN DESCRIPTION

- COOLING DEVICE DESCRIPTION
  - TYPE: VM (AIRESEARCH BASELINED)
  - LUBRICATION: DRY
  - WORKING FLUID: HELIUM
  - VOLUME: 23.5" X 6" X 6"
  - WEIGHT: 17 LBS
  - PRESSURE VARIATION: 1000/885 PSIA
  - HIGH TEMPERATURE: 853°K
  - LOW TEMPERATURE: 77°K
  - SUMP TEMPERATURE: 300°K
  - COOL DOWN TIME: 15 MINUTES
  - POWER: 65 WATTS

- HEAT REJECTION SYSTEM
  - TYPE: HEAT PIPE RADIATOR
  - WEIGHT: 13 LBS (INCLUDING HEADERS)
  - VOLUME: 31" X 31" X 1"
  - HEAT PIPES: .375" OD AMMONIA/ALUMINUM (4)
  - COATING: SILVER BACKED TEFLON (α_S = 0.08; ϵ = 0.8)

- WEIGHT SUMMARY (LBS)
  - VM MACHINE   17 LB
  - ADIABATIC HEAT PIPE SECTION   3
  - RADIATOR   10
  - POWER @ .55 LB/WATT   36
  - TOTAL   66 LB

62
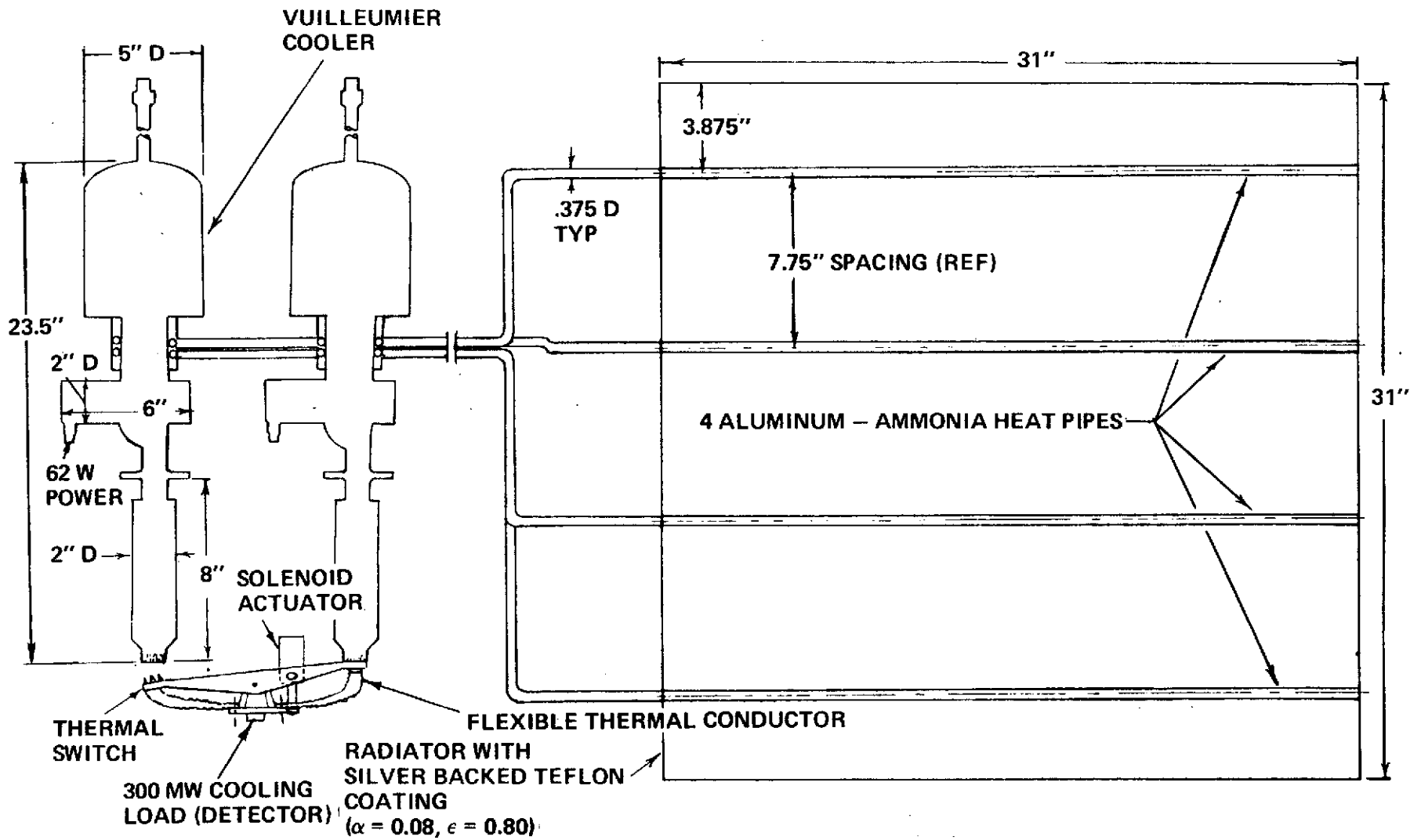
FIGURE 6-21

CATEGORY III ALTERNATE CONCEPTUAL DESIGN

TABLE 6-5

CATEGORY III ALTERNATE
CONCEPTUAL DESIGN DESCRIPTION

- COOLING DEVICE : SAME AS CATEGORY III (TABLE 6-4) DESIGN EXCEPT
  TWO VM UNITS REQUIRED

  – ADDITIONAL WEIGHT: 17 LBS

- **HEAT REJECTION: SAME AS CATEGORY III DESIGN** (TABLE 6-4)

- **HEAT LOAD INTERFACE REQUIREMENT**
  - **THERMAL SWITCH REQUIRED TO ISOLATE INACTIVE UNIT**
  - **SOLENOID ACTUATED**
  - **CONDUCTING CABLE PROBABLY REQUIRED**
  - **ESTIMATED WEIGHT: 5 LB**

- **WEIGHT SUMMARY (LBS)**

– VM MACHINES	**34**
– ADIABATIC HEAT PIPES	**4**
– RADIATOR	**10**
– POWER @ 0.55 LB/WATT	36
– THERMAL SWITCH	**5**
– TOTAL	89

6.3   Category IV

Category IV is 50 MW of cooling at 195°K for 2 years. The duty cycle is intermittent at 5 to 8 days per month. Thermoelectric coolers were selected for this category. Passive radiators are also good candidates and they are discussed in the next conceptual design, Section 6.4.

Thermoelectric (T/E) coolers employ the Peltier effect to provide the necessary heat transfer. The refrigerator is illustrated in Figure 6-22. The cooling occurs at the cold space where the detector is maintained at a temperature of T_c. Cooling is provided by passing an electric current through two dissimilar materials with a common junction. The two legs form a couple consisting of one "n" type and one "p" type material. Heat is pumped from the cold junction and dissipated with the electrical power at the hot junction at a temperature of T_h. Direct current with controlled current and voltage powers the unit. The power required for a couple is a function of the Figure of Merit, Z. Z is determined from the basic properties and is calculated as shown in Figure 6-22. The power penalty represents the largest component of weight for the system; and good material properties are needed to minimize that penalty.

Figure 6-23 presents the maximum coefficient of performance (COP) of a single stage T/E cooler as a function of the cold junction temperature and Figure of Merit. COP is the ratio of cooling rate at T_c to the power consumed; hence the power penalty is directly proportioned to this factor. Z's for Bi/Te alloys couples are typically between 2 and $3(10^{-3})$ °K^{-1}; with these materials a single stage cooler can not achieve cooling at 195°K. Bi/Sb alloys can have higher Z's but the COP is still very low.

A multistage cooler increases the cooling region of a given material. Figure 6-24 presents theoretical performance for one, two and three stage coolers. For a typical Figure of Merit of 0.002 °K^{-1}, maximum COP is plotted as a function of the cold junction temperature and number of stages. Multiple stages always increases the maximum COP; and the theoretical maximum always occurs with an infinite number. However, practical limits prohibit building a cooler with very many stages, and for 195°K, 4 stages is about optimum.

Figure 6-24(b) presents the maximum temperature difference across a couple as a function of Z and number of stages. (This maximum temperature lift is also proportional to the maximum COP). Again the largest number of stages will yield the highest performance for all materials (Z's). Again the number of
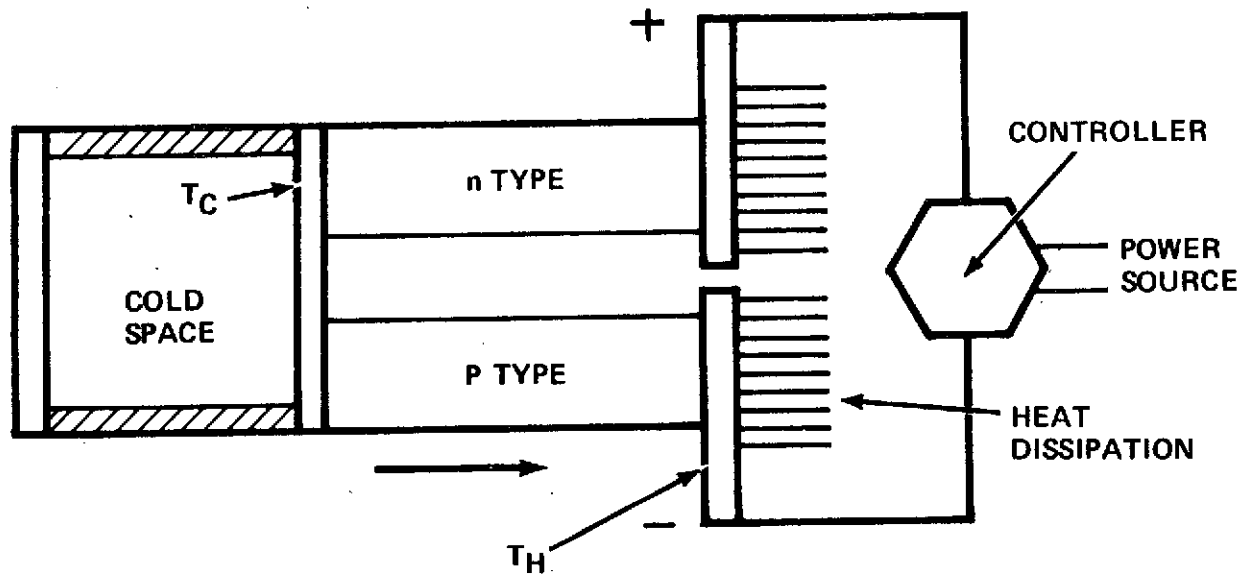
FIGURE OF MERIT

- OF A MATERIAL:

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

- OF A COOLER:

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

WHERE:

s = SEEBECK COEFFICIENT FOR A SINGLE MATERIAL, V/K

ρ = ELECTRICAL RESISTIVITY, OHM—CM

k = THERMAL CONDUCTIVITY, W/CM—K

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

FIGURE 6-22

THERMOELECTRIC COOLING
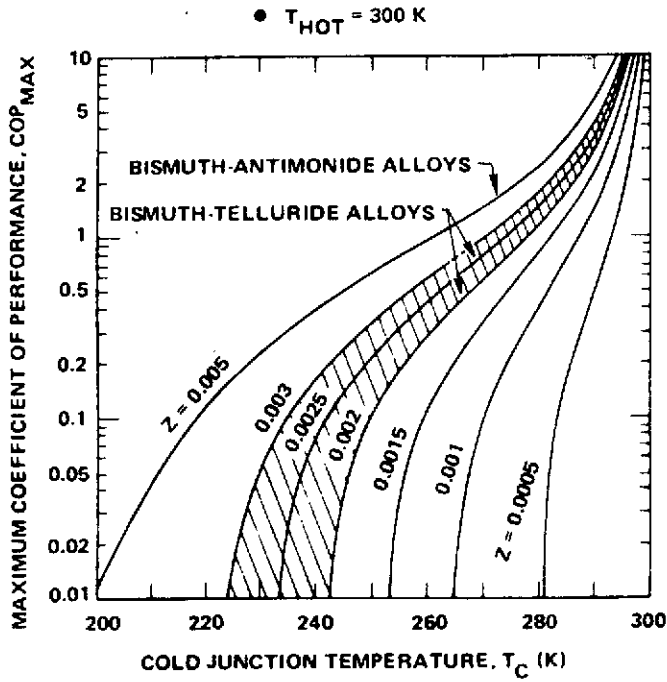
THEORETICAL PERFORMANCE OF SINGLE
STAGE T/E COOLERS

FIGURE 6-23

(A) MAXIMUM COEFFICIENT OF PERFORMANCE
VS COLD JUNCTION TEMPERATURE

(B) MAXIMUM TEMPERATURE LIFT VS FIGURE
OF MERIT
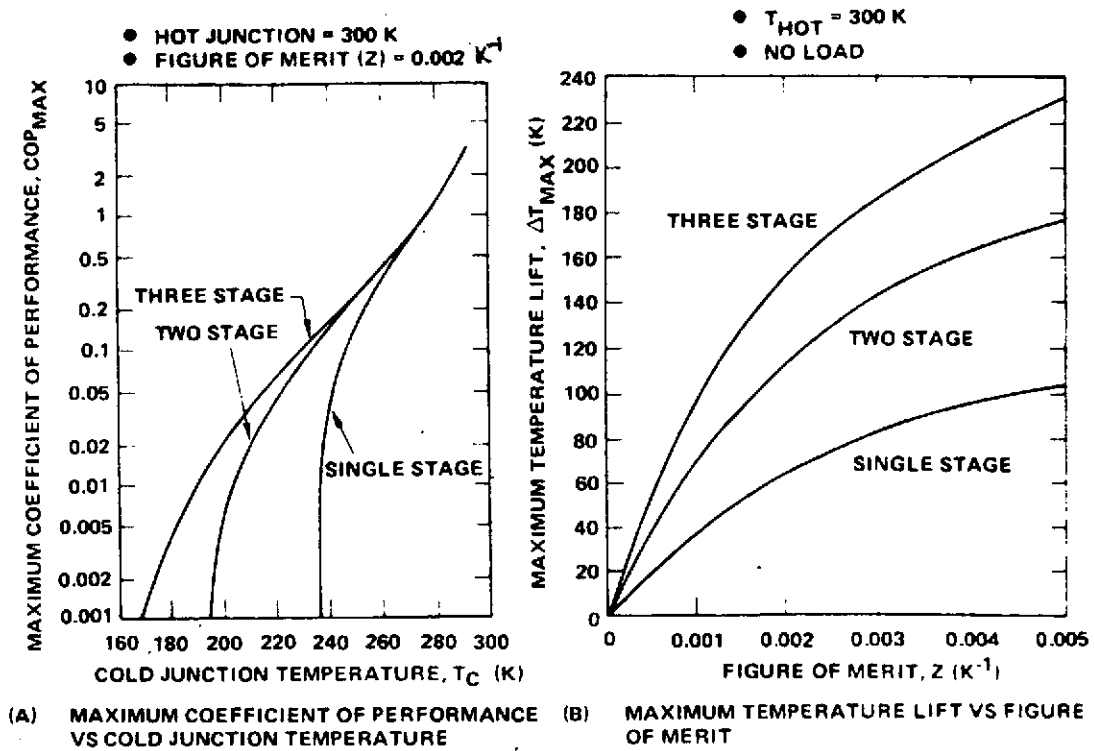
THEORETICAL PERFORMANCE OF MULTISTAGE
T/E COOLERS

FIGURE 6-24

stages will be limited by the ability to efficiently construct a larger number.

Figure 6-25 presents measured data on Figure of Merit for N and P type materials. Three sources for materials were considered commercial, Borg Warner Thermoelectrics and Lockheed Missiles and Space Co. The commercial materials are available in large quantities today; off-the-shelf hardware in a number of standardized designs can be purchased. The other two are still in development and are currently produced by laboratory methods. However, for space application and for the time period of interest, sufficient quantities could be produced even if current manufacturing techniques must be used.

Bi/Te alloys are employed in the commercial and BWTE materials. Basically the same alloys are used in both, but an improved manufacturing technique provides the improvement in Figure of Merit. For the 10 to 20% increase in Z, a 40% to 100% improvement in COP (see Appendix B, the COP for the 6 stage Marlow Industries and Nuclear Systems are 0.006 to 0.010 at 195°K versus the 0.014 for the 6 stage Borg Warner unit). The higher performance material can thus have very significant reduction in power requirements. This advantage will probably more than justify the higher costs for a spacecraft application.

The LMSC materials are Bismuth-Antimonide (Bi/Sb) alloys. The P-type material has a figure of merit approximately equivalent to the BWTE material. The N-type materials are distinctly different. Very high Z's can be obtained but only at the very low temperatures. At 200°K and higher temperatures the BWTE materials have better properties than the LMSC. Since this study is concerned with the 195°K and higher temperature range, the LMSC material has little promise.

The N-type Bi/Sb are affected by a magnetic field, but P-type is not significantly affected. With the proper magnetic field applied, the N-type Figure of Merit can be increased to $3.5(10)^{-3}$ °K^{-1} at 300°K, $5.5(10)^{-3}$°K^{-1} @ 250°K and $6.2(10)^{-3}$°K^{-1} @ 190°K (Ref. 12). The magnetic field strengths reportedly can be obtainable by permanent magnetics. This Z is greatly in excess of that of the BWTE material, which is not affected significantly by magnetic fields. Thus the power requirements can be greatly reduced with a magnetically enhanced Bi/Sb cooled. However, this technology is not well developed and significant questions have yet to be solved. For those latter reasons the BWTE material was chosen for the conceptual design over the magnetically enhanced LMSC material.

P—TYPE FIGURE OF MERIT MEASUREMENT DATA

*N—TYPE FIGURE OF MERIT MEASUREMENT DATA
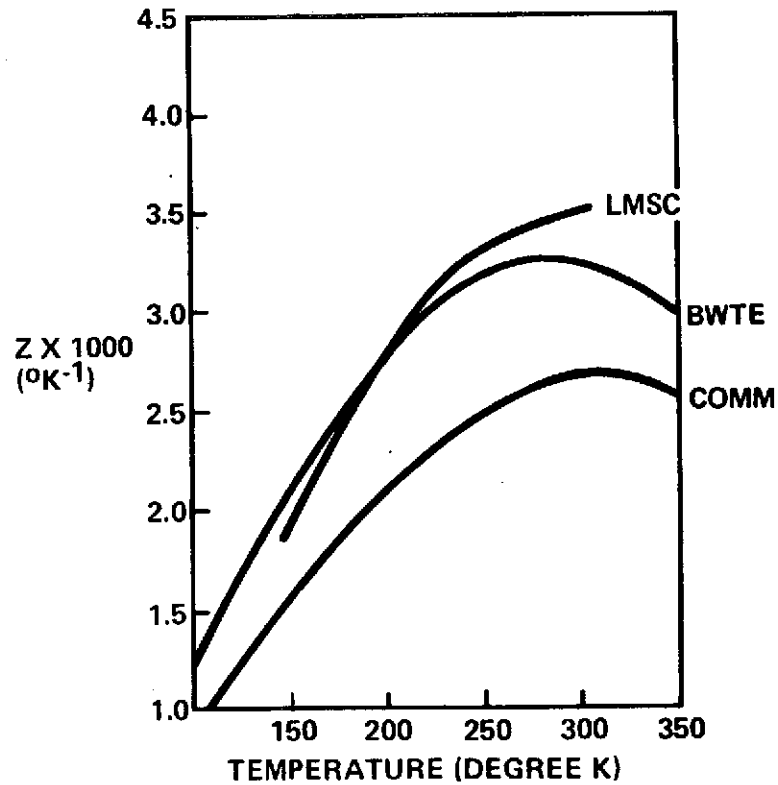
LMSC: LOCKHEED MISSILES AND SPACE CO.

BWTE: BORG-WARNER THERMOELECTRICS

COMM: COMMERCIAL
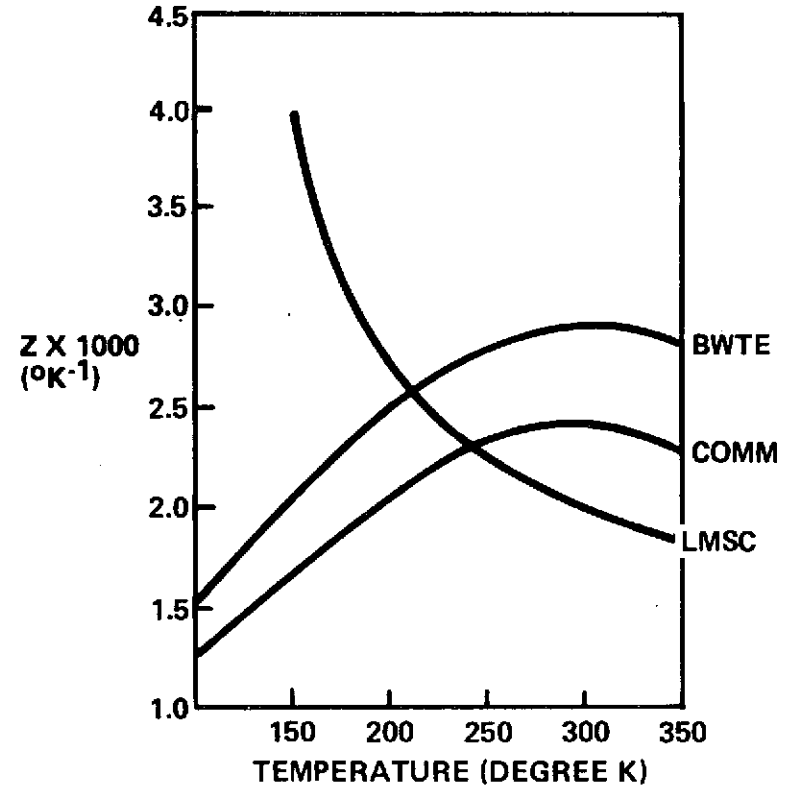
*MAGNETIC ENHANCEMENT IS POSSIBLE WITH N—TYPE MATERIALS ONLY; A "Z" UP TO $3.5 (10)^{-3}$ $^{o}K^{-1}$ HAS BEEN MEASURED AT $300^{o}K$ WITH THE LMSC MATERIAL

FIGURE 6-25

MATERIAL PROPERTIES

69

Figure 6-26 presents a conceptual design of a T/E cooler for Category IV requirements. The photograph on the right is of a prototype unit made by BWTE. The COP of this unit was measured as 0.0243 at 195°K and radiator at 300°K. The detector in this design directly mounts to the cooler. The cooler unit has an envelope of 0.4" x 0.67' x 0.75" and is connected to the radiator either by a flexible conductive cable or by direct mounting, if tolerances permit. Mounting holes are provided to install the radiator on the outside surface of the spacecraft. Silver-backed Teflon is the coating and the radiating area is sufficiently large to provide heat rejection in any orbit and any location on the spacecraft provided the radiator is not radiantly heated by other spacecraft surfaces. To increase the effectiveness of the radiator the surrounding surfaces should also have a low solar absorptance and high emittance coating; insulation from the surrounding surfaces is not necessary.

The temperature of the detector is maintained within tolerances by the control module. That module controls voltage and current supplied to the cooler and senses the temperature of the detector via a thermistor mounted on the cold junction of the T/E cooler. When either the heat load of the detector or the radiation environment change, the control module responds in one of two ways as follows:

1) changing the current and voltage supplied to the cooler

or 2) Increasing (or decreasing) the heat dissipation in a resistor mounted at the T/E cold junction (requires 10 to 50 milliwatts)

The weights of all component are presented at the bottom of Figure 6-26. The total weight including penalties is 1.87 lbs of which the vast majority is the power penalty for 2.06 watts (1.33 lbs at 0.55 lb/watt). The next largest weight is the controller, at approximately 0.5 lbs for a constant current supply and temperature control via heat dissipation in a resistor on the cold junction. The T/E unit itself is almost negligible by comparison to the other items.

Figure 6-27 presents the approximate effect of cooling capacity upon the system weight. Total system weights, both with and without the power penalty are presented. When the power penalty is different from 0.55 lb/watt, the total system weight is calculated by adding the electrical power weight equivalent to the lower curve. These data were linearly scaled from the conceptual design;
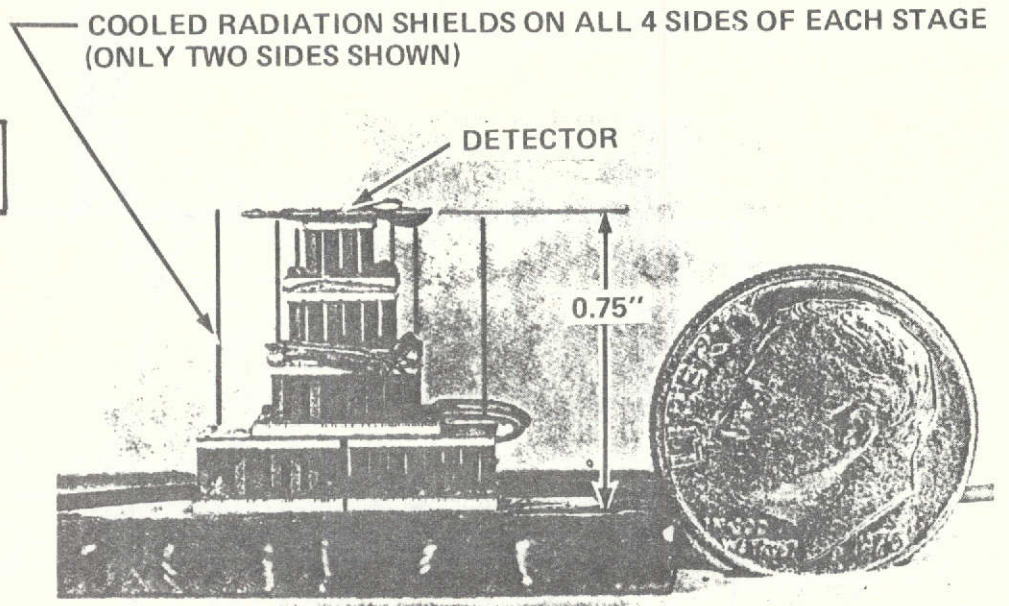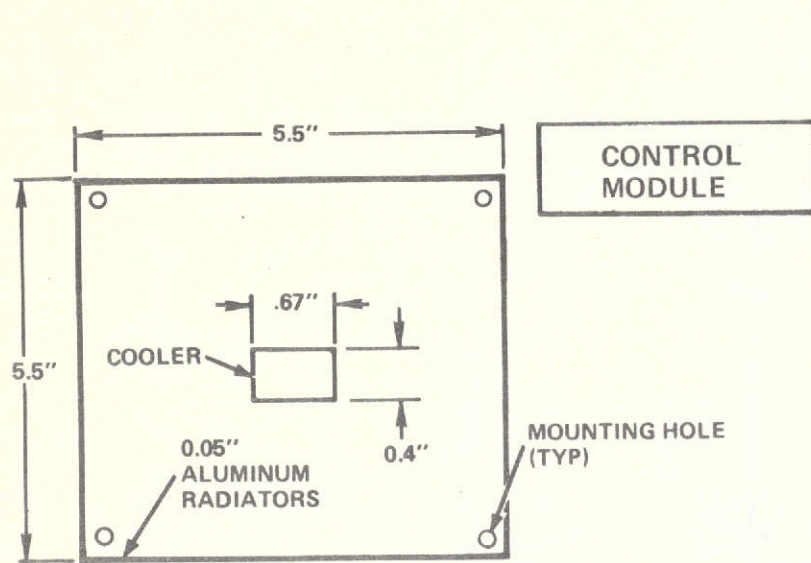
70

CONTROL
MODULE

5.5"

5.5"

.67"

COOLER

0.4"

0.05"
ALUMINUM
RADIATORS

MOUNTING HOLE
(TYP)

DETECTOR

0.75"

BORG WARNER

WEIGHT ANALYSIS		
THERMOELECTRIC COOLER:		0.036 LBS
RADIATOR (ALUMINUM)	:	0.150 LBS
POWER 2.06 WATTS	:	1.133 LBS (@ 0.55 LBS/WATT)
CONTROLS	:	0.5 LBS (APPROXIMATE)
MOUNTS	:	0.02 LBS
RADIATION SHIELD	:	0.03 LBS
TOTAL	:	1.87 LBS

FIGURE 6-26

CONCEPTUAL DESIGN OF A 50 MW 195°K
T/E COOLER
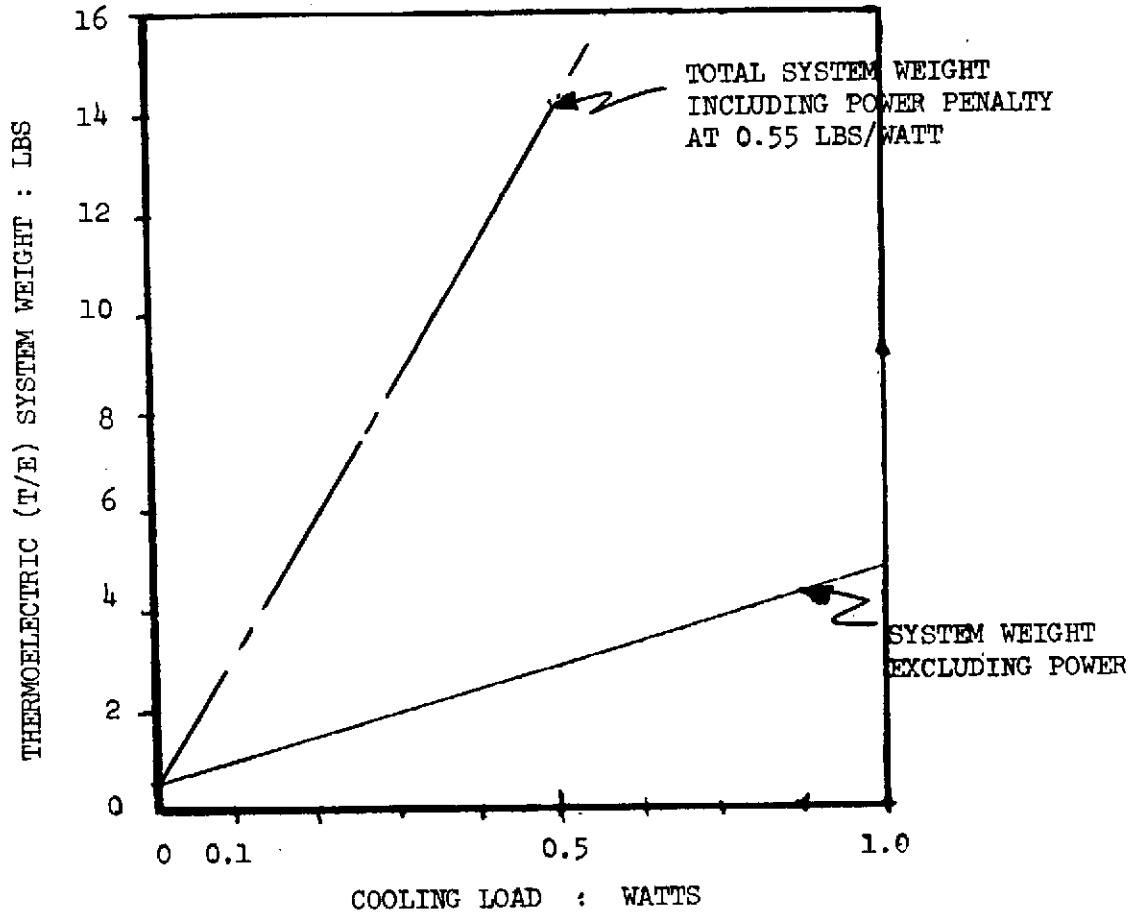
FIGURE 6-27

195°K T/E SYSTEM WEIGHT VS CAPACITY

the radiator weight may increase due to the need for increased thickness and possibly active heat transfer. For the latter reason the data are not good with cooling loads in excess of 1.0 watt.

T/E coolers are not strongly affected by life. Since there are no moving parts, the reliability of these units are very high even with durations up to 5 years. In spacecraft power penalties must be assigned for the maximum rate and there is no advantage for intermittent over continuous duty. As mentioned earlier the T/E cooler may be mounted with any orbit and at any location on the spacecraft. This characteristic greatly reduced design installation problems but may incur large weight penalties at high cooling loads. Passive radiators can provide lower penalties but incur orientation and orbit limitations. Those devices are discussed in the next section.

6.4   Category V and VI

The trade study described in Section 5.0 suggested that radiators would best meet the requirements of Categories V and VI. Conceptual designs were performed for these categories which are described in Section 6.4.2 and 6.4.3. A general discussion of radiators is given in Section 6.4.1.

6.4.1   Radiators

Two types of radiators were considered for the requirements of Categories V and VI. These were (1) shielded radiative coolers and (2) flat radiator panels with advanced coatings to obtain the low temperatures. Several configurations have been conceived for obtaining low temperatures in space[4 and 13 thru 17] using shielded radiators with some as low as 77°K. Since the requirements of Category V are not severe, the two stage cooler approach described in Reference 4 and shown in Figure 6-30 was considered best. The weight of this cooler is approximately 35 pounds per square foot of cold stage radiator area at 77°K. The amount of heat rejection obtainable from a shielded radiator may be estimated from Figure 6-28 for known emissivity and area of the cold stage.

The heat rejection of the flat plate radiators may be estimated by the relationship:

$$Q/A = n[\sigma \varepsilon T^4 - \alpha (\frac{Q_{abs}}{A})]$$

73
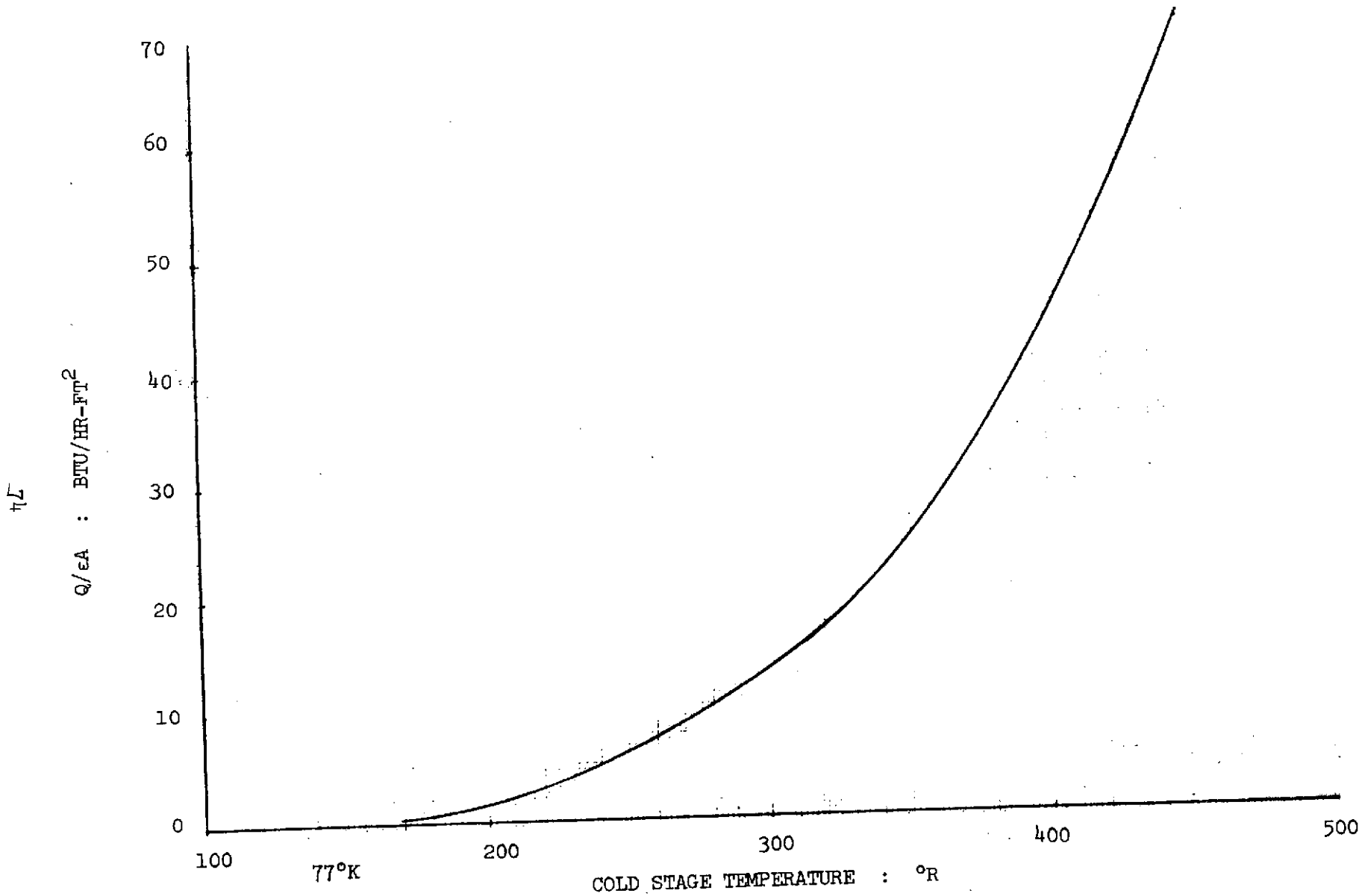
FIGURE 6-28 : SHIELDED RADIATOR PERFORMANCE

where: Q/A   is heat rejection per unit area

η   is the fin effectiveness

σ   is the Stefan-Boltzman constant

ε   is the radiator coating emissivity

α   is the radiator coating absorptivity

T   is the radiator temperature

$(\frac{Q_{abs}}{A})$ is the absorbed heat per unit area

The approximate weight can be estimated by $w = 3 + A$ for both the 195°K and 300°K radiators where w is weight in pounds and A is the area in square feet.

6.4.2   Category V

The requirements for Category V are 500 milliwatts cooling load at 195°K with a 2 year continuous operating life. The trade study described in Section 5.0 indicated a radiator would be the best type cooler for these requirements. Thus a radiator was baselined. The orbital design environment was not firmly defined for the current study. However, it is anticipated that the PMS satellite would be in a low earth orbit and may be in a sun synchronous orbit. Thus designs were performed for two cases: (1) a sun synchronous orbit and (2) a general low earth orbit. Each of these are discussed separately below.

6.4.2.1   Design for Sun Synchronous Orbit

The design of a radiator is strongly dependent upon its radiant environment which in turn depends upon the orbit being flown and the orientation of the radiator in orbit. One candidate orbit for the pollution monitoring satellites is a sun synchronous orbit. This orbit which is shown schematically in Figure 6-29 maintains a constant angle, θ, between the orbit plane and the Earth-Sun line throughout the year. The orbital inclination for the sun synchronous orbit is 99°. A conceptual design for a passive radiator for the sun synchronous orbit is shown in Figure 6-30. This radiator is the Arthur D. Little Radiator Cooler Model 101 which was designed for rejection of 10 milliwatts of cooling at 100°K. However, its projected performance at 195°K meets the requirement of Category V. The cooler consists of two radiator stages and a reflecting cone. The external radiator stage attaches (thermally) to the cone outer opening and extends out to a diameter of 11.88". The internal radiator, which is isolated from the reflecting cone, is approximately 3.95" diameter. The two radiator surfaces are assumed to be coated with silver backed teflon, giving a solar absorptance, α, of 0.08 and
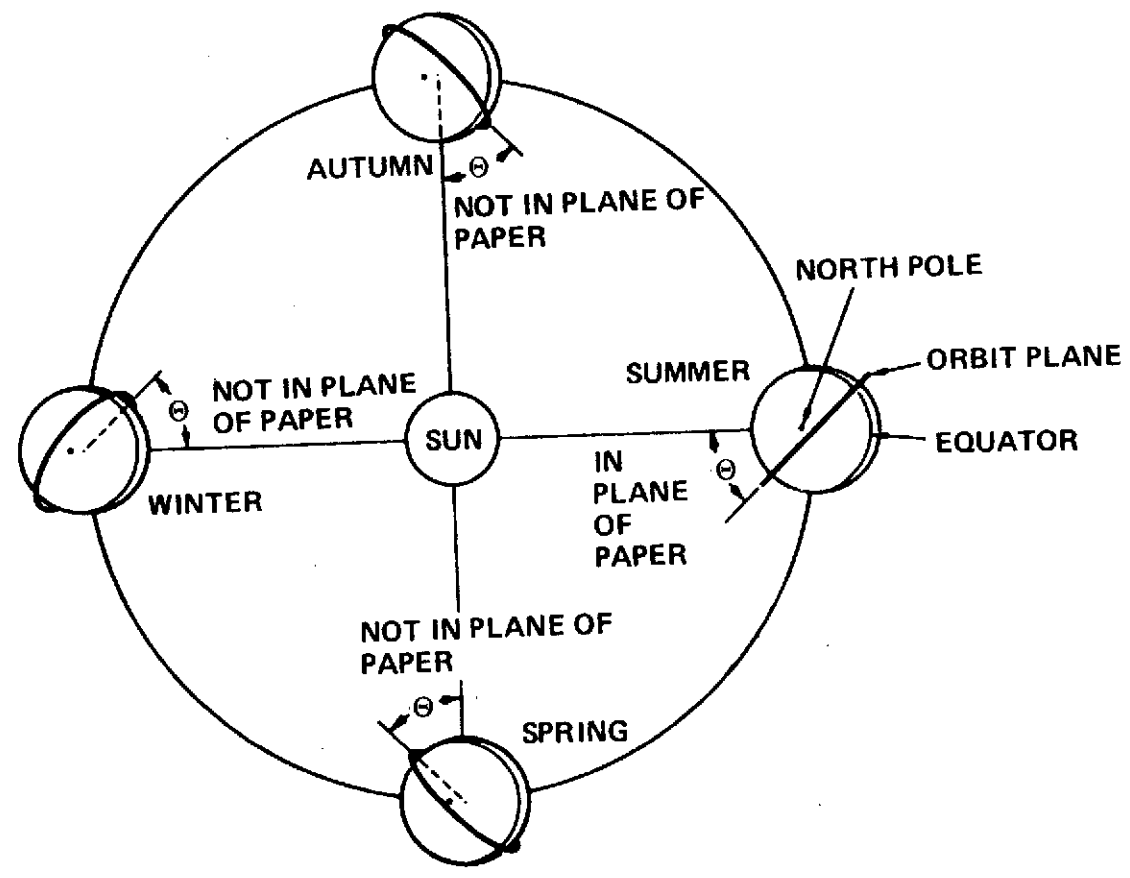
FIGURE 6-29

SUN SYNCHRONOUS ORBIT
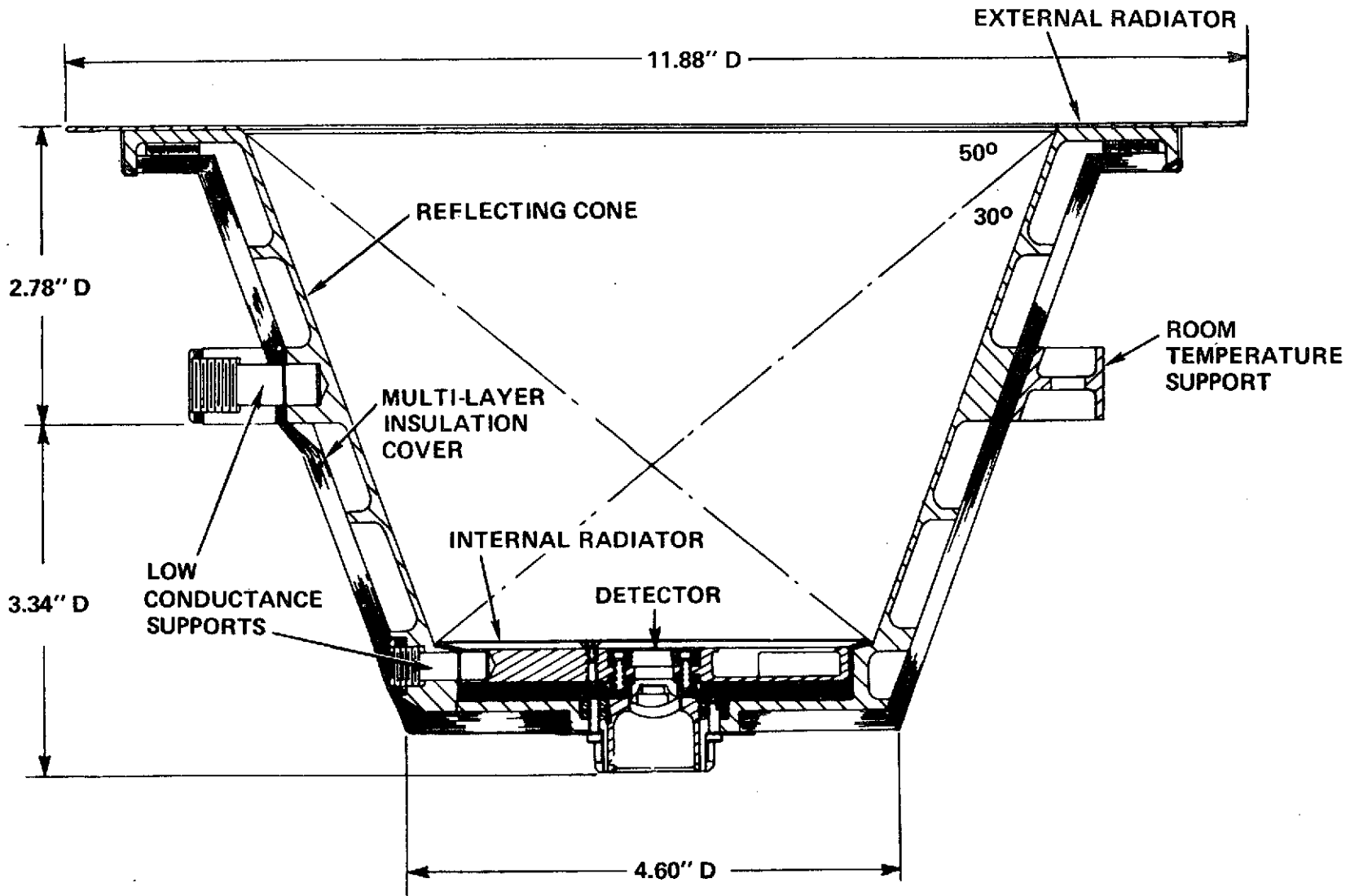
FIGURE 6-30

SHIELDED PASSIVE RADIATOR DESIGN
Arthur D. Little Design

emittance, ε, of 0.8. The reflecting cone has a low α & ε: α = 0.1, and ε = 0.02. The weight of the radiator is approximately 3.5 pounds and the system has a very small power requirement for controlling the radiator temperature.

The shielded passive radiator has a limitation on the direction it may face in orbit. There are two sources of energy which constrain its orientation. The Earth radiation requires that the maximum angle from the earth-satellite line toward the horizon, α_{max}, be 80° or less. The Solar radiation dictates that the maximum angle, γ_{max}, between the Earth-Sun line and the direction the radiator faces must be larger than approximately 50°. Figure 6-31 illustrates the allowable orientation range.

6.4.2.2   Design For Earth Oriented Satellites

Earth oriented satellites may be in an orbit with any solar inclination angle. One side always faces the earth and at least one side is always shielded from the earth's infrared radiation. A radiator placed on the shielded side will receive direct solar but no infrared. Using Optical Solar Reflectors (OSR's) very low equivalent sink temperatures can be obtained in flat panels without shielding. Such a cooler will be usable in any orbit as long as no infrared radiation falls on the surface.

Figure 6-32 presents a conceptual design of an OSR radiator. It consist of an 8.45 inch square aluminum plate, approximately 0.1 inches thick which is isolated from the spacecraft structure by low conductance mount. The surrounding spacecraft structure is coated with a low α/ε coating. The radiator panel is coated with optical solar reflectors described by Reference 18 which has an α/ε of .050/.81. A high thermal conductance cold finger connects the radiator to the cold finger detector heat load. This cold finger could be a solid bar of copper or aluminum or it could be a heat pipe. Control is achieved by an electrical resistance heater when the temperature is too cold and by a small thermoelectric cooler for the cases where the temperature is slightly high. The maximum power required for this control is 2.3 watts for the heater power. (The thermoelectric cooler requires 1 watt max). The total weight including the equivalent weight of the power is about 3.9 pounds. A summary of the design is given in Table 6-6.

6.4.3   Category VI

The design requirements for Category VI are 1000 milliwatts maximum cooling load at 300°K with a two year life. The trade study described in Section 5.0 indicated that a semi-passive radiator would be the best approach for this cooler requirement.

78

FIGURE 6-31

ORBITAL ORIENTATION RANGE FOR
PASSIVE RADIATOR

500 N. MI.
ALTITUDE

SUN

ORBIT PLANE

Θ

$\alpha_{MAX} = 80^{o}$
$\gamma_{MIN} = 50$

60°

γ_{MIN}

α_{MAX}

79

SPACECRAFT STRUCTURE
(LOW α/ϵ COATING)

3"

8.48"

LOW CONDUCTANCE
MOUNTS

RADIATOR PANEL WITH
OPTICAL SOLAR REFLECTOR
COATING

COLD FINGER TO DETECTOR
HEAT LOAD — EITHER CONDUCTING
ROD OR HEAT PIPE (INSULATED)

2.3 WATT
RESISTANCE
HEATER

THERMOELECTRIC DEVICE

TEMPERATURE
SENSOR

DETECTOR HEAT LOAD
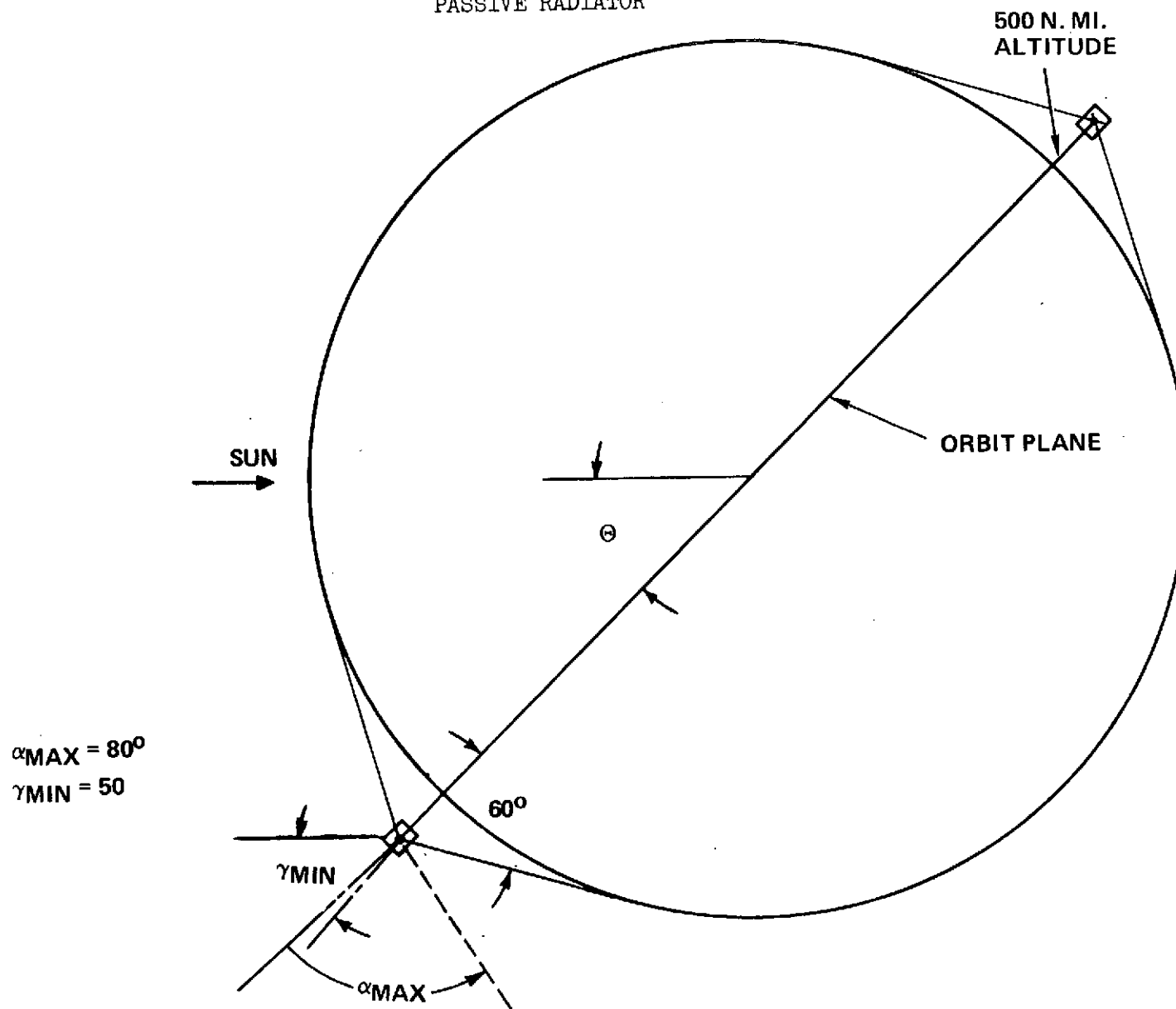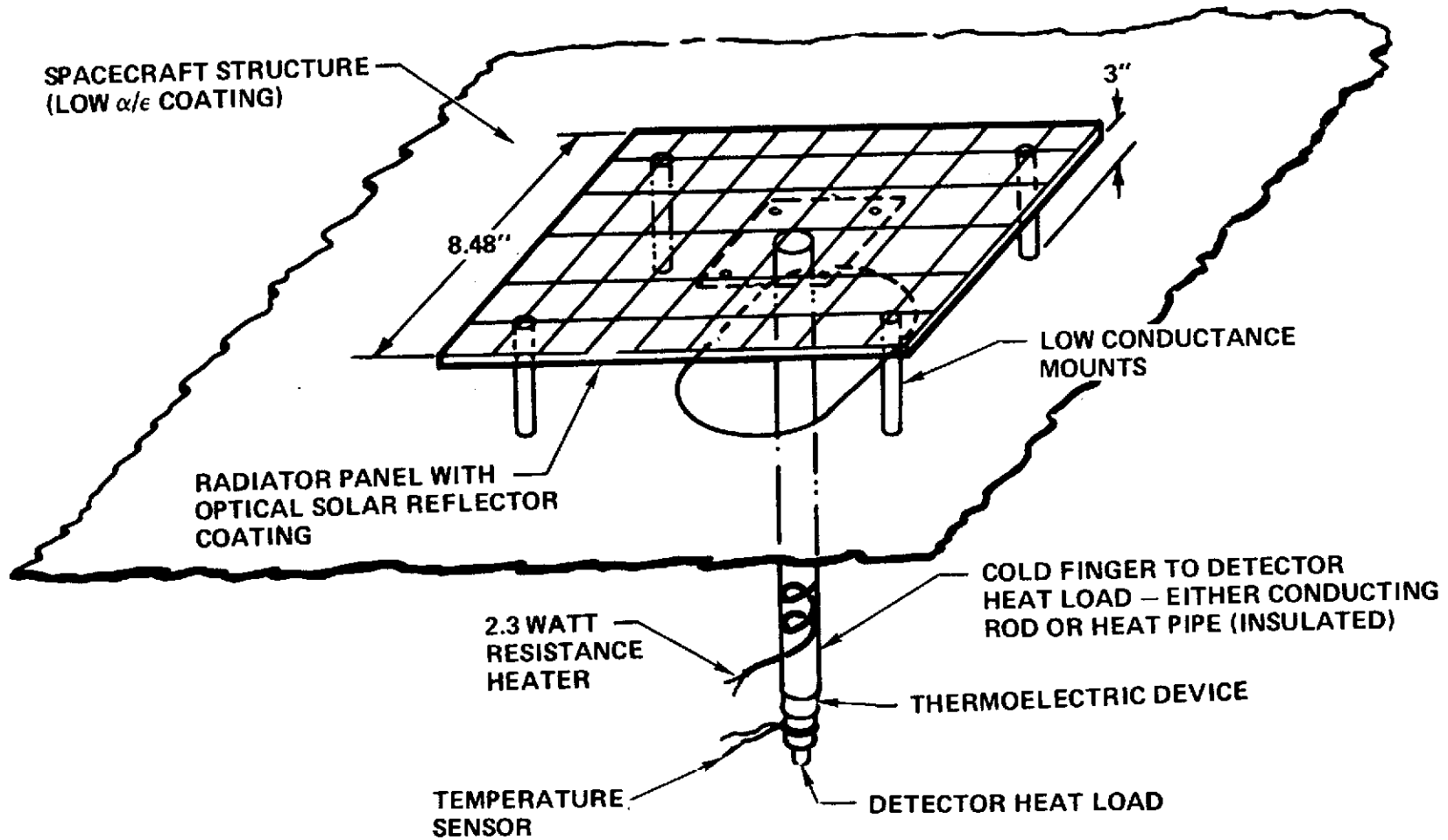
FIGURE 6-32

CONCEPTUAL DESIGN OF 500 MW 195°K
SEMI-PASSIVE
RADIATOR

TABLE 6-6

CATEGORY V CONCEPTUAL DESIGN SUMMARY
FOR SEMIPASSIVE RADIATOR

- AREA: 0.5 FT^2 (8.48" X 8.48")

- PLATE THICKNESS: 0.06 IN.

- DESIGN ENVIRONMENT:
  - MAXIMUM: FULL SUN
  - MINIMUM: NO INCIDENT HEAT

- COATING: OPTICAL SOLAR REFLECTORS ($\alpha = 0.05$; $\epsilon = 0.81$)

- HEATER/THERMOELECTRIC POWER: 2.3 WATTS MAX

- CONTROL: THERMOELECTRIC FOR COOLING (1 WATT INPUT)
  RESISTANCE HEATER FOR WARMING (2.3 WATTS INPUT)

- WEIGHT SUMMARY

— RADIATOR AND MOUNTS	.63
— COLD FINGER, HEATER, AND T/E DEVICE	1.00
— CONTROLLER (ESTIMATE)	1.00
— POWER	1.27
— TOTAL	3.90 LBS

The radiator design for this requirement is shown in Figure 6-33. It consists of a 3" square aluminum plate coated with silver backed Teflon (α = .08, ϵ = .8). The radiator plate is isolated from the spacecraft structure with low conductance mounts. Temperature control is achieved with a resistance heater and a cold finger (either solid metal or heat pipe) conducts the heat load from the detector. The maximum power required for control is approximately 3.2 watts. The total weight of the system is 2.8 pounds. The design environment assumed was that of full earth. This is the most severe requirement and thus no orientation restrictions are imposed. Table 6-7 summarizes the Category VI conceptual design.
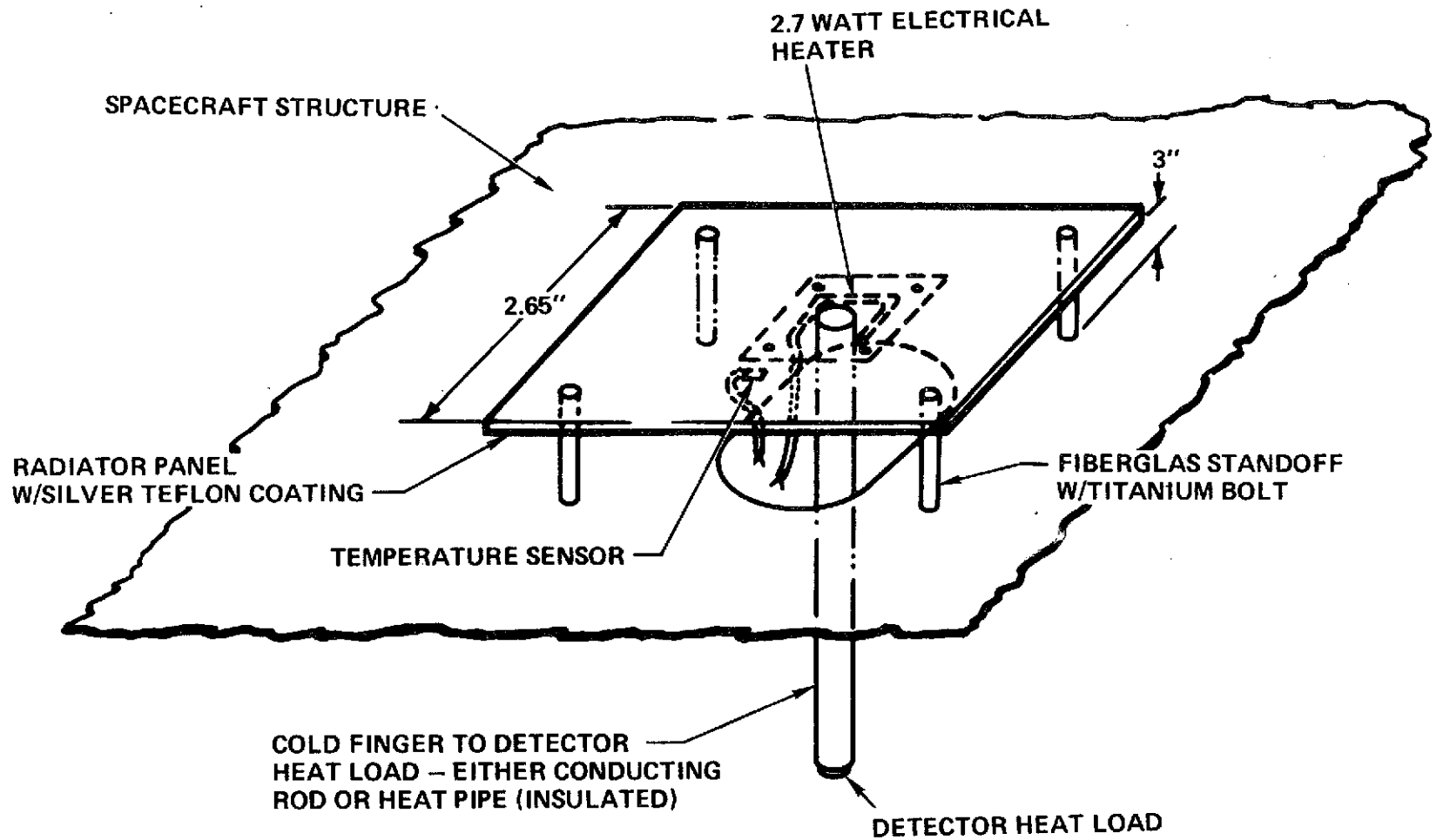
FIGURE 6-33

CONCEPTUAL DESIGN OF 1000 MW 300°K
RADIATOR

TABLE 6-7

CATEGORY VI
CONCEPTUAL DESIGN DESCRIPTION

- COOLER TYPE: SEMI PASSIVE RADIATOR
- AREA: 0.063" FT^2 (3.01" SQUARE)
- COATING: SILVER BACKED TEFLON ($\alpha = 0.08$; $\epsilon = 0.8$)
- PLATE THICKNESS: 0.06"
- APPROXIMATE HEAT LEAK THROUGH MOUNTS: 1.0 WATT
- DESIGN ENVIRONMENT: ANY ORBITAL ORIENTATION
- MOUNTS: FIBERGLAS STANDOFF WITH TITANIUM BOLT (4 MOUNTS)
- TEMPERATURE CONTROL: 2.7 WATT RESISTANCE HEATER
- MAXIMUM POWER: 3.2 WATTS
- COLD FINGER: 1/2 IN. DIA. COPPER BAR
- WEIGHT SUMMARY (LBS)
  - RADIATOR   .050
  - MOUNTS (4)   .25
  - COLD FINGER   .76
  - POWER   1.75
  - TOTAL   2.81

7.0   ADVANCED CONCEPTS

The previous sections have described conceptual designs which could be developed for a near term application, the PMS. In this section advance concepts are presented; these techniques could improve the state-of-the-art in cryogenic cooling but not necessarily in the time frame of the PMS. These concepts were generated in the course of this study and no development has been conducted. The concepts are modified J-T coolers, redundant mechanical refrigerators, combined systems and liquid cryogen storage systems. The modified J-T cooler has previously been discussed. The others are described in the following paragraphs.

7.1   Redundant Mechanical Refrigerators

Redundancy is a well established method of increasing the life and reliability of a mechanical system. For very long satellite missions maintenance is not practical; employing redundant coolers can provide the necessary life without the need for a state-of-the-art improvement. Existing designs in VM coolers can thus be employed without the need for expensive redesign and higher cost hardware.

Integrating the redundant unit into the system imposes several problems. The concept is obviously heavier and larger than non-redundant, but the primary penalties for a cryogenic refrigerator are for the power penalty and the heat rejection system. Both of those items inherently have high reliability and can be employed as common elements with two or more mechanical refrigerators minimizing the impact on weight and volume. Integrating the refrigerators with the detector requires special consideration. In a remote sensing application the detector can normally be mounted at only one location due to constraints imposed by the optical system. Several approaches are available to meet that requirement and are discsssed in the following paragraphs.

Common Thermal Contact

The simplest approach is to place both refrigerators in thermal contact with each other and the detector. No development is required and the approach is easy to accomplish with flexible conductors such as copper wire braids or heat pipes. The heat load on the active refrigerator is greatly increased by heat transfer from the inactive unit. Both units must be made larger as well as the power supply and radiator. These penalties may or may not be acceptable depending upon the magnitude of the heat leak.

85

Thermal Switch

Isolating the inactive unit requires a thermal switch to make contact with the detector. Figure 7-1 presents a concept for a thermal switch; this approach was previously mentioned in Section 6.2.5, the alternative conceptual design for Category III. The power supply and radiator can be used by either cooler. Both VM's are continuous connected to the radiator by heat pipes. As shown, VM #2 is active, when that unit fails a control system redirects power to VM #1 and the solenoid actuator. The lever arm moves breaking contact with VM #2 and making contact with VM #1. To maintain the switch position with good thermal contact many options are available. Some are as follows:

- o Continuous power on the solenoid
- o A motor and screw drive
- o Springs and latching mechanism

Development is needed to select and build a good cryogenic thermal switch; the probability of success is very good. The concept also has the problem of increased heat load on the refrigerator (and hence size) but the impact is anticipated to be much smaller than the common thermal contact approach.

Optical Means

The redundant refrigerator may also be equiped with a second detector; and both isolated from the primary units. Two independent units are thus provided. A means of directing the light signal from the item being sensed to the redundant detector is needed. A mirror or mirrors which are moved into the light path to redirect the signal can be used for this purpose; then only a minor change in the optical system is needed. Preflight, in place calibration of both detectors and cooler sets can resolve the small differences in response characteristics. Optical means of switching is not as general as the other two since a good location for the switch mirror may not be available in some optical systems. In addition optical switching may be more expensive for a high quality mounting mirror and a second detector.

Redundant cryogenic refrigerator will be needed only on applications with long durations. Current technology in VM coolers has demonstrated the capability for long life, but this conclusion was based on an extrapolation of partial data. Two to 5 year life with a complete system has not yet been proven. To minimize the risk of a failure, redundant units can be employed. The impact

VUILLEUMIER
COOLERS

#1 #2

HEAT PIPES

SOLENOID
ACTUATOR

THERMAL
SWITCH

DETECTOR

FLEXIBLE THERMAL CONDUCTOR

RADIATOR
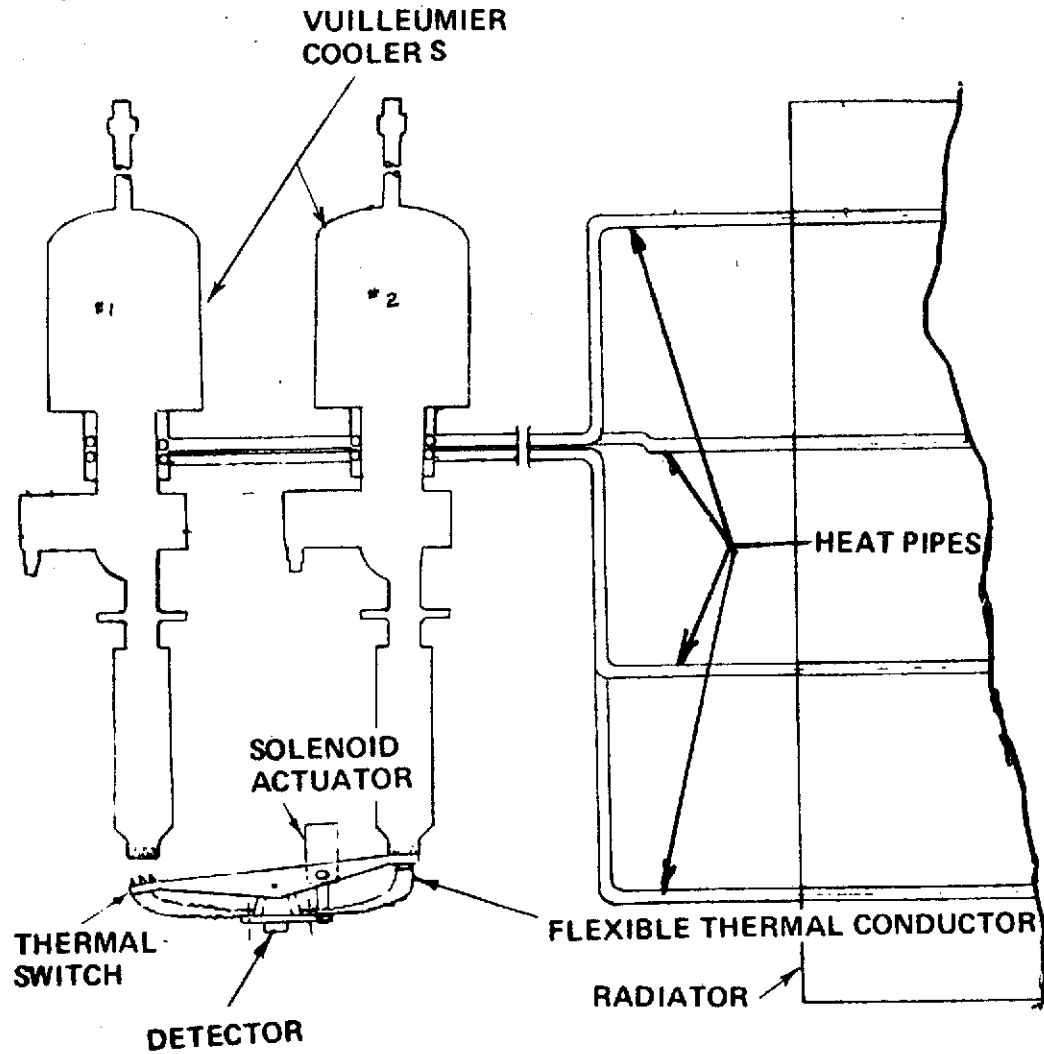
FIGURE 7-1

REDUNDANT COOLERS WITH THERMAL SWITCH

on weight is anticipated to be on the order of a 30% increase over a non-redundant design (based on Category III requirements). Since most applications will not require redundancy, that development may be a less expensive approach (in both development and recurring costs) than developing a single machine for long life.

7.2   Combined Systems

Previous developments in cryogenic coolers have concentrated on a single type of cooler. Combining different cooler types in one cooler can have weight and volume advantages; potentially costs can also be saved.

Figure 7-2 presents an example of a combined system employing a T/E and a solid cryogen. The T/E cooler maintains an inner shield at 170°K instead of 1st stage solid cryogen. The interior of the shield is a CH_4 solid cryogen cooler as described previously. The exterior of the tank is a vacuum jacket to obtain a low pressure for good performance of the insulation. The outer jacket also serves as the radiator to reject heat inside the spacecraft. No controls are needed since temperature tolerances are not critical. Assuming a 6 stage Borg Warner (see Appendix B), approximately 50 watts of power are required; the total system equivalent weight including the power penalty is about 60 to 70 lbs for a Category I application (2 years, 250 watt/hr at 77°K). This weight compares to approximately 100 lbs for a pure solid cryogen (CH_4/NH_3) system for the same application.

Passive radiators could also be used in a solid cryogen system. When the orientation of the spacecraft permits, the passive radiator may be lighter than T/E since no power is required.

Combined systems are not limited to solid cryogens; the modified J-T previously discussed is a combined system. Another alternative is the use of passive radiators in lieu of T/E in the modified J-T. Previously mentioned in the use of passive radiators is the use of T/E to provide temperature control and limited refrigeration in orbits with relative warm sink temperatures. Other approaches are also possible; but additional work is needed to prove the validity of these concepts.

7.3   Liquid Cryogen Storage Systems

Normal liquid cryogen systems require both storage and use of the cryogen at the same temperature. With such a system there is a continuous loss of the cryogen due to heat leak into the storage container. That problem
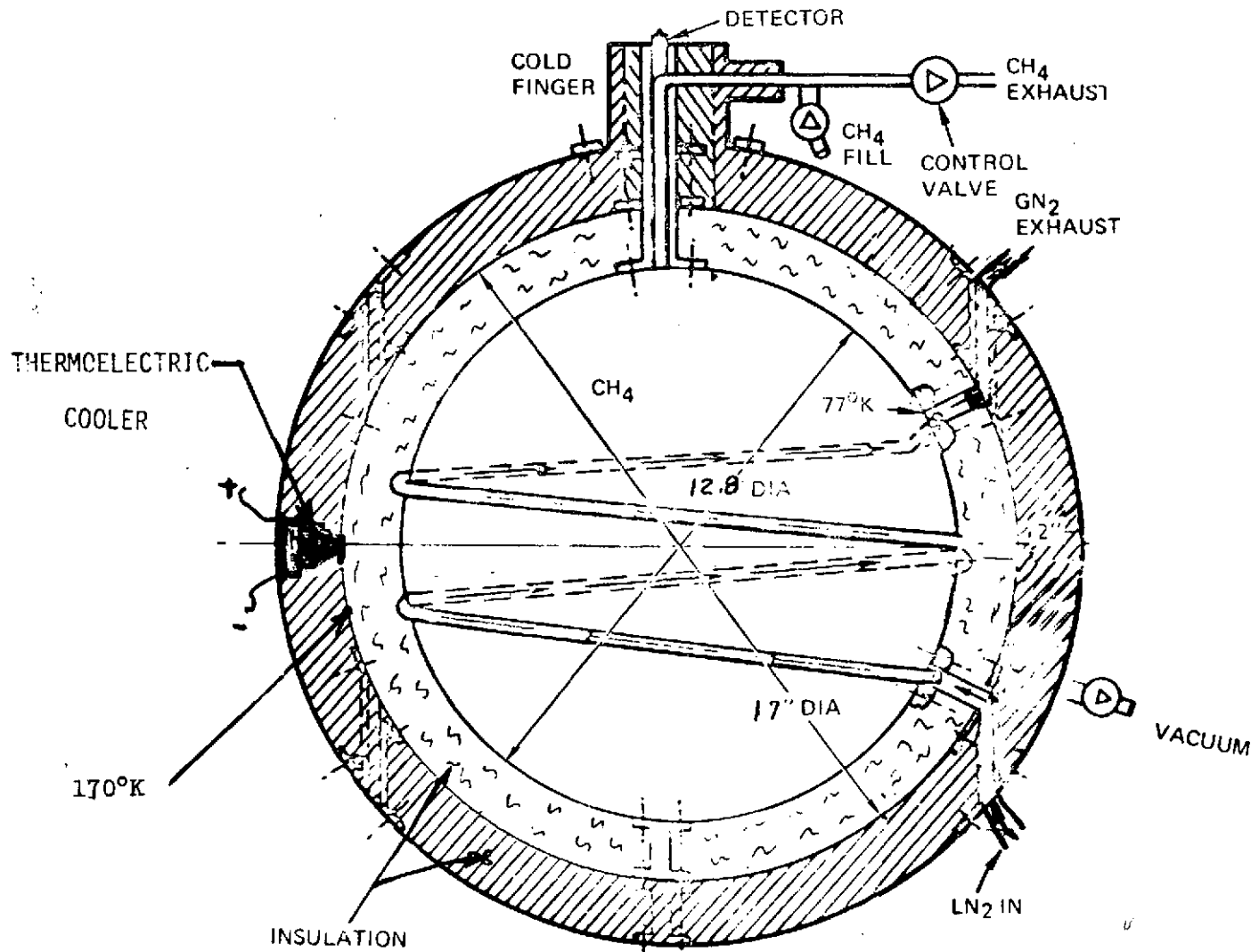
FIGURE 7-2

CONCEPTUAL DESIGN OF A 50 MW SOLID CRYOGEN T/E COOLER

can be minimized by storing the cryogen at a high temperature to reduce
the rate of heat transfer, and then flashing the liquid to low pressure and
temperature will produce the desired cooling. Figure 7-3 presents a schematic
of such a concept. That concept employs NH_3 stored at room temperature. The
NH_3 is withdrawn from the storage tank and flashed in a miniature flash evaporator.
The evaporation of the liquid NH_3 maintains an insulated methane tank at
150°K. The CH_4 is stored at high pressure so that no loss occurs during storage.
Liquid is withdrawn and flashed at low pressure, cooling a detector at 77°K.

The cooling effect per unit mass and volume of expendable are smaller
that using the same cryogens as solids operating at the same conditions. For
CH_4 and NH_3 the cooling effects are as follows:

	CH₄ @ 77°K		NH₃ @ 150°K	
	Δh (Watt-hr/lb)	ρΔh (Watt-hr/ft³)	Δh (Watt-hr/lb)	ρΔh (Watt-hr/ft³)
Solid	77.3 (77°K Storage)	2505	229 (150°K Storage)	11427
Liquid	35 (150°K Storage)	906	112 (300°K Storage)	4166

The solids have more than twice the cooling effect of the liquids for both
cryogens. However because the storage temperatures are lower with the solids,
the heat leakage is also larger. The latter effect can be significantly larger
than the detector heat load, and a liquid system can be smaller than a solid
cryogen system. Figure 7-3 presents weight and volume data for a liquid system
sized for Category I requirements (two years, 250 watt-hr of cooling at 77°K);
the estimated weight is only 58 lbs compared to 100 lbs for a solid system.
The volume is even smaller for the liquid system, 15" dia. vs 21" dia spheres
for CH_4 and 14" dia vs 18" dia spheres for NH_3.

The miniature flash evaporator is the only major development item
necessary to achieve the advantages. The major problem is to flash a liquid
at high pressure to a very low pressure (order of 0.2 psia to 0.1 mmHg) with
solid being formed at the low pressure. Preventing loss of the solid prior
to its sublimation is anticipated to be a major problem; but based on VSD's
experience with large flash evaporators (see Ref. 19) the problem is solvable.
The expansion valve may be either a simple orifice with an upstream control valve
or the same type of valve as used in a J-T cryostat. The remainder of the system

C - 2
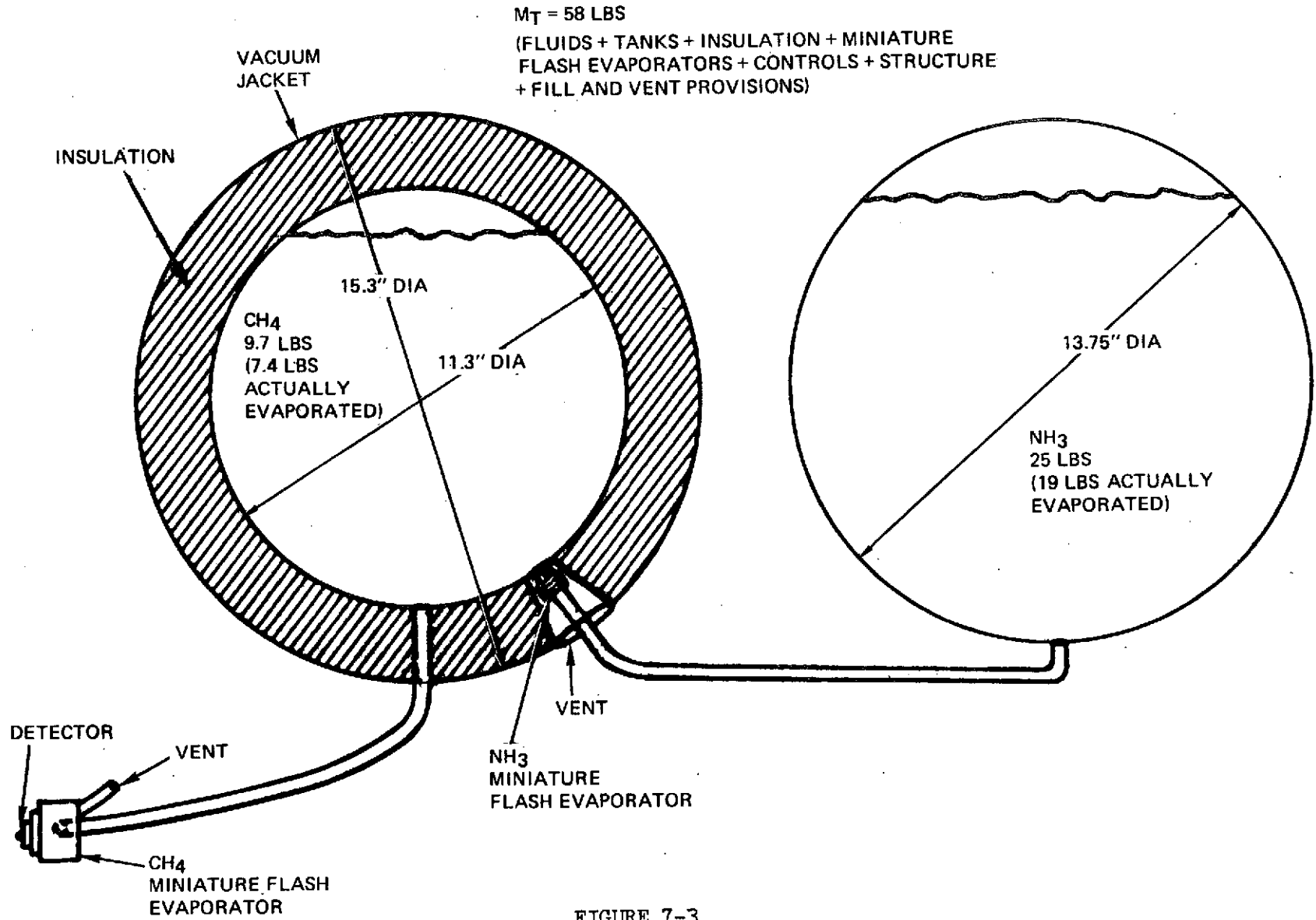
FIGURE 7-3

LIQUID CRYOGEN STORAGE SYSTEM

employs state-of-the-art zero "g" liquid storage tanks with minimal technical risks.

8.0   CONCLUSIONS

Cryogenic coolers will be available within the time period to satisfy all requirements considered herein for satellite remote sensing instrumentation. The state-of-the-art and on-going developments are sufficient to meet the needs of all identified categories. Joule-Thomson and thermoelectric coolers, and radiators have all been developed. The Joule-Thomson coolers are available in standardized units at relatively low cost. Thermoelectric coolers are also available in standardized size in the commercial materials. The more advanced thermoelectric materials can produce significant advantages and cooler should be available, although at higher cost than the commercial materials. Low temperature passive radiators have been used in spaceflight. These radiators are available in either shielded (for very low temperatures) and unshielded designs (T $>$195°K). Neither design requires development but the shielded radiators are more expensive and larger in volume.

The Vuilleumier cooler, solid cryogen systems and advanced thermoelectric materials are not yet fully developed. Research is being conducted by NASA-Goddard, the Army Night Vision Laboratory at Ft. Belvoir, and the Air Force Flight Dynamics Lab. The activities of these government agencies will assure the basic technology base. Production of the required hardware is feasible within the specified time herein considered. Some expenditure of development effort can be expected for the requirements of a specific application since standardized hardware will not be available.

The selection of a cooler for a given application is strongly dependent upon the duration or life of the mission, the temperature and required cooling capacity. For the sensor requirements as defined herein, the following coolers are most promising for the indicated temperature range:

	NON-EXPENDABLE	EXPENDABLE
77°K	Vuilleumier Stirling	Joule-Thomson Solid Cryogens
110 to 140°K	Vuilleumier Shielded Radiator	Liquid Cryogens Solid Cryogens
195°K	Thermoelectrics Passive Radiators – Shielded – Unshielded	–
243°K	Passive Radiators	–
300°K	Passive Radiators	–

The expendable systems are best suited to missions of one year or less, but do not require heat rejection to space. The VM, Stirling, and thermoelectric refrigerators require a radiator in addition to the cooler. These radiators typically reject heat at about 300°K and do not impose any orbit or orientation restrictions upon the spacecraft. Passive radiators at 110-140°K, 195°K, and 243°K can only be employed with orientation constraints.

Six categories were defined to obtain representative designs of cryogenic coolers. The categories and coolers selected for conceptual design were as follows:

I   77°K, 50 MW, 2 years as continuous for one month followed by 5 to 8 selected days per month (175 to 257 watt-hous) (Joule-Thomson, and Solid Cryogen)

II   77°K, 1000 MW, 2 years as continuous for one month followed by 10 days continuous per 6 week period (4731 watt-hours) (Vuilleumier)

III 77°K, 300 MW, 2 years continuous for the entire period (5260 watt-hours) (Vuilleumier)

IV   195°K, 50 MW, 2 years as continuous for one month followed by 5 to 8 selected days per month (Thermoelectrics)

V   195°K, 500 MW, 2 years as continuous for the entire period (Shielded Radiator and Passive Unshielded Radiator)

VI   300°K, 1000 MW, 2 years as continuous for the entire period (Passive Unshielded Unrestricted Radiator)

The selections were not necessarily optimum for each category. The trade-off is strongly dependent upon the constraints of the mission and the relative importance of the constraints, weight, volume, launch costs, and equipment costs. Trade-off data for weight and volume have been included with each cooler type. The other factors are mission dependent and must be defined by the person making the choice for flight hardware.

9.0   RECOMMENDATIONS

The basic technology needed for satellite remote sensing applications is available. However, in some systems standardized hardware will not be available by 1976 and planning for a limited amount of development of hardware is recommended. The specific coolers where development is needed are VM's, solid cryogens, T/E's, and shielded passive radiators (to a much less degree).

Significant advances in cryogenic technology can be made to reduce cost and/or weight and volume. The recommended areas for development are:

1) Modified Joule-Thomson Coolers

2) Redundant Mechanical Refrigerators

3) Combined Coolers

. Thermoelectric-Solid Cryogens

. Radiator-Solid Cryogens

4) Liquid Cryogens with Miniature Flash Evaporator

The modified Joule-Thomson device is a straightforward application of two well known coolers and only a demonstration is needed. Redundant mechanical refrigerators may or may not need a thermal switch; a study to evaluate the potential and a demonstration are recommended. Combined coolers employ existing technology, but additional study on integration problems is recommended. Liquid cryogens could reduce weight and volume over solid cryogen systems for long duration and reduce cost by standardized and cheaper designs. The development of a miniature flash evaporator is needed to obtain those advantages.

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APPENDIX A

LITERATURE SURVEY DOCUMENTS

A listing of the documents obtained during the literature search is given in this Appendix. The documents are categorized by cooler types.

A.   Claude and Reverse Brayton Cycles

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APPENDIX B

CHARACTERISTICS OF EXISTING SPACE APPLICABLE CRYOGENIC COOLERS

This Appendix presents the results of an industry survey conducted during the early portion of the Cryogenic Cooler Study. The survey included cooling systems with a cooling capacity in the range of 0.05 watts to 5 watts at temperatures of 60°K to 300°K. Primary emphasis was placed on systems with potential for long life (1 to 2 years) operation in the environment of space.

The results are tabulated in Tables B-1 thru B-6 as summarized below:

COOLING TEMPERATURE	COOLER TYPE	TABLE NO.
77°K	Vuilleumier	B-1
"	Stirling	B-2
"	Solvay	B-3
"	Joule-Thomson	B-4
195°K	Thermoelectric	B-5
"	Passive Radiators	B-6

TABLE B-1

VUILLEUMIER - SPACE APPLICABLE

		HUGHES	HUGHES	HUGHES	HUGHES	AIRESEARCH	AIRESEARCH	AIRESEARCH
1.	MANUFACTURER	HUGHES	HUGHES	HUGHES	HUGHES	VM Cooler	VM Cooler	VM Cooler
2.	SYSTEM DESCRIPTION							
	TRADE NAME	2 Stage VM	Modular VM	Modular VM	Modular VM	Fractional Watt Cooler	Long Life, 5 W Cooler	IR&D 3 W Cooler
	MODEL							
	I.D. NUMBER	15 VM-A						
	OPERATING TEMP.	13.5/60°K	77°K	77°K	77°K	65°K	75°K	77°K
	TYPE CYCLE	VM				VM	VM	VM
	WORKING FLUID	Helium				Helium	Helium	Helium
	LUBRICATION					Dry	Dry	Dry
	POWER REQMTS							
	LOAD	400 W	80W	150W	850W	80W	270 W	350 W
	VOLTS/PHASE/FREQ							
	HEAT SINK REQMTS							
	MEDIUM	Liquid				Liquid or heat pipe	Liquid or Heat Pipe	Liquid
	TEMPERATURE	325°K				335°K	335°K	335°K
	STATE OF DEVELOPMENT	Successfully flown				In Design	Pre-production proto built	Lab unit built
	APPLICABILITY TO SATELLITE COOLING	Directly Applicable						
3.	OPERATIONAL FEATURES							
	HIGH PRESSURE							
	LOW PRESSURE							
	MINIMUM TEMP.	13.5°K/60°K	77°K	77°K	77°K	65°K	75°K	77°K
	COOL DOWN TIME		2 min.	3.3 min.	7 min.	15 min.	30 min.	30 min.
	ATTITUDE REQUIREMENTS	None				None	None	None
4.	PHYSICAL CHARACTERISTICS							
	COST							
	RELIABILITY					17,500 hr (2 yr)	2 yr design/ 5 yr goal	2500 hr design/5000hr goal
	MAINTENANCE INTERVAL	3600 hr.					proto:40 lb; flight:20lb	
	FIXED WEIGHT - EXPANDER	19.5 lb				16.5 lb.		
	- COMPRESSOR							
	- RADIATOR	0						
	EXPENDABLE WEIGHT							
	VOLUME	310 in³(7.25"D x 12.5" comp)(0.9"D x 8" comp finger)	3"x3-1/2"x4" Comp .18"D x 1.13" Cold Finger	3" x 3-1/2" x 4" Comp. .3"D x 1-1/2" ColdFinger	12-1/2x10-1/2x7 Comp .85"D x 3.6" Cold Finger	846 in³(23.5" x6"x6")	11.52 in³(18"x8"x8")	864 in³ (18"x8"x6")
5.	CYCLE PARAMETERS	Stage 1 / Stage 2			Stage 1 / Stage 2			
	COLD PLATE TEMP.	60°K / 13.5°K	77°K	77°K	77°K / 25°K	65°K	75°K	77°K
	REFRIGERATION CAP.	3W / .15W	0.5W	2.0W	2.5W / 1W	.25W	5 W	3 W
	POWER INPUT	400 W				80W	270 W	350 W
	COP	.0075				.003125	.0185	.00857
	% CARNOT EFF.					66	Proto: 8; Flight: 4	
	LBS./WATT OUTPUT	6.1g					230	288
	IN³/WATT OUTPUT	98.4				3384		
6.	GOVN. CONTRACT NR.							
	GOVN. CONTACT/LOCATION							
7.	COMMENTS							

B-2

STIRLING CYCLE - SPACE APPLICABLE

1.	MANUFACTURER	North American Phillips Corp.

2.	SYSTEM DESCRIPTION
	  TRADE NAME
	  MODEL
	  I.D. NUMBER
	  OPERATING TEMP.
	  TYPE CYCLE
	  WORKING FLUID
	  LUBRICATION
	  POWER REQMTS
	    LOAD
	    VOLTS/PHASE/FREQ
	  HEAT SINK REQMTS
	    MEDIUM
	    TEMPERATURE
	  STATE OF DEVELOPMENT
	  APPLICABILITY TO SATELLITE COOLING
		$77°K$
		Stirling
		Helium
		Dry
		
		30 Watts
		
		$125°F$
		Developed
		Good

3.	OPERATIONAL FEATURES
	  HIGH PRESSURE
	  LOW PRESSURE
	  MINIMUM TEMP.
	  COOL DOWN TIME
	  ATTITUDE REQUIREMENTS
		$77°K$
		5 min.
		None

4.	PHYSICAL CHARACTERISTICS
	  COST
	  RELIABILITY
	  MAINTENANCE INTERVAL
	  FIXED WEIGHT - EXPANDER
	           - COMPRESSOR
	           - RADIATOR
	  EXPENDABLE WEIGHT
	  VOLUME
		8000 hrs.
		
		10 hrs.
		
		290.7 in.^{3}

5.	CYCLE PARAMETERS
	  COLD PLATE TEMP.
	  REFRIGERATION CAP.
	  POWER INPUT
	  COP
	  % CARNOT EFF.
	  LBS./WATT OUTPUT
	  IN^3/WATT OUTPUT
		$77°K$
		1 W
		30 W
		.0333
		10
		290.7

6.	GOVN. CONTRACT NR. GOVN. CONTACT/LOCATION	Applied Physics Lab
7.	COMMENTS	1. Rhombic drive

SOLVAY CYCLE - SPACE APPLICABLE

1.	MANUFACTURER	Air Products and Chemicals, Inc.
2.	SYSTEM DESCRIPTION TRADE NAME MODEL I.D. NUMBER OPERATING TEMP. TYPE CYCLE WORKING FLUID LUBRICATION POWER REQMTS LOAD VOLTS/PHASE/FREQ HEAT SINK REQMTS MEDIUM TEMPERATURE STATE OF DEVELOPMENT APPLICABILITY TO SATELLITE COOLING	Displex MS-1003 77°K Solvay Dry Lubrication 560 watts 115/3/400 Water or Air 135°F Developed Marginal
3.	OPERATIONAL FEATURES HIGH PRESSURE LOW PRESSURE MINIMUM TEMP. COOL DOWN TIME ATTITUDE REQUIREMENTS	 77°K 5 min. None
4.	PHYSICAL CHARACTERISTICS COST RELIABILITY MAINTENANCE INTERVAL FIXED WEIGHT - EXPANDER - COMPRESSOR - RADIATOR EXPENDABLE WEIGHT VOLUME	 3100 hr MTBF 1200 hrs. 0.5 lbs 14.5 lbs 1.3"x1.3"x4" Expander 1.75"x1.75"x5.75" Valve 7.25"x5.43"x8.56" Compr.
5.	CYCLE PARAMETERS COLD PLATE TEMP. REFRIGERATION CAP. POWER INPUT COP % CARNOT EFF. LBS./WATT OUTPUT IN³/WATT OUTPUT	 77°K 1 watt 560 watts .0018 15 361.4
6.	GOVN. CONTRACT NR. GOVN. CONTACT/LOCATION	
7.	COMMENTS	1200 hr maintenance interval is limiting factor for space application. Compr. seals & bears and expander valve, disc, & displacer seal control life.

		Air Products and Chemicals, Inc.	Air Products and Chemicals, Inc.	Air Products and Chemicals, Inc.	Air Products and Chemicals, Inc.	Air Products and Chemicals, Inc.
1.	MANUFACTURER					
2.	SYSTEM DESCRIPTION					
	TRADE NAME	Variflow Cryostat	Variflow Cryostat	Variflow Cryostat	Variflow Cryostat	Cryo-Tip
	MODEL	VF-2020A	VF-2020A	VF-2020A	VF-2020A	AC-1
	I.D. NUMBER					
	OPERATING TEMP.	77°K	80°K	145°K	196°K	68°K
	TYPE CYCLE	J-T	J-T	J-T	J-T	J-T
	WORKING FLUID	Nitrogen	Argon	Halocarbon 14	Halocarbon 13	Nitrogen
	LUBRICATION	None	None	None	None	None
	POWER REQMTS					
	LOAD					.85 CFM
	VOLTS/PHASE/FREQ	3-6000 psi supply	3-6000 psi supply	3-6000 psi supply	3-6000 psi supply	
	HEAT SINK REQMTS					
	MEDIUM	None	None	None	None	None
	TEMPERATURE					
	STATE OF DEVELOPMENT	Developed	Developed	Developed	Developed	Developed
	APPLICABILITY TO SATELLITE COOLING	Excellent	Excellent	Excellent	Excellent	Excellent
3.	OPERATIONAL FEATURES					
	HIGH PRESSURE	6000 psi	6000 psi	6000 psi	6000 psi	
	LOW PRESSURE					
	MINIMUM TEMP.	77°K	80°K	145°K	196°K	68°K
	COOL DOWN TIME					5 min.
	ATTITUDE REQUIREMENTS					None
4.	PHYSICAL CHARACTERISTICS					
	COST					
	RELIABILITY					
	MAINTENANCE INTERVAL					
	FIXED WEIGHT - EXPANDER					
	- COMPRESSOR					
	- RADIATOR					
	EXPENDABLE WEIGHT					
	VOLUME	.25" diam. x 2" long	.25" diam. x 2" long	.25" diam. x 2" long		
5.	CYCLE PARAMETERS					
	COLD PLATE TEMP.	77°K	80°K	145°K	196°K	80°K
	REFRIGERATION CAP.	200 m.w.	200 m.w.	200 m.w.	200 m.w.	7 watts
	POWER INPUT					
	COP					
	% CARNOT EFF.					
	LBS./WATT OUTPUT					
	IN^3/WATT OUTPUT					
6.	GOVN. CONTRACT NR.					
	GOVN. CONTACT/LOCATION					
7.	COMMENTS					

1.	MANUFACTURER	Air Products and Chemicals, Inc.	Air Products and Chemicals, Inc.	Bendix	Bendix	Bendix	Welch Scientific C.
2.	SYSTEM DESCRIPTION						
	TRADE NAME	Cryo-Tip	Heli-Tran				
	MODEL	AC-2	LT-3-110	3271011-0101	3271003-0101	3271003-0201	3150
	I.D. NUMBER						
	OPERATING TEMP.	16°K	2°K	87°K	77°K	77°K	100°K
	TYPE CYCLE	J-T	J-T	J-T	J-T	J-T	J-T
	WORKING FLUID	Nitrogen/Hydrogen	Helium/Nitrogen	Argon	Nitrogen	Nitrogen	Air
	LUBRICATION	None	None	None	None	None	None
	POWER REQMTS						
	LOAD	.95 CFM/ .85 CFM	.75 1/hr	6000 psi supply	6000 psi supply	6000 psi supply	2.5 CFM @ 85 psi
	VOLTS/PHASE/FREQ						
	HEAT SINK REQMTS						
	MEDIUM	None	None	None	None	None	None
	TEMPERATURE						
	STATE OF DEVELOPMENT	Developed	Developed	Developed	Developed	Developed	Developed
	APPLICABILITY TO SATELLITE COOLING	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent
3.	OPERATIONAL FEATURES						
	HIGH PRESSURE						
	LOW PRESSURE						
	MINIMUM TEMP.	16°K	4.2°K	87°K	77°K	77°K	130°K
	COOL DOWN TIME	10 min.	10 min.				
	ATTITUDE REQUIREMENTS	None	None	None	None	None	None
4.	PHYSICAL CHARACTERISTICS						
	COST						
	RELIABILITY						
	MAINTENANCE INTERVAL						
	FIXED WEIGHT - EXPANDER						12 lbs.
	- COMPRESSOR						
	- RADIATOR						
	EXPENDABLE WEIGHT						
	VOLUME						
5.	CYCLE PARAMETERS						
	COLD PLATE TEMP.	23°K	4.2°K 20°K 50°K	87°K	77°K	77°K	100°K 110°K
	REFRIGERATION CAP.	4 watts	1.5W 8W 20W	1 W	1 W	1 W	0 W 2 W
	POWER INPUT						
	COP						
	% CARNOT EFF.						
	LBS./WATT OUTPUT						
	IN³/WATT OUTPUT						
6.	GOVN. CONTRACT NR.						
	GOVN. CONTACT/LOCATION						
7.	COMMENTS						

TABLE B-5

THERMOELECTRICS - SPACE APPLICABLE

1.	MANUFACTURER	Marlow Industries	Marlow Industries	Marlow Industries	Nuclear System, Inc.	Nuclear Systems, Inc.
2.	SYSTEM DESCRIPTION					
	TRADE NAME					
	MODEL	M1-6040	M1-6040	M1-6040	LT-700	LT-700
	I.D. NUMBER				6304-01	6304-01
	OPERATING TEMP.	170°K	170°K	170°	165°K	165°K
	TYPE CYCLE	6 Stage Thermoelectric	6 Stage Thermoelectric	6 Stage Thermoelectric	6 Stage Thermoelectric	6 Stage Thermoelectric
	WORKING FLUID					
	LUBRICATION					
	POWER REQMTS					
	LOAD	30 watts	12 watts	8.2 watts	26.4 watts	13.5 watts
	VOLTS/PHASE/FREQ	6 VDC	4 VDC	3.4 VDC	6 VDC	3 VDC
	HEAT SINK REQMTS					
	MEDIUM	Radiator or HX	Radiator or HX	Radiator or HX	Radiator or HX	Radiator or HX
	TEMPERATURE	300°K	300°K	300°K	300°K	300°K
	STATE OF DEVELOPMENT	Developed	Developed	Developed	Developed	Developed
	APPLICABILITY TO SATELLITE COOLING	Excellent	Excellent	Excellent	Excellent	Excellent
3.	OPERATIONAL FEATURES					
	HIGH PRESSURE					
	LOW PRESSURE					
	MINIMUM TEMP.	160°K	177°K	190°K	160°K	174°K
	COOL DOWN TIME					
	ATTITUDE REQUIREMENTS	None	None	None	None	None
4.	PHYSICAL CHARACTERISTICS					
	COST					
	RELIABILITY					
	MAINTENANCE INTERVAL					
	FIXED WEIGHT - EXPANDER					
	- COMPRESSOR					
	- RADIATOR					
	EXPENDABLE WEIGHT					
	VOLUME	0.414 in³	0.414 in³	0.414 in³	0.5 in³	0.5 in³

5.	CYCLE PARAMETERS																
	COLD PLATE TEMP.	170°K	180°K	195°K	186°K	190°K	195°K	190°K	195°K	205°K	173°K	181°K	195°K	180°K	190°K	195°K
	REFRIGERATION CAP.	.05W	0.1W	0.18W	.05W	.07W	.098W	0W	.05W	.087W	.05W	.10W	.19W	.05W	.085W	.141W
	POWER INPUT	30W	30W	30W	12W	12W	12W	8.16W	8.16W	8.16W	24.6W	24.6W	24.6W	13.5W	13.5W	13.5W
	COP	.00167	.00333	.00667	.0042	.0058	.0096	0	.00613	.011	.0020	.0041	.0077	.0037	.0063	.010
	% CARNOT EFF.															
	LBS./WATT OUTPUT										.56	.28	.147	.56	.329	.198
	IN³/WATT OUTPUT	8.28	4.14	2.07	8.28	5.91	4.22		8.28	4.76	10.0	5.0	2.63	10.0	5.88	3.55

6.	GOVN. CONTRACT NR.					
	GOVN. CONTACT/LOCATION					
7.	COMMENTS					

THERMOELECTRICS - SPACE APPLICABLE

1.	MANUFACTURER	Nuclear Systems, Inc.	Borg-Warner	Borg-Warner	Borg-Warner

2.	SYSTEM DESCRIPTION
	  TRADE NAME
	  MODEL
	  I.D. NUMBER
	  OPERATING TEMP.
	  TYPE CYCLE
	  WORKING FLUID
	  LUBRICATION
	  POWER REQMTS
	    LOAD
	    VOLTS/PHASE/FREQ
	  HEAT SINK REQMTS
	    MEDIUM
	    TEMPERATURE
	  STATE OF DEVELOPMENT
	  APPLICABILITY TO SATELLITE COOLING
		6304-01
		165°K
		6 Stage Thermoelectric
		
		6.4 Watts
		3.2 VDC
		
		Radiator or HX
		300°K
		Developed
		Excellent
			193°K
			4 Stage Thermoelectric
			
			2 Watts
			6 VDC
			
			Radiator or HX
			300°K
			Developed
			Excellent
				170°K
				6 Stage Thermoelectric
				
				10.3 Watts
				6.3 VDC
				
				Radiator or HX
				300°K
				Developed
				Excellent
					145°K
					8 Stage Thermoelectric
					
					39 Watts
					
					Radiator or HX
					325°K
					Developed
					Excellent

3.	OPERATIONAL FEATURES
	  HIGH PRESSURE
	  LOW PRESSURE
	  MINIMUM TEMP.
	  COOL DOWN TIME
	  ATTITUDE REQUIREMENTS
		189°K
		
		None
			182°K
			
			None
				165°K
				
				None
					145°K
					
					None

4.	PHYSICAL CHARACTERISTICS
	  COST
	  RELIABILITY
	  MAINTENANCE INTERVAL
	  FIXED WEIGHT - EXPANDER
	           - COMPRESSOR
	           - RADIATOR
	  EXPENDABLE WEIGHT
	  VOLUME
		0.5 in^3	0.2 in^3
			.4" x .67" x .75"
				1.25 in^3
				1" x 1" x 1.25"
					2.55 in^3
					1.35" x 1.35" x 1.4"

5.	CYCLE PARAMETERS
	  COLD PLATE TEMP.
	  REFRIGERATION CAP.
	  POWER INPUT
	  COP
	  % CARNOT EFF.
	  LBS./WATT OUTPUT
	  IN^3/WATT OUTPUT
		189°K 195°K 205°K
		0W .05W .11W
		6.4W 6.4W 6.4W
		0 .0078 .0172
		
		     .56 .25
		    10.0 4.54
			193°K 200°K 209°K
			.052W .070W .10W
			2.06W 2.06W 2.06W
			.025 .034 .0485
			
			.54 .40 .28
			3.85 2.86 2.0
				170°K 181°K 193°
				.057W .10W .141W
				10.3W 10.3W 10.3W
				.0055 .0097 .014
				
				.70 .40 .28
				22.9 12.5 8.86
					144.3°K
					.04W
					39W
					.001
					
					4.625
					63.75

6.	GOVN. CONTRACT NR. GOVN. CONTACT/LOCATION				
7.	COMMENTS				

1.	MANUFACTURER	Santa Barbara Research Center	Philco Ford	Philco Ford	Arthur D. Little
2.	SYSTEM DESCRIPTION 　TRADE NAME 　MODEL 　I.D. NUMBER 　OPERATING TEMP. 　TYPE CYCLE 　WORKING FLUID 　LUBRICATION 　POWER REQMTS 　　LOAD 　　VOLTS/PHASE/FREQ 　HEAT SINK REQMTS 　　MEDIUM 　　TEMPERATURE STATE OF DEVELOPMENT APPLICABILITY TO SATELLITE COOLING	SMS/VISSR Cooler >125°K Radiative Cooler 0 W Radiation to Space Space Qualified	4 Staged Radiator Cooler >120°K 0 W Development Model Tested	3 Staged Parabolic Cooler >111°K 0 W Development Model Tested	2 Staged Cooler >77°K 0 W Analyzed
3.	OPERATIONAL FEATURES 　HIGH PRESSURE 　LOW PRESSURE 　MINIMUM TEMP. 　COOL DOWN TIME 　ATTITUDE REQUIREMENTS	77°K 60 hrs. No Direct Sun/Earth Rad	120°K No Direct Sun/Earth Rad	111°K No Direct Sun/Earth Rad	77°K No Direct Sun/Earth Rad
4.	PHYSICAL CHARACTERISTICS 　COST 　RELIABILITY 　MAINTENANCE INTERVAL 　FIXED WEIGHT – EXPANDER 　　　　　　– COMPRESSOR 　　　　　　– RADIATOR 　EXPENDABLE WEIGHT 　VOLUME	3 to 5 years .7 lb 0 2 ft³ (18"D x 14")	3 to 5 years 1.3 lb 0 .18 ft³ (11.46"D x 3")	3 to 5 years 2.57 lb	3 to 5 years 2.5 lb 0 1.45ft³ (18.5"D x 9.34")
5.	CYCLE PARAMETERS 　COLD PLATE TEMP. 　REFRIGERATION CAP. 　POWER INPUT 　COP 　% CARNOT EFF. 　LBS./WATT OUTPUT 　IN³/WATT OUTPUT	77°K　125°K　165°K 0W　.05W　1.0W ∞　140　7 ∞　69000　3456	120°K　131°K　139°K 0　.1W　.183W ∞　13.　7.10 ∞　3110　1699.		77°K　140°K　195°K　243°K 0W　.235W　.935W　2.35W ∞　10.6　2.7　1.06 ∞　10662　2680　1066
6.	GOVN. CONTRACT NR. GOVN. CONTACT/LOCATION				
7.	COMMENTS	Three Units Delivered – – Flight of VISSR Inst. on Synchronous Satellite Due Spring '74	Data obtained from AIAA Paper 70-854		Data obtained from ASME paper 71-AV-30 – Later data is probably avail- able from ADL

APPENDIX C

SATELLITE INCIDENT RADIATION

This Appendix presents data to determine the magnitude of all
radiation incident upon a satellite. The data have been taken directly from
"Handbook of Military Infrared Technology", edited by Wolfe, William L.,
Office of Naval Research, Department of the Navy, Washington, D. C., 1965.,
pages 812 through 821.

For additional data, including thermal radiation properties of
selected materials, the reader is referred to Section 20 of the above reference.

20.7.2. Satellites. A satellite, *i.e.*, a spacecraft orbiting a planet, is irradiated from three external sources:

(1) The sun (solar radiation)

(2) Reflected sunlight off the planet (albedo radiation)

(3) The planet surface acting as a blackbody (planet radiation)
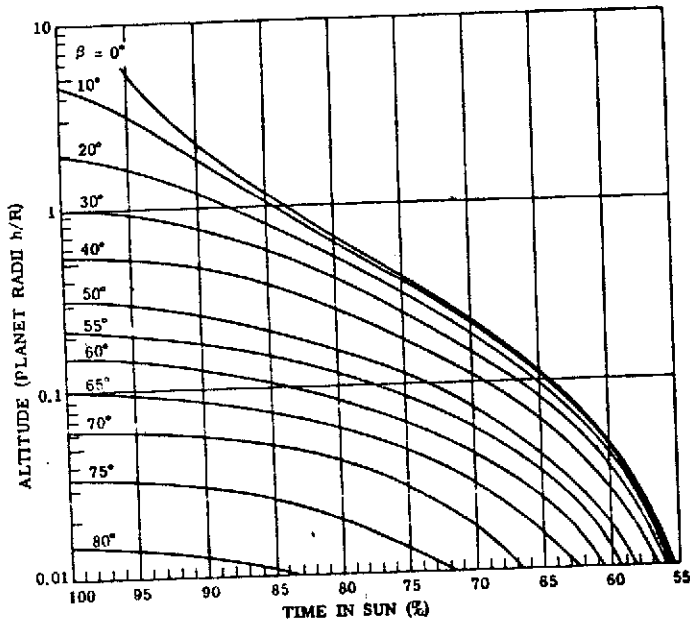
FIG. 20-29*b*. Percentage of time in sun *vs.* altitude with orbit inclination β as parameter for vehicle in circular orbit.

20.7.2.1. *Incident Solar Radiation.* The solar radiation power in a planet's orbit is obtained from Table 20-3.

In orbit, a satellite in the shadow of a planet is occulted from the sun. This shadowing, and time in the sunlight, varies according to the angle β between the satellite's orbital plane and the sun-planet plane (the ecliptic, when considering the earth, see Fig. 20-29*b*). The angle β is determined by the incident launch angle *i*. The orbit of an earth satellite precesses; *i.e.*, the normal to the orbital plane will generate a cone about the earth's axis at a constant half-angle of *i*. A 500-mi circular orbit at $i = 32°$ precesses approximately 6° per day. Because of the precession of the orbit and the rotation of the earth about the sun, the angle β varies continuously from a maximum (equatorial inclination $+i$) to a minimum (equatorial inclination $-i$). The maximum and minimum average solar heat load over the orbit are determined, respectively, when β is a maximum and when $\beta = 0°$.

Consequently, the orbital average solar heat flux incident at a point on a spatially oriented satellite's surface is

$$H_s = S \cos \theta \times (\% \text{ time in sun}) \qquad (20\text{-}10)$$

where θ is the angle between the surface normal and the sun-line.

20.7.2.2. *Incident Albedo Radiation.* The parameters in the orbit geometry (Fig. 20-30) used in defining the incident albedo flux are:

θ_s = angle between the planet-sun line and the planet radius vector to the satellite

γ = angle between the satellite surface normal and the planet radius vector to the satellite surface

ν = orbit angle about the planet

ϕ_c = one of the attitude parameters, the angle of rotation about the planet radius vector to the surface normal

ϕ_r = 0 when the normal lies in the plane containing the planet-surface vector and the planet-sun vector

ϕ_r, θ_s, and γ may vary as the satellite traverses its orbit; therefore, to obtain average albedo heat fluxes, computations of geometric factors must be made at each orbit angle ν and integrated over the orbit.

Figures 20-31 through 20-41 are used to determine the geometric factor F_a for albedo radiation incident to a sphere, cylinder, and flat plate [13].
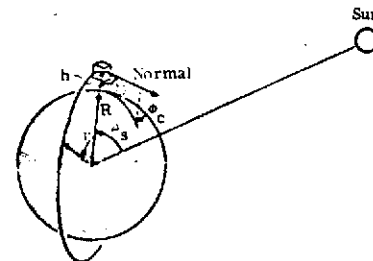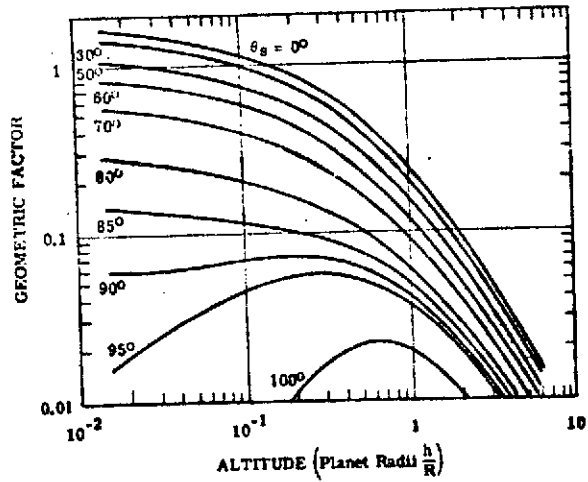

FIG. 20-30. Sun-vehicle geometry.

Fig. 20-31. Geometric factor for planetary albedo radiation incident to a sphere, *versus* altitude, with angle of sun as a parameter.
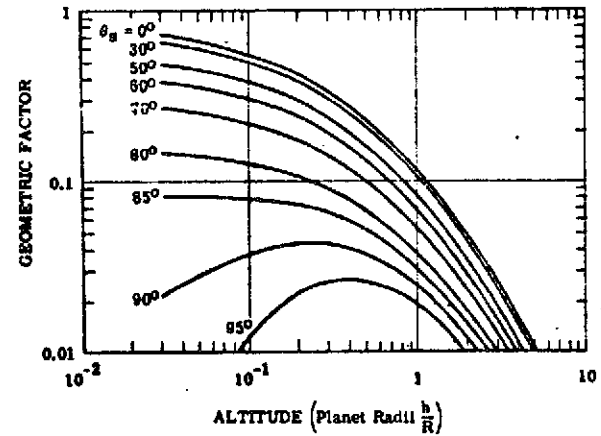

Fig. 20-33. Geometric factor for albedo to hemisphere, *versus* altitude, with angle of sun as parameter ($\gamma = 90°$, $\phi_r = 0°$).

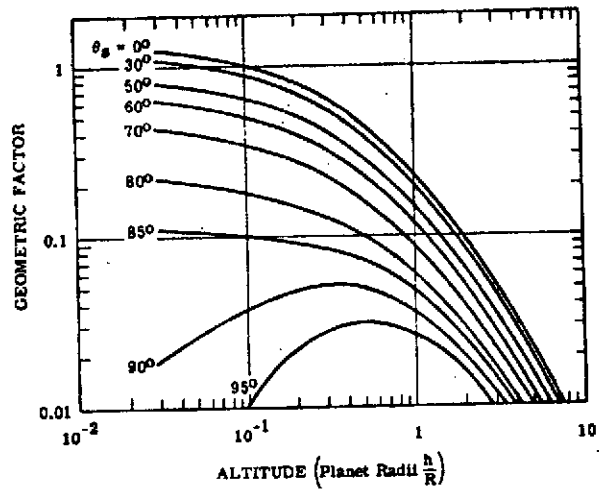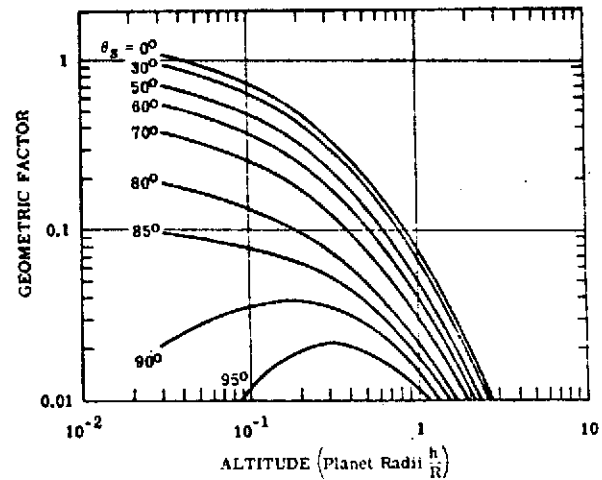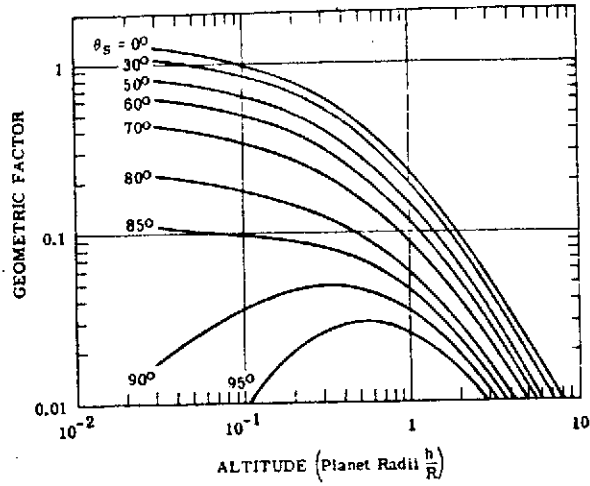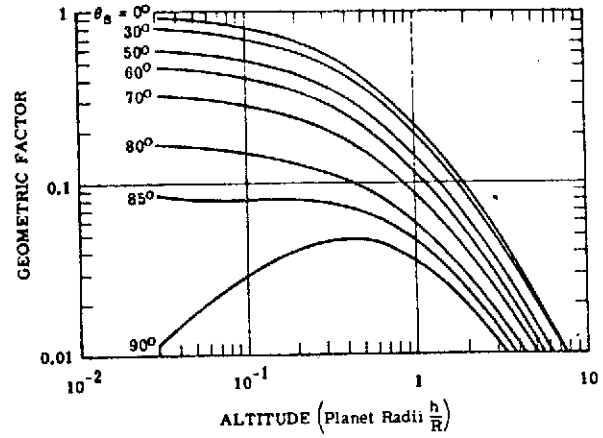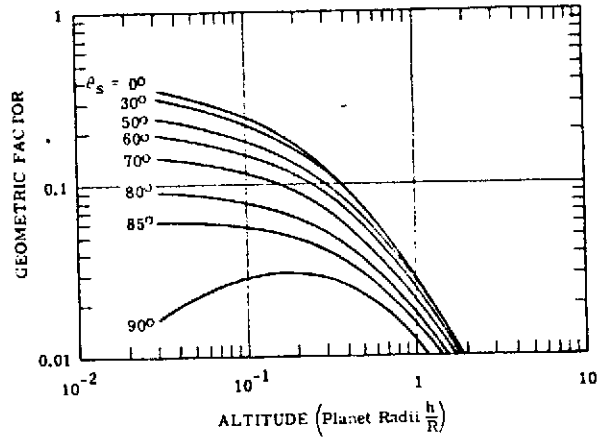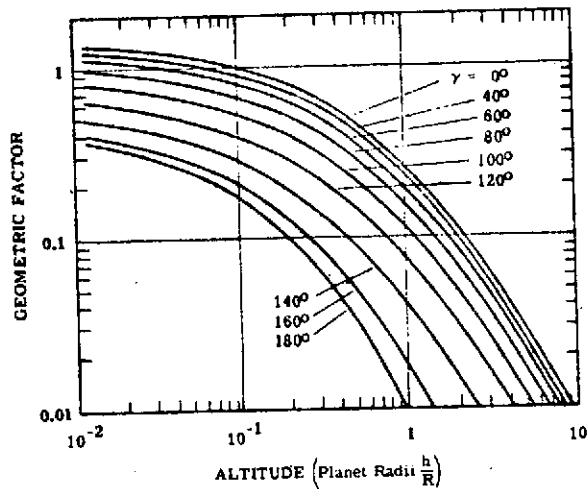Fig. 20-32. Geometric factor for albedo to hemisphere, *versus* altitude, with angle of sun as parameter ($\gamma = 0°$, $\phi_r = 0°$).

Fig. 20-34. Geometric factor albedo to cylinder, *versus* altitude, with angle of sun as parameter ($\gamma = 0°$, $\phi_r = 0°$).

FIG. 20-35. Geometric factor for albedo to cylinder, *versus* altitude, with angle of sun as parameter ($\gamma = 90°$, $\phi_c = 0°$).

FIG. 20-37. Geometric factor for albedo to one side of flat plate, *versus* altitude, with angle of sun as parameter ($\gamma = 0°$, $\phi_c = 0°$).

FIG. 20-36. Geometric factor for albedo to one side of flat plate, *versus* altitude, with angle of sun as parameter ($\gamma = 90°$, $\phi_c = 0°$).
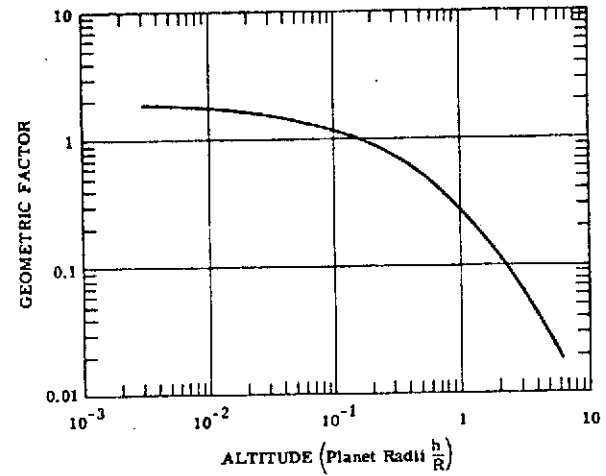
FIG. 20-38. Geometric factor for planetary thermal radiation incident to sphere, *versus* altitude.

FIG. 20-39. Geometric factor for planetary thermal radiation incident to hemisphere, *versus* altitude, with attitude angle as parameter.
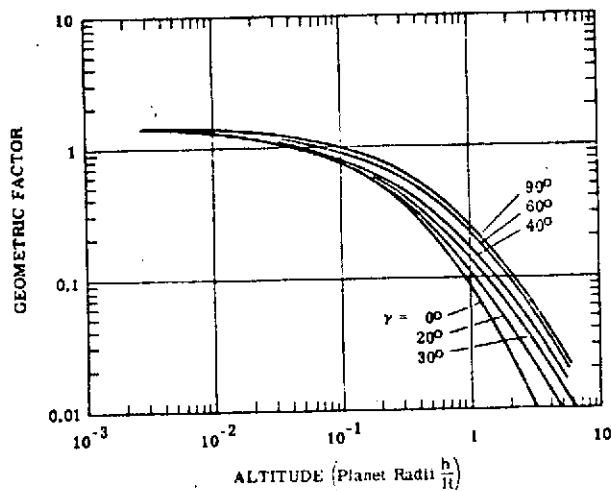
FIG. 20-40. Geometric factor for planetary thermal radiation to cylinder, *versus* altitude, with attitude angle as parameter.
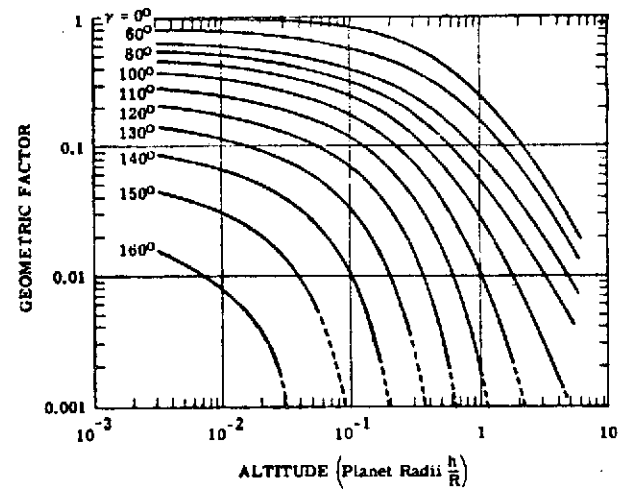

FIG. 20-41. Geometric factor for planetary thermal radiation incident to flat plate, *versus* altitude, with attitude angle as parameter.

From these, the instantaneous albedo heat flux density indicated by the prime is obtained:

$$H'_a = SaF_a$$

where a is the albedo factor (from Table 20-3).

Average heat fluxes per orbit are obtained from

$$H_a = Sa \sum_{n=1}^{n=k} F_a \Bigg]_{\nu_n}$$

where $\nu_n = 0°, 10°, 20°, \ldots 350°$

$n = 1, 2, 3, \ldots k$

To obtain the total power incident, $P = HA$, one can project the surface area for A in Fig. 20-42 through 20-45. For the flat-plate heat fluxes, the cosine relation is incorporated in the curves, so that the full flat-plate area can be used.

20.7.2.3. *Incident Planetary Radiation.* Similar geometric factors [13] are given below for planetary radiation to a sphere, cylinder, and flat plate using the geometry outlined above. The angle γ varies with orbit angle, and for average orbital planet flux the geometric factor must be averaged over the orbit.

$$H = H_p = \epsilon_p \sigma T_p^4 \sum_{n=1}^{n=k} \frac{F_R]_{\nu_n}}{n}$$

$$= W_p \sum_{n=1}^{n=k} \frac{F_R]_{\nu_n}}{n}$$

Generally the planet emissivity ϵ_p is assumed to be 1. T_p is obtained from Table 20-3, and $\epsilon_p \sigma T_p^4$ becomes the planet emittance W_p. The instantaneous planet heat flux is:
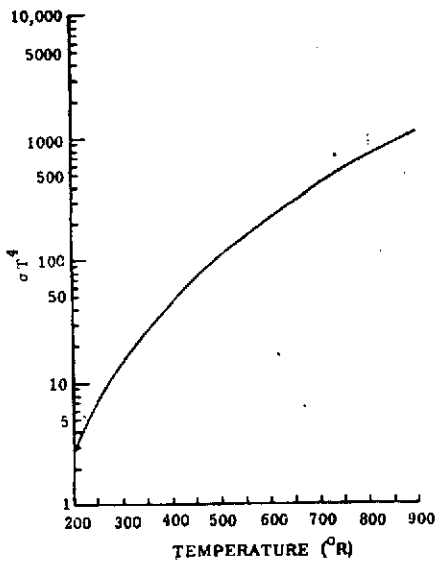
$$H'_p = W_p F_R$$

FIG. 20-42. Space radiation to an insulated thin plate, 200°-900°R.
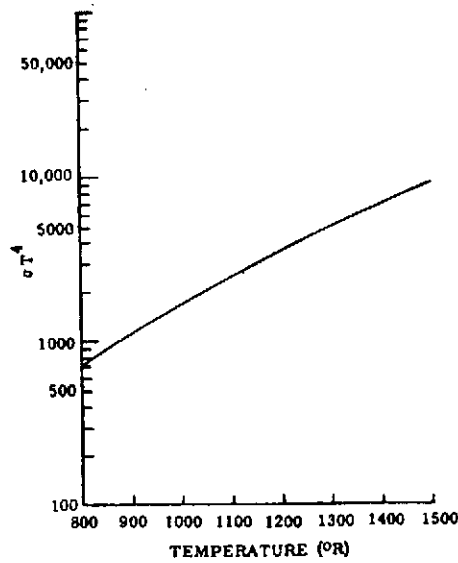

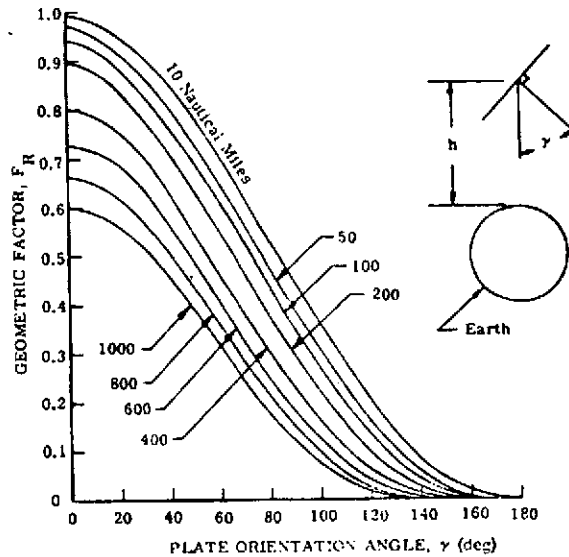FIG. 20-43. Space radiation to an insulated thin plate, 800°-1500°R.

FIG. 20-44. Geometric factor for earth radiation to a flat plate, *versus* plate orientation and distance above earth surface. y is the plate orientation angle, the true angle between the plate normal and the earth-plate line. h = plate distance above the earth's surface, in nautical miles. F_R = geometric factor for diffuse radiation from the earth to the plate. The flat plate irradiance is given by: $H_p = F_R W_p$ Btu (hr ft²)⁻¹. For an average yearly albedo of 36%, H_p = 70.9 Btu (hr ft²)⁻¹.
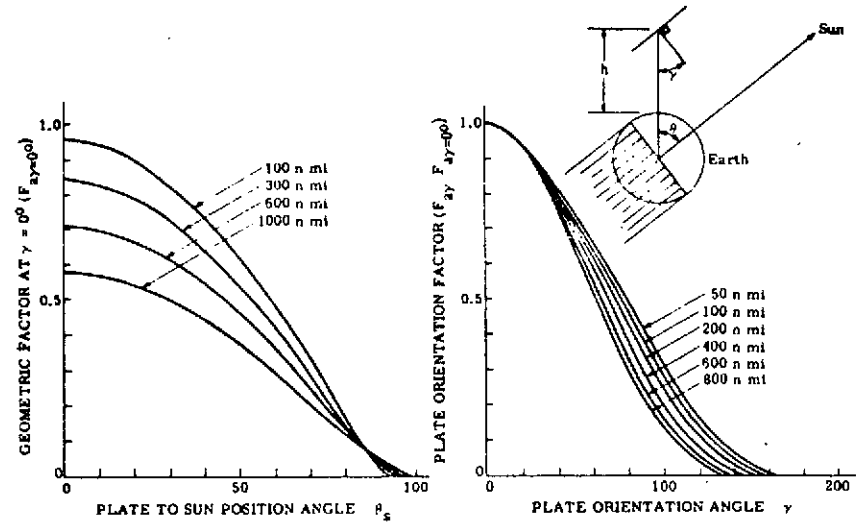

FIG. 20-45. Geometric factors for earth albedo to a flat plate, *versus* plate orientation and distance above earth's surface. $F_{a\gamma=0}$ is the geometric factor from the earth to a plate parallel to the earth surface. $F_{a\gamma}/F_{a\gamma=0}$ is the plate orientation correction factor as a function of altitude. γ is the true angle between plate normal and earth-plate line. θ_s is the true angle between earth-plate and earth-sun lines. h is the plate distance above earth's surface, in nautical miles. The flat plate irradiance is given by: $H = F_{\gamma=0} (F_\gamma/F_{\gamma=0})H_a$. For the average yearly earth albedo of 36%, H_a = 159.3 Btu (hr ft²)⁻¹.

20.7.2.4. Total Incident Space Radiation. If one sums the heat fluxes of solar, albedo, and planet radiation, the instantaneous heat flux to a satellite's surface at a point is calculated from:

$$H' = S \cos \theta + aSF_a + W_p F_R$$

as a function of ϕ_c, θ, θ_s, and γ.

The orbital average heat flux to a surface is calculated from:

$$H = S \cos \theta + aS \sum_{n=1}^{n=k} \frac{F_a]_{\nu_n}}{n} + W_p \sum_{n=1}^{n=k} \frac{F_R]_{\nu_n}}{n} = H_s + H_a + H_p$$

Once the irradiances from space are known, along with the corresponding spacecraft surface thermal properties of α_s, α_a, and ϵ_{sfc}, the surface temperature (T_{sfc}) of a perfectly insulated material may be computed from:

$$\epsilon_{sfc} \sigma T_{sfc}^4 = H_s \alpha_s + H_a \alpha_a + H_p \alpha_p$$

Generally, the absorption of the materials for albedo radiation is the same as the absorption for the solar spectrum, and for the case where the planet temperature and skin temperature are similar, the surface absorption for planet radiation (α_p) is the same as the surface emissivity; i.e., $\alpha_p = \epsilon_{sfc}$. The above equation then becomes

$$\sigma T_{sfc}^4 = (H_s + H_a) \frac{\alpha_s}{\epsilon_{sfc}} + H_p$$