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CONCEPTS AND METHODS OF REFRIGERATION FOR SUPERCONDUCTING POWER TRANSMISSION CABLES

Final Report

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June 1976

Work performed under Contract E(04-3)-1061

AiResearch Manufacturing Company
Torrance, California

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ENERGY RESEARCH AND
DEVELOPMENT ADMINISTRATION

Division of Electric Energy Systems

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CONCEPTS AND METHODS OF REFRIGERATION FOR SUPERCONDUCTING POWER TRANSMISSION CABLES

FINAL REPORT

Scott A. Manatt, Paul G. Wapato, John Stanko, and John P. Baumgartner

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JUNE 1976

Prepared for

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OFFICE OF CONSERVATION
UNDER CONTRACT NO. E(04-3)-1061**

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FOREWORD

This study of concepts and methods of cryogenic refrigeration for superconducting power transmission cables was conducted by AiResearch Manufacturing company of California, a division of The Garrett Corporation, under United States Energy Research and Development Administration Contract E(04-3)-1061. The work was sponsored by the Electric Energy Systems Division under the USERDA assistant administrator for conservation; Mr. Nelson P. Laquna was the program technical coordinator. The contract was administrated by the USERDA San Francisco operations office.

The effective date of the contract extended from March 24, 1975 through October 15, 1976. Mr. Scott A. Manatt of AiResearch was the program manager and principal investigator, Mr. Paul G. Wapato of AiResearch was the associate investigator, and Mr. Robert L. Brindley of AiResearch was the program staff reliability engineer. Other AiResearch personnel who made significant contributions include Mr. J. Baumgartner, Mr. F. Bescoby, Mr. J. Denk, Dr. A. Saur, Mr. J. Stanko, and Dr. R. Wright.

This technical report has been assigned AiResearch report number 76-12854. The report was submitted by the authors in June 1976.

ABSTRACT

An analysis of refrigeration system requirements for the superconducting power transmission cables currently under study at the three principal U.S. cable development centers indicates the need for cable system design considering the interrelated performance of the various cable system elements to successfully develop these systems for commercial operation in the 1990's. Hardware alternatives, compatible with the application timeframe, are evaluated to establish reliable refrigerator system design to a composite requirement typical of the range of refrigerator requirements presented by the cables currently under development. In addition, a methodology is presented and utilized to establish an estimate of hardware reliability and to evaluate redundancy requirements to a specific refrigerator system reliability allocation for a cable system used in previous cable development center studies. Reliability and performance of several compressor systems are evaluated. The oil-flooded screw compressor is felt to be the superior positive displacement compressor; however, the efficiency advantage resulting from preliminary design analysis of a multistage compliant foil process gas bearing centrifugal compressor system indicates a potential major operating cost reduction and the elimination of the need for oil lubrication and its subsequent cleanup requirements. Heat exchangers and expansion engines are evaluated. A preliminary design for a compliant foil process gas bearing permanent magnet turboalternator operating entirely at expansion turbine temperatures is discussed.

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SECTION 1

INTRODUCTION AND SUMMARY

Future trends in energy generation and consumption in the United States are expected to develop larger electrical power generation centers located farther from load consumption centers. Increased concern for the aesthetic environment combined with the increasing cost of right-of-way indicates that overhead transmission will be replaced by underground power transmission in populated areas. Current design in underground power transmission cables limit their electrical capacity such that many parallel circuits would be required. If substantial use is to be made of underground power transmission cables, economic considerations show the need for cables with vastly increased electrical transmission capacity.

As an answer to the need for high-capacity power transmission, cables using superconductors to carry electrical current with low losses are under development in the United States* in Western Europe** in Japan*** and in the Soviet Union. In the United States, various elements of superconducting power transmission cable systems are being developed under both government and private sponsorship. The three principal U.S. centers involved in cable system development are the Union Carbide Corporation (UCC) the Brookhaven National Laboratory (BNL), and the Los Alamos Scientific Laboratory (LASL). All three centers are operating under sponsorship of the U.S. Energy Research and Development Administration (ERDA); UCC's work is jointly sponsored by the Electrical Power Research Institute (EPRI).

Among the unique requirements of a superconducting power transmission cable is the need to maintain superconductor temperature within a few degrees of absolute zero. A study of the concepts and methods of refrigeration for superconducting power transmission cables has been undertaken to evaluate this element of a cable system for application in the 1990's. As a result of this study, AikResearch has concluded that the refrigerator system should not be expected to limit successful application of superconducting power transmission cable systems and believes that a refrigerator system should be optimized as a part of the overall cable system. The interrelationship of cable, cryostat and refrigerator system elements are such that their individual suboptimization should be iterated with the overall cable system design.

The cable systems currently being developed at the three principal U.S. cable development centers differ in the choice of superconductor, dielectric insulation, operating voltage, etc. These differences have affected the optimum refrigerator system interface requirement; thus it can be expected that the various cable systems will require refrigerators engineered to their specific requirements. The study has shown that the various emerging cable

* Belanger, B.C., "Superconducting Power Transmission Programme in the U.S.A.," Cryogenics, February 1975.

** Bogner, G., "Cryopower Transmission in Europe," Cryogenics, February 1975.

*** Horigome, T., "The Present State of R&D for Superconducting Transmission in Japan," Cryogenics, February 1975.

system designs are sufficiently unique that a single off-the-shelf refrigerator system design can not be expected to be suitable for all cooling requirements. Rather until or unless cable system optimization results in common refrigerator system requirements, its refrigerator will continue to require separate suboptimization to the specific application.

To evaluate the design of key subsystems and components, AiResearch has established a baseline refrigerator system design which approximates the interface requirements of both the BNL and LASL cable systems. The baseline multi-expander reversed Brayton refrigerator system, discussed in Section 3, permitted the generation of specific requirements for a typical compressor subsystem, cold box heat exchanger and expansion engine. The baseline refrigerator system, while not fully optimized as a part of a specific cable system, results in subsystem and component requirements sufficiently representative of optimized design to allow detail conclusions to be established.

In Section 4, various design approaches are considered for the compressor subsystem, the largest cold box heat exchanger, and the coldest expansion engine.

Compressor subsystem study included the oil-flooded Lysholm screw compressor, and several alternative multistage centrifugal compressors. BNL's screw compressor testing shows the potential for these machines as replacements for piston-type positive displacement compressors where long periods of uninterrupted operation is required. Their use of oil and overall operating efficiency are considered as drawbacks, however.

The development of multistage centrifugal compressors, supported by compliant foil process gas bearings offers an attractive alternative to positive displacement compressors. Both magnetically coupled gearbox driven assembly and modular motor/2-stage compressor designs were evaluated. For the relatively short 43-mile cable system used as a design baseline (Plan 3A of the BNL-Long Island Lighting Company study*), estimated energy cost savings for centrifugal compressor subsystems are expected to be as high as \$2.5 million per year when compared with screw compressors.

In dynamic aircraft environment, compliant foil process gas bearings have demonstrated mean-time-between-failure of over 60,000 hours, without scheduled maintenance, in environmental control system application. The steady-state, static environment of a refrigerator substation is considered to be an ideal application for the helium gas bearing centrifugal compressor resulting in a long-life, reliable, efficient compressor subsystem. AiResearch has recommended USERDA development of multistage compressor subsystems.

The comparisons of aluminum and stainless steel plate-fin heat exchangers for the cold box subsystem has shown that units of approximately the same size would result. Although initially more expensive due to the material cost, stainless steel units can be expected to be more corrosion resistant than dip-brazed aluminum heat exchangers. For production quantities, formed-tube-plate construction is discussed in terms of advantages for reduction of initial cost and reduced stress concentration.

*Forsyth, E.B., Mulligan, G.A., Beck, J.W., and Williams, J.W., The Technical and Economic Feasibility of Superconducting Power Transmission: A Case Study, IEEE Paper T 74 462-8.

The baseline refrigerator system design includes four expansion engines, two for helium bypass, one remotely located at the extreme end of the cable segment, and a final expansion engine at the return to the cold box from the cable. Design evaluation shows that the turboexpander is also an encouraging application for the hydrodynamic compliant foil process gas bearing. A preliminary design of a turboalternator is presented which results in the virtual elimination of conduction losses by operating the entire turboalternator at expansion turbine temperatures; the only conductor heat path is via wires for electrical dissipation of expansion engine work external to the cold box. Overall isentropic expansion efficiencies are expected to be about 65 percent resulting in improved cycle efficiency. Recommendations for development of a compliant foil process gas bearing supported turboalternator are included for USERDA consideration.

AiResearch believes that reliability engineering considerations should become a major part of cable system design. Reliability requirements for the refrigerator system should be an outcome of the recommended cable system optimization.

Reliability engineering considerations have been used in the study to evaluate redundancy requirements in the baseline system design. The reliability study, reported in Section 5, requires failure rate and maintenance data relative to hardware experience in the application under consideration. Unfortunately, very little organized reliability engineering data exists for helium refrigeration/liquesfaction systems. Based on referenced sources, AiResearch has projected reliability/maintainability data for various compressor and cold box subsystem components for use in this study. This data, presented in Appendix A, has allowed an evaluation of principal redundancy required to meet the baseline system reliability goals coordinated with USERDA and the cable development centers.

It is felt that the application under consideration and the consequence of cable system unreliability will be a significant factor in its eventual acceptance for commercial service. Accordingly, AiResearch has recommended that a reliability engineering data collection, storage and retrieval program be initiated both at the cable system and refrigeration system levels. The importance of reliability engineering to future systems design requires that these preparatory steps to the establishment of useful data be undertaken in the near future.

The final section of the report summarizes the conclusions and recommendations. Recommendation for future study and development are considered at various levels from the entire cable system to detail component designs.

SECTION 2

CABLE SYSTEM DEVELOPMENT

Although this report is primarily concerned with the evaluation of refrigerators and their components for superconducting power transmission cable systems it is useful to summarize the status of development of these systems. Since this subject is summarized in depth in publications by those more directly involved in the activity, this section is intended to briefly note the relevant background and the activities of those cable developers in the United States with whom AiResearch has held discussions in the course of this study.

POWER TRANSMISSION

Since the 1940's electrical power usage in the United States has grown at a rate of approximately 7.5 percent over the previous year's consumption.* Projections in the past were based largely on an extrapolation of this annually compounding rate. The drastic changes in fuel cost of the 1970's will no doubt result in changes in actual power consumption when compared to previous estimates, yet it seems certain that the industrial expansion of the U.S. will continue to require additional electrical power generation. Increased costs will undoubtedly result in a more rapid development of alternate energy sources and more efficient generation, transmission and distribution systems.

Current electrical power networks in the U.S. represent an immensely complicated power system. Electrical power generated by many sources have, in recent years become integrated into vast Intertie networks which may transmit power over hundreds of miles from generation sites to final distribution for consumption. These networks are supplied by numerous smaller utility companies; with few exceptions, power generation is matched to demand. Although underground cables for power distribution are not uncommon, high power transmission lines are currently almost exclusively above ground installations due primarily to economic advantages.

In the remaining quarter of this century, it is likely that as old energy sources become depleted and more expensive, and as additional capacity is required, the needs of electrical power in the U.S. can be best met by new larger power stations compatibly designed for the vast transmission networks. Relatively small generation sources in urban areas can be expected to be replaced by these larger generation sources coming into the system. The sites for the generation stations are likely to be located primarily by fuel considerations and will, in general, be further from their main use centers. With the development of new generation capacity must come changes to the available transmission systems to more efficiently carry these increased loads.

*Forsyth, E.B., Underground Power Transmission by Superconducting Cable, BNL Report No. 50325, March 1972.

Overhead lines will probably remain the least expensive method of power transmission, except where the cost of real estate in the cities overwhelms other costs. The trend toward environmentally motivated legislation may, however, be expected to restrict the use of cheap overhead transmission in areas of population growth for aesthetic reasons. It is this consideration that is expected to cause the need for high capacity underground transmission cables. Within the end of the century, the major interties can be expected to carry 30×10^6 KW between regional systems. Allowing for reliability considerations, each of several circuits could easily be expected to carry $5-10 \times 10^6$ KW. The problem for future transmission systems is then to develop efficient low cost means for meeting these requirements using underground cables.

The most commonly used cable in the U.S. systems consists of paper insulated conductors operating in a welded steel jacket filled with oil operating at several atmospheres pressure - the high pressure oil filled (HPOF) cable. These cables are limited by the thermal considerations of heat dissipation. The electrical losses must be conducted by the oil to the surrounding back fill, often of special sand offering increased thermal conductivity. Electrical capacity at the currently used 345 KV voltage is limited to about 500 MVA. HPOF cables being developed for operation at 550 KV may increase this limit to about 900 MVA. However at 550 KV, with a length of only about 20 miles, the entire rated current is required for charging the system, thus practical use of HPOF cables will require expensive reactive correction at intervals along their transmission length. Approaches to increase the capacity by axial circulation and external cooling of the oil are expected to result in perhaps 1200 MVA capacities.

In addition to the efforts to increase capacity by improvements to HPOF cables, activity is underway to develop new concepts including compressed gas cables, cryoresistive cables, and superconducting cables. The use of electro-negative sulfur hexafluoride (SF_6) compressed gas cables has been suggested as a means to increase the cable power capacity ratings. Designs for capacities to 10,000 MVA using compressed gas insulation cables have been proposed. Another alternative cable is the cryoresistive cable where the conductor is refrigerated to a convenient temperature (e.g. 78°K) to increase the current carrying capacity by decreasing the conductor resistivity. Here the resistance of the conductor is substantially decreased, but the conductor remains "normal"

SUPERCONDUCTING CABLES

The future development of HPOF cables can be expected to be limited to about 900 MVA for cellulose insulation or to about 1000 MVA if synthetic insulation is used. Forced cooling may increase synthetic insulated HPOF cable capacity to as high as 2000 MVA; however, uncorrected reactive power generation may limit their useful transmission length. The concept of the superconducting power cable can substantially increase the current carrying capability over alternate approaches; critical length may be increased to to several hundred miles.

In order to further evaluate the power transmission potential offered by superconductors, three major efforts to evaluate alternate approaches are underway in the United States. The remainder of this section is devoted to a brief discussion of three principal U.S. cable development center programs* which have been used to establish the refrigerator requirements discussed in Section 3.

Rigid Ac Cable

The earliest of the three U.S. programs is being conducted by the Union Carbide Corporation (UCC). Originally begun in 1968 under the sponsorship of the Electric Research Council (ERC), the initial program was aimed at investigating ac losses in pure niobium superconductor. The present program, jointly sponsored by the Electric Power Research Institute (EPRI - the affiliation of utility companies which superseded ERC) and the U.S. Energy Research and Development Administration (ERDA), uses rigid tubular niobium conductors bonded to copper-Invar substrates. These conductors are enclosed in a cryostat sealed at the factory. The entire rigid assembly would be manufactured in lengths compatible with field transportation; at each junction, the superconductors for all three phases are spliced in the field and a vacuum insulated sleeve used to complete the insulation is also field installed; the field cryostat welds do not, however, effect the cryostat vacuum.

The principal UCC development program has been designed for a 138 KV system with a capacity rating of 3400 MVA. Of the cable center activities considered in the course of this study, the UCC approach is unique in the use of niobium as a conductor, thereby requiring refrigeration to lower temperatures as shown in Section 3. In addition, since helium gas is also used as the dielectric insulation, cable gas density must be maintained to prevent dielectric breakdown.

UCC cable development has progressed further than those under development at either of the National Laboratories. Designs for both the cable and the cryostat have been fabricated and tested. Work conducted at UCC includes joint development of such system elements as pothead terminations.

Flexible Ac Cable

Based on the understanding of ac superconductors resulting from previous development for high energy physics devices, in 1971 Brookhaven National Laboratory (BNL) began a study of underground superconducting power transmission on a National Science Foundation grant under an interagency agreement with the U.S. Atomic Energy Commission.

The cable development at Brookhaven differs substantially from that at UCC. The cable itself is designed to make maximum use of fabrication techniques developed for present HPOF cables. The conductor consists of niobium-tin (Nb_3Sn) tapes with copper layers used for fault stability. The electrical insulation is composed of a helium impregnated wrap of plastic tape enabling

*Belanger, B.C., "Superconducting Power Transmission Programme in the U.S.A.," Cryogenics, February 1975.

the power cable to be sufficiently flexible to be wound onto reels for transportation to the field. The flexible cable is then pulled into rigid cryostat sections, previously welded into place in the field, in a manner similar to existing cable installation.

The use of higher critical temperature niobium-tin superconductor allows increased refrigeration temperatures. In addition, the BNL studies have resulted in elimination of the complication of cryostat shielding at the expense of additional heat loads at the cable temperatures and correspondingly larger refrigeration requirements.

Brookhaven's selection of a flexible design is intended to reduce installation cost and reduce the number of cable splices. BNL's use of Nb₃Sn tape results from their efforts which showed that the hysteresis losses associated with these tapes are not too high to prevent the design of power transmission cables. In addition to the development of the superconducting tapes and cable assembly techniques, BNL has sponsored the development and test of several cryostats, and is presently working with elements of the refrigeration system.

Although several cable approaches are currently under consideration at BNL, the Brookhaven-Long Island Lighting Company (LILCO) joint study has identified a number of specifics in a tradeoff study of a 43 mile transmission requirement. This study identified a number of options, among which is a two-circuit approach using superconductors for the entire transmission distance. This approach, termed system 3A in the BNL-LILCO study*, utilizes two 345-KV cable circuits, each with a 2400-MVA normal rating.

Flexible Dc Cable

Electrical power generation for industrial use is a product of the 20th century. Early development was marked by local systems differing in the voltage, type (i.e. ac or dc) and frequency of electrical power. As machinery was developed and distributed, it became necessary to standardize. In the United States, electricity is, with few exceptions, generated as three-phase, 60-Hz power, although the voltages for transmission may vary according to specific needs. Since the final distribution and use of electrical power is in the form of alternating current, the use of direct current transmission is limited to distances where the savings in losses are sufficient to pay for the losses and initial cost of conversion equipment where it interfaces with ac networks. An example of the use of dc power transmission is the 750-KV Pacific intertie which delivers electrical power from hydroelectric sources on the Columbia river to distribution systems in southern California.

Since ac superconducting cables have small losses associated with hysteresis and insulation dielectric, the use of dc cables offers a potential promise of near loss-free power transmissions. Under sponsorship of the United States Energy Research and Development Administration (ERDA), Los Alamos Scientific Laboratory

*Forsyth, E.B., Mulligan, G.A., Beck, J.W., and Williams, J.A., The Technical and Economic Feasibility of Superconductor Power Transmission: A Case Study, IEEE Transaction Paper T 74 462-8.

(LASL) has been developing a dc superconducting power transmission cable. Although the use of dc power may eventually allow the use of materials with losses which eliminate them from consideration for ac application, the emerging LASL design presently uses multifilamentary niobium-tin superconductor, as does the BNL flexible ac cable, but generally allows a somewhat higher maximum operating temperature. Also like the BNL effort, the LASL dc cable is intended to be sufficiently flexible to be wound onto a spool for delivery to the field where it will be pulled into the assembled cryostat in much the same manner as required by the BNL design.

The ERDA sponsored LASL activity, begun in 1972, is the newest of the three cable development programs. Many of the basic considerations of cable modeling remain to be finalized, however the use of 100 KV to ground seems to be the favored operating voltage at this writing. Capacities are likewise to be fully defined, however the range of 1 to 10 GW is being considered. Both single and multiple parallel cryostats are being considered.

Although a relatively recent entrant into superconducting power transmission cable development (compared to UCC and BNL), LASL has provided a number of innovations. These include the adoption of the more flexible multifilamentary Nb_3Sn conductor instead of the Nb_3Sn tape used at BNL and the use of the remote expander cable cooling concept. This approach (developed by J. Dean and the BNL staff prior to his association with LASL) makes use of an expansion engine following the helium "go" run to cool the stream prior to entry into the cable helium return passage. The need for thermal isolation between the "go" and "return" passages is thereby removed, simplifying cable and cryostat design and reducing system cost.

SECTION 3

REFRIGERATOR SYSTEM REQUIREMENTS

A knowledge of the unusual low-temperature electrical conductivity properties of some materials closely followed the successful liquefaction of helium gas some 65 years ago. The unexpected sudden drop of electrical resistance to immeasurably small values, as the temperature of the conductor approaches absolute zero, provided a possibility of electrical devices operating without resistive losses. For many years the practical engineering problems associated with useful application of the superconductor phenomenon has prevented its use in all but laboratory environments. In recent years, additional interest has provided the need for superconductor and supporting system development for energy storage, high field electrical magnets, motors and generators, fusion reactors and for power transmission applications.

The needs of each application are somewhat specific in terms of superconductor development, leading to parallel efforts at government and private laboratories aimed at providing the data and systems engineering required for these applications. One requirement, however, is common in the emerging array of superconducting equipment developments: the need to provide refrigeration required to maintain these devices at temperatures within a few degrees of absolute zero.

For many experimental laboratory programs, the cost and complexity of a closed refrigeration system required to maintain superconductors at the required fluid temperature is not justified for short-term projects. Here, expendable fluid cooling is a convenient means of providing refrigeration if a coolant with the proper boiling temperature can be found. Helium, with a normal boiling temperature slightly over 4 K, has been used for this purpose. The helium is liquefied in a remotely located liquefier and stored and transported in a low-loss vacuum insulated vessel until required. It is then used for refrigeration in an open cycle where heat causes vaporization resulting in loss of the helium coolant from the system. Temperature adjustments, within a limited range, are achieved by controlling the vapor pressure.

When a superconductor is required to operate for a significant period of time, direct coupling of the refrigeration load to the source of refrigeration is generally more economical, provided the refrigerator can be expected to reliably provide the cooling required for the duration needed. Refrigerators using helium gas as the coolant (throughout the cycle or at the low temperature end of the cycle) have been successfully manufactured and are in wide use where cooling to these low temperatures is required. The majority of these devices are for small (relative to a power transmission cable) loads; continuous operation for years, as required for utility application has not been the principal design goal. Although each new application generally requires some custom engineering to optimize the refrigerator to the requirements of the application, "off-the-shelf" models with a number of options are available from system manufacturers who have endeavored to supply helium liquefiers and refrigerators for the needs of potential users. Small laboratory machines are

designed as liquefiers to produce helium for local storage and use. Many are used as closed cycle refrigerators enabling the use of the sensible heat capacity of the helium gas to improve the cycle efficiency on dedicated systems.

A second class of systems, perhaps more similarly related to the needs of utility applications, exists. These are the large capacity commercial helium liquefiers which are used to produce liquid helium for sale to government and commercial customers. Here the system requirements are for continuous or near continuous operation typical of process industry equipment.

CABLE SYSTEM MODEL

The need for low-temperature refrigeration will require highly efficient, highly reliable subsystems and components and must carefully consider the requirements of the overall cable system. Since the first commercial application of superconducting power transmission is expected to be in the 1990's, this study has been concerned primarily with the selection of the hardware components to best satisfy requirements at that time. As a result, the study has not limited itself to currently available components and subsystems; where development of a new class of hardware is considered in the best interest of the success of this mode of power transmission, its use has been considered.

The AiResearch study emphasis has been on system, subsystem and component performance requirements for 1990's, including both cycle efficiency and refrigerator reliability. In order to establish detail subsystem and component requirements, it was first necessary to establish a typical "baseline" refrigerator system. Since the design of the refrigerator must provide for the needs of the cable, it was decided that the most useful baseline for the study would result from a consideration of the current requirements generated at the three principal U.S. cable development centers discussed in Section 2.

To carry out the baseline system, subsystem and component evaluation efforts, it was necessary to define the performance and reliability requirements typical of a model cable system. Since individual component problem statements were needed for a model system refrigerator, an assumed cold box configuration had to be adopted. In addition to configuration, ground rules related to acceptable refrigeration system operation were needed to guide the reliability and maintainability studies. These were mutually defined by ERDA and AiResearch personnel.

For the purposes of this study, only the refrigerators associated with the cable itself were considered; it is anticipated that the special requirements associated with cable terminations, etc. can best be met with systems optimally designed for those special applications.

Performance Requirements

To evaluate the requirements of each of the three specific U.S. cable development centers, discussions were held with key technical and program personnel with the assistance of the ERDA technical coordinators. The basic refrigeration system related requirements were established in June of 1975

for the purpose of preliminary analysis; follow-up data reflecting the progress in cable development was also requested again in December 1975. The data received from the cable development centers is shown in Table 3-1.

Establishment of subsystem and component requirements to be used as a basis for the hardware studies described later in this report, called for an assumed configuration for the refrigerator systems to be established first. Since the cable studies underway at BNL, LASL and Union Carbide show clearly that the cable, enclosure and the refrigerator require an integrated system design, the assumed cold box design must be capable of the mode of operation upon which the cable studies are based. As a first step in defining the refrigerator system and cold box requirements, BNL, LASL and the Linde Division of Union Carbide Corporation were queried with regard to cable interface requirements and preferred cold box subsystem configuration. Initial system analyses were based on the responses received in June 1975, as shown in Table 3-1. Due to the progress being made at the cable development centers, it was necessary to update this data.

Updated responses were received from BNL and LASL, both showing changes in cable concepts which avoid costly helium-cooled heat shields in the cable enclosure. Both also made use of the "remote expander" concept, in which a work-producing expander is installed at the end of the cable "go" stream at the far end of the cable segment. The cable coolant stream is cooled by this far end expansion process before the fluid is routed to the cable return flow passages. No update was received from Union Carbide; however, it is assumed that the Linde enclosure concept remains unchanged from previous estimates, requiring use of a 4.6 K helium stream for enclosure heat shield cooling.

Reliability Requirements

In addition to the established baseline refrigeration system performance requirements, the specific needs of the intended application required that a reliability criteria also be imposed on the cable system refrigerators. Further, the reliability criteria for a single refrigerator system are only meaningful as they relate to the operation of the entire power transmission cable system. Since individual refrigerator system requirements are really a function of the entire network requirements, it was first necessary to determine "typical" requirements for the entire system. To establish requirements compatible with the cable system, the requirements were again coordinated with current cable systems used as baselines for other studies; the use of the BNL-LILCO 3A system* as a baseline was mutually agreed upon as a model to establish these requirements. As a result of discussions with BNL cable development program personnel, the following requirements were established for the cable system refrigeration reliability based on an apportionment of overall system requirements which, in turn, are based on experience from installed HPOF cables:

*Forsyth, E.B., Mulligan, G.A., Beck, J.W., and Williams, J.A., The Technical and Economic Feasibility of Superconducting Power Transmission: A Case Study, IEEE Paper T 74 462-8.

TABLE 3-1
REFRIGERATION SYSTEM REQUIREMENTS¹

Requirements	June 1975			December 1975	
	UCC ²	BNL ³	LASL	BNL ⁴	LASL ⁵
Cable Design Service					
Voltage, kv	138 (ac)	345 (ac)	+100 (dc)	345 (ac)	
Rating, normal/peak, mva	3400/6800	2400/4800	5000/10,000	2400/4800	
Conductor Requirements					
Conductor	Nb on Cu-Invar	Nb ₃ Sn tape	Nb ₃ Sn wire	Nb ₃ Sn tape	
Maximum operating temperature, °K	5.1	8.5	10-13	8.5	
Dielectric Density Requirements	4 atm @ 5.1 °K	50 kg/m ³	10 atm	50 kg/m ³	
Overall Average Thermal Load (exclusive of any shields)					
Ambient heat leak, w/km	260	55	250	480	
Hysteresis losses, w/km	54	42	-	42	
Dielectric losses, w/km	<u>6</u>	<u>165</u>	<u>-</u>	<u>165</u>	
Total, w/km	320	262	250	687	
Shield Configuration, (if any)					
Inlet temperature, °K	4.8	75	-	-	
Temperature rise, °K	296	25			
Flow rate, gm/sec	1.75/km	80.3	0	0	
Cable Interface Conditions					
Flow rate, gm/sec	200-400	312.2	70-140	180	
Inlet pressure, atm	4-6	15	10-12	15	
Inlet temperature, °K	4.8	6.75	10	7.0	
Outlet pressure, atm	4-6	14	9-11	6.7	
Outlet temperature, °K	5.1	8.75	13	8.6	
Assumed Refrigerator Spacing					
Substation separation, km	8	12	10	12	

- (1) Requirements for one circuit half-way to adjacent refrigeration substation in one direction.
- (2) UCC did not reply to December 1975 update request.
- (3) Assume use of shielded Cryenco cryostat.
- (4) Assume use of unshielded BNL-II cryostat.
- (5) LASL's December response indicated only that June data remained substantially unchanged.

- All interfacing nonrefrigeration system components (e.g., cable, cryostat) or those on whose operation the refrigerator depends (e.g., the supply of power, heat rejection, etc.) have separately apportioned allowable failure rates. The study considers only failures of refrigerator system parts, components, etc.
- A failure is defined as the occurrence of a condition (caused by the refrigerator systems) which results in either of the parallel circuits: (1) losing its rated current capacity, i.e., to go normal; or (2) to require the current to be switched off for its protection.
- Due to stored thermal capacity, any segment* of either circuit may be without active refrigeration for up to 8 hr, providing the refrigeration system maintains the minimum acceptable helium pressures in both the "go" and "return" flow channels. If helium at the correct supply conditions cannot be supplied within this required period, a failure is assumed to have occurred.
- For calculation purposes, a "failure," i.e., lack of refrigeration in any segment for over 8 hr, is allocated to a single circuit's refrigeration systems at a maximum frequency of once a year.
- Two (or more) such "failures" resulting in a simultaneous loss of refrigeration for over 8 hr in segments of both circuits will result in a requirement to shut off power to both circuits. This condition is permissible no more than once every 10 years.

ERDA concurrence completed the definition of acceptable system reliability requirements.

Establishment of Baseline Refrigerator

1. Substation Configuration

As can be seen in the data of Table 3-1, the various cable system designs require control of both helium pressure and temperature within the cryostat. This requirement is due to the need to control the conductor temperature within superconducting limits and to maintain the required dielectric strength in the conductor insulation. These interface requirements result from cable system optimization; the refrigerator system suboptimization can not shift these requirements without iterating the effect on the overall cable system.

An example of cable system optimization imposed requirements can be seen in the assumed refrigerator station spacing of Table 3-1. Initial refrigerator cost can be reduced by decreasing the number of required refrigerator systems; operating costs can be expected to be reduced by decreases in the low-temperature refrigeration load. Unfortunately, however, adverse cost effects on other interrelated cable system elements result in counterbalancing cost increases. Reduction in the number of refrigerators servicing a cable circuit will require increased helium flow if interface temperatures and pressures are to be maintained. Increased helium flow requires increased flow passage areas in both

*A segment is herein defined as the length of cable between adjacent refrigeration substations.

the cable and the cryostat to control pressure drop thereby significantly impacting the cost of these elements. Similarly, decreasing the low temperature load requirements require changes to cable and/or cryostat design which tend to increase this cost.

Based on the present level of cable system optimization performed by the cable development centers, an optimum refrigerator spacing has been established. From the refrigerator system point of reference, this divides the cable into a number of individually cooled segments reaching from a refrigerator substation to its remote expander substation located midway between adjacent refrigerator substations. From this information, a "typical" refrigerator system may be defined for any of the three cable development center performance requirements. The selection of the BNL-LILCO study system baseline provides the means to apportion individual refrigerator system reliability requirements from the allocation to overall cable system refrigerators.

The basic refrigerator-remote expander substation arrangement for a cable circuit is shown in Figure 3-1A. Refrigerator substations are spaced along the cable at the optimized intervals, remote expander stations are located midway between the stations. Each refrigerator system provides the cooling required for half the distance between adjacent refrigerator substations. The BNL-LILCO cable system will require two parallel cable circuits. The arrangement of refrigerator systems (RS) selected for the baseline design is shown in Figure 3-1B. (Support equipment includes power substations, heat rejection system, makeup gas supply, etc.).

A number of alternate refrigerator substation arrangements are possible. Figure 3-2A shows an arrangement using double-sized refrigerator systems each providing refrigeration to remote expander substations in both directions from the refrigerator substation. Figure 3-2B shows yet another alternate using double-sized compressor subsystems (C_S) but baseline sized cold box subsystem (CB_S). Numerous other arrangements can be considered, including a single refrigerator system or major subsystem for both cable circuits; however, this would violate circuit independent operation reliability criteria. For the purposes of this study, the arrangement of Figure 3-1B was used as a design baseline (although some additional analysis of compressor subsystems compatible with the arrangements of Figure 3-2 is presented as is an assessment of reliability implications of the arrangement shown in Figure 3-2A).

Since two of the three cable efforts appeared to be satisfied by similar cold box configurations, while the third calls for a much different refrigerator concept, it was decided to choose a "typical" baseline system for component studies based on the BNL-LASL remote expander approach;* the selection of this system approach was coordinated with ERDA personnel. (The Linde conductor selection will call for a cold box arrangement similar to the Linde LTR refrigeration cycle** to provide the very cold, high-pressure fluid required to maintain conditions below the critical temperature of pure niobium. For the

* Dean, J.W. and Laquer, H.L., Conceptual Design of a Dc Superconducting Power Transmission Line, submitted to IEEE Power Engineering Society, 1976 Conference on Underground Transmission and Distribution.

** Terbot, J., A New Helium Refrigerator for Superconducting Cable Systems, Paper R-2, 1975 Cryogenic Engineering Conference, Kingston, Ontario, 1975.

A. BASELINE SYSTEM CIRCUIT

B. BASELINE REFRIGERATOR SYSTEM ARRANGEMENT

S-6957

Figure 3-1. Baseline Refrigerator Substation Arrangement

A. DOUBLE CAPACITY REFRIGERATOR SYSTEM

B. DOUBLE COMPRESSOR SUBSYSTEM

S-6956

Figure 3-2. Alternate Refrigerator Substation Arrangements

definition of a baseline system for study, the individual refrigerator capacity was chosen to provide 4000 watts of cooling with a cable stream return temperature of 8.5 K.

The BNL-LILCO baseline system, used for the study, must run some 43 miles from the presumed Long Island Shoreham nuclear generator site to the Ruland Road power substation near the Nassau-Suffolk County border. The selection of the nominal 4000 watt baseline refrigerator capacity will require six refrigerator substations.

The capacity allows cooling of 5.8 km of cable for the BNL requirements or 8 km for the LASL requirements based on the estimates of thermal loads shown in Table 3-1. As noted, the enclosure heat leak in both cases does not rely on helium cooled shields in the cryostat. The cable return temperature is slightly below the 8.6 K value considered by BNL and is well below the 11 K temperature allowed by the LASL cable. Thus, the selected baseline refrigerator design possesses large margin for the LASL application.

2. Baseline System Cycle Optimization

Analyses of cycles satisfying the above conditions were conducted with an existing computer program which treats a system using two bypass expanders and a direct expander following the refrigeration load. This arrangement allows the desired cable inlet and exit pressure and temperature conditions to be established. Analysis of the remote expander process between the go and return portions of the cable stream was matched to the computer output. It was found that, with a remote expander of 60 percent overall efficiency, a 4000 watt load could be supplied at a cable stream flow rate which would imply a load of 3050 watts to the computer program model. The program was therefore run with the proper cable interface conditions and the apparent load of 3050 watts, optimizing the two-bypass expander inlet temperatures and bypass flow ratios with respect to input power.

The input parameters used in the analysis are noted in Table 3-2. The secondary cooling loads indicated were estimated to provide capability for heat shielding within the cold box. All expander efficiencies are overall values, including the losses due to the turbine loading device. Only the balanced-flow heat exchanger effectivenesses are specified; the effectiveness of the unbalanced-flow units at each bypass expander process are determined by the cycle energy balance.

3. Baseline System Performance

The optimized baseline refrigerator system flow sheet resulting from the procedure described above is presented in Figure 3-3, where it is shown that a total compressor flow rate of 252.4 gm per sec is required to satisfy the 4000 watt cable load. The cable stream flow rate is 189.7 gm per sec. The final efficiency adopted for the remote expander is 61 percent; the cable interface boundary conditions and the 0.5 atm pressure loss allowed in the

TABLE 3-2
BASELINE SYSTEM INPUT PARAMETERS

Refrigeration capacity, watt	4000
Cable inlet temperature, K	7.0
Cable inlet pressure, atm	15.0
Cable return temperature, K	8.5
Cable return pressure, atm	6.7
Aftercooler discharge temperature, K	300
Secondary cooling load 1, watt	200
Secondary cooling load 2, watt	100
Balanced heat exchanger effectiveness	0.98
Expander 1 efficiency, percent	70
Expander 2 efficiency, percent	70
Expander 3 efficiency, percent	60
Remote expander efficiency, percent	60
Heat exchanger $\frac{\Delta P}{P_{IN}}$ (each pass)	
Heat exchanger 1	0.025
Heat exchanger 2	0.015
Heat exchanger 3	0.025
Heat exchanger 4	0.015
Heat exchanger 5	0.010

Figure 3-3. Refrigerator System Model Schematic Diagram

s-6913

go and return passes result in the 4000 watt load being divided into 1618 w for the go pass and 2382 w for the return. This appears to be a reasonable apportionment for the purposes of the baseline system established. Cycle optimization resulted in bypass expander inlet temperature of 93.96 K and 29.77 K for the Number 1 and 2 units, respectively. Cold box shield exit temperatures are then 89.00 K and 28.60 K.

EQUIPMENT MANUFACTURER SURVEY

Prior to initiating the component design analysis subsequently conducted by the AiResearch turbomachinery, heat transfer, and reliability engineering departments, AiResearch solicited opinions and hardware data from qualified manufacturers of both components and cryogenic refrigerator systems. These data were desired to document the current state-of-the-art in hardware efficiency and reliability, to establish the performance range of available equipment, and to gain information relating to trends that established manufacturers believe will become the basis for hardware to be designed for future cable system application. Manufacturers were asked for estimates of the type of hardware required to best satisfy the requirements of superconducting power transmission cables to be developed for use in the early 1990's. Brief estimated performance requirements were provided for key subsystems and components, and respondents were asked to provide specific hardware details if they so desired.

The questionnaire, asking for opinions on the characteristics and development efforts associated with a hypothetical cable refrigerator systems for 1990, was directed to 12 U.S. and foreign commercial organizations active in helium refrigeration and liquefaction; although each was contacted by local Garrett representatives either in person or by phone, only seven completed or partially completed systems opinion questionnaires were received. No additional data relating to the specific requirements were received, although three respondents sent catalog/brochure type data. Two systems manufacturers declined to provide data. One of these objected to the requirement that their data bear no restrictions upon publication or dissemination; the other stated a policy whereby they do not do free engineering and emphasizing the overall complexity of the system. Three systems manufacturers did not reply at all.

In addition to the 12 systems manufacturers, all of which produce at least some components, AiResearch contacted thirty seven additional manufacturers who supply cryogenic refrigerator components or components which could be utilized in these systems. Again, response was not overwhelming. In subsequent conversations, many indicated that the application time frame prevented prompt attention being given to this subject; i.e. today's business today. This is felt to be significant in that it is an indication that the cable developers and/or refrigerator system manufacturers to be associated with the cables in the 1990's must begin to outline development needs today. Once these are established, development effort in areas which do not appear to be progressing at a rate commensurate with cable system requirements should be identified for supplemental development. A summary of the survey responses is shown in Table 3-3. (Table 3-3 does not include an AiResearch response.)

TABLE 3-3
EQUIPMENT MANUFACTURER SURVEY

	EQUIPMENT MANUFACTURED			
	SYSTEMS	COMPONENTS		
		COMPRESSOR	EXPANDER	HEAT EXCHANGER
NUMBER SURVEYED	12	17	4	13
RESPONSES				
SPECIFIC DATA	0	0	0	0
CATALOG/BROCHURE DATA	3	2	0	3
NO EQUIPMENT IN RANGE		2	1	
"PROPRIETARY DATA"	1			1
NOT IN BUSINESS		2	1	1
NO "FREE ENGINEERING"	1			
NO REPLY	3	12	2	8
SYSTEM PROJECTION	7	-	-	-

s-6955

The range of responses obtained for overall system characteristics showed that respondents felt that the high-pressure cold helium coolant requirement would be best satisfied by a refrigerator producing cold supercritical helium rather than by a separate supercritical transport loop cooled by a liquid helium refrigerator. The range foreseen by the various respondents for the other characteristics was very broad, as can be seen in Table 3-4.

TABLE 3-4
SYSTEM SURVEY RESULTS

Characteristic	Response Range
Thermodynamic performance (1/COP)	340 to 640
Cold box volume, cubic meters	5 to 850
Capital cost, \$ millions	1 to 10
Operator support per day, man-hours	0 to 8
Scheduled maintenance interval, hours	720 to 20,000
Labor per scheduled maintenance, man-hours	15 to 240

Various candidate subsystem and component types suggested as possible system alternates are shown in Table 3-5. The manufacturers were asked to indicate those types of subsystems and components they projected for most likely 1990's application. In addition, their estimate of required development effort for selected component types was requested. Responses from the seven completed or partially completed questionnaires are shown in Table 3-6.

TABLE 3-5
SUGGESTED SUBSYSTEMS AND COMPONENTS

Application	Suggested Types
Compressor	Reciprocating Rotary positive displacement Centrifugal aerodynamic Axial aerodynamic
Heat exchanger	Aluminum plate-fin Stainless steel plate-fin Wound tubular coil
Expansion engine	Reciprocating Rotating positive displacement Gas bearing turbine Oil-lubricated turbine
Expander loading	Aerodynamic External oil bearing Mechanical Alternator

TABLE 3-6
PROBABLE COMPONENT TYPES

	COMPRESSOR	EXPANDER	HEAT EXCHANGER
<u>COMPONENT TYPES</u>			
MOST FAVORED	CENTRIFUGAL	TURBINE/ BLOWER	ALUMINUM PLATE-FIN
NEXT MOST FAVORED	SCREW	TURBINE/ ALTERNATOR	—
<u>DEVELOPMENT EFFORT</u>			
PRESENT ADEQUATE	—	2	2
MAJOR EFFORT	3	1	—

SECTION 4

SUBSYSTEM AND COMPONENT DESIGN EVALUATIONS

Since the expected first commercial application for superconducting power transmission cables is in the 1990's, from the outset the study has concentrated on evaluating hardware solutions for the refrigeration systems, subsystems and components which can be developed for that time frame. This has required an assessment of the current state-of-the-art, an evaluation of the advances expected from specific development required for this application and from nonrelated developments, and a determination of those areas where hardware development is required for cable system refrigerators.

Although numerous additional areas should also be considered in some depth prior to final selection of refrigerator configurations, this study has, of necessity, limited the scope of detail analysis to those components which may require the longest lead-time to begin specific development necessary for the refrigerators to reliably support the superconducting cables.

SUBSYSTEM AND COMPONENT REQUIREMENTS

The key elements selected for design analyses during this study are the compressor subsystem, the expanders, and the heat exchangers. During the study, these components were examined in sufficient detail to evaluate the expected performance from various design approaches. From the baseline refrigerator system defined in Section 3, specific units were identified for the purposes of this study.

A number of alternative compressor subsystem configurations were considered to supply helium gas at the baseline design flow rates and pressures noted on the schematic diagram of Figure 3-3. The design requirements used in these studies are shown in Table 4-1. From these requirements, various turbomachinery compressor approaches were analyzed that are expected to provide low maintenance/high reliability helium gas compression. For example, the use of process gas bearing turbomachinery to eliminate the presence of oil in the helium flow was considered as an additional advantage for turbocompressors.

Additionally, the use of a "screw" compressor of the type currently available was considered. Although this type of compressor requires oil to be mixed with the helium, good reliability for this type of rotary positive displacement machine has been shown. Data available from Brookhaven National Laboratory (BNL) was evaluated.

Although aluminum is presently the principal material of construction for high effectiveness plate-fin heat exchangers, it exhibits a greater sensitivity to corrosion than does stainless steel. Additionally, although stainless steel requires greater heat transfer surface than aluminum due to its reduced thermal conductivity, this very property offers potential reduction in axial conduction, an advantage in high effectiveness heat exchangers.

TABLE 4-1
COMPRESSOR SUBSYSTEM REQUIREMENTS

- Mass Flow Rate: 252.4 gm/sec
- Inlet Pressure: 1.20 atm
- Outlet Pressure (following aftercooler): 16.42 atm
- Outlet Temperature (following aftercooler): 300 K
- Assume Cooling Water at 288.9 K (60F) Available

The No. 1 heat exchanger from the baseline system (Figure 3-3) was selected for design analysis. This heat exchanger requires the largest heat transfer rates of the five in the baseline cold box arrangement. In addition to using the full flow from the compressor, this unit must produce the largest terminal temperature difference. The design requirements used for the study are shown in Table 4-2.

TABLE 4-2
HEAT EXCHANGER REQUIREMENTS

- High Pressure Side
 - Inlet pressure: 16.42 atm
 - Inlet temperature: 300.00 K
 - Outlet pressure: 16.00 atm
 - Outlet temperature: 93.69 K
 - Mass flow rate: 252.4 g/sec
- Low Pressure Side
 - Inlet pressure: 1.23 atm
 - Inlet temperature: 89.00 K
 - Outlet pressure: 1.20 atm
 - Outlet temperature: 295.78 K
 - Mass flow rate: 252.4 g/sec
- (Thermodynamic Effectiveness): 0.980

The expander selected for design investigation is the No. 3 unit, which expands the cable "return" stream as it re-enters the cold box (Figure 3-3) to a pressure slightly above the compressor inlet pressure. Selection of this unit for detail study is based on its unique requirements and criticality to refrigerator cycle operation. The unit must operate at very low temperatures (i.e. the cable return temperature which is limited by the properties of the selected superconductor). Performance is critical; the exhaust temperature from this inline expander directly effects the helium temperature at the exit from heat exchanger No. 5, which is the controlling temperature at the cable's interface.

Analysis of this component was limited to turboexpanders. The design of a compliant foil process gas bearing turboalternator has been considered to virtually eliminate the thermal losses associated with units requiring warm-end loading; compliant foil process gas bearing support offers the elimination of losses associated with hydrostatic gas bearings. Design requirements used for this component are summarized in Table 4-3.

TABLE 4-3
EXPANDER REQUIREMENTS

•	Inlet Conditions
	Pressure: 6.70 atm
	Temperature: 8.50 K
•	Exhaust Conditions
	Pressure: 1.34 atm
	Temperature: 5.13 K*
•	Mass Flow Rate: 189.7 g/sec

*assumed overall efficiency = 60%

COMPRESSOR SUBSYSTEM

Three different types of compressor systems were studied using the baseline compressor requirements of Table 4-1. The oil-injected screw compressor, which is currently under test at Brookhaven National Laboratory (BNL) was studied first since this equipment is developed and appears to be a viable candidate for cable refrigeration systems. However, the screw compressor is less efficient and requires oil removal from the process gas (helium) to an impurity level less than 0.1 PPM. While tests to date at BNL indicate that these impurity levels can be achieved with commercially-available oil clean-up equipment, long term reliability of this equipment has not been determined.

The potential problem with oil clean-up was the incentive to investigate compressor systems and expanders utilizing process gas bearings which eliminate the need for oil or oil clean-up. The two types of compressors studied in this respect were the axial and centrifugal compressors; however, the large number of stages and resulting mechanical complexity of the axial compressor deemed this design impractical. A study of the multistage centrifugal compressor indicated that this type of equipment could be developed for cable refrigeration systems.

Screw Compressor Study

The oil injected screw compressor appears to be most viable of the positive displacement class of compressors for helium refrigerator systems for the following reasons:

- High volumetric and mechanical efficiency relative to reciprocating units
- Uniform gas flow
- Pure rotating motion with even torque
- Fewer moving parts than reciprocating units
- Oil injection effective in lubricating unit and cooling helium
- Useful lifetime consistent with cable refrigeration requirements

For the baseline compressor requirements studied in this effort, the compressor must deliver 252.4 gm/sec (33.4 lb/min) of helium at an overall pressure ratio of 13.7 to 1. Two Dunham-Bush Model 2516 screw compressors operated in parallel can handle this capacity requirement (3150 SCFM). According to Dunham-Bush engineering, a large, single-stage screw compressor could easily be designed and built to deliver 3150 SCFM (or more) at a pressure ratio up to 15 to 1 or slightly higher. The English firm of James Howden is presently building single-stage screw compressors with this flow rate and pressure ratio capability. Therefore, existing screw compressor units are capable of meeting the capacity requirement for helium refrigerator systems needed for cable refrigeration.

Figure 4-1 presents a cross section of a hermetically sealed screw compressor which shows the simplicity of the unit. Oil and gas enter the suction port and these fluids are compressed nearly isothermally by the meshing (but noncontacting) lobes of the male and female rotors. The compressed mixture is discharged around the motor thereby lubricating the bearings and cooling the motor. Figure 4-2 presents a simplified schematic of the complete screw compressor subsystem.

During the study, AiResearch coordinated the screw compressor study with both BNL, to establish current operating experience, and Dunham-Bush, the compressor's manufacturer. In visits to both facilities, insight into the operating experience and manufacturing considerations were gained. For example, during a visit to the BNL compressor test facilities, data from the Dunham-Bush model 1610 was provided for analysis. Some of this data is presented in Table 4-4 for operating pressures and temperatures roughly equivalent to study baseline design requirements. Calculated adiabatic and isothermal efficiencies are included in this table for comparison with theoretical efficiencies computed for multi-stage centrifugal design presented later.

A. CROSS-SECTIONAL DIAGRAM

B. VOLUME AS A FUNCTION OF ROTOR POSITION

F-23735

Figure 4-1. Volume as a Function of Rotor Position
Hermetically Sealed Screw Compressor

Figure 4-2. Schematic of Diagram Screw Compressor Subsystem

TABLE 4-4
BNL SCREW COMPRESSOR TEST RESULTS

$P_{in},$ P_{sia}	P_R	$T_{in},$ $^{\circ}R$	$T_{out},$ $^{\circ}R$	Q_{SCFM}	$W_{He},$ lbm/min	$W_{oil},$ lbm/min	$P_{tot},$ hp	$P_{sh}^*,$ hp	$\eta_{overall}$ (isothermal), %	$\eta_{iso}^{**},$ %	$\eta_{ada}^{**},$ %
15	10.03	574	590	245.2	2.61	163.6	94.8	80.6	42	50	82
15	10.00	574	611	224.6	2.38	101.9	87.5	74.4	42	50	80
15	12.00	567	608	236.4	2.51	116.3	101	85.8	41	48	82
15	11.93	560	585	249.8	2.65	206.2	104	88.4	41	49	82
15	15.00	562	589	240.5	2.55	228.6	120	102	38	45	80
20	11.30	561	594	323.8	3.43	224.6	130	110	42	49	80
20	11.25	561	593	322.9	3.42	235.8	130	110	42	49	83

* $P_{sh} = 85\% P_{TOTAL}$ (estimate from Dunham-Bush)

**BASED ON P_{sh}

where:

$$\eta_{iso} = \frac{W_{He} R T_{in} \ln(P_R)}{P_{sh} (33,000)}$$

$$\eta_{ada} = \frac{W_{He} C_P T_{in} \left(\frac{P_R}{Y} - 1 \right)}{P_{sh} (33,000)}$$

W_{He} = helium mass flow rate, lb/min

R = gas constant for helium, 386 lbf-ft/lbm- $^{\circ}R$

T_{in} = helium inlet temperature, $^{\circ}R$

T_{out} = helium outlet temperature, $^{\circ}R$

C_P = specific heat of helium at constant pressure, 1.242 Btu/lbm- $^{\circ}R$ (965 lbf-ft/lbm- $^{\circ}R$)

P_R = pressure ratio

$C_P/C_V = \gamma = 1.66$

NOTE: Adiabatic and isothermal efficiencies presented in the above table were calculated in the same manner as the efficiencies shown for the impellers in the next section (Turbomachinery Compressor Study). The above values are approximately 20% higher than the BNL values partly due to including bearing losses with motor losses in obtaining shaft horsepower (P_{sh}).

Over 2,000 hours of operating time has been accumulated on the screw compressor at BNL. Fifteen hundred of these 2,000 hours represent operation with the oil clean-up system. No significant problems or failures have been encountered during this testing and hydrocarbon impurity levels of less than 0.1 PPM have been achieved after the 7th stage of purification. The 7-stage oil clean-up system has required no maintenance during this 1,500 hours of operation.

Dunham-Bush personnel stated that one of their screw compressors with 60,000 operating hours (6.85 years) was inspected and showed no visible signs of wear. Some tarnish from oil breakdown was found on the rotors, but this tarnish has no detrimental effect on operation of the unit. Furthermore, Dunham-Bush believes that minor design changes to existing screw compressor units would optimize the performance of the compressor and provide long term maintenance-free operation with helium. These design changes would include:

- Improved lubrication system to increase motor cooling
- Improved porting and motor matching for helium compression
- Improved casing and flange seals to minimize helium leakage
- Improved electrical controls and alarm system

Implications of Oil Lubricated Compressors

The 7 stages of oil clean-up described in Figure 4-2 probably represent the major technical drawback to the oil-injected screw compressor. About 98 percent of the oil/gas mixture (by weight) is oil, and this oil must be reduced to less than 0.1 PPM for cable refrigeration systems. To date, tests at Brookhaven indicate this degree of oil clean-up is achieved by using commercially available oil clean-up equipment. The establishment of the trade-offs of various levels of oil contamination has not been clearly established. As the temperature of the helium gas leaving the compressor subsystems begins to decrease in the cold box, the residual oil can be expected to collect at various locations within the cold box. The use of added adsorption elements, operating at cold temperatures within the cold box, can be expected to result in the removal of an unknown fraction of this oil, the balance accumulating at cold surfaces within the cold box heat exchangers or expanders.

Deposition of contaminants in a heat exchanger would produce a flow restriction and eventual plugging; deposition in an expansion turbine could produce either plugging or rotor unbalance. In either case, the effect would be cumulative since oil lubricated compressor continually add oil to the helium flow entering the cold box.

In order to estimate the amount of oil accumulation within the cold box, it is only necessary to establish an impurity level and the interval at which the cold box may be taken off-line for oil decontamination. To establish the accumulation of impurities the compressor exit flow of 252.4 gm/sec can be used. The contamination accumulation within the system can then be seen by reference

to Figure 4-3. Reduction of oil concentration to 0.1 PPM can be expected to result in about 800 grams/year of oil accumulation in the cold box. The use of a double size cold box (for cable cooling in both directions from a refrigerator system substation) can be expected to accumulate oil at twice this rate.

Contaminants in the helium coolant can result from leaks in the refrigeration system and from impurities present in the helium as supplied, as well as from failure of the purification system to remove all the compressor oil. In terms of instrumentation to measure the contamination levels, it is convenient to consider three categories of impurity. These are hydrocarbon compounds, water vapor, and everything else. A brief survey of various types of instruments for monitoring each of these impurity types is presented in Table 4-5.

TABLE 4-5
CONTAMINANT INSTRUMENTATION SUMMARY

Instrument or Measurement Method	Detection Capability for			Response Time
	Hydrocarbon	Water Vapor	Other	
Flame Ionization Detector	1 PPB	NO	NO	1 sec
Dew Point Indicator	Interference Possible	5 PPM	NO	15 sec
Electrolytic Cell (P_2O_5)	Interference Possible	100 PPB	NO	15 sec
Optical or Infrared Absorption Spectrograph	5 PPM	5 PPM	5 PPM	15 sec
Gas Chromatograph	5 PPM	5 PPM	5 PPM	10 min
Mass Spectrometer	1 PPM	1 PPM	1 PPM	1 sec to 10 min

The first column of Table 4-5 is a list of instrument types or measurement techniques considered in the survey. The second, third, and fourth columns present the estimated detection sensitivity for each type of contaminant. If the method is inappropriate or if the sensitivity is insufficient to warrant consideration for a given impurity, that conclusion is expressed by the word "NO". In some cases, the presence of one impurity in sufficient concentration will interfere with the ability of the instrument to detect another impurity, a condition noted by the comment "interference possible" in the appropriate box in the table. The fifth column presents an estimate of the response time or measurement cycle time for the instrument. The data presented in Table 4-5

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Figure 4-3. Baseline Refrigerator Cold Box Contaminant Accumulation

represent performance that can be achieved at present by good practice and does not necessarily indicate the ultimate capability of any given instrument.

The instruments listed in Table 4-5 are commercially available. It can be seen that only one case, that of the flame ionization detector for hydrocarbon monitoring, does the sensitivity claimed for the method satisfy detecting contamination at the level of one part per billion (10^{-9}). In all other cases, sensitivities lie in the range of 10^{-7} to 10^{-5} . It is clear that if systems to greatly reduce contaminant concentration are developed, monitoring contaminants at a level of 10^{-9} will require an improvement by a factor of between 100 and 10,000 in the sensitivities of these instruments. Alternatively, instruments of a type different from those commercially available at present will be required.

Currently available gas analysis equipment reliability for automatic unattended operation/or extended periods of time has not been established. Such instruments as a gas chromatograph or mass spectrometer give best results when operated continuously. A good vacuum system is an integral part of such an instrument. When the instrument is first turned on, it generally requires two or three days of continuous pumping and outgassing to remove absorbed impurities. It is not inconceivable, therefore, that such an instrument could be designed to operate continuously for an indefinite period.

If an essential monitoring instrument malfunctions, the failure should be apparent. For example, thermocouple signal conditioning circuits are designed to indicate upscale if an open circuit occurs in the thermocouple leads. None of the instruments in Table 4-5 have that inherent property. The mass spectrometer is better than most of the others in that respect, because if it is operating properly, it will always indicate the presence of helium.

All of these instruments require that a sample be withdrawn from the helium coolant stream. In most cases, the instrument accepts a continuous stream. The gas chromatograph operates instead on a batch principle. In some cases, the sample is not contaminated by the analyzing apparatus and could, therefore, be returned to helium refrigeration loop. In the case of the flame ionization detector, the sample is contaminated with combustion products and should be discarded. In the case of the mass spectrometer, the sample is exhausted through a vacuum pump and can not conveniently be recovered. Recovery of the sample from a gas chromatograph may or may not be practical, depending on the type of detector employed. In any case, the question of recovering the sample is related to the overall requirements of helium gas make-up in operating the system. A typical sample flow requirement is that of the P205 cell, in which the standard flow rate through the cell is 100 cc per minute.

Turbomachinery Compressor Study

The use of turbomachinery for helium gas compression offers potential reduction in operating costs due to improved overall compressor subsystem efficiency. In addition, the use of compliant foil process gas bearings offer the advantage of compressor operation without the need for oil lubrication. Since all turbomachinery approaches considered in this study utilize the hydrodynamic process gas bearing, it is useful to first discuss the history of their application.

1. Compliant Foil Process Gas Bearings

AiResearch has sponsored a gas bearing research and development program continuously since 1957 when the potential advantages of process fluid bearings for high-speed turbomachinery were first realized. In 1962, AiResearch demonstrated gas-lubricated compliant foil bearings that supported a high-speed rotating shaft at speeds ranging up to 600,000 rpm. These foil bearings also are forgiving of contamination, thermal gradients, dimensional tolerances, shock and vibration. These features have allowed the very successful use of compliant foil bearings in commercial aircraft as well as fighter aircraft applications.

Since 1962, the research and development of compliant foil gas bearings have resulted in sophisticated analytical tools designed to yield basic data for nondimensional analysis. These tools have been refined to correlate with laboratory test data and now allow various designs to be accurately calculated and evaluated before the hardware is fabricated.

Both in production and development of high-speed turbomachinery, the AiResearch foil bearings have demonstrated their extraordinary long life and reliability in severe environments. In addition to aircraft applications previously mentioned, they also have been applied successfully to a number of closed Brayton-cycle turbogenerators for long-duration space power for NASA, and in small turboalternator expanders in closed Brayton-cycle cryogenic refrigeration systems for space. The large power turbines use two cylindrical journal bearings and double acting thrust bearings. Smaller turbogenerator expanders have been designed to use two conical bearings that combine radial and thrust capability.

The purpose of the turbomachinery study was to evaluate the application of various designs to the baseline system. This study did not produce a final optimized design, such an effort is recommended as discussed in Section 6.

An axial compressor for cable refrigeration systems has been ruled impractical because of the approximately 40 stages of compression that would be required resulting in an impractical design due to mechanical complexity. Multistage centrifugal compressors using AiResearch process gas bearings and meeting baseline system requirements have been studied in some detail. This design approach appeared attractive for the following reasons:

- Adiabatic efficiencies are higher than those for positive displacement compressors, which results in electric power savings
- Oil clean-up is not required
- Reliability and lifetime of the gas bearings is excellent

One of the types of design considered for this study makes use of 2-stage modules. Each module consists of hermetically-sealed housing containing a motor-driven rotating assembly which has two compressor impellers at one end of the shaft and the motor rotor and thrust bearing at the other end. The rotating assembly, supported on gas bearings, is shown schematically in Figure 4-4.

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Figure 4-4. Schematic Diagram of 2-Stage Centrifugal Compressor Rotating Assembly

Figure 4-5 presents a schematic of a multistage modular compressor system. Seven 2-stage modules are shown in this figure, representing what is currently felt to be a very feasible design meeting baseline systems requirements. Other compressor designs having fewer stages were considered in this study. The number of stages can be reduced when the motor speed and horsepower ratings are increased. Further motor design effort is required to optimize the number of modules utilized.

2. Parametric Study of Impellers

Five possible centrifugal compressor designs which meet the requirements of the baseline system and two for a double capacity compressor were analyzed. The study evaluated designs operating at various tip speeds and different specific speeds. Specific speed generally refers to a grouping of variables which expresses the dynamic similarity characteristics of a turbomachine. Since, for a particular type of machine, peak performance occurs at a certain optimum condition of dynamic similarity, geometric similarity (by classes of machines) can be inferred from a specific speed. Although specific speed is widely employed in the field of turbomachinery, there is no common definition generally used by all workers. Both dimensional and dimensionless expressions bear the name "specific speed".

The definition of specific speed used is the dimensionless form defined as Equation 4-1.

$$N_s = \frac{N (V_{in})^{0.5}}{60 (g_c H_{ad})^{0.75}} \quad (4-1)$$

where

N = Rotational speed, rpm

V_{in} = Stage inlet volumetric flow rate, cu ft per sec

H_{ad} = adiabatic head rise, ft-lbf per lbf

g_c = 32.2 ft-lbm/sec-lbf

The inlet volume flow and head rise are stagewise variables, thus the specific speed of each stage must be individually determined.

Tables 4-6 through 4-10 present stage parameters for 5 possible centrifugal compressor designs which meet the baseline refrigerator requirements. Tables 4-11 and 4-12 show stage parameters for twice the flow rate required by the baseline model (i.e., 66.8 lbf/min). This size of compressor would reduce the number of refrigerator systems required for cable refrigeration.

The calculations done in preparing the tables have been performed stage by stage utilizing an AiResearch computer program. The isentropic (i.e., adiabatic) and isothermal efficiencies given at the bottom of each table are defined as follows:

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Figure 4-5. Multi-Stage Modular Centrifugal Compressor Subsystem

TABLE 4-6

OPEN IMPELLER UNITS, 2000 FPS TIP SPEED
45,000 RPM

Stage	P_{in} , psia	P_{out} , psia	D_{imp} , inch	N_s	η_{AD} , %	Impeller, hp
1	17.64	24.76	10.22	.0824	81.00	93.0
2	24.51	34.32	10.22	.0701	79.16	95.76
3	33.98	47.57	10.22	.0595	76.74	98.78
4	47.09	65.93	10.22	.0506	73.73	102.82
5	65.27	91.37	10.22	.0430	70.06	108.19
6	90.46	126.64	10.22	.0365	65.72	115.34
7	125.38	175.53	10.22	.0310	60.66	125.00
8	173.77	243.28	10.22	.0263	54.83	138.26

$$W_{He} = 33.4 \text{ lbm/min}$$

$$\text{TOTAL IMPELLER POWER} = 877.14 \text{ HORSEPOWER}$$

$$\eta_{ad} = 111.38\%$$

$$\eta_{iso} = 62.18\%$$

TABLE 4-7
OPEN IMPELLER UNITS, 2000 FPS TIP SPEED
38,000 RPM SPEED

Stage	P _{in} , psia	P _{out} , psia	D _{imp} , inch	N _s	η _{AD} , %	Impeller, hp
1	17.64	24.76	12.10	.0696	78.74	95.64
2	24.51	14.32	12.10	.0592	76.33	99.32
3	33.98	47.57	12.10	.0503	73.26	103.48
4	47.09	65.93	12.10	.0427	69.52	109.04
5	65.27	91.37	12.10	.0363	65.12	116.40
6	90.46	126.64	12.10	.0308	60.00	126.35
7	125.38	175.53	12.10	.0262	54.12	140.08
8	173.77	243.28	12.10	.0222	47.39	160.00

$$W_{He} = 33.4 \text{ lbm/min}$$

$$\text{TOTAL IMPELLER POWER} = 950.26 \text{ HORSEPOWER}$$

$$\eta_{ad} = 102.81\%$$

$$\eta_{iso} = 57.39\%$$

TABLE 4-8
OPEN IMPELLERS, 1800 FPS TIP SPEED
50,000 RPM

Stage	P _{in} , psia	P _{out} , psia	D _{imp} , inch	N _s	η _{AD} , %	Impeller, hp
1	17.64	23.30	8.25	.1073	83.72	72.86
2	23.06	30.34	8.25	.0945	82.90	73.58
3	30.04	39.51	8.25	.0828	81.73	74.68
4	39.12	51.46	8.25	.0726	80.25	76.01
5	50.94	67.01	8.25	.0636	78.41	77.80
6	66.34	87.27	8.25	.0557	75.03	81.33
7	86.40	113.66	8.25	.0488	73.53	83.02
8	112.52	148.02	8.25	.0428	70.43	86.60
9	146.52	192.77	8.25	.0375	66.87	91.23
10	190.84	243.81	7.78	.0359	65.55	82.63

$$W_{He} = 33.4 \text{ lbm/min}$$

$$\text{TOTAL IMPELLER POWER} = 799.69 \text{ HORSEPOWER}$$

$$\eta_{ad} = 122.17\%$$

$$\eta_{iso} = 68.26\%$$

TABLE 4-9

38,000 RPM - 12 STAGES, 6 MODULES

Stage	P _{in} , psia	P _{out} , psia	D _{imp} , in.	N _s	η _{AD} , %	Impeller, hp	
UNSHROUDED	1	17.64	23.27	10.86	.0815	82.20	74.23
	2	23.06	30.34	10.86	.0718	81.00	75.32
	3	30.04	39.51	10.86	.0630	79.30	76.93
	4	39.12	51.46	10.86	.0550	77.12	79.11
SHROUDED	5	50.12	62.51	9.31	.0610	83.00	54.06
	6	61.88	75.93	9.31	.0550	83.00	54.06
	7	75.17	92.23	9.31	.0500	82.50	54.39
	8	91.31	112.04	9.31	.0454	81.50	55.05
	9	110.92	136.09	9.31	.0412	80.50	55.79
	10	134.73	165.32	9.31	.0374	79.30	56.58
	11	163.66	199.76	9.16	.0346	78.40	55.70
	12	197.76	241.37	9.16	.0315	77.20	56.56

$$W_{He} = 33.4 \text{ lbm/min}$$

$$\text{TOTAL IMPELLER POWER} = 747.78 \text{ HORSEPOWER}$$

$$\eta_{ad} = 130.61\%$$

$$\eta_{iso} = 72.72\%$$

TABLE 4-10

50,000 RPM - 14 STAGES, 7 MODULES
 UNSHROUDED - 6 STAGES: OTHERS SHROUDED

Stage	P _{in} , psia	P _{out} , psia	D _{imp} , in.	N _s	η _{AD} , %	Impeller , hp	
UNSHROUDED	1	17.64	21.49	6.90	.1403	84.00	50.83
	2	21.28	25.92	6.95	.1273	84.00	51.50
	3	25.67	31.27	6.95	.1159	84.00	51.50
	4	30.96	37.71	6.95	.1056	84.00	51.50
	5	37.33	45.48	6.95	.0961	84.00	51.50
	6	45.03	54.86	6.95	.0875	84.00	51.50
SHROUDED	7	54.31	66.17	6.95	.0797	84.00	51.50
	8	65.51	79.82	6.95	.0726	84.00	51.50
	9	79.02	96.27	6.95	.0661	83.99	51.50
	10	95.31	116.12	6.95	.0602	83.35	51.90
	11	114.96	140.06	6.95	.0548	82.62	52.35
	12	138.65	168.93	6.95	.0499	81.78	52.89
	13	167.24	203.75	6.95	.0454	80.86	53.50
	14	201.72	243.30	6.76	.0431	80.47	50.92

$$W_{He} = 33.4 \text{ lbm/min}$$

$$\text{TOTAL IMPELLER POWER} = 724.39 \text{ HORSEPOWER}$$

$$\eta_{ad} = 125.53\%$$

$$\eta_{iso} = 75.29\%$$

TABLE 4-11

35,300 RPM - 14 STAGES, 7 MODULES
 UNSHROUDED - 6 STAGES; OTHERS SHROUDED

Stage	P _{in} , psia	P _{out} , psia	D _{imp} , in.	N _s	η_{AD} , %	Impeller, hp	
UNSHROUDED	1	17.64	21.49	9.76	.1403	84.00	101.66
	2	21.28	25.92	9.83	.1273	84.00	103.00
	3	25.66	31.27	9.83	.1159	84.00	103.00
	4	30.95	37.71	9.83	.1056	84.00	103.00
	5	37.33	45.49	9.83	.0961	84.00	103.00
	6	45.03	54.86	9.83	.0875	84.00	103.00
SHROUDED	7	54.31	66.17	9.83	.0797	84.00	103.00
	8	65.51	79.82	9.83	.0726	84.00	103.00
	9	79.02	96.27	9.83	.0661	83.99	103.00
	10	95.31	116.12	9.83	.0602	83.35	103.80
	11	114.96	140.06	9.83	.0548	82.62	105.70
	12	138.65	168.93	9.83	.0499	81.78	105.80
	13	167.24	203.75	9.83	.0454	80.86	107.00
	14	201.72	243.30	9.56	.0431	80.47	101.80

$$W_{He} = 66.8 \text{ lbm/min}$$

$$\text{TOTAL IMPELLER POWER} = 1449.76 \text{ HORSEPOWER}$$

$$\eta_{ad} = 125.53\%$$

$$\eta_{iso} = 75.24\%$$

TABLE 4-12
27,000 RPM - 12 STAGES, 6 MODULES

Stage	P _{in} , psia	P _{out} , psia	D _{imp} , in.	N _s	η _{AD} , %	Impeller, hp	
UNSHROUDED	1	17.64	23.30	15.35	.0815	82.20	148.00
	2	23.06	30.34	10.86	.0718	81.00	150.60
	3	30.04	39.51	10.86	.0630	79.30	153.86
	4	39.12	51.46	10.86	.0550	77.12	158.22
SHROUDED	5	50.94	62.51	13.17	.0610	83.00	108.10
	6	61.88	75.93	13.17	.0550	83.00	108.10
	7	75.17	92.23	13.17	.0500	82.50	108.80
	8	91.31	112.04	13.17	.0454	81.50	110.10
	9	110.92	136.09	13.17	.0412	80.50	111.60
	10	134.73	165.32	13.00	.0374	79.30	113.20
	11	163.66	199.76	13.00	.0346	78.40	111.40
	12	197.76	241.37	13.00	.0315	77.20	113.12

$$W_{He} = 66.8 \text{ lbm/min}$$

$$\text{TOTAL IMPELLER POWER} = 1495.1 \text{ HORSEPOWER}$$

$$\eta_{ad} = 130.61\%$$

$$\eta_{iso} = 72.74$$

$$\eta_{ad} = \frac{W C_p T_{in} (P_R^{\frac{\gamma-1}{\gamma}} - 1)}{\text{shaft power} (33,000)} \quad (4-2)$$

$$\eta_{iso} = \frac{W R T_{in} \ln(P_R)}{\text{shaft power} (33,000)} \quad (4-3)$$

where

W = mass flow rate, lb per min

C_p = specific heat capacity at constant pressure, 965 lbf-ft/lbm-°R

T_{in} = absolute compressor inlet temperature, °R

P_R = overall pressure ratio

R = gas constant, 386 lbf-ft/lbm °R

Y = ratio of specific heat capacities = 1.66

Since the compression of helium requires large energy input, the stage design is limited by a maximum tip speed determined, for modular design, by the currently available high-speed motors. Further developments in motor speeds could shift the speed limitations to the impellers. For these tables, the following impeller tip speeds have been assumed:

- 2000 fps for unshrouded impeller
- 1600 fps for shrouded impellers

These set the maximum pressure ratios that the compressor stage can deliver, as follows (for 300°K inlet temperature):

1.397 for 2000 fps tip speed

1.316 for 1800 fps tip speed

1.245 for 1600 fps tip speed

The number of stages necessary to deliver the required overall pressure ratio are determined by the above values. A minimum of 8 stages are needed at a 2000 fps tip speed, while 10 stages are necessary at 1800 fps. Titanium alloys will allow the tip speed limit to be increased to possibly as high as 2700 fps; a compressor designed for 2400 fps resulted in a 6-stage design. Matching modular motor design with the impeller requirements will be discussed later in the report.

3. Compressor Drive Motors

Two types of high frequency motors were considered for the use with the centrifugal impellers, and these design are:

1. Induction
2. Permanent magnet

a. 150 Hp Induction Motor Design

Preliminary design of a 150 hp, 46,000 rpm induction motor was conducted to evaluate the feasibility for long life superconductive cable refrigerator system application. Other motor sizes can be extrapolated from this basic design when stress and thermal conditions are considered. Design of a high power density, high speed induction motor involves close coordination of the electromagnetic design with thermal, stress propermagnetic, insulation and structural materials selection.

The baseline design described in the following paragraphs is based on existing similar units, however, advanced construction approach must be used to allow the rotor tip speed of 600 ft/sec.

At the high operating speed, the power output can be obtained with a relatively small motor. The small size and weight is favorable from the structural design standpoint, however, the compactness introduces a premium on achieving effective thermal protection of the unit due to the high power density. Electromechanical design optimization alleviates the thermal problem by:

- (1) Appropriate selection of magnetic materials for maximum saturation density and minimum loss
- (2) Design of insulation system for long life at high temperature
- (3) Proper trade-off of electric and magnetic loading
- (4) Appropriate balance of structure and magnetics and the integration of the cooling circuits

Thermal protection and long operational life is assured by the use of a cooling system requiring external pumping and heat exchanger equipment with back up thermal and protectors.

The design objective for the baseline motor is as follows:

Speed	46,600 rpm
Output power	150 hp
Input power	120/207, 3 phase, 800 Hz

Design life

20,000 hr

Duty cycle

Continuous

The above motor design was selected because it provides the power and speed required for the two stage compressor design presented in Table 4-10.

Figure 4-6 shows the typical construction features of an induction motor. The rotor is fabricated from electrical steel laminations with die cast conductors and endings. No slip rings are required. The stator is also fabricated from electrical steel laminations, with a three phase winding connected to the power source. The induction motor is well suited to high speed, high temperature operation.

Motor design optimization was achieved using an AiResearch proprietary computer program. This program predicts performance of polyphase induction motors of the squirrel cage rotor variety at supply frequencies up to 3200 Hz. Input data are primarily motor geometry, power supply voltage and frequency, ambient temperatures and pressure, winding and iron data. Print out includes resistances, reactances, flux densities in the various parts of the magnetic circuit, loss breakdown and weight breakdown. In addition, motor performance parameters such as speed, torque, efficiency, output horsepower, power factor and current are plotted for easy reference. The calculations are performed for a selection slip range to give maximum accuracy at the desired condition. (Induction motor magnetic spring rate was calculated by using the method developed by A. Covo, as presented in "Unbalanced Magnetic Pull In Induction Motors with Eccentric Rotors," AIEE transactions, 1954, Pages 1421 to 1425.)

The final design performance and dimensional summary is presented in Table 4-13. Performance curves are shown on Figure 4-7 for low slip operating range of 0.000 to 0.007 per unit. Induction motors similar to those required which have been designed and manufactured by AiResearch are:

10 HP - 88,000 rpm Helium Compressor Drive Motor

10 HP - 60,000 rpm Front Compressor Drive Motor

40 HP - 24,000 rpm Freon Compressor Drive Motor

350 HP - 14,000 rpm Traction Motor

Starting Considerations: (1) Motor may be started with a special variable frequency motor - generator (M-G) set rated at 200 KVA. Low frequency and voltage application is necessary initially to accelerate the rotor assembly to full gas bearing lift off. After this point, the bearing drag is negligible, and only load torque (and accelerating torque) needs to be supplied by the M-G set.

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Figure 4-6. Typical Induction Motor Cross Section

Figure 4-7. Induction Motor Performance Curves

TABLE 4-13
INDUCTION MOTOR DESIGN DATA

RATING	
Output Power	150 HP
Output Speed	45,777 RPM
Power Factor	96.7 per cent
Efficiency	94.7 per cent
Voltage	120 V-L-N, 3 phase
Frequency	767 Hz
Current	342 amps/phase
DIMENSIONAL SUMMARY (INCHES)	
Stator Stack O.D.	5.38
Stator Bore	3.00
Stack Length	7.00
Stator Coil Extension	2.00
Rotor O.D.	2.97
Air gap	0.015
Rotor I.D.	1.19

- (2) Solid state variable voltage - variable frequency power supply with a built in current limit is another possible way to start and run the motor. If the same power supply is used for start and run, oversizing of the components will be necessary, however, proper sequencing of several motors connected to a large power supply, or paralleling a special starting power supply can minimize the hardware complexity and cost.

Cooling System: Cold water circulates on outer housing integral heat exchanger, helium cooling on inside.

b. 350 Hp Induction Motor Design

An analysis was done of a 350 hp, 27,000 rpm induction motor which would meet the impeller requirements shown in Table 4-12. The stage design in Table 4-12 is for a compressor system with twice the baseline flow rate of Table 4-1. Such a compressor system is consistent with designs being considered for cable refrigeration systems. Higher flow rates allow this reduction in speed since impeller performance is maintained (see discussion of specific speed). Aerodynamically, the high flow rate is desirable. The larger motor design allows the use of fewer compressor modules at a refrigerator substation and will tend to reduce initial equipment cost.

c. Permanent Magnet Motor Design

The development of the new rare earth cobalt magnet materials has made possible the development of large permanent motors and generators for a variety of new applications. Figure 4-8 shows the construction features of a typical brushless permanent magnet motor. The rotor contains an array of permanent magnets and iron pole pieces enclosed in a thin cylindrical sleeve. The stator is similar to that of an induction motor. No brushes are required; however, electromagnetic rotor position indicators are necessary for most applications.

Preliminary electromagnetic and stress design was conducted for a 150 hp, 46,000 rpm permanent magnet rotor synchronous machine. The design was based on other proposed units for 45 kw and 1 mw aircraft generators. Configuration optimization was achieved using a second proprietary AiResearch computer program. This computer program combines magnetic and structural design to optimize the rotor configuration for the high energy rare earth cobalt magnets. The program is quite versatile, with a capacity to create and analyze machine geometry based on a limited number of user inputs specifying materials, tip speed, rating data, and desired rotor configuration.

Design data for the baseline PM motor is summarized in Table 4-14.

Rare earth cobalt permanent magnet motors similar to that required, which have designed and manufactured by AiResearch, are:

3 HP - 10,000 rpm Turbine Starter Motor

7 HP - 3,000 rpm Servo Motor

75 HP - 30,000 rpm Drive Motor (scheduled for completion Sept 76)

1,000 KW - rpm alternator (can be operated as motor--scheduled for completion Dec 76)

d. Comparison of Induction and Permanent Magnet Motor Designs

Numerical parameters of the two designs are compared side by side in Table 4-15. As stated above, the induction motor can be electrically started and run with a properly sized variable speed motor-generator set or with a current limited variable voltage-variable frequency solid-state inverter. Electrically starting the permanent magnet motor with a motor-generator set would require a more complicated switching and synchronizing mechanism but, after lock-in, the permanent magnet motor will operate satisfactorily with an M-G set. The solid-state inverter power supply for a permanent magnet motor is simpler and less expensive than that for an induction motor because the inverter SCR'S are commutated with the motor back voltage.

S-91666 -A

Figure 4-8. Permanent Magnet Motor Cross Design

TABLE 4-14

PERMANENT MOTOR DESIGN DATA

<u>Electromagnetic Efficiency</u>		95.84%
<u>Weight Summary, Lbs</u>		
Stator		
	Core	3.47
	Teeth	2.94
	Copper	<u>2.39</u>
	Subtotal	8.80
Rotor		
	Damper	0.3
	Poles	1.61
	Magnets	<u>2.35</u>
	Subtotal	4.26
	Total	13.06
116.6 kw/156.36 hp		
240 vac 170.5 amps, -.95 P.F. (Input Frequency 3067 Hz)		
8 poles, 46,000 rpm		
<u>Dimensions, In.</u>		
	Stack O.D.	3.96
	Bore	2.52
	Rotor O.D.	2.47
	Rotor Poles, I.D.	1.60
	Rotor Length	6.12
	Pole width	0.99

TABLE 4-15

COMPARISON OF INDUCTION AND PERMANENT MAGNET MOTOR DATA

Comparison Criteria	Induction Motor	Permanent Magnet Motor
Rotor diameter, in.	2.79	2.475
Rotor length, in.	7.00	6.12
Housing length, in.	11.00	9.00
Air gap, in.	0.015	0.036
Stator O.D., in.	5.38	3.96
Stator current density, amp/in. ²	10,550	12,106
Air gap flux density, K1/in. ²	33.3	41.8
Loss density, watts/in. ²	26.8	35
Magnetic spring rate lbf/in. ²	13,363	6309
High frequency motor efficiency, %	92	94
Power supply		
Output frequency, Hz	800	3067
60 Hz M-G set efficiency, %	94 x 95	94 x 95
Solid-state converter efficiency, %	93	86

e. Gearbox Drive

A third type of centrifugal compressor drive utilizes a single 60 Hertz motor and a speed step-up gearbox with the required number of output shafts to drive single or multiple stage centrifugal compressors. The gearbox uses oil and in order to provide a hermetic system either a permanent magnetic coupling or an electromagnetic coupling would be used. The centrifugal compressors would operate on compliant foil gas process bearings and the helium circuit would be oilless and hermetically sealed.

Figure 4-9 illustrates the gearbox driven configuration.

Figure 4-10 illustrates the compressor and compliant foil process gas bearing arrangements. The impeller aerodynamic design is based on Table 4-10, the same impeller characteristics that were used in the permanent magnet or inductor motor driven modules discussed earlier in this report. The actual number of stages to be used in the output shaft would require additional study to determine the optimum selection of stages and horsepower per output shaft.

Figure 4-11 shows a typical electromagnetic coupling. The unit contains a stationary electrically excited number which magnetizes two sets of magnetic poles, with both sets free to rotate and separated from each other by a non-magnetic stationary barrier. Torque is transmitted through the stationary barrier by the magnetic field, which can be de-energized to decouple. The unit must be stopped to re-engage the coupling.

AiResearch has built a 40 hp, 39,000 rpm coupling of this type.

Figure 4-12 shows a typical permanent magnet coupling. Two independently rotatable members containing an assembly of permanent magnets are separated by a stationary barrier. Torque is transmitted through the magnetic barrier by the magnetic field.

AiResearch has designed and built several couplings of this type:

1 HP - 5,000 rpm Radial Gap Pump Coupling

5 HP - 40,000 rpm Axial Gap Coupling

20 HP - 90,000 rpm Radial Gap Coupling between a compressor and gearbox

Subsystem Definition and Evaluation

As seen in the turbomachinery compressor design studies above, compliant process gas bearing supported centrifugal compressors require consideration of the aerodynamic and structural characteristics of the compressor, the electric drive motors and the speed conversion equipment either electrical (e.g., M-G set or solid-state frequency converter) or mechanical (e.g., gearbox). The advantage of the mechanical gearbox approach is found in the reduction of the number of electric motors which tends to reduce the initial cost and increase reliability. In addition, by proper selection of gear ratio, each stage (or multistage shaft) can be driven at different rotational speeds allowing further compressor aerodynamic optimization.

OIL LESS CENTRIFUGAL COMPRESSOR
ON COMPLIANT FOIL GAS BEARINGS
(10 TO 14 STAGES)

S-6945

Figure 4-9. Gearbox Driven Centrifugal Compressor Subsystem

s-6953

Figure 4-10. Gearbox Driven Centrifugal Compressor

Figure 4-11. Electromagnetic Coupling

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Figure 4-12. Permanent Magnet Coupling

The primary disadvantage of this arrangement is found in the gearbox which must be oil lubricated. Since one of the primary advantages of centrifugal helium compression is the use of oil-less process gas bearings, this requires a separation of the gearbox from the compressor stages. Both shaft seals and hermetic magnetic coupling were considered for helium/oil isolation. Based on calculated helium loss rates, hermetic magnetic couplings (as shown in Figures 4-8, 4-11 and 4-12) are considered the preferred solution, although less efficiency is achieved than would be with labyrinth seals on the compressor shafts.

As a result of the compressor study, four feasible conceptual system arrangements have been compared to establish efficiency estimates. These are summarized in Table 4-16 below for the baseline system compressor.

Energy cost considerations of compressor operation are found to be substantial at the required power levels. In order to compare costs for the four systems described in Table 4-16, input electrical power requirements were estimated using the relationship developed below.

$$\text{Input power} = \left(\frac{\text{Isothermal Compression Power}}{\eta_{\text{isothermal}}} \right) \times \left(\frac{1}{\eta_{\text{comp mech}}} \right) \times \left(\frac{1}{\eta_{\text{drive}}} \right) \times \left(\frac{1}{\eta_{\text{motor}}} \right) \times \left(\frac{1}{\eta_{\text{power cond}}} \right)$$

where isothermal compression power, kw = $\frac{W R T_{in} L_n(P_r)}{33,000} \times 0.746$

Calculations of compressor isothermal power requirements and the use of appropriate efficiencies in Equation 4-4 allow the establishment of electrical power requirements for each of the four compressor subsystems. These data are shown in Table 4-17.

TABLE 4-16
COMPRESSOR SUBSYSTEM ARRANGEMENTS

Compressor Type	Motor Type	Drive Type	Frequency Converter	Number of Stages
Oil injected screw	60 Hz	Direct	None	1
Centrifugal	High frequency induction	Direct	M-G set	2 x 7
Centrifugal	High frequency permanent magnet	Direct	M-G set	2 x 7
Centrifugal	60 Hz	Gear box	None	14

TABLE 4-17

COMPRESSOR SUBSYSTEM POWER COMPARISON

Compressor Subsystem	Efficiencies, percent						Input power, kw
	η isothermal	η comp mech	η drive	η motor	η power cond	η overall (isothermal)	
60 Hz-screw						40 ⁽¹⁾	1015
Induction-centrifugal	75 ⁽²⁾	97 ⁽³⁾	100	92	95x95 ⁽⁴⁾	62	655
Parmanent magnet-centrifugal	75	97	100	94	95x95	64	635
60 Hz-centrifugal	75	97	95x97 ⁽⁵⁾	95	100	66	616

Notes: (1) 40 percent based on interpolated data from Table 4-4

(2) 75 percent based on 50,000 rpm design from Table 4-10

(3) 3 percent compliant process gas bearing loss

(4) 95 percent motor efficiency, 95 percent generator efficiency

(5) 95 percent gearbox efficiency, 3 percent coupling windage and hysteresis losses

Reliability Assessment of Centrifugal and Screw Compressor Subsystem

A reliability trade study was performed on a baseline compressor subsystem incorporating compliant foil process gas bearing centrifugal and helical screw compression techniques. Component failure rate studies present simplified schematics of the baseline compressor subsystem for the screw compressor and a modular centrifugal gas bearing compressor, respectively. Both subsystem approaches were quantified using failure rate and repair time data from the Appendix A. The quantitative results are summarized below:

	Screw Compressor Subsystem	Gas Bearing Centrifugal Compressor Subsystem
1a. Mean-Time-Between-Failures	1,342 hours	1,245 hours
1b. Mean-Time-Between-Failures (Requiring more than eight hours to repair and restore to service)	38,684 hours	48,497 hours

	Screw Compressor Subsystem	Gas Bearing Centrifugal Compressor Subsystem
2a. Mean-Time-To-Repair (All failures)	6.66 hours	5.84 hours
2b. Mean-Time-To-Repair (Failures requiring more than eight hours for restoration)	11.23 hours	11.28 hours

The above quantitative results show that the gas bearing centrifugal compressor subsystem has a higher MTBF associated with failures requiring more than eight hours to restore but that otherwise the two subsystem concepts appear relatively comparable. The tabulated summary given below shows the contributions of the major hardware groups within the subsystems to subsystem unreliability in terms of equipment failure rates (this summary was taken from the tabulations of failure rate and repair time data presented in Table 5-1 of the reliability engineering analysis (Section 5) and Table 4-18 for the screw compressor subsystem and the modular centrifuges gas bearing compressor subsystem, respectively.

	Screw Compressor Subsystem $\lambda \times 10^6$	Gas Bearing Centrifugal Compressor Subsystem $\lambda \times 10^6$
Compressor and support equipment	85 (0.8)*	376.4 (7.32)
Contaminant removal and monitoring equipments	600.15 (25.05)	265.05 (13.30)
Controls	60 (--)	162 (--)

This summary shows that the screw compressor subsystem enjoys a considerable advantage in terms of compressor and control equipment reliability but that this advantage is lost when the required contaminant removal and purity monitoring equipment are considered. The limitations to the data used to quantify these subsystems are discussed in Appendix A. The area where these limitations impose the greatest restraints on this type of a comparison lie in the contaminant removal and purity monitoring equipment.

* The parenthetical failure rates are those associated with failures requiring more than eight hours for restoration.

TABLE 4-18

SUMMARY OF FAILURE RATES AND REPAIR TIMES
FOR A GAS BEARING CENTRIFUGAL COMPRESSOR SUBSYSTEM

Item Description	Failure Rates and Repair Time		Refrigerator Failure Rates and Down Times for Repairs			Refrigerator Failure Rates and Expected Down Times for Repairs Requiring More Than 8 Hours			
	1 Item Failure Rate $\lambda \times 10^6$ Hours	2 Est. Repair Time τ Hours	3 Qty Req'd N	4 Refrig Failure Rate $N\lambda \times 10^6$ Hours	5 Down Time Per 10^6 Hrs For Repair $D_T = \tau N\lambda \times 10^6$	6 Est. of Percent of Repairs Not Completed in 8 Hours	7 Refrig Failure Rate $\lambda = \% N\lambda \times 10^6$ Hours	8 Est. Repair Time τ Hours	9 Down Time Per 10^6 Hrs For Repair $D_T = \% \tau N\lambda \times 10^6$
Power Conditioner	100	4	1	100	400	--	--	--	--
Compressor Module	10	8	8	80	640	0.05	4	11	44
Gas Stream Coolers	2	8	33	66	528	0.05	3.3	11	363
Pressure Regulating Valve	4	3	2	8	24	--	--	--	--
Primary Filter Adsorber	15	8	1	15	120	0.05	0.75	11	8.25
Secondary Filter Adsorber	0.05	128	1	0.05	6.4	1	0.05	128	6.4
Module Inlet Filter	0.05	8	8	0.40	3.2	0.05	0.02	11	0.22
H ₂ O Contamination Monitor	100	8	1	100	800	0.05	5	11	55
Gas Contamination Monitor	150	8	1	150	1200	0.05	7.5	11	82.5
Bearing Gas Support System	100	4	1	100	400	--	--	--	--
Plumbing	22	4	1Set	22	88	--	--	--	--
Basic Subsystem Controls, Instruments, Interlocks and Alarm	162	3	1Set	162	486	--	--	--	--
Compressor Subsystem									
Failure Rate = $\Sigma N\lambda$			803.45/ 10^6 Hour			$\Sigma \% N\lambda$		20.62/ 10^6 Hours	
MTBF = $(\Sigma N\lambda)^{-1}$			1,245 Hours			$(\Sigma \% N\lambda)^{-1}$		48,497 Hours	
Mean-Time-To-Repair (MTTR) = $\Sigma D_T / \Sigma N\lambda$			5.84 Hours			$\Sigma D_T / \Sigma \% N\lambda$		11.28 Hours	
Repair Rate = $N = 1/\text{MTTR}$			0.1712					0.0887	

COLD BOX SUBSYSTEM

Those components between the ambient temperature compressor subsystem and the cold refrigeration load (i.e. the cable and cryostat) produce the refrigeration effect. In the simplified schematic diagram of Figure 3-3, the principal components, the work removing expanders and the heat exchangers, can be seen to successively reduce the temperature from the assumed 300 K ambient to a 7 K cable inlet temperature. Cycle efficiency requires that ambient heat transfer to these cold components be reduced as much as can be practically achieved. This means that a vacuum contains (or a number of adjacent vacuum containers for maintainability reasons) will enclose components to virtually eliminate gas convective heat transfer. Thus, these components will be packaged as a subsystem; the "cold box" subsystem refers to the package of these refrigeration effect producing components. In this subsection, the design of a heat exchanger and an expander are examined.

Heat Transfer Equipment

Superconducting transmission cable helium refrigerator systems will employ large, high-effectiveness heat exchangers. Heat exchangers now used for similar applications are plate-fin aluminum units fabricated by dip brazing. The reliability of current exchangers is generally considered low; leakage, increasing with time, is often cited in describing the performance of these units. The reliability requirements that will be necessary for transmission cable refrigeration systems make it certain that great improvements must be made in the exchangers. This section discusses how such improvements can be made.

1. Current Practice

It is well known that the dip brazing process produces heat exchangers which are prone to leakage during their service life due to corrosion of the aluminum caused or promoted by salts which were not removed from the core after brazing. There are numerous instances where the corrosion produces small leaks which are insignificant to the operation of the heat exchanger or the system. However, nearly absolute leak tightness is required for these high effectiveness applications.

In this process, the entire heat exchanger stack-up is immersed in a bath of molten salts (predominantly chloride salts) at approximately 1100 F. This bath melts the braze alloy and acts as a flux to promote distribution and bonding of the braze material. After brazing, all manufacturers subject their heat exchangers to extensive cleaning procedures to remove the salts. These procedures include high velocity rinses with very hot water. During these rinses, water effluents are monitored for chloride content. Typically an exchanger is not accepted until chloride concentration of the rinse is as low as 10 ppm. However, extensive experience with such heat exchangers indicates that total salt removal is practically impossible, or at best, not verifiable.

The problem of salt removal hinges on the accessibility of the interior of the heat exchanger core to rinse water and to inspection. If a core is small, uses widely spaced fins, and has no interior turns, the probability of complete salt removal is very good. On the other hand, large cores, closely spaced fins, and internal turns are likely to provide regions that can entrap braze salts, and impede the entry of rinse water. If the heat-exchanger uses interrupted fins to provide high heat-transfer coefficients, the difficulty of salt removal is increased.

High-effectiveness cryogenic heat exchangers are usually large counterflow units employing interrupted plate-fin surfaces. Figures 4-13(a) and 4-13(b) show two possible flow arrangements for plate-fin counterflow heat exchangers. It can be seen that at least one fluid stream must be turned within the core. Figure 4-13(c) shows the type of rectangular off-set fins which AiResearch would employ; other manufacturers might use louvered or wavy fins. In summary, cryogenic heat exchangers are large, employ tightly-spaced interrupted fin surfaces, and contain internal turns. Subsequently, the probability of complete salt removal is low, and the demonstration of adequate reliability for this application would be difficult. Therefore, it is concluded that the risk of salt-promoted corrosion will render dip brazing unsuitable for the manufacture of cryogenic heat exchangers for superconducting cable refrigeration systems.

2. Alternative Designs

There are two major alternatives to dip-brazed aluminum heat exchangers: furnace-brazed stainless-steel exchangers and fluxless-brazed aluminum exchangers. Both alternatives require considerably more manufacturing facilities than dip brazing. The facility for dip brazing is essentially a bath of molten salt. The bath can be fabricated of a variety of materials. It can be sunken in the floor at the manufacturing facility and can be left essentially uncovered. In contrast, the alternatives require a specially designed braze furnace capable of vacuum or inert gas operation. Heat application to the exchanger must be carefully controlled. In a vacuum furnace the heat application is nearly all by radiation; thus, this process may be very slow. With inert-gas brazing, the furnace contains blowers and manifolds to direct heated gas through the unit(s) to be brazed.

AiResearch has not conducted an extensive facilities survey. However, to our knowledge, there are no facilities in the United States to furnace braze heat exchangers of the size needed for cable refrigerators. In the time period of concern for this application, it is highly probable that proper facilities will become available. This assertion is based upon the emphasis on increased fuel economy and thermal efficiency in all energy-related processes. For example, in an application for large-scale industrial gas turbine engines, AiResearch is currently adapting compact heat-transfer technology to very large recuperators. A large braze facility is being developed for the manufacture of plate-fin heat exchangers weighing over 2000 lb each. This facility will probably be able to handle the large cryogenic exchangers needed for superconducting cable refrigeration systems. Undoubtedly other facilities will become available during the 1980's.

a. COUNTERFLOW EXCHANGER WITH TRIANGULAR END (ENTRY/EXIT) SECTIONS

b. COUNTERFLOW EXCHANGER WITH STRAIGHT-THRU PATH FOR ONE FLUID

c. RECTANGULAR OFFSET PLATE-FIN SURFACE

S-6951

Figure 4-13. Typical Counterflow Plate-Fin Heat Exchanger Configurations

3. Thermal-Design Comparison of Alternative Concepts

It is desirable to generate a comparison of the stainless and aluminum heat exchangers for this application. A design optimization study was performed for heat exchangers constructed of both stainless steel and aluminum. In addition, the effect of using nickel fins in an otherwise all-stainless-steel unit was considered. The optimization was carried out to the level of the heat exchanger core, and did not include manifolds, mounting brackets, insulation, etc. The desired result from the study is a basic comparison of the heat exchangers made from the two materials. The design optimization was carried out over a wide range of heat-transfer surface parameters and pressure drops. The actual computations were performed by existing AiResearch counterflow plate-fin design program. The best designs were then checked thoroughly by use of a detailed finite-difference performance prediction program.

Before presenting the design-optimization results, a short discussion of the design features of the concepts may be helpful. The ratio of stainless-steel density to that of aluminum is 2.86. If the same physical designs were produced in the two metals, the weight ratio of the two units would be very close to 2.86. However, this approach would ignore the strength differences between the materials, as well as the very large differences in thermal conductivity; average values over the heat exchanger No. 1 temperature range are 7 and 65 Btu/hr-ft²-°F/ft, for stainless and aluminum, respectively. Compared to aluminum, thinner gage stainless steel can be used to provide the required strength. For the heat exchanger plates, this is advantageous; less material would be required. Smaller gages also allow more dense fin matrixes to be produced. However, due to its lower conductivity, and thus lower fin effectiveness, it is possible that thinner fins would not be desirable from the heat transfer point of view. With respect to longitudinal conduction in the core, the major secondary effect limiting the effectiveness of the exchanger, stainless steel offers a distinct advantage. It has often been the case that because of significantly less longitudinal conduction, a stainless steel unit can be designed to be only slightly heavier (say 1.5 times) than an aluminum unit. Thus, at the outset there could be some optimism regarding stainless-steel designs.

Table 4-19 provides the basic results of the core designs optimizations. The actual design configuration includes triangular end sections for fluid entry and exit as shown in Figure 4-13(a). This configuration provides good pressure-drop performance and good flow distribution in the core. During the optimization, the total pressure drop of each design was taken to be 0.272 atm (4.0 psi) rather than the 0.45 atm (6.6 psi) shown in Table 4-2. This was done to enhance overall system optimization. Also the partitioning of the pressure drop between low- and high-pressure sides was done to produce lighter weight designs. Allowance was made for manifolds and ducting.

As can be seen from Table 4-19, the basic sizes of the two designs are very nearly the same. The aluminum design uses taller fins and fewer fin passages: 25 for the aluminum, 47 for the stainless steel. Because of lower conductivity and fin effectiveness, the optimized stainless-steel design uses shorter fins, but many more passages. Its stack height is however less than

TABLE 4-19

COMPARISON OF ALUMINUM AND
STAINLESS STEEL HEAT EXCHANGER DESIGNS

	Heat Exchanger Material			
	Aluminum		Stainless Steel	
Flow passage width, in.	15.00		15.00	
Core length, in.	101.80		120.26	
Number of passages (each stream)	25		47	
Plate-fin stack height, in.	9.99		8.80	
Core weight, lb*	261.7		665.0	
Fluid stream	Hot	Cold	Hot	Cold
Core pressure drop, psi*	1.66	1.35	1.29	1.94
Core fin configuration				
Rectangular fins per in.	24	24	20	16
Fin height, in.	0.075	0.300	0.050	0.125
Fin offset length, in.	0.111	0.111	0.100	0.125
Fin thickness, in.	0.004	0.004	0.003	0.003
*Not including manifolds				

that of the aluminum design: 8.80 inches for the stainless steel as opposed to 9.99 inches for aluminum. The overall weight ratio is 2.54, which is not very different from the basic metal density ratio of 2.86. Thus, the expected reduction of the longitudinal conduction effect did not materialize. This is somewhat simply explained by noting that, in the design procedure, no restriction was placed on core length. Thus, the aluminum design came out with a longer core, resulting in very little longitudinal conduction. Therefore, the use of stainless steel, with respect to longitudinal conduction, has little room to provide improvement. The use of nickel fins in stainless cores did produce slightly lower weight designs. However, the difference did not seem to justify the additional cost involved.

4. Overall Comparison of Stainless-Steel and Aluminum Designs

Table 4-19 shows that a stainless-steel unit would require significantly more material than an aluminum unit. And since the unit price of stainless steel is much more than aluminum, exchangers of the former would have a much higher cost for raw material. It is expected that manufacturing costs for the aluminum units would be greater than for stainless steel; however, it is unlikely that these costs would overcome the material cost differential.

At the present time, the techniques for brazing stainless-steel heat exchangers are well developed. Basically, the same can be said for fluxless-brazed aluminum heat exchangers, but more care is required in the braze operation. Each technique would require extensive tooling and braze fixtures. The stainless-steel braze alloy actually melts and can flow to some extent. With fluxless brazing of aluminum, the braze material flows much less. In fact, the resulting joints are much like a diffusional bonding process. Because of this, aluminum brazing would require fixturing to be somewhat more precise than that for stainless steel. This can be illustrated by assuming that, for a stainless steel unit, the fin sheets in a particular region have not been compressed firmly to the plate. That is, the flat areas of the fin sheets, as shown in Figure 4-13(c), are loosely in contact with the plates rather than being fully seated on the plates. Since the braze alloy used for stainless does flow appreciably, it is highly probable that the braze material will fill the interstices between fins and plates. Thus, good mechanical and thermal bonds would be created. On the other hand, with fluxless brazing of aluminum, due to the relative immobility of the braze alloy the fin-sheet flats should be well seated to the plates. The fixture for aluminum brazing must ensure, therefore, good mechanical contact throughout the core stack.

Braze furnace cleanliness is much more important in brazing aluminum due to the greater potential of oxidation of the material at braze temperatures. To insure cleanliness, it may be necessary to provide a special enclosure for the stacked aluminum heat exchanger and its holding fixture to be used during the brazing cycle.

5. Heat Exchanger Configuration Alternatives

The design of a high-performance heat exchanger requires careful selection of the heat transfer matrix as well as the materials of construction and the application of sophisticated thermal, fluid, and structural design techniques to yield a minimum-cost solution for the unique cable system performance, environment, and life requirements.

Concentrated developments in the aerospace heat exchanger field have yielded a variety of efficient, highly compact heat-transfer matrices of plate-fin, tubular, and other types. Where fluid pressures are not excessively high, the plate-fin type of heat exchanger generally exhibits a lower weight and volume solution.

In this type of construction, heat transfer takes place between parallel flat plates or channels that incorporate finned secondary surfaces. Considerable flexibility is afforded with this type of geometry because the passage

heights and fin form can be varied independently to give an optimized unit for any particular application. The simplest forms of the secondary surfaces are the plain rectangular and triangular fins, but these are rarely used in high-performance heat exchangers because of their relatively poor heat transfer characteristics, which result in units of excessive volume and weight. Several simple modifications can be made to these geometries to enhance their heat-transfer characteristics; the most common is the offsetting of the fins in the direction of flow so that the boundary layer is constantly interrupted. This enhances the heat-transfer coefficient considerably, with only a slight increase in pressure drop.

Plate-fin heat exchanger fabrication involves stacking of the plates (referred to as tube plates) and the chosen fin stock to obtain the desired matrix. In header bar construction, the side walls of each fluid passage are formed by a solid bar having a height equal to the fin height and length equal to the passage length. When the stack of tube plates, fin stock and header bars is complete, the matrix is brazed, using suitable alloys, to obtain an integral structure referred to as a heat exchanger core. Manifolds and duct connections are then welded to the core to obtain the completed heat exchanger. This approach allows great flexibility in design and is well suited to applications where the number of units to be produced is small. However, production of many units of the same design does not yield great reduction in fabrication costs. A further disadvantage is the concentration of material at the tube plate boundaries; here, the thin tube plates join a relatively rigid wall formed by the stack of header bars, leading to reduction in life capability under cyclic thermal conditions.

These two problems are ameliorated by the formed tube plated approach; this concept avoids use of header bars by employing tube plates formed to define the flow passage rather than flat tube plates. A typical formed tube plate is shown in Figure 4-14(a). The large tray-like recess formed in the center of the plate allows the fin stock to be installed as shown in Figure 4-14(b). The example shown is a counterflow recuperator; thus, both offset fins and plain turning section fins are placed in the recess. Note that the tube plate stamping shown here also produces inlet and outlet manifolds for each pass. This feature of the formed tube plate concept avoids the weld-over-braze operations necessary in attaching manifolds to header bar core assemblies.

In this case, core stacking merely involves stacking of the alternate mating formed plates with the desired fin material sandwiched between each pair of plates. Brazing takes place at the formed lip around the plate periphery, resulting in a sheet material-to-sheet material joint. The brazed core requires only addition of duct connections to the integral manifolds to obtain a complete heat exchanger, as shown in Figure 4-15.

The disadvantage of the approach lies in the tool design and fabrication effort required to produce the tube plate stampings. While the investment needed renders the formed tube plate unattractive for small volume production, improved structural efficiency and reduced assembly costs are available for applications involving sufficient volume to amortize the tube plate tooling.

a. Formed Tube Plate

Figure 4-14. Formed Tube Plate Exchanger Detail Parts

Figure 4-15. Typical Formed Tube Plate Counterflow Heat Exchanger

The heat exchanger shown in Figure 4-14 and 4-15, is a recuperator for a closed Brayton cycle power system. This unit was designed for 90 percent effectiveness on the air side and a combined pressure drop on both sides of 2.5 percent. The hot gas inlet temperature is 1257 F. Design point pressures are 53 and 30 psia on the high and low pressure sides respectively. A complete finite difference thermal and stress analysis of the formed tube plate recuperator was performed. The unit is capable of withstanding 1000 start-stop cycles and 30,000 hours of operation at maximum temperature and pressure conditions. The specified start transient temperature ramp is a linear rise from 100 to 1257 F in 3.5 minutes. Note that these conditions are considerably more stressful than encountered in helium refrigerator service.

Turboexpander

Two turboexpander design approaches have been examined:

- (1) Turbine wheel immersed in the cold box with bearings and loading device outside the cold box (cold/warm version)
- (2) Turboexpander completely within the cold box (cold/cold version)

The first approach has been successfully reduced to production hardware; no detailed study was undertaken on this approach. However, a cold/cold version may offer an advantage over existing units by substantially reducing thermal losses. Such a unit would operate on compliant foil process gas bearings with loading provided by an alternator dissipating its power output external to the cold box. Figure 4-16 is a schematic drawing of this cold/cold version turboexpander. The turboalternator design considered for the cold/cold approach is discussed below.

1. Component Design

a. Turbine Wheel

Turbine aerodynamic analysis for the sample baseline system requirements of Table 4-3 shows that a 1.06 in. dia. radial wheel operating at a speed of 80,000 rpm should yield an adiabatic efficiency of 70 percent at the specified inlet conditions and flow rate. Turbine shaft power output would be 2.45 hp.

b. Alternator

The alternator consists of two basic electromagnetic components: the rotor and the stator as seen in Figure 4-17. The rotor is made up of a hollow cylindrical magnet mounted on the shaft of magnetic steel and supported by a high strength nonmagnetic sleeve. Nonmagnetic end pieces are installed at either end of the magnet cylinder to complete the rotor shaft. The stator consists of a core of silicon steel laminations with skewed slots to minimize rotor cogging and a set of three phase windings of magnet wire.

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Figure 4-17. Permanent Magnetic Alternator Cross Section

In order to minimize rotor material problems in a cryogenic environment, a hollow cylindrical magnet configuration is chosen to eliminate bonding material between magnet poles and interpoles. The samarium cobalt magnet cylinder is magnetized tangentially with two poles. The shaft has a hexagonal cross-section to avoid its slipping from the magnet cylinder due to dimensional discrepancy between them at extremely low temperature.

All significant electromagnetic losses occur in the stator assembly where they can be more conveniently handled. Since the alternator will be working at cryogenic temperature, the stator winding resistance is very low, resulting in low copper loss of about 8 watts. Due to the increase in eddy current in the iron core as temperature decreases, the iron loss of 45 watts is about 15 percent higher than that at room ambient. The total loss including stray load loss and windage loss is 72 watts, which results in an alternator overall efficiency of 95 percent.

The dimensional and selected performance data are summarized in Table 4-20.

c. Gas Bearings

The turboexpander rotating assembly is supported by compliant foil process gas bearings of the same type discussed in the compressor subsystem subsection above. These particular gas bearings appear attractive for this application because the flexible foils compensate for differential dimensional changes in the bearings that will occur when the unit is cooled down to near absolute zero. In addition, the bearing material is compatible with low-temperature operation. Figures 4-18 and 4-19 represent some of the preliminary work done on the gas bearings. Figure 4-18 shows the reduction in bearing load capacity at low temperatures and serves as a guideline for sizing the journal bearings. Figure 4-19 indicates a benefit of operating the bearings at low temperature, i.e., a substantial reduction in bearing losses at the operating temperature of 9 R. Total bearing loss (journal bearings and thrust bearing) is estimated to be 58 watts, only 3 percent of the output power of the turbine wheel.

2. Turboalternator Efficiency

Based on the concept of operating the entire rotating assembly at process gas temperatures, the conductive thermal losses can be expected to be significantly reduced over those of turboexpanders requiring warm-end loading. In addition, the use of compliant foil process gas bearings eliminates losses associated with the helium flow used for support of hydrodynamic bearing types. The overall adiabatic efficiency of this type of turboalternator operating at the required conditions can be expected to be about 65 percent ($0.70 \times 0.95 \times 0.97$).

In the baseline cold box optimization resulting in the flows and temperatures shown in Figure 3-3, an efficiency of only 60 percent (see Table 4-3) was assumed for expander number 3. Establishment of this increased efficiency would slightly shift the state points and flows shown in Figure 3-3 resulting in an improved overall cold box efficiency. Moreover, this is a strong indication that the overall efficiency of the three remaining turboexpanders shown in Figure 3-3, can also be expected to operate at slightly increased efficiencies, thereby further increasing cold box efficiency.

Figure 4-18. Effect of Temperature on Foil Journal Bearing Load Capacity

Figure 4-19. Foil Journal Bearing Power Loss as a Function of Temperature

TABLE 4-20
ALTERNATOR DESIGN DATA

<u>Dimensional Data</u>	
Rotor OD, in.	0.826
Rotor ID, in.	0.395
Stator OD, in.	1.958
Stator ID, in.	0.856
Stack Length, in.	1.186
Housing CD, in.	2.037
Housing Length, in.	3.270
Rotor Weight, lb	0.197
Stator Weight, lb	0.737
Electromagnetic Weight, lb	0.934
Total Alternator Weight, lb	1.25
<u>Performance Data</u>	
Rated Voltage per phase, v	110
Power Factor	0.85
Efficiency, percent	95
<u>Material</u>	
Rotor Magnet	Samarium Cobalt
Rotor Shaft	A 4340 Steel
Rotor Sleaving	Inconel 718
Stator Laminations	M-19 Silicon Steel

3. Turboalternator Experience

The design approach suggested above differs substantially from the cold/warm turboexpander currently in use. The concept is not, however, untried. AIResearch has developed a number of high-speed turboalternators that use compliant foil process gas bearings. These were 200,000 rpm cryogenic turboalternators of 6 to 60 watt capacity with two conical foil bearings. A gas turbine of 30 hp at 52,000 rpm with two conical foil bearings and one cylindrical foil journal bearing also was tested successfully. Currently, a miniature heart engine turbopump is under development for operation with hexafluorobenzene. The bearing losses have been measured on the dynamometer to be 0.6 w at 200,000 rpm. This unit has successfully completed a 13,610 hour endurance test running at 200,000 rpm.

SECTION 5

RELIABILITY ENGINEERING ANALYSIS

Reliability, in its simplest form, means that a system will perform the task for which it was intended. It is a means to quantitatively measure how well that task is accomplished. Reliability engineering, however, is much more than a science of measurement.

In the aerospace industry, reliability engineering has developed into a supportive science of its own; most aerospace firms have established departments for these technical specialists whose support is required on virtually all programs. Indeed, the reliability engineering requirements which accompany a major aerospace system account for a substantial fraction of the eventual nonrecurring costs. Perhaps one of the major historical reasons that this science has successfully developed within the aerospace industry has been the potential for disastrous consequences resulting from unreliable systems. Once developed, however, the cost effectiveness of reliability engineering has firmly established its value to both system and component design. Higher initial costs of reliability engineering tend to reduce later costs associated with correction of problems encountered in field applications; it is the correction of these hardware problems during production and post-production phases that increase system costs to several times initial estimates.

The first task of reliability engineering is to assist the designer to assure that reliability is designed into a system, not added on. Reliability engineering should be involved beginning with the design concept, and in a major program, remain active through the system and component design and through production phases. An effective reliability support remains actively involved even after hardware has been shipped, field data reports concerning hardware performance and maintenance action provide an effective feed-back which is useful for product improvements.

Application of the concept of reliability engineering can be expected to result in significant benefit to the successful development of the cable system and its refrigerator system. Utility applications, like aerospace application, do not tolerate unreliable system operation. Since it is expected that a new class of hardware will be required to satisfy cable refrigeration needs, it is important that reliability engineering become a part of the design effort beginning with the initial conceptual design phase. Since this report represents a study of concepts, AiResearch has chosen to include reliability engineering evaluations in this study.

Due to the fact that historical practice in the manufacture of helium level liquefiers and refrigerators has not been to establish and document reliability and maintainability performance data, AiResearch has begun the system reliability studies by established estimates of various component data from experience with similar hardware, adjusted for the specific application. This data, presented in Appendix A, is but a start.

Two basic nonexclusive means exist to provide high reliability. These are: the selection of subsystems and components which demonstrate characteristically high reliability and improved maintainability, and providing for redundancy where the initial selection of components and subsystems does not provide sufficiently high reliability. The study reported in this section utilizes both approaches.

The purpose of the study was to evaluate candidate refrigerators against the reliability requirements for a superconducting cable system. These requirements, noted in Section 3, are expressed in terms of minimum acceptable times between failure; a failure is defined as the loss of active refrigeration to a segment of a cable for more than eight hours. A minimum time between failures of one year is specified for each cable circuit, and a minimum time of ten years is specified for the simultaneous occurrence of a failure in both circuits, as discussed in Section 3.

To evaluate the candidates against a set of requirements based on this definition it was necessary to incorporate both reliability and maintenance factors into the quantitative models. Both equipment failure rate and repair time data from the Component Reliability and Maintenance Data studies of Appendix A were utilized. The impact of various corrective maintenance policies and the number of repair crews involved were also considered. Insufficient data on the preventive maintenance requirements of the individual refrigerator equipments precluded the inclusion of this aspect of factors contributing to refrigerator down time into the studies.

The scope of the study included the evaluation of three refrigerator concepts. These were:

1. The baseline refrigerator (Figure 3-3) consists of the minimum number of components to cool the cables.
2. The baseline refrigerator as modified by the application of reliability engineering improvement techniques at the refrigerator and refrigerator substation level. This refrigerator is called the optimized baseline refrigerator in this study.
3. A double-size, double-capacity refrigerator incorporating the same reliability improvement techniques applied to the optimized baseline refrigerator. This refrigerator is called the double-size refrigerator in this study.

This study emphasized the application of reliability improvement techniques at the highest level which appeared to be feasible and consistent with both cost effectiveness and the reliability requirements. The most prominent example of this is the incorporation of a standby redundant cold box subsystem and intrastation compressor and cold box switching subsystem at each refrigerator substation.

SYSTEM STUDY MODELING PROCESS

The process described in this section is applicable to situations where redundancy allows continued system operation while corrective maintenance is being performed. The process which was used in these studies is based on the techniques developed in "System Reliability Engineering," by G. H. Sandler, Prentice Hall Inc., 1963.

For a maintained system, the system can be defined as being in various operating or failed states. The modeling process reflects how the system passes back and forth from each of these possible states. The fundamental assumption upon which this process is based is that the future state of the system depends only upon its immediate past history. This process is called a Markov Process. In these studies, the parameter which was assessed was the mean time for the system to transition to a failed state from the state in which all equipments were operating. The values of mean time between failure (MTBF) calculated in the system studies reflect this time.

The information required to construct a model based on this transitioning process is (1) the equipment failure distribution, (2) the equipment repair distribution, (3) the system configuration, (4) the repair (corrective maintenance) policy, and (5) the possible states which the system can be in.

Equipment Failure and Repair Distributions

It was assumed in these studies that the equipment failure and repair distributions can be described by a negative exponential distribution: that is that the equipment failure rate (λ) and repair rate (μ) are constant. The application of this assumption to equipment failure rates is based on the fact that preventive maintenance and/or replacement of failed items is employed, and thus the system will be composed of a large number of equipments of mixed ages. Although many equipment components will exhibit some other failure distribution due to infant mortality or wearout, experience indicates that an exponential failure distribution provides a good model for a system with a large population of equipments of varied ages. The application of a negative exponential distribution to the equipment repair rates is based on the fact that the refrigerator is composed of a large number of equipments which will fail relatively frequently and which have a relative short and constant repair time. The items which have long and perhaps varied repair times such as a cold box heat exchanger have a very low failure rate and thus do not have the same impact on system downtime.

With the assumption of the negative exponential failure distribution, the probability of equipment failure in the interval of time $[0, t]$ is $F(t) = 1 - e^{-\lambda t}$. The probability of failure in the interval of time t to $t + dt$, given that the equipment has not failed prior to t , is λdt . In a similar manner, the probability that a repair has been completed by time t is $R(t) = 1 - e^{-\mu t}$. The probability that the repair will be completed in the interval t to $t + dt$, given that the repair has not been completed at time t , is μdt . If $t = 1$ hour these probabilities are λ and μ .

Other Parameters

A description of the system is essential to define the various possible states which the system can be in the interval of time t to $t + dt$ under consideration. These states are mutually exclusive, that is, a system cannot be in two states at the same time.

Definition of the repair policy is needed since this policy may impact on the repair rate, and hence the probability that a system will be in a given state. If the repair policy is to repair this equipment immediately, the repair rate is $\mu = 1/\tau$. If the repair policy is to defer repair on this item for-say-100 hours to reduce the number of maintenance crews required to maintain the system then the repair rate is $\mu = 1/(100 + \tau)$.

Example of Model Construction

Figure B-3 (Appendix B) provides an example of the construction of a model for a redundant system composed of two elements having identical failure rates λ , and repair rates μ .

Part I of this figure defines the system and implies a repair policy where items are repaired or replaced as they fail.

Part II of this figure describes the states in which the system can be.

Part III of this figure presents the probability matrix, P , for the system passing among its various states. The states 0, 1, and 2 of the columns are the input states at time t . The states 0, 1, and 2 of the rows are the output states at time $t + dt$. For example, the probability that the system in state 0 will stay in state 0 is $1 - 2\lambda$, which is the probability that neither element fails during the interval t to $t + dt$.^{*} The probability that one of these elements will fail and the system transition to state 1 in this interval is 2λ . The probability that the system in state 1 will have an element repaired and transition to state 0 is μ . The probability that the second element will fail and the system transition to state 2 is λ . The probability that neither the failed element is repaired nor the second element fails is $1 - (\lambda + \mu)$, which is the probability that the system remains in state 1 in this interval. The sum of the probabilities of the possible transitions of a system starting in a given state is the sum of the probabilities of the system transitioning from or remaining in that state (i.e. the sum of the probabilities in that row), which must always equal 1.

The probability matrix, Q , represents the probabilities of the system transitioning among its various non-failed states. It has been shown that the mean time the system will remain in a non-failed state given that it started in a non-failed state can be found from the probability matrix $(I - Q)^{-1}$, where I is an identity matrix. The matrix $(I - Q)^{-1}$ is known as the fundamental matrix. From the fundamental matrix the mean time to system failure from any non-failed state is the sum of the terms in the row corresponding to that state. The MTBF of the system in this example is found in Part IV of Figure B-3 by this method.

^{*}The probability that both elements fail, λ^2 , in this interval is negligible and is not reflected in the matrices.

RELIABILITY ANALYSIS SUMMARY

Baseline Refrigerator

The baseline refrigerator was found to be unable to provide the level of reliability required. The assessed mean-time-between-failures of a system utilizing the baseline refrigerator was found to be 1068 hours between failures for a single cable circuit (1 year = 8,760 hours MTBF required) and 12,659 hours between failures for the two circuit cable system (10 years = 87,600 hours MTBF requirement). These assessed values were based on the availability of a repair crew to work on items when they failed, and hence that there would be no queuing of refrigerators or expanders awaiting repair.

Optimized Baseline Refrigerator

The optimized baseline refrigerator was established by sufficient modification to the baseline refrigerator to achieve the required level of reliability when the cable refrigeration system was supported by an "instrumentation" repair crew and a "general" repair crew operating under a "fix as soon as practical" corrective maintenance policy. A repair crew was considered to be an optimized number of technical and maintenance personnel to perform the corrective maintenance actions which might be expected to occur at a station. Under the "fix as soon as practical" corrective maintenance policy the "instrumentation" repair crew would be available to attend to and restore process stream purity monitoring and vacuum maintenance equipments within the repair times determined by the studies of Appendix A. The "general" repair crew could defer corrective maintenance on other items having a redundant back-up for 126 hours.

The assessed MTBF's for a system employing the optimized baseline refrigerator and the above corrective maintenance provisions are 12,152 hours between failures for a cable circuit, and more than 3.416×10^6 hours for simultaneous failures in both cable circuits.

Double-Size Refrigerator

The study of the double size refrigerator was conducted under the assumption that there would be no significant changes to the failure rate and maintenance data used in the prior studies. Given that this assumption is reasonably correct, the fact that only twelve double size refrigerators are required to support the cable system (compared with twenty four refrigerators for the other two candidates) offers a considerable advantage in terms of reduced corrective maintenance actions. This advantage, however, is not reflected in the cable circuit and cable system MTBF's. These values are approximately the same as those obtained for the optimized baseline. The cable circuit MTBF was 13,192 hours (12,152 hours optimized baseline), and the dual cable circuit MTBF was greater than 3.410×10^6 hours (3.416×10^6 hours optimized baseline). This relative equality between the optimized baseline and double-size refrigeration systems lies due to the use of stand-by redundancy and compressor and cold box switching at the refrigerator stations. The differences between the two systems is in the refrigerator substations, and the substation contribution to cable circuit and cable system unreliabilities is small in comparison to the unreliabilities contributed by the expander substations, where both refrigerators are the same.

Refrigerator Comparison

The refrigerator to be eventually selected from this and future studies is the one which is the most cost effective in terms of: (1) system acquisition costs, (2) maintenance costs, (3) failure consequence costs, and (4) operating costs. These costs are related to this study in terms of the quantity of equipments and maintenance support necessary to attain and maintain the reliability required of the system. The number of equipments and a corrective maintenance policy have been identified for the three system concepts. The determination of these cost factors was not a part of these studies.

An optimized baseline system offers somewhat more flexibility in terms of intra-substation compressor and cold box scheduling to maintain active refrigeration to one cable circuit for preventive maintenance or for a multiple refrigerator failure condition. This flexibility is due to the fact that the optimized baseline refrigerator substation could tolerate the loss of three cold box subsystems and two compressor subsystems without impairing this capability. A double-size refrigerator substation can tolerate the loss of only two cold box and one compressor subsystems under these conditions. This advantage of the optimized baseline does not appear great in an analysis considering the loss of both cable circuits simultaneously because of refrigerator failures. In terms of failure consequence costs (i.e., the costs to the areas receiving power from the cable system), and actual experience by a repair maintenance crew, this advantage might be much greater than indicated.

The double-size refrigerator system concept has one half of the number of refrigerators. The determination of the cost/size relationship for the acquisition, maintenance, and energy costs may show that the double-size refrigerator is the most cost effective of the candidates studied, particularly if subsequent studies show that additional redundancy will be required (a comparison of the initial costs based upon Strobbridge's data is included in Section 6).

SYSTEM LEVEL PROBLEM AREAS DISCLOSED BY ANALYSIS

Time Allowed for Corrective Action

The reliability requirements are based on the allowance of eight hours following the loss of active refrigeration to assemble the crew at the site, diagnose the anomaly, identify the cause and correct it, verify the correction, and restore active refrigeration to the cable segment. During the time that the cable segment is without active refrigeration the thermal gradient across the cable insulation system will increase. The criticality of this increase to the cable, and the time required to restore the cable insulation system to its steady-state operating conditions were not defined at the time of this analysis. Since a second failure in a refrigerator might occur before these steady-state conditions had been restored there might not be eight hours available to implement and complete a second corrective action.

In the system studies two cases were considered. The first case assumed that only those failures which required more than eight hours for repair and restoration were critical. The second case assumed that every failure was critical. This was done to establish upper and lower reliability bounds to facilitate further system studies should the eight hour allowable corrective action time prove to be optimistic. For this study, only the upper bound was evaluated in relation to the reliability requirements.

Intrastation Switching

The concept of intrastation switching of compressors and cold boxes to maintain active refrigeration for both reliability and preventive maintenance considerations has much to recommend it. For example, the addition of a single stand-by cold box subsystem to each refrigerator station is much more cost effective in terms of acquisition and maintenance costs than the addition of one stand-by cold box for each circuit or for each cable segment serviced by the substation.

The negative side of this concept is that the switching subsystem can become a limiting factor in the potential reliability level attainable at the substation. In this event, no amount of reliability improvements added to the individual refrigerators can compensate for an unreliable switching subsystem. The incorporation of reliability improvement techniques to this subsystem may result in considerable additional equipments, perhaps to the point where the cost effectiveness of the concept might no longer be justified. A poor design might result in the switching subsystem becoming a single failure point, where failure means the loss of active refrigeration from the entire substation.

BASELINE REFRIGERATOR

System Description

The cable refrigerator system consists of twenty-four refrigerators and twenty-four remote expanders grouped in six refrigerator substations and seven remote expander substations. Figures 5-1, 5-2, and 5-3 show the distribution of these substations along the cable system, and the arrangement of the refrigerators and remote expanders within their respective substations. As shown in these figures, each cable circuit is divided into twelve separately refrigerated segments. The refrigerators for each segment in the cable circuit comprise a cable circuit refrigerator subsystem. A single refrigerator and remote expander comprise the refrigerator for each cable circuit segment in this system.

The baseline cable segment refrigerator consists of a compressor subsystem, a coldbox subsystem, and a remote expander subsystem. Figure 5-4 illustrates the general arrangement of these subsystems. A screw compressor was chosen for the baseline compressor subsystem analysis; it was anticipated that most operating experience in the near future will be accumulated by a screw compressor subsystem. A gas bearing turboexpander was selected for the baseline coldbox and remote expander subsystems analysis because of the high reliability reflected in the component reliability and maintenance data reported in Appendix A. Figures A-1, A-3, and A-4 in Appendix A illustrate the details of these subsystems.

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Figure 5-1. Refrigeration Substation (RS) and Remote Expander Substation (ES) Distribution for One, Typical, Superconducting Cable Circuit

TOTAL NO. OF REFRIGERATORS REQUIRED FOR BASELINE CABLE
SYSTEM = 24 (6 SUBSTATIONS WITH 4 REFRIGERATORS PER SUBSTATION)

Figure 5-2. Minimum Refrigerator Equipment Required
for a Baseline Refrigerator Substation

REQUIRED NO. OF REMOTE EXPANDERS:

$$(4/\text{FULL SUBSTATION}) \times (5 \text{ FULL SUBSTATIONS}) + (2/\text{TERMINAL SUBSTATIONS}) \times (2 \text{ TERMINAL SUBSTATIONS}) = 24 \text{ REMOTE EXPANDERS REQUIRED}$$

* SUBSTATION SHOWN IS TYPICAL FOR ALL REMOTE EXPANDER SUBSTATIONS EXCEPT THOSE AT THE CIRCUIT TERMINALS

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Figure 5-3. Minimum Expander Equipment Required for a Baseline Remote Expander Substation

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Figure 5-4. Basic Configuration of a Refrigerator and Remote Expander to Cool One Cable Circuit Segment

Summary of Results and Conclusions

Four parameters were assessed. These parameters and their assessed values are:

- 1a. Mean-time-between-failures resulting in the shutdown of a cable circuit because of the loss of active refrigeration for more than eight hours in any of its segments: assessed $MTBF_1 = 1,068$ hours.
- 1b. Mean-time-between-failures for all failures (including those contributing to $MTBF_1$) resulting in loss of active refrigeration in any segment of a cable circuit: assessed $MTBF_2 = 67$ hours.
- 2a. Mean-time-between-failures resulting in the shutdown of both cable circuits because of the simultaneous loss of active refrigeration for more than eight hours in one or more segments in both cable circuits: assessed $MTBF_3 = 12,659$ hours.
- 2b. Mean-time-between-failures for all failures (including those contributing to $MTBF_3$) resulting in the simultaneous loss of active refrigeration in one or more segments in both cable circuits: assessed $MTBF_4 = 336$ hours.

The required MTBF's are: (1) an MTBF of 8,760 hours between cable circuit shutdowns because of a lack of active refrigeration for more than eight hours in any cable segment; and (2) an MTBF of 87,600 hours between cable system shutdowns because of a simultaneous lack of active refrigeration for more than eight hours in any segment of both cables. The assessed MTBF's given above show that the baseline refrigerator is not capable of meeting either requirement.

Baseline Cable Segment Refrigerator

The baseline cable segment refrigerator consists of an in-series arrangement of the compressor, coldbox, and remote expander subsystems. Each item within these subsystems must function properly or the refrigerator must be shut down while corrective action is taken. The cable segment serviced by this refrigerator is thus without active refrigeration during this period of time. Therefore:

$$\lambda = \sum_{i=1}^n \lambda_i$$

the failure rate associated with the loss of active refrigeration	the sum of the failure rates of the individual components
---	---

And:

$MTBF = 1/\lambda$ = the mean-time-between-failure where a failure is defined as the loss of active refrigeration for the cable segment

Both $MTBF_{1R}$ (MTBF for failures resulting in the loss of active refrigeration to the cable segment for more than eight hours) and $MTBF_{2R}$ (the MTBF for all failures resulting in the loss of active refrigeration to the cable segment) were determined by this method. The subscript R is added to these MTBF designations to avoid confusion with the similar MTBF parameters for the cable circuit refrigerator and cable refrigerator systems.

There are two exceptions to the series relationship of the baseline refrigerator components. Redundancy has been provided in two areas described below.

- Coldbox sensors which will require more than eight hours to replace, illustrated and described in Figure B-1*
- Protection of the coldbox from contamination caused by failure of the filter/adsorber in the compressor subsystem, illustrated and described in Figure B-2

Table 5-1 presents a summary of the failure rates and repair times for each of the baseline refrigerator components. The failure rates associated with the redundant coldbox sensors are included in this table. The filter/adsorber redundancy--as defined in Figure B-2--has no impact on the data presented.

The failure rates presented in columns 4 and 7 of Table 5-1 were summed and the reciprocals of these sums taken to determine the $MTBF_{1R}$ and $MTBF_{2R}$ respectively: i.e.:

$$MTBF_{1R} = \left[\sum N \lambda \text{ from col. 7} \right]^{-1} = \left[78.05/10^6 \right]^{-1} = 12,812 \text{ hours}$$

$$MTBF_{2R} = \left[\sum N \lambda \text{ from col. 4} \right]^{-1} = \left[1244.29/10^6 \right]^{-1} = 804 \text{ hours}$$

Baseline Cable Circuit Refrigerator System

The baseline cable circuit refrigerator system consists of the in-series arrangement of twelve cable segment refrigerators. Therefore:

$$MTBF_1 = \left[12 \left(\frac{1}{MTBF_{1R}} \right) \right]^{-1} = \left[12 \sum N \lambda \text{ from col. 7} \right]^{-1} = \left[936.6/10^6 \right]^{-1} \\ = 1,068 \text{ hours}$$

$$MTBF_2 = \left[12 \left(\frac{1}{MTBF_{2R}} \right) \right]^{-1} = \left[12 \sum N \lambda \text{ from col. 4} \right]^{-1} = \left[14,931.48/10^6 \right]^{-1} \\ = 67 \text{ hours}$$

The λ and associated analysis are presented in Appendix B.

SUMMARY OF FAILURE RATES AND REPAIR TIMES FOR THE BASELINE REFRIGERATOR

Item Ref No.	Item Description	Failure Rates and Repair Times		Refrigerator Failure Rates and Down Times for Repairs			Refrigerator Failure Rates and Expected Down Times for Repairs Requiring More Than 8 Hours			
		1	2	3	4	5	6	7	8	9
		Item Failure Rate, $\lambda \times 10^6$ Hours	Est Repair Time τ Hours	Qty Reqd in Baseline N	Refrig. Failure Rate $N\lambda \times 10^6$ Hours	Down Time Per 10^6 Hours For Repair $D_T = \tau N\lambda \times 10^6$	Est. of Percent of Repairs Not Completed in 8 Hours	Refrig. Failure Rate $\lambda = \tau N\lambda \times 10^6$ Hours	Est. Repair Time τ Hours	Down Time Per 10^6 Hours for Repair of Failures $D_T = \tau N\lambda \times 10^6$
1	Compressor Subsystem									
1.1	Screw Compressor	6	8	2	12	96	0.05	0.6	11	6.6
1.2	Oil Pump/Filter	7	3	2	14	42	---	---	---	---
1.3	Helium/Liquid Oil Separation	50	4	2	100	400	---	---	---	---
1.4	Oil Cooler	2	6	2	4	24	---	---	---	---
1.5	Oil Pressure Regulator Valve	5	3	2	10	30	---	---	---	---
1.6	Helium After Cooler	2	8	2	4	32	.05	.2	11	2.2
1.7	Helium Pressure Regulator Valve	5	3	3	15	45	---	---	---	---
1.8	Comp. Inlet Filter	.05	3	2	.1	0.3	---	---	---	---
1.9	Primary Filter/Adsorber	100	8	1	100	800	.05	5	11	55
1.10	Oil Contamination Monitor	150	8	1	150	1200	.05	7.5	11	82.5
1.11	H ₂ O Contamination Monitor	100	8	1	100	800	.05	5	11	55
1.12	Gas Contamination Monitor	150	8	1	150	1200	.05	7.5	11	82.5
1.13	Secondary Filter/Adsorber	.05	128	1	.05	6.4	1	.05	128	6.4
1.14	Plumbing	26	4	1	26	104	---	---	---	---
1.15	Controls, Instrumentation and Interlocks	60	3	1 set	60	180	---	---	---	---
				Total	745.16	4959.7	Total	25.85		290.2
Compressor Subsystem										
Failure Rate = $\Sigma N \lambda$				745.15/10 ⁶ Hour			25.85/10 ⁶			
Mean-Time-to-Repair (MTTR) = $\Sigma D_T / \Sigma N\lambda$				6.66 Hours			11.23 Hours			
Repair Rate = $\mu = 1/\text{MTTR}$				0.1502 Hour			.0890 Hour			

TABLE 5-1 (Continued)

Item Ref. No.	Item Description	Failure Rates and Repair Times		Refrigerator Failure Rates and Down Times for Repairs			Refrigerator Failure Rates and Expected Down Times for Repairs Requiring More Than 8 Hours			
		1	2	3	4	5	6	7	8	9
		Item Failure Rate $\lambda \times 10^6$ Hours	Est Repair Time τ Hours	Qty Reqd in Baseline N	Refrig. Failure Rate $\lambda \times 10^6$ Hours	Down Time Per 10^6 Hours For Repair $D_T = \tau \lambda \times 10^6$	Est. of Percent of Repairs Not Completed in 8 Hours	Refrig. Failure Rate $\lambda = \lambda \times 10^6$ Hours	Est. Repair Time τ Hours	Down Time Per 10^6 Hours for Repair of Failures $D_T = \tau \lambda \times 10^6$
11	Cold Box Subsystem									
11.1	Main Cold Box	2	128	1	2	256	1	2	128	256
11.2	Filter/Adsorber Vacuum Module	1	24	1	1	24	1	1	24	24
11.3	Expander Vacuum Module	0.5	6	3	1.5	9	.025	.0375	9	0.34
11.4	Pressure Relief Valve Vac Module	0.5	8	1	1	8	.16	.16	11	1.76
11.5	Heat Shield	0.5	128	2	1	128	1	1	128	128
11.6	Heat Exchanger No. 1	1.25	128	1	1.25	160	1	1.25	128	160
11.7	Heat Exchanger No. 2	1	128	1	1.	128	1	1	128	128
11.8	Heat Exchangers Nos. 3 to 5	0.6 ea.	128	3	1.8	230.4	1	1.8	128	230.4
11.9	Filter Adsorber	5	128	1	5	640	1	5	128	640
11.10	Throttling Valve	8	6	2	16	96	.025	0.4	9	3.6
11.11	Turbo Expander	10	6	3	30	180	.025	0.75	9	6.75
11.12	Press. Relief/Reg. Valve	5	8	2	10	80	.16	1.6	11	17.6
11.13	Main Cold Box Plumbing	0.55	128	1 set	0.55	70.4	1	0.55	128	70.4
11.14	Vacuum Module Plumbing	6.1	8	1 set	6.1	48.8	.16	0.976	11	10.74
11.15	Main Cold Box Sensors	0.33*	128	40 groups	13.2	1689.6	1	13.2	128	1689.6
11.16	Turbo Expander Sensors	0.33*	6	12 groups	4	24	.025	0.1	9	0.9
11.17	Cold Box/Cable Interface Sensors	0.33	8	4 groups	1.32	10.56	.16	0.2112	11	2.32
11.18	Vacuum Pump and Assoc. Equipment & Instruments	200	8	1	200	1600	.05	10.	11	110
11.19	Controls and Sensors (not included above)	20	3	1 set	20	60	---	---	---	---
				Total	316.72	4597.96	Total	41.03		3480.41
	Cold Box Subsystem									
	Failure Rate = $\Sigma \lambda$					316.72/10 ⁶ Hour		41.03/10 ⁶ Hour		
	Mean-Time-to-Repair (MTTR) = $\Sigma D_T / \Sigma \lambda$					17.185 Hours		84.83 Hours		
	Repair Rate = $\mu = N = 1/MTTR$					.0582 Hour		.0118 Hour		

TABLE 5-1 (Continued)

Item Ref No.	Item Description	Failure Rates and Repair Times		Refrigerator Failure Rates and Down Times for Repairs			Refrigerator Failure Rates and Expected Down Times for Repairs Requiring More Than 8 Hours			
		1	2	3	4	5	6	7	8	9
		Item Failure Rate $\lambda \times 10^6$ Hours	Est Repair Time τ Hours	Qty Reqd in Baseline N	Refrig. Failure Rate $N\lambda \times 10^6$ Hours	Down Time Per 10^6 Hours For Repair $D_T = \tau N \lambda \times 10^6$	Est. of Percent of Repairs Not Completed in 8 Hours	Refrig. Failure Rate $\lambda = \%N\lambda \times 10^6$ Hours	Est. Repair Time τ Hours	Down Time Per 10^6 Hours for Repair of Failures $D_T = \tau N \lambda \times 10^6$
III	Remote Expander Subsystem									
III.1	Vacuum Box	1	40	1	1	40	1	1	40	40
III.2	Turbo Expander	10	6	1	10	60	.025	0.25	9	2.25
III.3	Metallic Particle Containment	50	6	1	50	300	.025	1.25	9	11.25
III.4	Press. Relief/Reg. Valve	5	6	2	10	60	.025	0.25	9	2.25
III.5	Vacuum Box Plumbing	2.1	40	1 set	2.1	84	1	2.1	40	84.
III.6	Vacuum Box Sensors	0.33	40	4 groups	1.32	52.8	1	1.32	40	52.8
III.7	Vacuum Pump and Assoc. Equipment and Instruments	100	8	1 set	100	800	.05	5	11	55
III.8	Controls, Instruments and Interlocks	8	3	1 set	8	24	---	---	---	---
				Total	182.42	1420.8	Total	11.17		247.55
Remote Expander Subsystem										
Failure Rate = $\Sigma N\lambda$				182.42/10 ⁶			11.17/10 ⁶			
Mean-Time-to-Repair (MTTR) = $\Sigma D_T / \Sigma N\lambda$				7.79 Hours			22.16 Hours			
Repair Rate = $\mu = 1/\text{MTTR}$				.1284 Hour			.0451			

Baseline Cable Refrigeration System

The baseline cable system refrigeration system consists of two baseline cable circuit system refrigeration systems. The reliability requirement for the cable system refrigeration is concerned with the shutdown of both cable circuits. For this reason the cable circuit refrigeration systems were considered to represent an active redundant arrangement of the two systems.

In addition to the cable segment refrigerator failure rates, three additional parameters are required for this analysis:

1. The rate at which the restoration of a failed refrigerator can be accomplished: i.e., the repair rate, μ .
2. The availability of a repair crew to accomplish the repair.
3. The time available to accomplish the repair before the cable system must be shutdown.

To determine the first parameter, the following operation is performed using the data from Table 5-1.

$$\mu_{R1} = \text{mean refrigerator repair rate} = \sum(N\lambda) / \sum(D_T)$$

Columns 4 and 5 of Table 5-1 are used to determine the mean refrigerator repair rate when all refrigerator failures are considered. Columns 7 and 9 of Table 5-1 are used to determine the mean refrigerator repair rate associated with only those failures which require more than eight hours of repair time. These parameters are:

$$\begin{aligned} \mu_{R1} &= \text{repair rate for failures requiring more than eight hours of repair time} \\ &= (78.05/10^6) / (4018.16/10^6) = .0194 \end{aligned}$$

$$\begin{aligned} \mu_{R2} &= \text{repair rate for all failures} \\ &= (1244.29/10^6) / (11823.26/10^6) = .1052 \end{aligned}$$

The second and third parameters, the availability of a repair crew and the time allowed for repair were evaluated in the following manner.

The Mean-Time-To-Repair (MTTR) a failure requiring more than eight hours of repair time is $1/\mu_1 = 52$ hours.

$$\text{The MTBF}_{2R} \text{ is 804 hours, and } \lambda_{2R} = \frac{1}{808} = 1244.29/10^6.$$

While one refrigerator is shutdown for 52 hours for a repair there is a $1 - \exp(-11\lambda_{R2} \times 52 \text{ hours}) = 0.509$ probability that one or more refrigerators in the same circuit will fail. There is a probability of $1 - \exp(-12\lambda_{R2} \times 52 \text{ hours}) = 0.540$ that one or more refrigerator failures will occur in the

second circuit. The probability that one or more additional failures will occur in the cable system during this time is:

$$P = 0.509 + 0.540 - (0.509)(0.540) = 0.774$$

Any one of these possible refrigerator failures could result in a cable circuit shutdown unless they occur near the end of the 52 hour time span. If one of these failures occurs in the second circuit (probability = 54 percent) both circuits may have to be shut down unless the repair crew concentrates all of its effort on keeping one circuit refrigeration system in operation.

The above assessment led to the conclusion that more than one repair crew will be required.

The baseline cable system refrigeration system was next assessed under the assumption that there would be a repair crew available to correct any refrigerator failure before the time available for repair became a critical factor.

Using this assumption, the cable circuit refrigeration system can be considered as a single item having a failure rate of $12(\lambda_R)$ and a repair rate of μ_R . Figure B-2 presents the reliability logic block diagram and model development utilized in this analysis where:

$$MTBF_3 = \frac{3\lambda_1 + \mu_{1R}}{2\lambda_1^2} = \frac{3(936.6/10^6) + .0194}{2(936.6/10^6)^2} = 12,659 \text{ hours}$$

and

$$MTBF_4 = \frac{3\lambda_2 + \mu_{2R}}{2\lambda_2^2} = \frac{3(14,931.48/10^6) + .1052}{2(14,931.48/10^6)^2} = 336 \text{ hours}$$

These MTBF's fall far short of the 87,600 hour MTBF requirement.

OPTIMIZED BASELINE REFRIGERATOR

System Description

The areas listed below were major contributors to the refrigerator down times listed in columns 5 and 9 of Table 5-1. These areas appear to lend themselves to the use of stand-by redundancy at the refrigerator level.

1. Liquid Oil/Helium Separation Equipment (Item 1.3 of Table 5-1) in the compressor subsystem. Illustrated and quantified in Figure B-4.
2. Filter Adsorbers (Item 1.9 of Table 5-1) in the compressor subsystem. Illustrated and quantified in Figure B-5.

3. Helium Purity Monitoring Equipment (Items 1.10, 1.11, and 1.12) of Table 5-1 in the compressor subsystem. Illustrated and quantified in Figure B-5.
4. Filter Adsorbers (Item 11.9 of Table 5-1) in the cold box subsystem. Illustrated and quantified in Figure B-6.
5. Vacuum Pump and Associated Equipment (Items 11.18 and 11.7 of Table 5-1) in the cold box and remote expander subsystems. Illustrated and quantified in Figure B-7.

Other changes to the baseline refrigerator which were made to reduce areas which made large contributions to refrigerator downtime are listed below.

1. Additional compartmentation of the cold box to facilitate the repair or replacement of HX1 (Item 11.6 of Table 5-1) and reduce the estimated down time associated with these operations from 128 hours to 40 hours. Illustrated in Figure B-8.
2. The addition of redundant helium purity monitoring equipment to monitor the status of the filter adsorbers in the cold box. Illustrated and quantified in Figure B-6.
3. A change in the cold box heat exchanger monitoring philosophy from the concept of monitoring the condition of each heat exchanger to the concept of monitoring the condition of these items as two units: (1) HX1, and (2) HX2 through HX5 (see Figure 3-3). This change will reduce the number of difficult to replace sensor groups in this area (Item 11.15 of Table 5-1). Illustrated in Figure B-8.
4. The addition of two additional sensors to the sensor groups described in 3 above. Illustrated and quantified in Figure B-8.

The cold box with its heat shields, heat exchangers, and plumbing was the largest contributor to refrigerator down times in excess of eight hours. It was not considered as a candidate for stand-by redundancy at the refrigerator level for two reasons. First, it will probably be the most expensive item in the refrigerator. Second, the feasibility of cooling a stand-by unit down to its required operating temperature within eight hours or less has not been determined. One method which appears feasible to protect the system against the loss of this critical item is to provide a spare cold box subsystem at each refrigerator substation. Because the screw compressors have the capability for variable compression capacities the spare cold box could be maintained at a nominal temperature which would allow a rapid cooldown to the operating temperature by the production of a small additional flow from one of the active compressors.

This concept of stand-by redundancy for the cold box subsystem will require a switch-over subsystem to supply the stand-by unit's helium requirement from any one of the four compressor subsystems. This switch-over subsystem would also be required to perform the switching required to allow the use of the stand-by unit in any one of the four refrigerators. An

extension of this switchover subsystem was included to allow the employment of any one of the four compressor subsystems or the five cold box subsystems (including the stand-by) to provide active refrigeration to any of the four cable segments serviced by the refrigeration station. This would enhance the capability of maintaining active refrigeration for one cable circuit in the event of multiple failures either within a refrigerator substation or along the cable system.

Figure B-9 illustrates these concepts of stand-by redundancy for the cold box subsystem and intrastation compressor and cold box subsystem switching. The basic mathematical models required for quantitative assessments are also derived in this figure.

Summary of Results and Conclusions

Four parameters were assessed. These parameters and their assessed values are:

- 1a. Mean-time-between-failures resulting in the shutdown of a cable circuit because of the loss of active refrigeration for more than eight hours in any of its segments: assessed $MTBF_1 = 12,152$ hours.
- 1b. Mean-time-between-failures for all failures (including those contributing to $MTBF_1$) resulting in loss of active refrigeration in any segment of a cable circuit: assessed $MTBF_2 = 359$ hours.
- 2a. Mean-time-between-failures resulting in the shutdown of both cable circuits because of the simultaneous loss of active refrigeration for more than eight hours in one or more segments in both cable circuits: assessed $MTBF_3 = 3,416,341$ hours.
- 2b. Mean-time-between-failures for all failures (including those contributing to $MTBF_3$) resulting in the simultaneous loss of active refrigeration in one or more segments in both cable circuits: assessed $MTBF_4 = 58,524$ hours.

The required MTBF's are: (1) an MTBF of 8,760 hours between cable circuit shutdowns because of a lack of active refrigeration for more than eight hours in any cable segment; and (2) an MTBF of 87,600 hours between cable system shutdowns because of a simultaneous lack of active refrigeration for more than eight hours in any segment of both cables. The values for $MTBF_1$ and $MTBF_3$ which are based upon the consideration of those failures requiring more than eight hours to repair and a "fix as soon as practical" corrective maintenance policy for stand-by redundant elements, exceed these requirements.

The assessment of $MTBF_3$ and $MTBF_4$ revealed that the reliability of the switch over subsystem becomes the ultimate limit for the reliability achievable for the concept of stand-by redundancy at the refrigeration station level for the cold box subsystem with intra station cold box and compressor switching capability. The implications of this limit are not considered to be important as long as (1) the switch over subsystem can be repaired within the time

allowable before the cable system must be shutdown for its protection, or (2) the switch over subsystem MTBF is not substantially greater than 525,600 hours.

Optimized Baseline Cable Segment Refrigerator

The analysis of the optimized baseline refrigerator was iterated to determine the effects of different corrective maintenance policies upon refrigerator reliability. These policies can briefly be described as (1) "fix only when refrigerator has failed", and (2) "fix as soon as practical". In evaluating these policies it was assumed that an item failure and its repair process has no impact on the other items in the refrigerator other than resulting in a refrigerator shutdown. That is, each item is independent from the other items in the refrigerator.

1. Fix-Only-When-Refrigerator-Failed Corrective Maintenance Policy

The "fix only when failed" policy is based upon the implementation of corrective action when the refrigerator has failed. For the in-series items in the refrigerator, the situation is the same as that described for the baseline (i.e. - any failure is a refrigerator failure). For the items with a stand-by redundant element, however, a switchover is accomplished either automatically or manually when a failure occurs, and the refrigerator continues to function until either the second element fails or the failure of a series item elsewhere within the system fails and results in refrigerator failure.

When refrigerator failure does occur, the repair crew can either (1) fix only the item causing refrigerator failure, or (2) fix the item causing refrigerator failure as well as any elements in redundant circuits which may have failed. In the first case the stand-by redundant circuits will continue without corrective maintenance until both elements in the circuit have failed. This situation is reflected in the calculations shown in Figures B-4, B-5, B-6 and B-7 where the repair rate, μ , has been assumed to be zero. Table 5-2 summarizes the failure rates of the refrigerator components under these conditions: i.e. - only those items causing a refrigerator failure are repaired. The MTBF's associated with this policy are:

$$MTBF_{1R} = 18,168 \text{ hours (compared with 12,812 hours for the baseline refrigerator)}$$

$$MTBF_{2R} = 1,017 \text{ hours (compared with 804 hours for the baseline refrigerator)}$$

Under this case for a "fix only when failed" policy not much improvement in reliability has been achieved.

Now for the second case, where all failed items are repaired before a refrigerator is returned to service, the mean time that corrective maintenance is deferred on the failed items with stand-by redundancy was assumed to be the mean time between failure of the series items. Taking the reciprocal of

TABLE 5-2
SUMMARY OF FAILURE RATES AND DOWN TIMES, CORRECTIVE MAINTENANCE
DEFERRED UNTIL REFRIGERATOR FAILURE, CASE (1)

Ref. No.	Item Description	Refrigerator Failure Rates and Expected Down Times for Repairs			Refrigerator Failure Rates and Expected Down Times for Repairs Requiring More Than 8 Hours			Source
		1	2	3	4	5	6	
		Estimated Repair Time τ Hours	Failure Rate $\lambda \times 10^6$ Hr	Down Time Per 10^6 Hr $D_T = \tau (\lambda \times 10^6)$	Estimated Repair Time τ Hours	Failure Rate $\lambda \times 10^6$ Hr	Down Time Per 10^6 Hr $D_T = \tau (\lambda \times 10^6)$	
I	Compressor Subsystem	--	145.1	553.45	--	0.8	8.8	Table 5-1
--	Items Unchanged from Baseline	--			--			
I.3*	Helium/Liquid oil separ.	4	50	200	--	--	--	Fig. B-4
I.9*	Filter Adsorber	8	50	400	11	2.5	27.5	Fig. B-5
I.10*	Purity Monitoring	8	200	1600	11	10	110	Fig. B-5
I.13	Secondard Filter Adsorber	40	0.05	2	40	0.05	2	Fig. B-8
			445.15	2755.45		13.35	148.3	
	Subsystem Failure Rate = $\Sigma \lambda$		445.15/10 ⁶	Hour		13.35/10 ⁶	Hour	
	Subsystem MTTR = $\Sigma D_T / \Sigma \lambda$		6.19	Hours		11.11	Hours	
II	Cold Box Subsystem	--			--			
--	Items Unchanged from Baseline	--	90.95	1245.80	--	10.274	853.59	Table 5-1
II.2	Filter Adsorber Vac Module	12	1	12	12	1	12	Fig. B-8
II.6	HX1	40	1.25	50	40	1.25	50	Fig. B-8
II.9*	Filter Adsorber	12	2.5	30	12	2.5	30	Fig. B-6
II.9A	Secondary Filter Adsorber	128	0.05	6.4	128	0.05	6.4	Fig. B-6
II.15	Main Cold Box Sensors	128	0.96	122	128	0.96	122	Fig. B-8
II.16	Turbo Expander Sensors	6	4.8	28.8	9	0.12	1.08	Fig. B-8
II.17	Cold Box/Cable Interface Sensors	8	1.92	15.36	11	0.307	3.379	Fig. B-8
II.17A	HX1 Module Sensors	40	0.96	38.4	40	0.96	38.4	Fig. B-8
II.17B	Filter Adsorber Module Sensors	12	0.96	11.52	12	0.96	11.52	Fig. B-8
II.18*	Vacuum Pump and Assoc Equipments	8	100	800	11	5	55	Fig. B-7
II.20*	Purity Monitoring	8	200	1600	11	10	110	Fig. B-6
			408.017	3982.524		33.381	1233.361	
	Subsystem Failure Rate = $\Sigma \lambda$		405.35/10 ⁶	Hour		33.38/10 ⁶	Hour	
	Subsystem MTTR		9.77	Hours		38.75	Hours	
III	Remote Expander Subsystem	--			--			
--	Items Unchanged from Baseline	--	81.1	568	--	4.85	139.75	Table 5-1
III.6	Vacuum Box Sensors	40	0.96	38.4	40	0.96	38.4	Fig. B-8
III.7*	Vacuum Pump and Assoc Equipments	8	30	400	11	2.5	27.5	Fig. B-7
			132.06	1006.4		8.31	205.65	
	Subsystem Failure Rate = $\Sigma \lambda$		132.06/10 ⁶	Hour		8.31/10 ⁶		
	Subsystem MTTR		7.62	Hours		24.75	Hours	
	Refrigerator Failure Rate		982.56/10 ⁶	Hour		55.04/10 ⁶	Hour	
	Refrigerator MTBF		1,017	Hours		18,168	Hours	

*Stand-by Redundant Equipments

the sum of the failure rates of the series items from column 2 of Table 5-2 (the stand-by redundant items are denoted by an asterisk), the MTBF_{2R} of the series items is:

$$MTBF_{2R} = \frac{1}{\sum \lambda_{series}} = \frac{1}{(330.06/10^6)} = 3,030 \text{ hours}$$

Therefore the mean-time-to-repair (MTTR) a failed element in a primary redundant circuit is 3,030 hours plus τ , the actual time to restore the failed element to service. The repair rate is thus:

$$\mu = \frac{1}{MTTR} = \frac{1}{3,030 + \tau}$$

Table 5-3 presents a summary of the refrigerator item failure rates under this condition. Only the stand-by redundant elements have been specifically identified on this table for clarity. Their failure rates are computed with the equations derived in Figures B-4 through B-7 with μ determined as shown above. The MTBF's associated with this policy are:

$$MTBF_{1R} = 24,155 \text{ hours}$$

$$MTBF_{2R} = 1,380 \text{ hours}$$

Therefore in this case a somewhat higher reliability is achieved than in the first "fix only when failed" situation.

2. Fix-As-Soon-As-Practical Corrective Maintenance Policy

Of the stand-by redundant circuits the largest contributors to system down time were the purity monitoring equipments. The equipment failure rate studies discuss the rationale behind the allocation of high failure rates to these equipments. For this assessment it is assumed that this equipment can be most efficiently be kept in an available state by one or more specially trained instrumentation technicians. Since there are eight sets of purity monitoring equipments in operation at each of the six refrigerator substations, one of these equipments will be needing repair or adjustment every:

$$MTBF = [6 \text{ substations} \times 8 \text{ sets/substation} \times 400/10^6 \text{ failures/hour set}] = 52 \text{ hours}$$

If this technician also repairs and adjusts the vacuum pump and associated equipments (which constitute another set of equipments in the same high failure rate category as the purity monitors), then he will be called upon every:

$$MTBF = [(24 \text{ expander vac equip set} \times 100/10^6) + (30 \text{ cold box vac equip sets} \times 200/10^6) + (48 \text{ purity monitor sets} \times 400/10^6)] = 36 \text{ hours}$$

This technician thus has thirty six hours to restore an equipment before another device will fail somewhere else along the cable system. This appears to be adequate to allow the assumption that the vacuum and purity monitoring equipments can be attended to and fixed within the estimated repair times, τ , listed on Table 5-2.

TABLE 5-3

SUMMARY OF FAILURE RATES AND DOWN TIMES, CORRECTIVE MAINTENANCE DEFERRED
UNTIL REFRIGERATOR FAILURE, CASE (2)

Ref. No.	Item Description	Refrigerator Failure Rates and Expected Down Times for Repairs				Refrigerator Failure Rates and Expected Down Times for Repairs Requiring More Than 8 Hours			Equation for Evaluating Failure Rate; $\lambda = [Equation]^{-1}$	Source
		1	2	3	4	5	6	7		
		Estimated Repair Time τ Hours	Repair Rate $\mu = [3030 + \tau]^{-1}$ Per Hr	Failure Rate $\lambda \times 10^6$ Hr	Down Time Per 10^6 Hr $D_T = \tau (\lambda \times 10^6)$	Estimated Repair Time τ Hr	Failure Rate $\lambda \times 10^6$ Hr	Down Time Per 10^6 Hr $D_T = \tau (\lambda \times 10^6)$		
I	Compressor Subsystem	--	--	145.15	555.45	--	0.85	10.80	--	
--	All items not incorporating stand-by redundancy	--	--	--	--	--	--	--	--	
I.3	Helium/liquid oil separ.	4	.00033	18.87	75.48	--	--	--	$4\lambda_p + \mu_p/4 \lambda_p^2$	Fig. B-4
I.9	Filter adsorber	8	.00033	18.87	150.96	11	0.94	10.34	$2\lambda + \mu/\lambda^2$	Fig. B-5
I.10	Purity monitoring	8	.00033	141.59	1132.72	11	7.08	77.88	$2\lambda + \mu/\lambda^2$	Fig. B-5
				324.48	1914.61		8.87	99.02		
	Subsystem Failure Rate = $\Sigma \lambda$			324.48/10 ⁶ Hour			8.87/10 ⁶ Hour			
	Subsystem MTTR = $\Sigma D_T / \Sigma \lambda$			5.90 Hours			11.16 Hours			
II	Cold Box Subsystem	--	--	102.85	1530.28	--	15.881	1098.369	--	
--	All items not incorporating stand-by redundancy	--	--	--	--	--	--	--	--	
II.9	Filter adsorber	12	.00033	.074	0.88	12	0.074	0.888	$2\lambda + \mu/\lambda^2$	Fig. B-6
II.18	Vacuum pump and assoc. equipments.	8	.00033	54.8	438.4	11	2.74	30.14	$2\lambda + \mu/\lambda^2$	Fig. B-7
II.20	Purity monitoring	8	.00033	141.59	1132.72	11	7.08	77.88	$2\lambda + \mu/\lambda^2$	Fig. B-6
				299.31	3102.29		25.78	1207.28		
	Subsystem Failure Rate = $\Sigma \lambda$			299.31/10 ⁶ Hour			25.78/10 ⁶ Hour			
	Subsystem MTTR = $\Sigma D_T / \Sigma \lambda$			10.37 Hour			46.83 Hour			
III	Remote Expander Subsystem	--	--	82.06	606.4	--	5.81	178.150	--	
--	All items not incorporating stand-by redundancy	--	--	--	--	--	--	--	--	
III.7	Vacuum pump and assoc equipments	8	.00033	18.87	150.96	11	0.94	10.34	$2\lambda + \mu/\lambda^2$	Fig. B-7
				100.93	757.36		6.75	188.49		
	Subsystem Failure Rate = $\Sigma \lambda$			100.93/10 ⁶ Hour			6.75/10 ⁶ Hour			
	Subsystem MTTR = $\Sigma D_T / \Sigma \lambda$			7.50 Hours			27.92 Hours			
	Refrigerator Failure Rate			724.72/10 ⁶ Hour			41.40/10 ⁶ Hour			
	Refrigerator MTBF			1380 Hours			24,155 Hours			

The other stand-by equipments are the liquid oil/helium separation equipments in the compressor subsystem, and the filter adsorbers in the compressor and cold box subsystems. It was assumed that these devices can be repaired and maintained by the same repair crew which handles the series items in the refrigerator system. In the previous discussion it was shown that the MTBF_{2R} of the series items in the refrigerator was 3,030 hours. Therefore for the cable system, the mean time between failure of the series equipments is:

$$\begin{aligned} \text{MTBF}_{2R \text{ series}} &= \frac{1}{24 \text{ refrigerator and expander sets } (\Sigma \lambda) \text{ items}} \\ &= \frac{1}{24(330.06/106)} = 126 \text{ hours} \end{aligned}$$

Therefore this "general" repair crew has an average time of one hundred and twenty six hours to spend correcting a refrigerator before another refrigerator will fail somewhere else along the cable system. If this crew is also responsible for the heat sink and other systems required to support the refrigerators, as well as preventive maintenance then they may well be employed for most of this period of time. It was assumed, therefore, that corrective maintenance on the filter adsorber and liquid oil/helium separation equipments would be deferred for approximately one hundred and twenty six hours. The repair rate is thus:

$$\mu = \frac{1}{126 + \tau}$$

Table 5-4 presents a summary of the refrigerator item failure rates under this condition where corrective action is taken immediately for purity monitoring and vacuum equipments, and within one hundred and twenty six hours for liquid oil/helium separation and filter adsorber equipments. The MTBF's associated with this policy are:

$$\text{MTBF}_1 = 43,937 \text{ hours}$$

$$\text{MTBF}_2 = 2,980 \text{ hours}$$

Thus this policy of restoring standby redundant circuits to full capability as soon as practical results in the greatest improvement in refrigeration MTBF. It is important to note however that this conclusion is derived under the assumption of item independence within the refrigerator. If subsequent system studies (i.e. - FMEA's) and/or operating experience show that the failure and repair process of one of these stand-by items disturbs refrigerator equilibrium to the point where lengthy down times are required for readjustment of the refrigerator before it can be restored to service, then a "fix only when refrigerator failed" corrective maintenance policy may be more appropriate (an illustration of a case where this latter policy is more appropriate is provided by the difficult to replace sensors in the cold box).

TABLE 5-4

SUMMARY OF FAILURE RATES AND DOWN TIMES; CORRECTIVE MAINTENANCE PERFORMED
AS SOON AS PRACTICAL

Ref. No.	Item Description	Refrigerator Failure Rates and Expected Down Times for Repairs				Refrigerator Failure Rates and Expected Down Times for Repairs Requiring More Than 8 Hours			Equation for Evaluat- ing Failure Rate: $\lambda =$ [Equation] ⁻¹	Source
		1	2	3	4	5	6	7		
		Estimated Repair Time τ Hours	Repair Rate μ	Failure Rate $\lambda \times 10^6$ Hr	Down Time Per 10^6 Hr $D_{TWT} (\lambda \times 10^6)$	Estimated Repair Time τ Hours	Failure Rate $\lambda \times 10^6$ Hr	Down Time Per 10^6 Hr $D_{TWT} (\lambda \times 10^6)$		
I	Compressor Subsystem All Items not Incorporat- ing stand by redundancy	--	--	145.15	555.45	--	0.85	10.80	--	
I.3	Helium/liquid oil separ.	4	.00769	1.27	5.08	--	--	--	$4\lambda_p + \mu_p / \lambda_p^2$	Fig. B-4
I.9	Filter Adsorber	8	.00746	1.31	1.05	11	0.066	0.726	$2\lambda + \mu / \lambda^2$	Fig. B-5
I.10	Purity Monitoring	8	.125	1.27	10.16	11	0.064	0.704	$2\lambda + \mu / \lambda^2$	Fig. B-5
				149.00	571.74		0.980	12.230		
	Subsystem Failure Rate = $\Sigma \lambda$			149/10 ⁶ Hour			0.98/10 ⁶ Hours			
	Subsystem MTTR = $\Sigma D_{TWT} / \Sigma \lambda$			3.84 Hours			12.48 Hours			
II	Cold Box Subsystem All Items not incorporat- ing stand-by redundancy	--	--	102.85	1530.28		15.881	1098.369	--	
II.9	Filter Adsorber	12	.00725	0.003	0.036	12	0.003	0.036	$2\lambda + \mu / \lambda^2$	Fig. B-6
II.18	Vacuum Pump and Assoc. Equipments	8	.125	0.319	2.552	11	0.016	0.176	$2\lambda + \mu / \lambda^2$	Fig. B-7
II.20	Purity Monitoring	8	.125	1.27	10.16	11	0.064	0.704	$2\lambda + \mu / \lambda^2$	Fig. B-6
				104.442	1543.028		15.964	1099.285		
	Subsystem Failure Rate = $\Sigma \lambda$			104.442/10 ⁶ Hour			15.964/10 ⁶ Hour			
	Subsystem MTTR = $\Sigma D_{TWT} / \Sigma \lambda$			14.77 Hours			68.85 Hours			
III	Remote Expander Subsystem All items not incorporat- ing stand-by redundancy	--	--	82.06	606.4	--	5.81	178.150	--	
III.7	Vacuum Pump and Assoc. Equipments	8	.125	0.080	0.064	11	0.004	0.044	$2\lambda + \mu / \lambda^2$	Fig. B-7
				82.140	607.04		5.814	178.194		
	Subsystem Failure Rate = $\Sigma \lambda$			82.140/10 ⁶ Hour			5.814/10 ⁶ Hour			
	Subsystem MTTR = $\Sigma D_{TWT} / \Sigma \lambda$			7.39 Hours			30.649 Hours			
	Refrigerator Failure Rate			335.58/10 ⁶			22.76/10 ⁶			
	Refrigerator MTBF			2980 Hours			43,937 Hours			

Optimized Baseline Cable System Refrigeration System

This analysis was performed under the assumption that a "fix as soon as practical" corrective maintenance policy is in effect. Figure B-11 presents the reliability logic diagram and the details of the analysis. The calculated MTBF values are:

$$MTBF_3 = 3,416,341 \text{ hours}$$

$$MTBF_4 = 58,524 \text{ hours}$$

The model used in this analysis is of interest because it points out a limiting factor for achievable reliability which can be obtained from the application of stand-by redundancy techniques at the refrigerator substation level. This model for a refrigerator substation =

MTBF of a refrigerator substation =

$$\left\{ \left[\frac{128 \lambda^3_{\text{COLD BOX}} + 30 \lambda^2_{\text{COLD BOX}} \mu_{\text{COLD BOX}} + 6 \lambda_{\text{COLD BOX}} \mu^2_{\text{COLD BOX}} + \mu^3_{\text{COLD BOX}}}{96 \lambda^4_{\text{COLD BOX}}} \right]^{-1} + \left[\frac{26 \lambda^2_{\text{COMP}} + 2 \lambda_{\text{COMP}} \mu_{\text{COMP}} + \mu^2_{\text{COMP}}}{24 \lambda^3_{\text{COMP}}} \right]^{-1} + \lambda_{\text{SWITCH OVER SUBSYSTEM}} \right\}^{-1}$$

Now as the terms for the cold box and compressor arrangement failure rates decrease through the use of intra-refrigerator reliability techniques we have:

The Limit of the Refrigerator Substation MTBF $\rightarrow$ MTBF of the switchover subsystem

As the Cold Box Arr $\lambda \rightarrow 0$

And the Comp Arr. $\lambda \rightarrow 0$

The MTBF of the Switchover Subsystem must be greater than 525,600 hours or the cable refrigeration system will never meet its ten year (87,600 hours) MTBF requirement. This minimum reliability requirement is determined as follows, assuming that the failure rates of the cold box, compressor, and remote expander arrangements are negligible.

$$87,600 \text{ hours} = [6 \text{ stations} \times \lambda_{\text{switch over subsystem}}]^{-1}$$

Solving for λ_{sw} we find that

$$\lambda_{\text{sw}} = 1.903/10^6,$$

and

$$MTBF = \left(\frac{1}{\lambda_{\text{sw}}} \right) = 525,600 \text{ hours.}$$

The analysis in Figure B-11 allocated a MTBF of 2×10^6 hours to the switch-over subsystem on the assumption that a relatively straight forward and easily repairable system of manual valves and welded piping will suffice to accomplish the switchover function. Subsequent studies may show that a vacuum box and/or an automated system are required or desirable, and that an MTBF substantially greater than 525,600 hours is not realistic. These studies may also show that the time available to repair a leaking switchover subsystem, for example, is considerably less than eight hours if there has been a significant build up of thermal inertia within the cable insulation as a result of prior refrigerator failures. In these events, the concept of using stand-by redundancy and switching at the refrigeration station level would require re-evaluation. The application of reliability improvement techniques to the switchover subsystem, for example, would need to be traded off against the abandonment of this concept in favor of the application of reliability improvement techniques at the cable circuit or cable segment refrigerator level.

DOUBLE-SIZE OPTIMIZED BASELINE REFRIGERATOR

System Description

The cable refrigerator system consists of twelve double-size refrigerators which have been optimized by the application of the reliability improvement techniques discussed in the optimized baseline cable segment refrigerator description in the previous subsection. These twelve refrigerators are grouped into six refrigerator substations, two refrigerators at each substation. Each substation incorporates a stand-by redundant cold box and an intrastation switchover subsystem which are analogous in application to those functions described in the previous subsection. There are twenty-four remote expanders arranged in seven remote expander substations as shown in Figures 5-1 and 5-3.

An illustration of the details of the arrangement of these double-size refrigerators in a refrigerator substation is given in Figure B-12.

Summary of Results and Conclusions

Four parameters were assessed. These parameters and their assessed values are:

- 1a. Mean-time-between-failures resulting in the shutdown of a cable circuit because of the loss of active refrigeration for more than eight hours in any of its segments: assessed $MTBF_1 = 13,192$ hours.
- 1b. Mean-time-between-failures for all failures (including those contributing to MTBF) resulting in loss of active refrigeration in any segment of a cable circuit: assessed $MTBF_2 = 521$ hours.
- 2a. Mean-time-between-failures resulting in the shutdown of both cable circuits because of the simultaneous loss of active refrigeration for more than eight hours in one or more segments in both cable circuits: assessed $MTBF_3 = 3,410,176$ hours.

- 2b. Mean-time-between-failures for all failures (including those contributing to MTBF₃) resulting in the simultaneous loss of active refrigeration in one or more segments in both cable circuits: assessed MTBF₄ = 55,222 hours.

It is of interest to compare the values given above with the corresponding values assessed for the optimized baseline refrigerator system. These values are:

	Optimized Baseline	Double-Sized Optimized Baseline	Requirement
MTBF ₁	12,152 hrs	13,192 hrs	8760 hrs
MTBF ₂	359 hrs	531 hrs	--
MTBF ₃	3,416,341 hrs	3,410,176 hrs	87,600 hrs
MTBF ₄	58,5245 hrs	55,222 hrs	--

A major assumption of this analysis was that the scaling factors for both the failure rate and the repair rate were both equal to one. That is, the same failure and repair rate data was used for the analyses of both the optimized and the double-size optimized refrigerator concepts. Given that this equality is reasonably accurate, the relative equality between the MTBF values compared above is surprising. There are twenty-four refrigerators in the optimized baseline concept compared to only twelve refrigerators in the double-size concept. The reason for this relative equality in cable circuit and cable system MTBF's is shown in Table 5-5, which sets forth the failure rate contributions of the major subsystems to each MTBF assessment.

From Table 5-5 it can be seen that the stand-by redundancy of the cold box and the intrasubstation switching capability reduce the impact of refrigerator subsystem unreliability to almost negligible proportions in comparison to remote expander subsystem unreliability in the assessment of MTBF₁, MTBF₃, and MTBF₄.

It is concluded, that with the concept of substation level stand-by redundancy and switching, that there is no appreciable advantage to be gained in cable circuit or cable system refrigerator reliability with the double-size refrigerator concept. There are, however, other considerations which show definite advantages and disadvantages associated with each concept. These considerations are:

1. Corrective maintenance frequency: there should be fewer refrigerator failures with the double-size refrigerator concept. The series MTBF's for refrigerator failures in a complete cable system refrigeration system are 164 hours for the optimized baseline, and 328 hours for the double-sized refrigerators.

TABLE 5-5

TOTAL EFFECTIVE FAILURE RATE CONTRIBUTIONS (INCLUDING REDUNDANCY)
OF THE SUBSYSTEMS TO CABLE CIRCUIT AND CABLE SYSTEM MTBF

	Refrigerators			Remote Expanders
	Compressors λ_3	Cold Boxes λ_2	Switchovers λ_3	
MTBF ₁				
Optimized Baseline	$1.62/10^6$	$23.52/10^6$	--	$139.4/10^6$
Double Size	$0.4/10^6$	$11.76/10^6$	--	$139.4/10^6$
MTBF ₂				
Optimized Baseline	$15.21/10^6$	$3576/10^6$	$3/10^6$	$1971.36/10^6$
Double Size	$3.83/10^6$	$1788/10^6$	$3/10^6$	$1971.36/10^6$
MTBF ₃				
Optimized Baseline	$10/10^{11}$	$21/10^{15}$	--	$2.9/10^7$
Double Size	$42/10^{11}$	$12/10^{11}$	--	$2.9/10^7$
MTBF ₄				
Optimized Baseline	$2.2/10^{10}$	$7/10^9$	$3/10^6$	$14.08/10^6$
Double Size	$5.9/10^9$	$1/10^6$	$3/10^6$	$14.08/10^6$

2. Operational flexibility: the optimized baseline refrigerator is a more efficient utilizer of the stand-by redundant cold box and intra-substation switching to maintain active refrigeration to one cable circuit during multiple failure conditions. An optimized baseline refrigerator substation can tolerate three cold box and two compressor subsystem failures without impairing its capacity to provide active refrigeration to one cable circuit. A double-size refrigerator substation can only tolerate two cold box and one compressor subsystem failures without impairing this capacity.

3. Reliability improvement techniques at the refrigerator level: subsequent studies may show that it is more cost effective to incorporate additional redundant elements in a double-size refrigerator than it would be with a optimum baseline refrigerator (i.e., to provide a spare compressor for each refrigerator at a station would require four compressors for the optimized baseline concept, and two compressors for the double-size concept.)
4. Equipment cost: preliminary evaluation of the twelve double-refrigerator concept shows a significant initial cost advantage over the 24 refrigerator baseline boxed or Strobridge's method. Consideration of redundancy requirements of the optimized baseline and double-size refrigerator may, however, tend to reduce the initial cost differences somewhat.

Analysis of a Double-Size Refrigerator

A one-to-one repair rate and failure rate scaling factor, and a "fix as soon as practical" corrective maintenance policy were assumed. There is no change to the quantitative assessment of the optimized baseline refrigerator.

1. Double-Size Cable Circuit Refrigeration System

Part I of Figure B-12 presents the details of this analysis.

2. Double-Size Cable System Refrigeration System

Part II of Figure B-12 presents the details of this analysis.

SECTION 6

CONCLUSIONS AND RECOMMENDATIONS

As a result of the investigation and analysis conducted as a part of this study, AiResearch has concluded that the refrigerator systems required for superconducting power transmission cables should not be expected to limit successful development, demonstration and subsequent application in the assumed 1990's timeframe. The technology of helium liquefaction and refrigeration systems is sufficiently advanced that major break-throughs in cycle design are neither expected nor required for this application. What is required is a careful application engineering of the existing system concepts to optimize the overall cost and performance to the specific requirements of a superconducting power transmission cable; refrigerator optimization can be expected to indicate the advantages for advances in the state-of-the-art of several refrigerator subsystems and components.

The refrigerator can not be optimized without consideration of the effects on the remainder of the power transmission cable system. As a result of coordination with the cable system development centers, the interrelationship of the performance of the cable itself with its enclosing cryostat and the refrigerator has indicated the need for coordinated effort in design and development of the various system elements. Since these three principal cable system elements are interdependently linked by performance, the overall cable system optimization must establish and control their interface requirements.

Although detail refrigerator system and component design must be a subtask of overall cable system optimization, certain general requirements, typical of application optimized refrigerators, have been established in Section 3; an assessment of key subsystem and component design approaches is discussed in Section 4 where potential operating efficiency advantages from new designs are noted. Reliability engineering principles have been evaluated to begin the establishment of system refrigerator design compatible with transmission cable applications requirements. This analysis, presented in Section 5, has included an initial assessment of the necessary reliability and maintainability data required for refrigerator design optimization. As a result of this study, certain specific conclusions and recommendations are offered.

CONCLUSIONS

A number of conclusions regarding the system requirements and design approach have been established. Many of these conclusions are based on the early program survey work and field visit to the principal U.S. cable system development centers. To concentrate the study's effort on refrigerator system reliability engineering and subsystem and component design approaches as reported in Sections 4 and 5, many baseline assumptions, such as the refrigerator substation arrangements and the refrigeration cycle to meet the cable interface, were established early in the program and are not the resultant detail tradeoffs reported in this document. In several cases, additional supporting work is felt desirable. This is addressed in the recommendation subsection of this section.

The conclusions reached under the study are categorized into each of the major design areas considered. Initially, the more general system conclusions are presented, followed by more specific subsystem and component hardware conclusions.

Refrigerator System Design

The refrigerator must be designed as a part of the overall cable system. While certain key hardware elements will probably be common to several specific cable applications, each cable system design will require new system engineering considerations. The cable system development currently underway at the three principal U.S. centers requires refrigerators differing in interface requirements. This indicates the probable need for separate refrigerator system optimization to each center's requirements. Moreover, within a development center, the refrigerator interface requirements for various cable systems under consideration may differ; such is the case with BNL's Long Island Lighting Company and Pennsylvania Utilities Company studies. Until such time as a "standard" (or several standard) cable design is adopted, this proliferation of unique cable/refrigerator interface requirements can be expected to continue. Even then, cable system reliability requirements may be expected to require differing redundancy arrangements, depending on total refrigeration system failure rate allocation, the number of refrigerators in parallel on a given circuit, etc.

A multi-expander reversed Brayton refrigeration cycle has been used to establish design baseline subsystem test and component requirements. The flow and pressure loss characteristics of evolving cable and cryostat design permit the use of the refrigerator compressor to circulate the helium coolant through the cryostat, thereby eliminating the need for a secondary cable coolant pump loop. The use of the refrigerator compressor for cable helium circulation is particularly attractive due to the elimination of additional components required by a separate circulation loop, the elimination of the coolant loop pump losses (which are added to the refrigerator's cold end load), and the elimination of the added necessity to produce refrigerator temperatures below cable requirements due to heat transfer from a separate coolant loop.

The BNL and LASL cable system designs are considering the tradeoff associated with temperature shielding within the cryostat. Their latest designs show the overall system tradeoff to favor simpler nonshielded, less costly cryostat designs at the expense of additional cold-end refrigerator load. An example of this evolution can be seen in Brookhaven's December survey reply (refer to Table 3-1) in which the use of the BNL-11 cryostat results in over 2.5 times the per unit length cold end refrigeration load expected from a shielded design. This change, while increasing the capacity of the refrigerator, somewhat simplifies its design by eliminating the requirement for intermediate temperature helium coolant circulation with its associated flow and pressure drop requirements. (The UCC cryostat design uses shields requiring the use of a small fraction of the cold helium from the cable. Its full range of sensible heat capacity to ambient temperature and its full range of pressure loss provided in the refrigerator compressor are used in these shields.) Shielding within the refrigerator cold box may, however, be advantageous. Refrigerator cold box shielding costs are expected to be minimal, and the pressure drop associated with cold box shields is compatible with the available

In-line capacity with only small additional penalties. If cryostat shields are eventually established as a result of cable system optimization, it is concluded that additional non-helium coolants, such as LN₂ should not be considered due to their separate refrigeration system requirements and the increased possibility for helium contamination.

For reasons of maintainability, it is concluded that separate vacuum compartmentalization of the cold box is desirable. As a minimum, separate (adjacent) vacuum spaces should be provided for the heat exchangers, for the expanders, and for low temperature contaminant removal adsorption traps. This arrangement will allow ambient pressure maintenance action to change the adsorption agent or replace a failed expander without the time required to warm and re-chill the entire cold box as noted in the reliability engineering discussion of Section 5. The use of standby redundant low-temperature adsorption traps, each maintained in a separate parallel vacuum enclosure, will permit maintenance of a contaminated adsorber without the need to shutdown the refrigerator.

At a typical refrigerator substation, a number of independent refrigerators may be operating. This number of units can be expected to depend on overall cable system considerations which include consideration of the number of parallel power cable circuits, their degree of independence in support equipment, and whether the refrigerator cools a section of cable circuit in a single or in both directions. Given the need to retain independent refrigeration in the two circuits of the BNL-LILCO cable system used as a baseline, the interface requirements show the need for six refrigerator substations. Initial refrigerator cost estimates, using T. R. Strobridge's data,* show the cost for a 24-refrigerator network to be about \$16 million while the initial cost for a double-size 12-refrigerator network is about \$12 million (based on loads at 7 K). The final selection of the optimum number of refrigerators for a cable system is the result of tradeoffs involving initial cost, operating costs, maintainability, and reliability.

Although detail substation arrangement is beyond the scope of this study, certain conclusions regarding refrigerator substation design were required to establish the baseline system presented in Section 3. Pressurized helium gas storage tanks for the supply of make-up gas, and storage of excess helium in the event of refrigerator shut-down, will probably be a substation facility; separate storage and supply units for each refrigerator at a substation is expected to be unnecessarily costly. Each substation's storage and supply system will require a separate helium compressor and supply purification arrangement to handle supply to, and return from, the refrigerators. The storage capacity of this peripheral equipment will be very large if storage of one or several circuit segment's helium inventory is required. Similarly, each substation can be expected to supply a single heat sink system for compressor heat rejection, and a single refrigerator power supply network. These facilities are not a part of the refrigerator, but effect its design at their interface. Special superconducting cable terminations will require carefully controlled temperature transition which can best be handled as a separate refrigeration problem. The special requirements of cable termination refrigeration will require additional study; the use of the substation refrigerator systems will probably not provide the best solution to this application.

*Strobridge, T.R., Cryogenic Refrigerators--An Updated Survey, NBS Technical Note 655.

Subsystem and Component Design

The conclusion that refrigeration system design must address reliability requirements commensurate with the cable system application is paramount to the approach taken by AiResearch in performance of the study. This has led to the design studies of Section 4 which emphasize the need for reliability considerations in the selection of component design approaches. Among the significant influences of reliability engineering on refrigerator system design are the achievement of high component reliability supplemented by redundancy where required, the maximization of maintenance intervals and reduction of time required for maintenance and the selection of components types which do not represent additional sources for contamination.

Subsystem and component performance impact the refrigerator system efficiency and cost. These in turn influence the design of the entire cable system. This pyramiding effect of efficiency has been considered in the establishment of conclusions which attempt to highlight significant cost and performance impact.

1. Helium Compressor Subsystem

The study showed that the most widely used type of helium compressor for refrigerators is the oil lubricated reciprocating positive displacement compressor. These units are relatively simple, rely on adapted technology developed for other applications (to the benefit of reduced development costs), and are generally suitable for low temperature experiment support in the laboratory; their primary application. Large liquefiers for the commercial production and sale of liquid helium also tend to use piston positive displacement compressors. However, some experience has been gained using turbocompressors to reduce maintenance and downtime.

The use of the Lysholm screw compressor for helium refrigeration is an ongoing experiment at Brookhaven National Laboratory. The compressors at BNL are basically freon vapor cycle compressors, adapted to helium gas from commercial refrigeration experience. AiResearch is impressed with the screw compressor operating experience at BNL; it must be concluded that the screw compressor is an attractive alternative to piston compressors. Operating efficiencies approaching 42 percent (isothermal) have been demonstrated, 2000 hours of operation has not resulted in a failure, and maintenance has not been a problem. Most significant is its availability today. Based on near term requirements, the use of oil flooded screw compressors appear to be the best helium compression technique available without major development.

Overall isothermal efficiencies of about 38 to 42 percent (Table 4-4) have been obtained at Brookhaven. These efficiencies include losses associated with the hermetically sealed 60 Hz motor. Since the oil/helium mixture must pass through the motor, windage losses are appreciable; Dunham-Bush has estimated net motor efficiencies to be about 85 percent. Decoupling of the motor from the compressor would be expected to increase screw compressor overall isothermal efficiency by perhaps 10 percent (to about 50 percent). Unfortunately, this would create a shaft sealing problem. An analysis of leakage makes a labyrinth seal look unattractive due to helium loss; the use of magnetic drive requires an

improvement in the state-of-the-art of magnet coupling capacity by approximately an order of magnitude.

The major operational drawback to screw compressors is the need to remove oil before the helium enters the cold box. Available techniques using molecular sieves or charcoal reduce the oil output to less than 0.1 PPM. However, as shown in the example of Section 4, even the accumulation of oil in parts-per-billion concentration levels can be expected to accumulate significant mass given sufficient time. The problem of their oil then becomes twofold: (1) where does it accumulate; and (2) what are the consequences of an oil removal system failure? Since all liquid and gaseous contaminants can be expected to solidify before the cable operating temperature is achieved, it is reasonable to expect accumulation in the cold box.

In addition, the failure of the oil separation subsystem, or oil migration through the cold box as a result of warm-up for maintenance or failure repair, may result in oil reaching the cable. The results of oil contamination in the dielectric may result in sufficient deterioration of its electrical insulation to allow local cable insulation failures which would require cable decontamination.

Oil-free screw compressors are currently under development in Japan and the U.S. From the data obtained during the study, it has become apparent that the oil performs three functions: (1) lubrication of contacting surfaces; (2) sealing of the screws to prevent excessive helium leakage, and (3) adsorption of the helium's heat of compression resulting in helium temperature profiles approaching isothermal conditions. While oil-free screw compressor development may result in operational machinery, it must be concluded that some of the advantages inherent in the oil-flooded screw compressor will be lost.

The use of process gas bearing supported centrifugal compressors can be expected to result in helium compression without the need for oil addition to the helium stream. The elements of a centrifugal helium compressor subsystem are analyzed in Section 4. Comparison of the expected operating efficiencies of three alternate oil-free centrifugal compressor subsystems are compared to demonstrated screw compressor efficiency in Table 4-17. Although centrifugal compressor subsystems are not currently available, the potential for reduced energy requirements provides a cost incentive for their development, particularly as the number of units in the field begins to increase. Based on the BNL-LILCO study used to establish the baseline design discussed in Section 3, 24 compressors will be required for this one short cable system.

From the input power requirements in Table 4-17, the energy cost savings associated with various compressor subsystem options are compared in Table 6-1.

The annual system wide savings in energy costs may exceed \$2.5 million; capitalized at 10 percent over a 25 year period, these costs savings are seen to range between \$20.6 and \$22.9 million. These savings are expected to be over 150 percent of the initial system cost estimates discussed in the system conclusions section based on \$.03/kwhr. Future energy costs may increase the potential savings.

TABLE 6-1

COMPRESSOR SUBSYSTEM ENERGY SAVINGS

Subsystem	Baseline Subsystem Input Power (kw)	Cable System Annual Savings (million kwhr) ⁽¹⁾	Cable System Annual Savings (\$ millions) ⁽²⁾
Screw - 60 Hz	1015	BASE	BASE
Centrifugal - Induction (800 Hz)	655	75.7	2.27
Centrifugal - Permanent Magnet (3067 Hz)	635	79.9	2.40
Centrifugal - 60 Hz	616	83.9	2.52
⁽¹⁾ Based on 24 compressor subsystems per BNL-LILCO baseline. ⁽²⁾ Energy costs at \$.03/kwhr.			

It is interesting to note that the use of multistage centrifugal compressors results in adiabatic efficiencies of well over 100 percent when their collective input power requirement is compared with that for a single stage isentropic compression. Even when all of the various electrical and mechanical efficiencies are considered, as in equation (4-4), the overall adiabatic compression efficiency remains high. Applying the mechanical efficiencies listed in Table 4-17 to 14 wheel centrifugal design of Table 4-10 results in an overall adiabatic efficiency of over 100 percent. This is, of course, a benefit of interstage cooling and implies an initial versus operating cost economic tradeoff in final determination of the optimum number of centrifugal stages. In addition, it serves as an ultimate comparison for single-stage positive displacement compressors. Without cooling, a single-stage positive displacement compressor would require greater energy (i.e. higher operating energy costs) even if it were 100 percent efficient (adiabatic) and had no mechanical losses. Since significant gas cooling in a single-stage compression process is difficult to achieve without the addition of a coolant in direct contact with the gas (e.g. the oil in the screw compressor), it is concluded that multistage centrifugal compression will provide a lower operating cost than is practical for single-stage positive displacement compressors.

It is concluded that the potential for energy cost savings justifies the costs of centrifugal compressor study and subsequent development.

2. Heat Exchangers

The high effectiveness heat exchangers required for cold box application can best be met using plate fin designs. Study details show that heat

exchangers made from stainless steel alloy will be approximately the same size as those fabricated from aluminum; the core weights are in almost the same proportions as material densities. This suggests that initial cost tradeoffs will favor aluminum. However, it is concluded that corrosion in dip-brazed aluminum plate-fin heat exchangers will preclude their use from reliability considerations. The development of furnace-brazed stainless steel or fluxless-brazed aluminum units in the required size range is expected to progress at a pace commensurate with the needs of the cable system application.

Where production rates are expected to be sufficiently high to cover the costs of added tooling, the consideration of formed tube plate units is suggested. This technique may result in reduced production costs and increased reliability.

3. Expanders

Reliability considerations show the desirability of using turboexpanders rather than positive displacement devices for the removal of work from the cold box. Since these engines must operate on a cold helium gas stream (down to 5-6 K), lubrication is not permitted on cold surfaces. The use of process gas bearing turboexpanders is concluded to be the most favorable approach.

In order to achieve high overall adiabatic efficiency, the design of a turboexpander directly coupled to a permanent magnet alternator was undertaken. The entire rotating assembly would operate at turbine temperatures; compliant foil process gas bearings would be used for support. The design of this component for the most critical expander of the baseline system design resulted in an estimated overall adiabatic efficiency of about 65 percent. (Using the turboalternator concept, work is removed from the cold box in the form of high frequency alternating current.) It is concluded that turboalternators are an attractive alternate to warm-end-loaded turboexpanders.

RECOMMENDATIONS

To begin commercial service in the 1990's it will be necessary to successfully demonstrate a segment of superconducting cable in an operational power transmission system early in the second half of the preceding decade. The demonstration, to be sufficiently meaningful to attract utility company investment, must be composed of at least prototype system and component hardware and must show acceptable performance, reliability and overall cost of ownership. This requirement will hold for virtually all system elements including the refrigerator system. This leaves less than 10 years for system study, hardware development, and demonstration system design and fabrication. Clearly, to meet this timetable hardware development to achieve the goal must be conducted in the remaining years of the 1970's.

The remainder of this section is devoted to a brief outline of the various study and hardware development efforts AiResearch recommends for USERDA consideration in support of this goal. The numerous recommendations are established at several levels of the overall cable system; they are summarized in Table 6-2, at the end of the section, where a recommended priority coding has been noted.

Cable System

1. System Optimization

The optimized cable system must include the interrelated effects of the various system elements on one-another. Changes to the cryostat design, for instance, may affect refrigerator requirements in terms of cold-end load, pressure loss requirements, segment cooling length, etc; uncoordinated refrigerator cost optimization would lead to extremely low cryostat ambient heat transfer rates and low-pressure loss flow passages at the expense of additional cryostat cost.

Since it is overall cable system life cycle costs which must be optimized, AiResearch recommends that full cable system level optimization be conducted which takes into account this tradeoff. This function can best be accomplished by the establishment of a system integrator who's function would be to iterate the various cable system designs to consider these major tradeoffs. From this function will come the final cable system element interface requirements to which selected contractors must design.

2. Reliability Engineering

AiResearch believes that there is a role for cable system reliability engineering which has yet to be fully established. It is recommended that system wide requirements be established and properly apportioned among the cable system elements. These apportioned requirements will then become a part of the design interface requirements to which hardware must be constructed.

To monitor performance, a cable system reliability/maintainability data bank should be initiated, maintained and utilized. The preliminary format presented in Appendix A of this report is recommended as a guideline.

Refrigerator Substation

1. Equipment Arrangement

Based on overall cable system reliability requirements, the arrangement of equipment at the substation level may be determined. It is recommended that overall cost/reliability tradeoffs be conducted to determine the desired degree of inter-refrigerator system dependency. Among the potential "common" interfacing equipment are power supplies, compressor cooling systems, gas supply, makeup and storage systems, switchover network, and monitor and control of critical functions.

2. Refrigerator System Capacity

Reliability and cost consideration will also dictate the selection of the number and capacity of refrigerators at a given substation. It is recommended that tradeoffs be conducted for single- and two-direction coolant loops considering the impact on system interdependence to allow the selection of the refrigerator capacity.

As a part of this task, the possibility of refrigerator system component or subsystem sharing should be considered. For example, it may be optimum to supply more than one substation cold box from a single compressor subsystem.

Refrigerator System

These recommendations are addressed to refrigeration system design; subsystem and component recommendations are separately discussed.

1. Cycle Optimization

As additional subsystem and component performance data becomes available, it is necessary to continually update cycle analyses. The iteration of system and subsystem/component design will tend to produce optimum refrigerator systems. Additionally, these results should be provided to the cable system integrator for consideration of overall cable system optimization.

AiResearch recommends a continuing effort in this regard with regular coordination at the cable system level.

2. Reliability/Maintainability Data Bank

Since refrigerator system design is a cost/reliability tradeoff where reliability is accomplished by subsystem and component selection, AiResearch recommends a continued and expanded effort be undertaken to build upon the data bank suggested in Appendix A of this report. Much of the helium level refrigerator/liquefaction system operation in the United States are at USERDA funded facilities. Under the direction of a reliability engineering manager, data storage/retrieval banks can be established. The program can then be expanded to other U.S. government controlled facilities, and eventually to pertinent domestic and foreign commercial sources.

This data will provide an invaluable tool in system design.

3. Cost Tradeoffs

In support of continuing cable system optimization, detail cost estimates are required for the refrigerator system. A well structured study must consider the capitalized life cycle costs. This study is recommended as a continuing effort. Included should be projections of development costs, initial hardware costs, installation costs, and operating costs (including maintenance and energy). This effort should also include assessments of required maintainability intervals and MTBF's.

4. Contamination Sensors

Since none of the instruments included in the brief survey is acceptable for incorporation in the helium refrigerator system in its present form, it is recommended that development of suitable instrumentation be undertaken. Such instrumentation must satisfy three requirements. First, it must exhibit the sensitivity needed to monitor contaminant concentrations in the range of

of parts per billion. Second, it must have the reliability and stability to operate unattended for a long time without failure and should be designed for routine servicing by electric utility technicians. Third, it must be inherently fail-safe to provide a certain and unambiguous indication of malfunction.

5. Establishment of System Coordinator

To coordinate development of the various subsystems and components which compose the refrigerator system, AiResearch recommends that a refrigerator system development coordination be established. The primary function of this contractor would be to coordinate subsystem and component optimization and development in much the same way that the elements of the overall cable system are coordinated by the cable system coordinator. Establishment of this refrigerator system coordinator will enable the centralization of refrigerator system development responsibility.

Compressor Subsystem

1. Screw Compressor

AiResearch believes that the screw compressor is the most practical of the positive displacement compressors currently available, although its use of oil is considered a definite drawback. It is recommended that experiments at Brookhaven be continued to acquire operational data. Since the optimum oil/helium mixture ratio has not been established, AiResearch recommends that these experiments be conducted to determine obtainable efficiencies.

Brookhaven's Fifth Avenue test facility is expected to generate additional valuable data with a larger screw machine than the one used to generate the data of Table 4-4. It is recommended that the refrigerator system coordinator continue to analyze this data and establish probable limits of oil-flooded screw compressor design with equipment manufacturers.

2. Centrifugal Compressor System

Calculations shown in Table 6-1 show significant operating cost advantages projected for centrifugal compressor subsystems amounting to millions of dollars, even for the short BNL-LILCO cable system used as a baseline for this study. The absence of oil offers an additional significant advantage. AiResearch believes that major improvements to overall cable system attractiveness can be accomplished by the successful development of centrifugal compressor subsystems. The following specific areas of hardware study and development are recommended for program consideration.

a. Compressor

An optimization study to select the recommended number of stages, wheel speed and geometry followed by a hardware development program is recommended. As a first step, critical stages should be identified and development hardware fabricated to test impeller adiabatic efficiency and compliant foil process gas bearing losses. The operation of selected stage(s) can be expected to generate sufficient data to allow refinements of the potential energy cost projections. In parallel with this effort, one or several drive mechanisms may be pursued.

Following the generation of data from single or multiple compressor stages, an analysis updating this study should be conducted; a compliant foil process gas bearing centrifugal compressor subsystem consisting of all stages with the appropriate drive should then be designed and tested.

b. Gearbox Drive

Gearbox technology is not considered to be a constraining feature, although development of gearbox driven compressors will require parallel development of magnetic couplings with sufficient capacity to transmit the required horsepower to the impellers. It is recommended that magnetic couplings of the required capacity be fabricated and tested.

c. Modular Motor

A series of individual motor driven modules, each with two centrifugal compressor stages, offers an alternate to the gearbox driven approach. This concept eliminates the need for mechanical drives, eliminates the need for development of magnetic couplings, and provides a completely hermetically sealed module. This may be expected to result in advantages for compressor failure repair; the time to replace a failed module would allow repair to be completed with less risk of cable circuit shutdown, and may eliminate the need for complete compressor subsystem redundancy.

AIRResearch recommends a detail study of both induction motor and permanent magnet motor design resulting in construction and test of a single module to be followed by a full subsystem design and operation.

d. Frequency Converter

The modular induction and permanent motor designs discussed in Section 4 will require input power frequency conversion to achieve the 46,600 rpm operating speeds. Selection of motor input frequency depends on the number of poles in the motor; as shown in Table 4-15, a motor input frequency of 800 Hz has been considered for the induction motor and 3067 Hz for the permanent magnet motor. In the study, two means of frequency conversion were discussed: use of a motor-generator set and use of a solid state frequency converter.

Each has inherent advantages and disadvantages. It is recommended that, if a modular approach is pursued, this power conditioning equipment be further studied from the reliability, efficiency and motor start-up considerations. Frequency conversion equipment should then be built and tested.

Heat Exchanger

Improvement in heat exchangers are expected to be largely in the fabrication techniques. While aluminum heat exchangers are expected to continue to be less costly at the time of initial purchase, their susceptibility to corrosion leakage is greater than for stainless steel units.

Newer assembly techniques, such as formed-tube-plate design (discussed in Section 4), are expected to be developed in a time frame commensurate with cable system development needs. At this time no sponsored development in this area is recommended.

Expander

Turboalternators operating on compliant foil process gas bearings are expected to show improved overall adiabatic efficiency over warm-end loaded units; this will result in improved cycle efficiency. The application is considered ideal for the use of hydrodynamic compliant foil process gas bearings.

Detail design and fabrication of the cold in-line expander, of the turboalternator type considered in this study, is recommended to evaluate performance. Comparison of the demonstrated efficiency with that achieved by existing designs will permit an assessment of potential cycle efficiency improvement.

TABLE 6-2

RECOMMENDATION SUMMARY AND PRIORITY

Cable System Level	Recommendation	Recommended Priority
Cable System	1. System optimization	S-2
	2. Reliability engineering	S-1
Refrigerator Substation	1. Equipment arrangement	S-3
	2. Refrigerator system capacity	S-2
Refrigerator System	1. Cycle optimization	S-2
	2. Reliability/maintainability data bank	S-1
	3. Cost tradeoffs	S-2
	4. Contamination sensors	S-1
	5. Establishment of system coordinator	S-2
Compressor Subsystem	1. Screw compressor	D-2
	2. Centrifugal compressor system	
	a. Compressor	D-1
	b. Gearbox drive	D-3
	c. Modular motor	D-3
Heat Exchangers	1. Formed tube-plate	D-4
	1. Cold/cold turboalternator	D-2
<p><u>Priority System</u></p> <ul style="list-style-type: none"> Study Recommendations <ul style="list-style-type: none"> S-1 should be established immediately to guide future efforts. S-2 should be established before system configuration for demonstration is finalized. S-3 would benefit program but is not essential at this time. Hardware Development Recommendations <ul style="list-style-type: none"> D-1 should begin as soon as possible to evaluate concept or provide needed data for early hardware selection where extensive development may be required. D-2 should be accomplished before selection of demonstration hardware types. D-3 supports hardware decision and can await results of other development programs D-4 no unique development required for the required timeframe.

APPENDIX A

COMPONENT RELIABILITY AND MAINTAINABILITY DATA

The purpose of this appendix is to begin the establishment of "best estimate" failure rates for each type of major component probable for use in helium refrigeration systems for superconducting power transmission cables. This task was extended to include two additional efforts:

1. The inclusion of "best estimate" maintenance data to support the system reliability engineering studies.
2. Document the findings revealed by the process of gathering and reducing the data.

Data from existing Government data banks and data compiled during field surveys of equipments similar to those envisioned for the helium refrigerator were gathered and analyzed statistically prior to inclusion in this study. Where data were unavailable, allocations were made if the parameter was required to allow the performance of the system reliability studies. In all cases the source of the data and its limitations are noted.

PROBLEM AREAS AND EQUIPMENT DESIGN RECOMMENDATIONS DISCLOSED BY FIELD SURVEY DATA

The review and evaluation of the field survey data which was accomplished for computational purposes revealed several areas which were notable by the consistency with which they appeared and/or by their suitability for incorporation as equipment design criteria in future hardware procurement. These areas are discussed in the following paragraphs.

Use of Water as a Process Stream Heat Sink

Water leakage and fouling and/or corrosion was a major contributor to the causes of failure noted in the surveys. This problem is considered to be an unwarranted contributor to refrigerator unreliability. Several alternate approaches are suggested for reliability improvement.

- Use of air as a heat sink
- Use of helium as an intermediate heat transfer agent between the process stream and the heat sink

Additional considerations for hardware design criteria are:

- Material selection should always consider the compatibility of the material with the heat sink.
- Process stream should be at a higher pressure than the heat sink so that leakage would not result in process stream contamination.

Turbine Expander Repair Times

Experienced laboratory personnel can apparently replace or repair a state-of-the-art turbine expander in a few hours. Future turbine expanders should also have this capability for quick repair or replacement imposed as a maintainability requirement.

Gas Bearing Turbine Expander Reliability

A major cause of failure of gas bearing turbine expanders was contamination resulting in wheel erosion, and bearing and seal failure. Another cause of failure was improper bearing gas pressure. Technically these would be classified as secondary failures induced by a primary failure in the gas stream contaminant removal or pressure regulating equipments. This suggests that the inherent reliability of gas bearing turbine expanders ought to be higher than the data indicates. The reliability of the bearing gas equipments and process stream contaminant removal equipments must be commensurate with the reliability of the expander.

Compressor Maintenance Turnaround Times

The maintenance turnaround times listed in the field surveys were usually several days to a week. A cable refrigeration system will require additional compressors (either mobile or incorporated at the substation) to maintain active refrigeration if these turnaround times cannot be reduced to within the eight hour time frame allowed for the interruption of active refrigeration. An eight hour time frame for major compressor overhaul should not be unrealistic if thought is given to it in the design of a refrigerator compressor. This is recommended as a maintainability requirement for future procurement.

Piston Ring Life

The piston ring life of reciprocating expanders and compressors represented in the surveys appear to be several thousand hours. This is not compatible with a refrigerator which must be operated on a twenty four hour a day basis unless (1) these rings can be readily changed or adjusted, or (2) one or more standby compressors are required for purposes other than piston ring maintenance.

Process Stream Purity

Process stream purity is emphasized both in the field surveys and in other reports relative to helium refrigeration as being a major contributor to refrigerator unreliability. The areas in the refrigerator affected by contaminants in the process stream are the cold box (freeze plugging) and the turbine expanders (wheel erosion). Another area which may be affected by the contamination problem is the cable itself. Pressure and temperature fluctuations in the refrigerator brought about by various failures or shut-downs for maintenance will induce contaminant migration through filters or screens. Given enough time these contaminants will eventually find their way into the cable. The criticality of such contamination is considered to depend upon the type of contaminant and the level of its accumulation which can be tolerated by the cable.

The following recommendations for refrigerator design criteria are made to minimize the amount of contaminants generated:

- Refrigerator should be designed to minimize the need for shutdowns for preventive or corrective maintenance.
- Use of oil, wear particle generating components (i.e., piston rings, nonmetallic valve seats, etc.) or any gas other than helium in equipments interfacing with the cold box process stream should be prohibited.
- Use of oil, water, or any gas other than helium in compressor sub-system components interfacing with the process stream should be accepted only if (1) contaminant removal equipments can provide a purity level allowing long term operation without shutdown for cold box clean-up operations, and (2) the down times and costs associated with contaminant cleanup are less than the costs associated with alternate means of performing the component function.
- Design of equipment and procedures for use in repair and maintenance actions should reflect the need to prevent the action from becoming a source of process stream contamination.

The following recommendations for tests and measurements to be included as part of the cable development test program are made to facilitate the assessment of the process stream purity problem.

- Measurement of contaminant carryover from the compressor to the cold box and from the cold box to the cable (or a simulated cable interface if a separate helium system is used to cool the cable) is recommended.
- Maintenance study is recommended to determine the magnitude of cold box cleanup (and other maintenance action) operations. This should be hardware oriented and involve the deliberate contamination of a cold box by a simulated failure of oil and other contaminant removal equipment. The length of time and procedures used involved to restore the cold box should be noted. The personnel used to perform the clean-up should be representative of the caliber likely to be employed by a utility company. National Laboratory personnel participation in the cleanup should be limited to an instructional capacity since their familiarity and expertise with these equipments might produce results more optimistic than utility experience will justify.
- Tolerance levels of the candidate cables to the type of contaminants to which they might be exposed should be assessed either analytically or by a test simulation using the contaminant carry-over measurement data obtained from the first recommendation.

DATA LIMITATIONS AND RECOMMENDATIONS FOR IMPROVEMENT

Analytical studies to suggest ways and means of designing a reliable and maintainable refrigeration system, and to predict what the reliability and maintainability of a refrigerator will be enhanced immensely by the employment of a standardized system for retaining and assessing the qualitative and quantitative failure and maintenance experience being accumulated by users of applicable refrigeration equipments.

DATA SOURCES

The following studies and data banks were surveyed for data which might be applicable to this task.

1. APCI field survey of laboratory helium and hydrogen refrigerator/liquefier operating experience.
2. Operating Experience-Helium Extraction and Liquefaction Expansion Devices, prepared by T.R. Strobridge and C.F. Sindt for the National Accelerator Laboratory, 1974.
3. Government-Industry-Data-Exchange Program (GIDEP): Summaries of Failure Rate Data, Volume 1, 1975; and Summaries of Replacement Rate Data, Volume 1, 1975.
4. Rome Air Development Center (RADC): Revision of Nonelectronic Reliability Notebook, RADC TR-74-268.

COMPONENT RELIABILITY AND MAINTENANCE DATA

The baseline refrigerator was defined in terms of its three major subsystems: the compressor subsystem, the cold box subsystem, and the remote expander subsystem. A data package for each subsystem was prepared which consists of:

1. Simplified subsystem schematic
2. Tabulation of failure rate and maintenance data
3. Tabulation of the data sources and source related notes

Reliability and Maintenance Data Packages

The reliability and maintenance data packages which were prepared are listed below along with the reference numbers of the figures and tables comprising each package.

1. Screw Compressor Subsystem: Figure A-1 and Tables A-1 and A-2
2. Centrifugal Gas Bearing Compressor Subsystem: Figure A-2 and Tables A-3 and A-4
3. Cold Box Subsystem Utilizing a Gas Bearing Turbine Expander: Figure A-3 and Tables A-5 and A-6
4. Gas Bearing Turbine Remote Expander Subsystem: Figure A-4 and Tables A-7 and A-8
5. Miscellaneous Alternate Refrigerator Equipments: Figures A-3 and A-5, and Tables A-9 and A-10

In addition to these packages, reliability estimates were prepared for the controls, instruments, and alarms for a basic compressor subsystem. These estimates are presented in Table A-11.

Definitions Relative to Data Presentation

1. Failure Rate per 10^6 Hours

These columns show the chi square 90 percent confidence interval within which the true failure rate of the equipment is expected to lie. The best-estimate is the quotient of the total number of recorded equipment failures divided by the total recorded equipment operating hours. Where variations from these definitions were utilized they are noted in the data source notes.

2. Data Source Notes

These notes, which are referenced in Tables A-1 and A-3, describe the source, the assumptions, and any inherent limitations to the tabulated data.

3. Time to Repair, τ

In most cases an estimated time to repair an item has been provided. This time represents the total elapsed time that it would take an optimized crew (e.g., the optimum number of repairmen for a specific job) to diagnose, correct, verify the correction, and restore a malfunctioning refrigerator to operation from a malfunction caused by the item under consideration. These estimates include the following assumptions:

- Station instrumentation can diagnose a problem and identify the abnormal item in less than one-half hour.

A-6

SKETCH A. A COMPRESSOR SUBSYSTEM UTILIZING A SINGLE SCREW COMPRESSOR

SKETCH B. A COMPRESSOR SUBSYSTEM AS ANALYZED IN THE BASELINE SYSTEM STUDY

FIGURE A-1. SIMPLIFIED SCHEMATICS OF SCREW COMPRESSOR SUBSYSTEMS.

TABLE A-1

RELIABILITY AND MAINTENANCE DATA SUMMARY SHEET

SUBSYSTEM: COMPRESSOR SUBSYSTEM, SCREW, COMPRESSOR
 SUBSYSTEM NO: 1A
 REFERENCE FIGURE: A-1

Equipment Description	Reliability Data				Maintenance Data				General Notes
	Failure Rate per 10 ⁶ Hours: $\lambda \times 10^6$			Data Source Notes*	Estimated Time to Repair r Hours	Time to Service Interruption for Scheduled Maintenance	Length of Time Service Interruption for Scheduled Maintenance	Data Source Notes	
	90% Confid Lower	Best Estimate	90% Confid Upper						
Compressor Unit (Compressor, Load/Unloading Valve, and Integral Drive Motor)	0.63	2.31	5.96	R1	8	50,000 Hrs	TBD	M1	On site repair required
Gas/Oil Separation									
Reservoir Assy	--	2		R4	4	TBD	TBD	M.E.	
Coalescer	--	4	--	R4	4	TBD	< 8	M.E.	
Drier and Demister	--	5	--	R4	4	TBD	< 8	M.E.	
Ancillary Equipment	--	10	--	R4	4	TBD	< 8	M.E.	
Oil Pump and Motor	0.63	2.31	5.96	R1	3	TBD	< 8	M.E.	Exchange Repair/O'Neil
Oil Filter	--	6	--	R4	3	TBD	< 1	M.E.	
Oil Cooler	0.58	1.24	2.33	R2	6	TBD	TBD	M.E.	
Gas Stream Aftercooler	0.58	1.24	2.33	R2	8	TBD	TBD	M.E.	
Compressor Inlet Metallic Particle Removal Device	--	0.05	--	R4	3	TBD	< 1	M.E.	
Pressure Regulator Valves	0.27	1.54	4.84	R1	8	TBD	< 8	M.E.	
Basic Subsystem Controls, Instruments, Interlocks and Alarms	--	30	--	R3	3	TBD	TBD	M.E.	
Primary Filter Adsorber	--	100	--	R5	8	TBD	TBD	M.E.	
Secondary Filter Adsorber	--	0.05	--	R5	128	TBD	TBD	M.E.	
Oil Contamination Monitor	--	150	--	R6	8	TBD	TBD	M2	
H ₂ O Contamination Monitor	--	100	--	R6	8	TBD	TBD	M.E.	
Gas Contamination Monitor	--	150	--	R6	8	TBD	TBD	M.E.	
Plumbing	--	26	--	R7	4	TBD	TBD	M.E.	

*Refers to sources defined in Table A-2

MAINTENANCE SOURCES

M.E. -- Maintenance estimate

M1 -- Repair time and scheduled maintenance parameters based on Dunham-Bush experience and recommendations

M2 -- Maintenance estimate. Repair time reflects time to remove adsorbant contaminants from the cold box

TABLE A-2

RELIABILITY SOURCE NOTES FOR TABLE A-1:
SCREW COMPRESSOR SUBSYSTEM

Symbol	Source
R1	<p>Failure rate data shown are based upon hardware manufacturer's cited field experience. No means of correlating user experience with the manufacturer's claims has been established at this time. Thus, the upper 90 percent confidence limit was chosen as the failure rate contribution of these items to the subsystem, shown in Table A-1. These values are assumed at this time to represent the best reliability level attainable by these types of equipment. The details pertinent to the manufacturer's data are as follows:</p> <p>(a) <u>Equipment Size</u>: helical screw rotary compressors ranging in motor size from 200 to 900 hp.</p> <p>(b) <u>Equipment Application</u>: 1) air conditioning, 2) industrial process, and 3) low temperature applications.</p> <p>(c) <u>Time Base</u>: approximately 1.3 million operating hours recorded on equipment elapsed time indicators in approximately ten years of service.</p> <p>(d) <u>Method of Failure Reporting</u>: Dunham-Bush maintenance personnel; according to Dunham-Bush, these personnel perform almost all customer maintenance.</p> <p>The details of the failure rates determined from this source are:</p> <p>Item 1: Compressor Assy</p> <p style="margin-left: 40px;">Compressor and load/unloading valve 1 failure in 1.3×10^6 hours</p> <p style="margin-left: 40px;">Integral motor (sealed) 2 failures in 1.3×10^6 hours</p> <p style="margin-left: 40px;">Best Estimate $\lambda = 3/1.3 \times 10^6 = 2.31 \times 10^{-6}$ per hour</p> <p>Item 3: Oil Pump and Motor (Gear Pump) 3 failures in 1.3×10^6 hours</p> <p style="margin-left: 40px;">Best Estimate $\lambda = 3/1.3 \times 10^6 = 2.31 \times 10^{-6}$ per hour</p> <p>Item 8: Pressure Regulator Valves 2 failures in 1.3×10^6 hours</p> <p style="margin-left: 40px;">Best Estimate $\lambda = 2/1.3 \times 10^6 = 1.54 \times 10^{-6}$ per hour</p>
R2	<p>Rome Air Development Center (RADC): <u>Revision of RADC Nonelectronic Reliability Notebook</u>, RADC TR-74-268. Statistically analyzed data for heat exchangers in ground facility applications. Although the data sources were not identified by RADC, the failure rate confidence limits shown appear reasonable for this application.</p> <p>A statistical reduction of the APCI field survey data shows an experience range of an MTBF of from 200,000 to 37,000 hours at a 90 percent confidence level (best estimate MTBF of approximately 78,000 hours). Water quality problems were a major contributor to the failures experienced. A carefully developed heat rejection system should be capable of minimizing coolant quality as a cause of heat exchanger failure.</p> <p>As shown in Table 1A, the RADC failure rate data was utilized.</p>
R3	<p>Reliability allocation based on a general list of these functions constructed using EPRI Final Report No. 282, <u>Feasibility Study of a Superconducting Cable Test Facility</u>, Prepared by T. R. Strobbridge, July 1975. Table A-11 presents the list which was constructed. Items 1 through 10 from this table were assumed to be required for this subsystem. No redundancy was assumed.</p>
R4	<p>Preliminary reliability allocation based upon a review of the oil separation equipment described on compressor subsystem schematics supplied by the Brookhaven National Laboratory for their 5th Avenue Test Facility. This installation will use a Dunham-Bush Screw Compressor and equipment similar to that described in data source R1.</p>

TABLE A-2 (Continued)

Symbol	Source
R5	<p>Preliminary reliability allocation. Failure rates were allocated for two cases.</p> <p>Case (i) <u>Primary Filter Adsorber</u>. The allocated failure rate represents the premature removal (or bakeout if applicable) rate of these devices associated with contamination breakthrough before attainment of the rated life. Failure data currently available on Garret adsorber products in airborne LN₂ refrigeration systems indicates the existence of this failure rate. These data show a relationship between the severity of the application environment of the compressor (oil lubricated reciprocating) and the failure rate, which has a range of from $1.4/10^3$ failures per hour to $5/10^3$ failures per hour. The extent that quality variations between adsorbers contribute to the failure rate is not known. The allocation made in this assessment of $\lambda = 1/10^4$ failures per hour reflects estimative judgment. Additional data from the National Laboratories and other users of filter adsorbers of a type and application more closely reflecting the anticipated application is of a high order of importance since the suitability of the screw compressor for use in this application depends upon the efficiency of the oil removal process.</p> <p>No allocation of this failure rate is made for the secondary filter adsorber because this device will only be required to adsorb contaminants during the brief period following detection of contaminant breakthrough by the purity monitors and refrigerator shutdown to regenerate or replace the primary device.</p> <p>Case (ii) <u>Secondary Filter Adsorber</u>. The allocated failure rate represents the rate associated with the loss of adsorbant from the filter adsorber because of a mechanical defect in the device. These failures, although infrequent, do occur. The allocated failure rate of $5/10^8$ failures per hour reflects an estimate of this rate of occurrence. It is included in the tabulation because of the extended period of refrigerator down time (estimated to be 128 hours) that will result from contamination of the cold box.</p>
R6	<p>Preliminary reliability allocation. The purity monitoring equipment which is currently available has been designed primarily for laboratory applications for which extensive failure rate and failure experience data was not available from the sources utilized by Garrett during this study. It is not considered that these devices will have a level of reliability commensurate with other items in the refrigerator. For this reason, relatively high failure rates have been allocated to these devices.</p>
R7	<p>Reliability allocation based on the allocation of 10 percent of the total failure rate of the oil and process stream equipments of the compressor subsystem shown in Figure 5-1 and quantified in Table 5-1 of the system studies.</p>

A-10

FIGURE A-2. CENTRIFUGAL GAS BEARING COMPRESSOR SUBSYSTEM

TABLE A-3

RELIABILITY AND MAINTENANCE DATA SUMMARY SHEET

SUBSYSTEM: COMPRESSOR SUBSYSTEM, GAS BEARING CENTRIFUGAL COMPRESSOR
 SUBSYSTEM NO: 1B
 REFERENCE FIGURE: A-2

Equipment Description	Reliability Data				Maintenance Data				General Notes
	Failure Rate per 10 ⁶ Hours: λ x 10 ⁶			Data Source Notes	Estimated Time to Repair τ Hours	Time to Service Interruption for Scheduled Maintenance	Length of Time Service Inter- rupted for Scheduled Maintenance	Data Source Notes	
	90% Confid Lower	Best Estimate	90% Confid Upper						
Power Conditioner	--	100	--	R1	4	TBD	TBD	M.E.	On site exchange of a failed module may be feasible On site exchange
Compressor Module	10	--	20	R2	8	TBD	TBD	M.E.	
Gas Stream Coolers	0.58	1.24	2.33	R3	8	TBD	TBD	M.E.	
Pressure Regulating Valve	1.072	2.055	3.585	R4	3	TBD	TBD	M.E.	
Primary Filter Adsorber	--	15	--	R5	8	TBD	TBD	M.E.	
Secondary Filter Adsorber	--	.05	--	R5	128	TBD	TBD	M.E.	
Module Inlet Metallic Particle Removal Device	--	.05	--	R5	8	TBD	TBD	M.E.	
Basic Subsystem Controls, Instruments, Interlocks and Alarms	--	162	--	R6	3	TBD	TBD	M.E.	
H ₂ O Contamination Monitor	--	100	--	R7	8	TBD	TBD	M.E.	
Gas Contamination Monitor	--	150	--	R7	8	TBD	TBD	M.E.	
Bearing Gas Support System	--	100	--	R8	4	TBD	TBD	M.E.	
Plumbing	--	22	--	R9	4	TBD	TBD	M.E.	

* Refers to sources defined in Table A-4.
 **M.E. = Maintenance Estimate

TABLE A-4

RELIABILITY SOURCE NOTES FOR TABLE A-3:
CENTRIFUGAL GAS BEARING COMPRESSOR SUBSYSTEM

Symbol	Source
R1	The failure rate assigned to the power conditioner in Table A-3 is a reliability estimate considered to be an upper limit for power supplies at the current state of the art. This failure rate represents an estimated-mean-time between failure (MTBF) of 10,000 operating hours for the power conditioner.
R2	<p>The failure rate for the compressor module given in Table A-3 is a reliability estimate, the range of failure rate values achievable for a fully developed device in production facility service is estimated to be between 50,000 and 100,000 operating hours. Since this is an estimate no statistical confidence can be assigned.</p> <p>The number of modules required has not been firmly established, for this analysis, eight modules has been assumed. The optimum estimated failure rate has been selected to allow a more favorable comparison with the helical screw compressor shown in Table A-1.</p>
R3	<p>Rome Air Development Center (RADC) Revision of RADC Nonelectronic Reliability Notebook, RADC TR-74-268. Statistically analyzed data for heat exchangers in ground facility applications. Although the data sources were not identified, the failure rate confidence limits shown appear reasonable for this application.</p> <p>A statistical reduction of the APCI field survey data shows an experience range of an MTBF of from 200,000 to 37,000 hours at a 90 percent confidence level (best estimate MTBF of approximately 78,000 hours). Water quality problems were a major contributor to the failures experienced. Carefully developed heat rejection subsystem should be capable of minimizing coolant quality as a cause of heat exchanger failure.</p> <p>As shown in Table A-3, the RADC failure rate data were utilized.</p>
R4	Rome Air Development Center (RADC) Revision of RADC Nonelectronic Reliability Notebook, RADC TR-74-268. Statistically analyzed data for pneumatic pressure regulating valves in ground facility applications. Although data sources and more definite hardware identification were not available from this source, the upper 90 percent confidence level failure rate bound (e.g., MTBF of 280,000 hours) appears to be a realizable goal for a simple pressure balance type of device operating in a clean, inert gas stream.
R5	<p>Preliminary reliability allocation. Failure rates were allocated for two cases.</p> <p>Case (i) <u>Primary Filter Adsorber</u>. The allocated failure rate represents the premature removal (or bakeout if applicable) rate of these devices associated with contamination breakthrough before attainment of the rated life. Failure data currently available on Garrett adsorber products in airborne LN₂ refrigeration systems indicates the existence of this failure rate. These data show a relationship between the severity of the application environment of the compressor (oil lubricated reciprocating) and the failure rate, which has a range of from $1.4/10^5$ failures per hour. The extent that quality variations between adsorbers contribute to the failure rate is not known. The allocation made in this assessment of $\lambda = 15/10^5$ failures per hour reflects estimative judgement of what this rate should be in comparison to a filter adsorber in a screw compressor subsystem.</p> <p>No allocation of this failure rate is made for a secondary filter adsorber because this device will only be required to adsorb contaminants during the brief period following detection of contaminant breakthrough by the purity monitors and refrigerator shutdown to regenerate or replace the primary device.</p> <p>Case (ii) <u>Secondary Filter Adsorber and Compressor Module Inlet Filter</u>. The allocated failure rate represents the rate associated with the loss of filtering or adsorbing material from the device because of a mechanical defect. These failures, although infrequent do occur. The allocated failure rate of $5/10^5$ failures per hour reflects an estimate of this rate of occurrence.</p>
R6	Preliminary reliability allocation based on a general list of these functions constructed using EPRI Final Report No. 282, <u>Feasibility Study of a Multipurpose Refrigerator for a Superconducting Cable Test Facility</u> , prepared by T. R. Strohbridge, July 1975. Table A-11 presents the list which was constructed. Items 1, 2, 3, 4, 6, 7, and 10 of this table were assumed to be required for this subsystem. No redundancy was assumed.
R7	Preliminary reliability allocation. The purity monitoring equipment which is currently available has been designed primarily for laboratory applications for which extensive failure rate and failure experience data was not available from the sources utilized by Garrett during this study. It is not considered that these devices will have a level of reliability commensurate with other items in the refrigerator. For this reason, relatively high failure rates have been allocated to these devices.
R8	Preliminary reliability allocation to account for pressure regulating, filtration, and other equipments which may be required for this system.
R9	Preliminary reliability allocation based on the allocation of 10 percent of the total failure rate of the process stream equipments of the compressor subsystem shown in Figure A-2 and quantified in Table A-3.

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FIGURE A-3 SIMPLIFIED SCHEMATIC OF BASELINE COLD BOX SUBSYSTEM

TABLE A-5

RELIABILITY AND MAINTENANCE DATA SUMMARY SHEET

SUBSYSTEM COLD BOX GAS BEARING EXPANDERS

SUBSYSTEM NO 11

REFERENCE FIGURE FIGURE A-3

Equipment Description	Reliability Data				Maintenance Data				General Notes
	Failure Rate Per 10 ⁶ Hours λ x 10 ⁶			Data Source Notes	Time to Repair 27 Hours	Interruption For Scheduled Maintenance	Length of Time Service Interrupted for Scheduled Maintenance	Data Source Notes	
	90% Confid Lower	Best Estimate	90% Confid Upper						
Main Cold Box	--	2	--	R1	128	TBD	TBD	M.E.--	
Filter Adsorber Vacuum Module	--	1	--	R1	24	TBD	TBD	M.E.	
Expander Vacuum Module	--	0.5	--	R1	6	TBD	TBD	M.E.	
Pressure Relief Valve Vacuum Module	--	0.5	--	R1	8	TBD	TBD	M.E.	
Cooled Heat Shield	--	0.5	--	R1	128	TBD	TBD	M.E.	
No. 1 Heat Exchanger	0.58	1.24	2.33	R2	128	TBD	TBD	M.E.	
No. 2 Heat Exchanger	0.58	1.24	2.33	R2	128	TBD	TBD	M.E.	
Heat Exchangers Nos 3 to 5 (Each)	0.58	1.24	2.33	R2	128	TBD	TBD	M.E.	
Filter Adsorber	--	5	--	R6	128	TBD	TBD	M.E.	
Gas Bearing Turbine Expander	10.07	29.48	67.45	R3	6	TBD	TBD	M.E.	
Motor Driven Throttling Valve	--	8	--	R4	6	TBD	TBD	M.E.	
Particulate Filter or Screen	--	0.05	--	R5	128	TBD	TBD	M.E.	
Pressure Regulating/Relief Valve	1.072	2.055	3.585	R7	8	TBD	TBD	M.E.	
Main Cold Box Plumbing	--	0.55	--	R8	128	TBD	TBD	M.E.	
Vacuum Module Plumbing	--	6.1	--	R8	8	TBD	TBD	M.E.	
Temperature/Pressure Sensor	--	0.5	--	R9	TBD	TBD	TBD	M.E.	
Vacuum Pump and Assoc. Equipment and Instruments	--	200	--	R10	8	TBD	TBD	M.E.	
Subsystem Controls, Instruments, Interlocks and Alarms (Sensors Mounted Inside Cold Box Not Included)	--	20	--	R11	3	TBD	TBD	M.E.	

* Refers to Sources defined in Table A-6
M.E. = Maintenance Estimate

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TABLE A-6

RELIABILITY SOURCE NOTES FOR TABLE A-5
COLD BOX SUBSYSTEM

Symbol	Source
R1	Preliminary reliability allocation for structural failures resulting in heat or vacuum leakage.
R2	<p>Rome Air Development Center (RADC): <u>Revision of RADC Nonelectronic Reliability Notebook</u>, RADC TR-74-268. Statistically analyzed data for heat exchangers in ground facility applications. Although the data sources were not identified, the failure rate confidence limits shown appear reasonable for this application.</p> <p>The subsystem failure rate contribution of the cold box heat exchangers has been allocated from within the RADC failure rate confidence limits. In the system studies, larger failure rates were assigned to heat exchangers 1 and 2 because of their susceptibility to contamination freeze plugging and the subsequent risk of damage resulting from unplugging measures.</p>
R3	Failure rate confidence limits calculated from the APCI field survey data on BOC, L'Air Liquide, and Sulzer gas bearing turboexpanders. The APCI data was censored to exclude what appeared to be secondary failures as the result of operator error (two failures within ten minutes traced to an improper regulator setting) or a serious system problem (ice build up in the machine). The predominant cause of failure in the final APCI sample was impurities in the process and bearing gas. The relationship of this failure cause to the laboratory environment (intermittent operation with temperature changes and pressure fluctuations in the process stream which could result in contaminant migration) can not be quantified at this time. This puts a limitation on any extrapolation of these data into a reliability prediction for production, steady-state operating equipments.
R4	Preliminary reliability allocation for motor driven throttling valve based on complexity (i.e., valve, motor, and motor control) and limited duty cycle.
R5	Preliminary reliability allocation for failure of the screen or filter resulting in contamination of the cold box by screen/filter particles. These particles would travel through the cold box return stream (e.g., away from the cable).
R6	Preliminary reliability allocation for the failure rate associated with the premature contaminant break through and adsorbent charge retention failure modes.
R7	Rome Air Development Center (RADC): <u>Revision of RADC Nonelectronic Reliability Notebook</u> , RADC TR-74-268. Statistically analyzed data for pneumatic pressure regulating valves in ground facility applications. Although data sources and more definitive hardware identification were not available from this source, the upper 90 percent confidence level failure rate bound (e.g., MTBF of 280,000 hours) appears to be a reliable goal for a pressure balance type of device.
R8	Preliminary reliability allocation based on the assignment of ten percent of the total failure rate of the applicable process stream equipments.
R9	Reliability allocation for a pressure or temperature sensor.
R10	Preliminary reliability allocation. Much (but not all) of the vacuum indicating and maintenance equipment has been designed primarily for laboratory applications for which only a limited amount of reliability data is available. These equipments may not have a level of reliability commensurate with other items in the refrigerator. For this reason a relatively high failure rate has been allocated.
R11	Preliminary reliability allocation based on a general list of these functions set forth in EPRI Final Report No. 282, <u>Feasibility Study of a Multipurpose Refrigerator for a Superconducting Cable Test Facility</u> , prepared by T.R. Strobridge, July 1975.

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FIG. 1 - A-4: SIMPLIFIED SCHEMATIC OF REMOTE EXPANSION SUBSYSTEM

TABLE A-7

RELIABILITY AND MAINTENANCE DATA SUMMARY SHEET

SUBSYSTEM: REMOTE EXPANDER, GAS BEARING TURBINE EXPANDER

SUBSYSTEM NO: III

REFERENCE FIGURE: A-4

Equipment Description	Reliability Data				Maintenance Data				General Notes
	Failure Rate Per 10 ⁶ Hours: $\lambda \times 10^6$			Data Source Notes ^a	Time to Repair 2 τ Hours	Interruption For Scheduled Maintenance	Length of Time Service Inter- rupted for Scheduled Maintenance	Data Source Notes	
	90% Confid Lower	Best Estimate	90% Confid Upper						
Vacuum Box	--	1	--	R1	40	TBD	TBD	M.E.**	
Gas Bearing Turbine Expander	10.07	29.48	67.45	R2	6	TBD	TBD	M.E.	
Metallic Particle Containment	--	50	--	R3	6	TBD	TBD	M.E.	
Pressure Regulating/Relief Valve	1.072	2.055	3.585	R4	6	TBD	TBD	M.E.	
Vacuum Box Plumbing	--	2.1	--	R5	40	TBD	TBD	M.E.	
Vacuum Pump and Assoc. Equipment and Instruments	--	100	--	R6	8	TBD	TBD	M.E.	
Subsystem Controls, Instruments and Interlocks	--	8	--	R7	3	TBD	TBD	M.E.	

* Refers to sources defined in Table A-8

***M.E. = Maintenance Estimate

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TABLE A-8
RELIABILITY SOURCE NOTES FOR TABLE A-7:
REMOTE EXPANDER SUBSYSTEM

Symbol	Source
R1	Preliminary reliability allocation for structural failures resulting in heat or vacuum leakage.
R2	Failure rate confidence limits calculated from the APCI field survey data on BOC, L'Air Liquide, and Sulzer gas bearing turboexpanders. The APCI data was censored to exclude what appeared to be secondary failures as the result of operator error (two failures within ten minutes traced to an improper regulator setting) or a serious system problem (ice build up in the machine). The predominant cause of failure in the final APCI sample was impurities in the process and bearing gas. The relationship of this failure cause to the laboratory environment (intermittent operation with temperature changes and pressure fluctuations in the process stream which could result in contaminant migration) can not be quantified at this time. This puts a limitation on any extrapolation of these data into a reliability prediction for production, steady-state operating equipments
R3	<p>Preliminary reliability allocation. This metallic particle containment device is considered to be a critical item since without it metallic particles generated by a turboexpander failure would pass into the cable. A screen or filter is not considered feasible at this time because of:</p> <ul style="list-style-type: none"> (i) Possible excessive pressure differential across the device. (ii) The possibility of a screen or filter failure introducing contaminants into the cable. <p>It was not determined if a device which would meet the requirements set forth below is currently available. This is recommended as an action item for a subsequent refrigerator study. The requirements foreseen for this device are:</p> <ul style="list-style-type: none"> (i) Capability of stopping the passage of <u>all</u> metallic particles into the cable without requiring an excessive pressure differential. (ii) The failure modes of this device must be such that it will fail in a manner which will not release metallic particles into the cable. That is, the device must be fail safe. (iii) The device must be reliable and capable of functioning unattended for extended periods of time.
R4	Rome Air Development Center (RADC): Revision of RADC Nonelectronic Reliability Notebook, RADC TR-74-268. Statistically analyzed data for pneumatic pressure regulating valves in ground facility applications. Although data sources and more definitive hardware identification were not available from this source, the upper 90 percent confidence level failure rate found (e.g., MTBF of 280,000 hours) appears to be a reasonable goal for a pressure balance type of device.
R5	Preliminary reliability allocation based on the assignment of ten percent of the total failure rate of the applicable process stream equipments.
R6	Preliminary reliability allocation. Much (but not all) of the vacuum indicating and maintenance equipment has been designed primarily for laboratory applications for which only a limited amount of reliability data is available. These equipments may not have a level of reliability commensurate with other items in the refrigerator. For this reason a relatively high failure rate has been allocated.
R7	Preliminary reliability allocation.

FIGURE A-5: SIMPLIFIED SCHEMATIC OF A TYPICAL TWO
STAGE RECIPROCATING COMPRESSOR
SUBSYSTEM

TABLE A-9

RELIABILITY AND MAINTENANCE DATA SUMMARY SHEET
FOR MISCELLANEOUS ALTERNATE REFRIGERATION EQUIPMENTS

	Reliability Data				Maintenance Data				
Equipment Description	Failure Rate Per 10 ⁶ Hours: $\lambda \times 10^6$			Data Source Notes*	Time to Repair τ Hours	Time to Service Interruption For Scheduled Maintenance	Length of Time Service Interrupted for Scheduled Maintenance	Data Source Notes	General Notes
	90% Confid Lower	Best Estimate	90% Confid Upper						
Reciprocating Dry Lubricated Expander	99.24	177	292.91	R1	TBD	2500 hours (rings)	TBD	M1**	Failure rate per expander cylinder
Reciprocating Dry Lubricated Compressor	73.16	123.69	196.66	R2	TBD	2500 hours (rings)	TBD	M1	Failure rate for a two stage compressor

* Refers to sources identified in Table A-10

** Maintenance estimate based on field survey data identified in source R2.

TABLE A-10

RELIABILITY SOURCE NOTES FOR TABLE A-9
MISCELLANEOUS ALTERNATE REFRIGERATION EQUIPMENTS

Symbol	Source																																			
R1	Failure rate confidence range is based on field survey data reported in <u>OPERATING EXPERIENCE - HELIUM EXTRACTION AND LIQUEFACTION EXPANSION DEVICES</u> prepared for the National Accelerator Laboratory in January 1974 by T.R. Strobridge and C.F. Sindt.																																			
R2	<p>Failure rate confidence range is based on the following APCI field survey data:</p> <table><tr><th>Laboratory</th><th>Compressor</th><th>No.</th><th>Total Stage Hours</th><th>No. Failures</th></tr><tr><td>CERL</td><td>2 Stage Alley</td><td>1</td><td>36,000</td><td>3</td></tr><tr><td>CULHAM</td><td>2 Stage Broom Walde</td><td>1</td><td>35,000</td><td>3</td></tr><tr><td>CERN</td><td>3 Stage Sulzer</td><td>2</td><td>48,000</td><td>4</td></tr><tr><td>McGILL</td><td>2 Stage Sulzer</td><td>1</td><td>31,200</td><td>0</td></tr><tr><td>LIVERMORE</td><td>2 Stage RIX</td><td>1</td><td>38,000</td><td>3</td></tr><tr><td>SIN</td><td>2 Stage Sulzer</td><td>1</td><td>22,000</td><td>0</td></tr></table>	Laboratory	Compressor	No.	Total Stage Hours	No. Failures	CERL	2 Stage Alley	1	36,000	3	CULHAM	2 Stage Broom Walde	1	35,000	3	CERN	3 Stage Sulzer	2	48,000	4	McGILL	2 Stage Sulzer	1	31,200	0	LIVERMORE	2 Stage RIX	1	38,000	3	SIN	2 Stage Sulzer	1	22,000	0
Laboratory	Compressor	No.	Total Stage Hours	No. Failures																																
CERL	2 Stage Alley	1	36,000	3																																
CULHAM	2 Stage Broom Walde	1	35,000	3																																
CERN	3 Stage Sulzer	2	48,000	4																																
McGILL	2 Stage Sulzer	1	31,200	0																																
LIVERMORE	2 Stage RIX	1	38,000	3																																
SIN	2 Stage Sulzer	1	22,000	0																																

TABLE A-11

RELIABILITY ESTIMATES FOR THE BASIC COMPRESSOR SUBSYSTEM
CONTROLS, INSTRUMENTS, INTERLOCKS AND ALARMS

Item No.	Description and Failure Rates of Constituent Items	Estimated Failure Rate $\lambda/10^6$ hr
1	Compressor Vibration: one per 2 stages Sensor Mech ($\lambda = 2/10^6$) Switch and Reset Mech ($\lambda = 1/10^6$)	3
2A*	Low Compressor Suction Temp. and Pressure Indicator ($\lambda = 1/10^6$) Pressure/Temp Sensors ($\lambda = 1/10^6$) Shutdown and Reset ($\lambda = 1/10^6$)	3
3A	High Compressor Discharge Temp: one per stage Indicator ($\lambda = 1/10^6$) Alarm Switch ($\lambda = 1.5/10^6$) Sensor ($\lambda = 0.5/10^6$)	3
4	Compressor Discharge Pressure: one per stage Sensor ($\lambda = 0.5/10^6$) Indicator ($\lambda = 1/10^6$)	1.5
5A	Compressor Oil Pressure Pressure Switch ($\lambda = 2/10^6$) Shutdown and Reset ($\lambda = 0.5/10^6$)	2.5
6A	Compressor Motor Overload Overload Sensor ($\lambda = 1.5/10^6$) Overload Relay ($\lambda = .05/10^6$)	1.55
7	Compressor Motor Controls and Protection Contactors ($\lambda = 1.5/10^6$) Starter relays, resets, etc. ($\lambda = 2.5/10^6$) Thermal and electrical protection ($\lambda = 1.5/10^6$)	5
8	Oil Pump Motor Controls and Protection	3
9	Compressor/Pump Shutdown Interlock	2.5
10	Failure and Warning Annunciation for "A" suffix Items Audible Alarm ($\lambda = 1.5/10^6$) Audible Alarm Silence Relay and Button ($\lambda = 0.5/10^6$) Failure/Warning Lights ($\lambda = 1/10^6$)	3

*Alarm annunciation of failure.

- Items which do not require a fork lift and special handling can be obtained from a warehouse and delivered to the site in one and one half hours or less.
- Items requiring a fork lift and special transportation can be obtained from a warehouse and delivered to the site in two and one half hours or less.

Based on the above assumptions, the time to repair includes from two to three hours to identify the cause of an abnormal refrigerator conditions and to collect and deliver the items needed for repair.

The determination of quantitative maintainability data for these equipment remains in a preliminary stage. A TBD in the table denotes an area which is "To Be Determined." Where quantitative data is presented, source notes are also presented.

FIGURE B-1: USE OF ACTIVE REDUNDANCY TO DEFER CORRECTIVE MAINTENANCE ON CRITICAL SENSORS WITH REPAIR/REPLACEMENT TIMES GREATER THAN 8 HOURS

The active redundancy of two identical sensors allow continued operation after first sensor failure.* The refrigerator would be shutdown for corrective maintenance at second failure.

Model of failure rate for arrangement:

$$\lambda_{\text{arrangement}} = \frac{2\lambda}{3} = \frac{2(0.5/10^6)}{3} = 0.33/10^6$$

The location and numbers of these arrangements are:

Cold Box Heat Exchangers	8 arrangements per HX
Cold Box Turboexpanders	4 arrangements per Exp.
Cold Box / Cable Interface	4 arrangements
Remote Turbo expander	2 arrangements
Cable / Remote Turbo expander Interface	2 arrangements

* Note - For this study it was assumed that all sensor failures are detectable

FIGURE B-2: USE OF STAND-BY REDUNDANCY FOR CRITICAL FAILURE MODE PROTECTION, CONTAMINANT BREAK THROUGH FROM THE FILTER ADSORBER TO THE COLD BOX

RELIABILITY LOGIC

The refrigerator is shutdown when the process stream purity monitors detect contaminant break through from the primary filter adsorber. The secondary filter adsorber prevents contamination of the cold box while the refrigerator is being shut down. It is assumed that shutdown will be rapid and that the possibility of contaminant break through from the secondary filter adsorber during this shutdown period is negligible.

It should be noted that this arrangement is for protection from a particular failure mode. The premature expenditure of the adsorbant in the primary filter adsorber will require refrigerator shutdown for an unscheduled corrective maintenance action. A failure of the secondary filter adsorber to retain its adsorbant will require refrigerator shutdown to clean up the cold box. The failure rates of these two items are reflected in Table 5-1.

FIGURE B-3: DEVELOPMENT OF THE MATHEMATICAL MODEL FOR DETERMINING THE MTBF OF THE BASELINE CABLE REFRIGERATOR SYSTEM

I. RELIABILITY LOGIC DIAGRAM

AT LEAST ONE OUT OF TWO MUST FUNCTION

II. THE SYSTEM WILL ALWAYS BE IN ONE OF THE THREE FOLLOWING STATES

State 0 - both elements are operating

State 1 - one element is operating. The other element has failed and is being repaired.

State 2 - both elements have failed and are being repaired. The cable system is out of service.

III. DEVELOPMENT OF THE FUNDAMENTAL MATRIX USING THE METHOD DESCRIBED IN THE SYSTEM STUDY MODELING DISCUSSION OF SECTION 5

ATTACHMENT TO FIGURE B-3

$$P = \begin{array}{c|ccc} & \text{State 0} & \text{State 1} & \text{State 2} \\ \hline \text{State 0} & 1-2\lambda & 2\lambda & 0 \\ \text{State 1} & \mu & 1-(\lambda+\mu) & \lambda \\ \text{State 2} & 0 & 2\mu & 1-2\mu \end{array}$$

where P is the probability transition matrix for the system moving among its various failed and non-failed states

$$Q = \begin{array}{c|cc} & \text{State 0} & \text{State 1} \\ \hline \text{State 0} & 1-2\lambda & 2\lambda \\ \text{State 1} & \mu & 1-(\lambda+\mu) \end{array} \quad \begin{array}{l} \text{the transition matrix for} \\ \text{the system moving} \\ \text{among its various} \\ \text{non-failed states.} \end{array}$$

$$I - Q = \begin{vmatrix} 1 & 0 \\ 0 & 1 \end{vmatrix} - \begin{vmatrix} 1-2\lambda & 2\lambda \\ \mu & 1-(\lambda+\mu) \end{vmatrix} = \begin{vmatrix} 2\lambda & -2\lambda \\ -\mu & \lambda+\mu \end{vmatrix}$$

the matrix required to determine the Fundamental Matrix

DATE _____

CALC. NO. _____ SHEET NO. 3 of 3

PART NO. _____

MODEL NO. _____

PREPARED BY _____

CHECKED BY _____

ATTACHMENT TO FIGURE B-3: CONTINUED

$$(I - \Phi)^{-1} = \begin{array}{c|cc} & \text{State 0} & \text{State 1} \\ \hline \text{State 0} & \frac{\lambda + \mu}{2\lambda^2} & \frac{1}{\lambda} \\ \text{State 1} & \frac{\mu}{2\lambda^2} & \frac{1}{\lambda} \end{array}$$

the Fundamental Matrix. The sum of each row is the mean time before system failure given that the system started in that state

IV MODEL FOR MTBF FOR THE SYSTEM STARTING FROM STATE 0 (BOTH ELEMENTS OPERATING)

$$\text{MTBF} = \frac{\lambda + \mu}{2\lambda^2} + \frac{1}{\lambda} = \frac{3\lambda + \mu}{2\lambda^2}$$

RELIABILITY LOGIC DIAGRAM
 (At least two elements must function)

SKETCH OF ADDED STAND-BY EQUIPMENT

FIGURE B4: ASSESSMENT OF ADDED STAND-BY REDUNDANCY TO THE COMPRESSOR HELIUM / LIQUID OIL SEPARATION EQUIPMENT

ATTACHMENT TO FIGURE B-4

1. Derivation of Math Model

From Reliab diagram the equipment can be in one of three states

0. Both elements in primary leg are operating, the stand-by element is off.
- 1 One element in the primary leg has failed and is being repaired. The other primary element and the stand-by element are providing liquid oil/helium separation
2. Two elements (either both primaries or one primary and the stand-by) have failed and are being repaired. The function has been lost.

The equipment transition matrices are:

$$P = \begin{matrix} & \begin{matrix} 0 & 1 & 2 \end{matrix} \\ \begin{matrix} 0 \\ 1 \\ 2 \end{matrix} & \begin{bmatrix} 1 - 2\lambda_p & 2\lambda_p & 0 \\ \mu & 1 - (\lambda_p + \lambda_s + \mu) & \lambda_p + \lambda_s \\ 0 & 2\mu & 1 - 2\mu \end{bmatrix} \end{matrix}$$

$$Q = \begin{bmatrix} 1 - 2\lambda_p & 2\lambda_p \\ \mu & 1 - (\lambda_p + \lambda_s + \mu) \end{bmatrix}, \quad I - Q = \begin{bmatrix} 2\lambda_p & -2\lambda_p \\ -\mu & \lambda_p + \lambda_s + \mu \end{bmatrix}$$

$$(I - Q)^{-1} = \begin{bmatrix} \frac{\lambda_p + \lambda_s + \mu_p}{2\lambda_p(\lambda_p + \lambda_s)} & \frac{1}{\lambda_p + \lambda_s} \\ \frac{\mu_p}{2\lambda_p(\lambda_p + \lambda_s)} & \frac{1}{\lambda_p + \lambda_s} \end{bmatrix}$$

ATTACHMENT TO FIGURE B4: CONTINUED

$$MTBF = \frac{\lambda_p + \lambda_s + \mu_p}{2\lambda_p(\lambda_p + \lambda_s)} + \frac{1}{\lambda_p + \lambda_s} = \frac{3\lambda_p + \lambda_s + \mu_p}{2\lambda_p(\lambda_p + \lambda_s)}$$

And since in this case $\lambda_p = \lambda_s$

$$MTBF = \frac{4\lambda_p + \mu_p}{4\lambda_p^2}$$

2. Calculation of New Failure Rate and Down Time Values for Item I.3 Oil Separation Equipment of Table 5-1

For Table 5-1 $\mu_p = 0$ since the repair rate will reflect not only the actual corrective action but also maintenance policies regarding the deferral of corrective action until preventive maintenance, other corrective actions, etc are required.

$$\therefore MTBF = \frac{1}{\lambda_p}$$

And the new values for Item I.3 of Table 51 are

Column 3: $N = 1$ set

Column 5: Re frig Failure Rate = $\lambda_p = 50/10^6$; enter 50 in col 5

Column 6: $DT = 50 \times 4 = \underline{200}$

DATE _____

PART NO. _____

PREPARED BY _____

CALC. NO. _____ SHEET NO. 1 of 4

MODEL NO. _____

CHECKED BY _____

SKETCH OF ADDED STAND-BY REDUNDANCY TO COMPRESSOR
FILTER/ADSORBER AND PURITY MONITORING FUNCTIONS

FIGURE B-5: ASSESSMENT OF ADDED STAND-BY REDUNDANCY
TO THE COMPRESSOR FILTER ADSORBER
AND PURITY MONITORING FUNCTIONS

ATTACHMENT TO FIGURE B-5

1. Reliability Logic Diagram (Applicable to either function)

2. Derivations of Math Model

The equipment can be in any one of three states :

State 0 Primary element operating, stand by element off.

State 1 Primary element failed and under repair, the stand-by element has been switched on to provide the equipment function.

State 2 Both the primary and stand-by elements have failed and are being repaired. The function has been lost!

The Transition Matrices Are :

$$P = \begin{matrix} & \begin{matrix} 0 & 1 & 2 \end{matrix} \\ \begin{matrix} 0 \\ 1 \\ 2 \end{matrix} & \begin{bmatrix} 1-\lambda & \lambda & 0 \\ \mu & 1-(\lambda+\mu) & 0 \\ 0 & \lambda & 1-\mu \end{bmatrix} \end{matrix}$$

$$Q = \begin{bmatrix} 1-\lambda & \lambda \\ \mu & 1-(\lambda+\mu) \end{bmatrix} \quad I-Q = \begin{bmatrix} \lambda & -\lambda \\ -\mu & \lambda+\mu \end{bmatrix}$$

$$(I-Q)^{-1} = \begin{bmatrix} \frac{\lambda+\mu}{\lambda^2} & \frac{1}{\lambda} \\ \frac{\mu}{\lambda^2} & \frac{1}{\lambda} \end{bmatrix} : \text{MTBF} = \frac{\lambda+\mu}{\lambda^2} + \frac{1}{\lambda} = \frac{2\lambda+\mu}{\lambda^2}$$

DATE _____

CALC. NO. _____ SHEET NO. 3 of 4

PART NO. _____

MODEL NO. _____

PREPARED BY _____

CHECKED BY _____

ATTACHMENT TO FIGURE B-5: CONTINUED

3. Calculation of Values for the Filter Adsorber Function, Item I.9 of Table 5-1.

For Table 1 $\mu = 0^*$ $\therefore$ MTBF = $\frac{2}{\lambda}$
 and the new values for Table 5-1 are

Col. 3: $N = \underline{1 \text{ set}}$ Col 4: Refrig Failure Rate = $\frac{\lambda}{2} = \frac{100/10^6}{2} = 50/10^6$ $\therefore$ enter 50 in column 4Col. 5: $D_T = 8(50) = \underline{400}$ Col 7: Ref. Failure Rate = $(.05)50 = \underline{2.5}$ Col 9: $D_T = (2.5)(11) = \underline{27.5}$

* No automatic regeneration capability for an expended adsorber has been assumed

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ATTACHMENT TO FIGURE B-5: CONTINUED

4. Calculation of new values for the Priority Monitoring Function, Items I.10, I.11, and I.12 of Table 5-1.

Combine these three items into one entry on Table 5-1, say Item I.10.

For Table 1 $\mu = 0 \therefore MTBF = \frac{2}{\lambda}$

The values for Table 5-1 are:

Column 1. $\lambda = 400/10^6$ enter 400

Column 2 $T = \underline{8}$

Column 3 $N = \underline{1 \text{ set}}$

Column 4 Refrig Failure Rate = $\frac{400/10^6}{2} = 200/10^6$
enter 200 in col 4

Column 5 $D_T = 8(200) = \underline{1600}$

Column 7 Refrig Failure Rate = $(.05)(200) = \underline{10}$

Column 9 $D_T = 11(10) = \underline{110}$

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FIGURE B-6: ASSESSMENT OF ADDED STAND-BY REDUNDANCY TO THE COLD BOX FILTER ADSORBER AND PURITY MONITORING FUNCTIONS

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ATTACHMENT TO FIGURE B-6

1. PURITY MONITORS

These monitors are identical to those used in the compressor subsystem. Their reliability logic is also the same.

Adding Item II.20, Purity Monitoring, to Table 5-1 the following data (taken from Figure B-5, Purity Monitors)

Column 1: $\lambda \times 10^6 = \underline{400}$

Column 2: $T = \underline{8}$

Column 3: $\underline{1}$ Set

Column 4: Ref. Failure Rate = 200

Column 5: $D_T = \underline{1600}$

Column 6: $\% = \underline{.05}$

Column 7: Ref. Failure Rate = 10

Column 8: $\tau = \underline{11}$

Column 9: $D_T = \underline{110}$

ATTACHMENT TO FIGURE B-6: CONTINUED

2. FILTER ADSORBERS

a) Reliability Logic Diagram

* The reduction of τ from 128 hours to 12 hours is the result of adding a secondary filter adsorber downstream of the primary filter adsorber for protection against contamination of the cold box from carryover from a prematurely expended filter adsorber (this situation is identical to that shown in Figure B-2).

b) From Figure B-5 the Math Model is .

$$MTBF = \frac{2\lambda + \nu}{\lambda^2}; \text{ and if } \nu = 0 \quad MTBF = \frac{2}{\lambda}$$

c) Calculation of Values of Filter Adsorber Function, Item II.9 of Table 5-1.

Column 4. Refrig Failure Rate = $\lambda/2 = \underline{2.5}$

Column. 3. $\tau = \underline{12 \text{ hours}}$

Column 5: $D_T = 12(2.5) = \underline{30}$

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ATTACHMENT TO FIGURE B6: CONTINUED

Column 7: Refrig Failure Rate = $1(2.5) = \underline{2.5}$

Column 8: $T = 12$ hours

Column 9: $D_T = \underline{30}$

FIGURE B-7: ASSESSMENT OF ADDED STAND-BY REDUNDANCY FOR THE VACUUM PUMPS AND ASSOCIATED EQUIPMENTS FOR THE COLD BOX AND REMOTE EXPANDER SUBSYSTEMS

1 RELIABILITY LOGIC DIAGRAM

PRIMARY VAC.
EQUIPMENTS
 λ_p^*
 $T = 8 \text{ hrs}$

STAND-BY VAC.
EQUIPMENTS
 λ_s^*
 $T = 8 \text{ hrs}$

failure rate of "failure to switch" failure mode considered negligible

* For the Cold Box S.S. Equipments
 $\lambda_p \& \lambda_s = 200/10^6$

For the Remote Expander S.S. Equipments
 $\lambda_p \& \lambda_s = 100/10^6$

2. MATH MODEL

From Part 2. of Figure B-5

$$MTBF = \frac{2\lambda + \mu}{\lambda^2}; \text{ and if } \mu = 0 \quad MTBF = \frac{2}{\lambda}$$

3. CALCULATION OF VALUES FOR THE COLD BOX S.S. VAC. EQUIPMENTS, ITEM II.18 OF TABLE 5-1

Column 3: $N = 1 \text{ set}$

Column 4: Refrig. Failure Rate = 100

Column 5: $DT = 8(100) = \underline{800}$

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ATTACHMENT TO FIGURE B-7: CONTINUED

Column 7: %Refrig. Failure Rate = $(.05)(100) = \underline{5}$

Column 9: $D_T = (5)(11) = \underline{55}$

4. CALCULATION OF VALUES FOR REMOTE EXPANDER S.S. VAC. EQUIPMENTS, ITEM III.7 OF TABLE 5-1.

Column 3: $N = \underline{1 \text{ set}}$

Column 4: Refrig Failure Rate = 50

Column 5: $D_T = 8 \times 50 = \underline{400}$

Column 7: %Refrig Failure Rate = $(.05)(50) = \underline{2.5}$

Column 9: $D_T = (2.5)(11) = \underline{27.5}$

LEGEND

○ = an arrangement of temperature and pressure sensors defined as follows

FIGURE B-8: ASSESSMENT OF COMPARTMENTATION AND SENSOR CHANGES TO THE BASELINE COLD BOX

ATTACHMENT TO FIGURE B-8

1. Math Model for Sensor Group

Since each sensor group will not be replaced until all four sensors have failed:

$$\lambda_{\text{group}} = \left[\frac{1}{\lambda} + \frac{1}{2\lambda} + \frac{1}{3\lambda} + \frac{1}{4\lambda} \right]^{-1} = \left[\frac{12+6+4+3}{12\lambda} \right]^{-1} = \left[\frac{25}{12\lambda} \right]^{-1}$$

Given a failure rate of $0.5/10^6$ for an individual sensor:

$$\lambda_{\text{group}} = \frac{12(0.5/10^6)}{25} = 0.24/10^6$$

2. Changes to Table 5-1.

The repair time τ is determined by the location of the sensor group within the cold box. The new values for Table 5-1 are:

Item II.15 Main Cold Box Sensors

Col 1 $\lambda = 0.24$; Col 2 $T = 128$; Col 3 $N = 4 \text{ groups}$

Col 4 $N\lambda = 0.96$; Col 5 $D_T = 122$; Col 6 $\% = 1$

Col 7 $\%N\lambda = 0.96$; Col 8 $\tau = 128$; Col 9 $D_T = 122$

Item II.16 Turboexpander Sensors

Col 1 $\lambda = 0.24$; Col 2 $\tau = 6$; Col 3 $N = 12+8 = 20 \text{ groups}$

Col 4 $N\lambda = 4.80$; Col 5 $D_T = 28.8$; Col 6 $\% = 0.025$

Col 7 $\%N\lambda = 0.12$; Col 8 $\tau = 9$; Col 9 $D_T = 1.08$

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ATTACHMENT TO FIGURE B-8: CONTINUED

Item II.17 Cold Box/Cable Interface Sensors

Col 1 $\lambda = \underline{0.24}$; Col 2 $\tau = \underline{8}$; Col 3 $N = \underline{8 \text{ groups}}$ Col 4 $N\lambda = \underline{1.92}$; Col 5 $D_T = \underline{15.36}$; Col 6 $\% = \underline{.16}$ Col 7 $\%N\lambda = \underline{0.307}$; Col 8 $\tau = \underline{11}$; Col 9 $D_T = \underline{3.379}$

Added Item II.17A HX 1 Module Sensors

Col 1 $\lambda = \underline{0.24}$; Col 2 $\tau = \underline{40}$; Col 3 $N = \underline{4 \text{ groups}}$ Col 4 $N\lambda = \underline{0.96}$; Col 5 $D_T = \underline{38.40}$; Col 6 $\% = \underline{1}$ Col 7 $\%N\lambda = \underline{0.96}$; Col 8 $\tau = \underline{40}$; Col 9 $D_T = \underline{38.40}$

Added Item II.17B Filter Adsorber Module Sensors

Col 1 $\lambda = \underline{0.24}$; Col 2 $\tau = \underline{12}$; Col 3 $N = \underline{4 \text{ groups}}$ Col 4 $N\lambda = \underline{0.96}$; Col 5 $D_T = \underline{11.52}$; Col 6 $\% = \underline{1}$ Col 7 $\%N\lambda = \underline{0.96}$; Col 8 $\tau = \underline{12}$; Col 9 $D_T = \underline{11.52}$

In addition the Remote Expander Subsystem incorporates four sensor groups. Therefore:

Item III.6

Col 1. $\lambda = \underline{0.24}$; Col 4. $N\lambda = \underline{0.96}$; Col 5. $D_T = \underline{38.4}$ Col 7. $\%N\lambda = \underline{0.96}$; Col 9. $D_T = \underline{38.4}$

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ATTACHMENT TO FIGURE B-8: CONTINUED

3. Changes to Repair Times From Compartmentation of the Cold Box

- a) Compartmentation of HX 1 will reduce estimated repair times associated with:
- on stream secondary filter adsorber in compressor S.S.
 - HX 1 sensors
 - HX 1

Estimated repair time reduction from 128 to 40 hours.

- b) Placing cold box filter adsorbers in separate, smaller vacuum modules. Estimated repair time reduction from 24 to 12 hours.

SKETCH 1

STAND-BY COLD BOX SUBSYSTEM MAINTAINED IN READINESS FOR FINAL COOLDOWN WITHIN EIGHT HOURS. ANY COMPRESSOR CAN PROVIDE THE HELIUM FLOW REQUIRED TO MAINTAIN STAND-BY READINESS

* = ITEM OUT OF SERVICE

SKETCH 2

ONE COLD BOX SUBSYSTEM OUT OF SERVICE. FINAL COOL DOWN AND SWITCH OVER HAVE BEEN ACCOMPLISHED TO RESTORE ACTIVE REFRIGERATION TO THE AFFECTED CABLE SEGMENT

* A DIRECT INTERFACE BETWEEN THESE TWO SUBSYSTEMS IS SHOWN FOR CLARITY. THE INTERFACE WOULD ACTUALLY BE ACCOMPLISHED THROUGH THE STATION SWITCH OVER SUBSYSTEM

SKETCH 3

TWO COMPRESSOR AND THREE COLD BOX SUBSYSTEMS OUT OF SERVICE. THE STATION SWITCH OVER SUBSYSTEM HAS SCHEDULED THE REMAINING SUBSYSTEMS TO MAINTAIN ACTIVE REFRIGERATION FOR TWO CABLE SEGMENTS ON THE SAME CIRCUIT TO AVOID A CABLE SYSTEM SHUTDOWN

FIGURE B-9: STAND-BY REDUNDANCY AND INTRA-STATION SWITCHING FOR A REFRIGERATOR SUBSTATION

ATTACHMENT TO FIGURE B-9

I. CABLE CIRCUIT RELIABILITY OF A REFRIGERATION STATION

A. RELIABILITY LOGIC DIAGRAM

B. DEVELOPMENT OF MATH MODEL FOR COLD BOX ARRANGEMENT

For this assessment assume the following.

- The failure rate of the stand-by element is negligible while the stand-by cold box is being maintained in a state of readiness
- The failure rate of the switchover subsystem is negligible for the "fail to switch" failure mode

ATTACHMENT TO FIGURE B-9: CONTINUED

c) The failure rate of the stand-by cold box equals the failure rate of a primary cold box when the stand-by has been cooled down and brought on stream.

d) The repair rate of the stand-by cold box equals the repair rate of a primary cold box.

Then, the Cold Box Subsystem arrangement can be in any one of the following states

State 0 - all four of the primary cold box subsystems are operating.

State 1 - one of the four primary subsystems has failed and is being repaired. The stand-by unit has been brought on-stream.

State 2 - two subsystems have failed. One is being repaired and one is awaiting repair. Three subsystems are on-stream. One cable segment is without active refrigeration.

The transition matrices are

$$P = \begin{matrix} & \begin{matrix} \text{State 0} & \text{State 1} & \text{State 2} \end{matrix} \\ \begin{matrix} \text{State 0} \\ \text{State 1} \\ \text{State 2} \end{matrix} & \left[\begin{array}{ccc} 1-4\lambda & 4\lambda & 0 \\ \mu & 1-(4\lambda+\mu) & 4\lambda \\ 0 & \mu & 1-\mu \end{array} \right] \end{matrix}$$

ATTACHMENT TO FIGURE B-9: CONTINUED

$$Q = \begin{bmatrix} 1-4\lambda & 4\lambda \\ \mu & 1-(4\lambda+\mu) \end{bmatrix}; I-Q = \begin{bmatrix} 4\lambda & -4\lambda \\ -\mu & 4\lambda+\mu \end{bmatrix}$$

$$(I-Q)^{-1} = \begin{bmatrix} \frac{4\lambda+\mu}{16\lambda^2} & \frac{1}{4\lambda} \\ \frac{\mu}{16\lambda^2} & \frac{1}{4\lambda} \end{bmatrix}$$

$$MTBF = \frac{4\lambda+\mu}{16\lambda^2} + \frac{1}{4\lambda} = \frac{4\lambda+\mu+4\lambda}{16\lambda^2} = \frac{8\lambda+\mu}{16\lambda^2}$$

For the cold box arrangement.

C) MATH MODEL FOR ASSESSING CABLE CIRCUIT FAILURE RATE OF A REFRIGERATOR STATION

$$\lambda_{RS_{CMT}} = \left[\frac{8\lambda_{CB} + \mu_{CB}}{16\lambda_{CB}^2} \right]^{-1} + 4\lambda_{COMP} + \lambda_{SWITCH}$$

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ATTACHMENT TO FIGURE B-9: CONTINUED

II CABLE SYSTEM RELIABILITY OF A REFRIGERATION STATION

A. RELIABILITY LOGIC DIAGRAM

ATTACHMENT TO FIGURE B-9: CONTINUED

B. DEVELOPMENT OF MATH MODEL FOR COLD BOX ARRANGEMENT.

Using the same assumptions set forth in PART I B, the cold box arrangement will be in one of the following states:

State 0 - all four of primary elements operating

State 1 - one primary has failed and is being repaired. The stand-by has been brought on stream. The stand-by now is considered as a primary element

State 2 - two primaries have failed. One is being repaired, one is awaiting repair. The three remaining primaries are providing active refrigeration to three cable segments.

State 3 - three primaries have failed. One is being repaired, and two are awaiting repair. The two remaining primaries have been scheduled by the switch over subsystem to maintain active refrigeration to two cable segments on one circuit.

State 4 - four primaries have failed. One is being repaired, and three are awaiting repair. One primary is operational. Two segments of one circuit and one segment of the other circuit are without active refrigeration.

State 5* - all five elements have failed. One is being repaired, and four are awaiting repair. All four cable circuit segments are without active refrigeration.

* State 5 is not considered in the reliability calculations since system failure is considered to occur at State 4. Thus MTBF is based on the time to reach State 4.

ATTACHMENT TO FIGURE B-9: CONTINUED

The transition matrices are:

$$P = \begin{array}{c|cccccc} & \text{State 0} & \text{State 1} & \text{State 2} & \text{State 3} & \text{State 4} & \text{State 5} \\ \hline \text{State 0} & 1-4\lambda & 4\lambda & 0 & 0 & 0 & 0 \\ \text{State 1} & \mu & 1-(4\lambda+\mu) & 4\lambda & 0 & 0 & 0 \\ \text{State 2} & 0 & \mu & 1-(3\lambda+\mu) & 3\lambda & 0 & 0 \\ \text{State 3} & 0 & 0 & \mu & 1-(2\lambda+\mu) & 2\lambda & 0 \\ \text{State 4} & 0 & 0 & 0 & \mu & 1-(\lambda+\mu) & \lambda \\ \text{State 5} & 0 & 0 & 0 & 0 & \mu & 1-\mu \end{array}$$

$$Q = \begin{array}{c|cccc} & & & & \\ \hline & 1-4\lambda & 4\lambda & 0 & 0 \\ & \mu & 1-(4\lambda+\mu) & 4\lambda & 0 \\ & 0 & \mu & 1-(3\lambda+\mu) & 3\lambda \\ & 0 & 0 & \mu & 1-(2\lambda+\mu) \end{array}$$

$$I-Q = \begin{array}{c|cccc} & & & & \\ \hline & 4\lambda & -4\lambda & 0 & 0 \\ & -\mu & 4\lambda+\mu & -4\lambda & 0 \\ & 0 & -\mu & 3\lambda+\mu & -3\lambda \\ & 0 & 0 & -\mu & 2\lambda+\mu \end{array}$$

ATTACHMENT TO FIGURE B-9: CONTINUED

$$(I-Q)^{-1} = \begin{array}{cccc} \frac{24\lambda^3 + 6\lambda^2\mu + 2\lambda\mu^2 + \mu^3}{96\lambda^4} & \frac{6\lambda^2 + 2\lambda\mu + \mu^2}{24\lambda^3} & \frac{2\lambda + \mu}{6\lambda^2} & \frac{1}{2\lambda} \\ \frac{6\lambda^2\mu + 2\lambda\mu^2 + \mu^3}{96\lambda^4} & \frac{6\lambda^2 + 2\lambda\mu + \mu^2}{24\lambda^3} & \frac{2\lambda + \mu}{6\lambda^2} & \frac{1}{2\lambda} \\ \frac{2\lambda\mu^2 + \mu^3}{96\lambda^4} & \frac{2\lambda\mu + \mu^2}{24\lambda^3} & \frac{2\lambda + \mu}{6\lambda^2} & \frac{1}{2\lambda} \\ \frac{\mu^3}{96\lambda^4} & \frac{\mu^2}{24\lambda^3} & \frac{\mu}{6\lambda^2} & \frac{1}{2\lambda} \end{array}$$

$$MTBF = \frac{24\lambda^3 + 6\lambda^2\mu + 2\lambda\mu^2 + \mu^3}{96\lambda^4} + \frac{6\lambda^2 + 2\lambda\mu + \mu^2}{24\lambda^3} + \frac{2\lambda + \mu}{6\lambda^2} + \frac{1}{2\lambda}$$

$$MTBF = \frac{128\lambda^3 + 30\lambda^2\mu + 6\lambda\mu^2 + \mu^3}{96\lambda^4} \quad \text{For Cold BOX SS. Arrangement}$$

ATTACHMENT TO FIGURE B-9: CONTINUED

C. DEVELOPMENT OF MATH MODEL FOR COMPRESSOR ARRANGEMENT

Using the same assumptions set forth in PART I B for the switchover subsystem, the compressor subsystems arrangement will be in one of the following states:

State 0 - all elements are operating

State 1 - one element has failed and is being repaired. Three elements are operating.

State 2 - two elements have failed. One is being repaired; one is awaiting repair. The switch over subsystem has scheduled the remaining two elements as necessary to maintain active refrigeration to two cable segments on one circuit.

State 3 - three elements have failed. One is being repaired; two are awaiting repair. One element is operational. Two segments of one cable circuit and one segment of the other cable circuit are without active refrigeration.

State 4* - four elements have failed. One is being repaired; three are awaiting repair. No elements are operational. All four cable circuit segments are without active refrigeration.

* State 4 is not considered in the MTBF calculation since system failure is considered to have occurred when State 3 is reached.

ATTACHMENT TO FIGURE B-9: CONTINUED

The transition matrices are:

$$P = \begin{array}{c|ccccc} & \text{State 0} & \text{State 1} & \text{State 2} & \text{State 3} & \text{State 4} \\ \hline \text{State 0} & 1-4\lambda & 4\lambda & 0 & 0 & 0 \\ \text{State 1} & \mu & 1-(3\lambda+\mu) & 3\lambda & 0 & 0 \\ \text{State 2} & 0 & \mu & 1-(2\lambda+\mu) & 2\lambda & 0 \\ \text{State 3} & 0 & 0 & \mu & 1-(\lambda+\mu) & \lambda \\ \text{State 4} & & & & \mu & 1-\mu \end{array}$$

$$Q = \begin{array}{c|ccc} & \text{State 0} & \text{State 1} & \text{State 2} \\ \hline & 1-4\lambda & 4\lambda & 0 \\ & \mu & 1-(3\lambda+\mu) & 3\lambda \\ & 0 & \mu & 1-(2\lambda+\mu) \end{array}$$

$$I-Q = \begin{array}{c|ccc} & \text{State 0} & \text{State 1} & \text{State 2} \\ \hline & 4\lambda & -4\lambda & 0 \\ & -\mu & 3\lambda+\mu & -3\lambda \\ & 0 & -\mu & 2\lambda+\mu \end{array}$$

$$(I-Q)^{-1} = \begin{array}{c|ccc} & \text{State 0} & \text{State 1} & \text{State 2} \\ \hline & \frac{6\lambda^2+2\lambda\mu+\mu^2}{24\lambda^3} & \frac{2\lambda+\mu}{6\lambda^2} & \frac{1}{2\lambda} \\ & \frac{2\lambda\mu+\mu^2}{24\lambda^3} & \frac{2\lambda+\mu}{6\lambda^2} & \frac{1}{2\lambda} \\ & \frac{\mu^2}{24\lambda^3} & \frac{\mu}{6\lambda^2} & \frac{1}{2\lambda} \end{array}$$

ATTACHMENT TO FIGURE B-9: CONTINUED

$$MTBF = \frac{6\lambda^2 + 2\lambda N + N^2}{24\lambda^3} + \frac{2\lambda + N}{6\lambda^2} + \frac{1}{2\lambda}$$

$$MTBF = \frac{26\lambda^2 + 6\lambda N + N^2}{24\lambda^3} \quad \text{For the Compressor S.S. Arrangement}$$

D. MATH MODEL FOR ASSESSING CABLE SYSTEM FAILURE RATE OF A REFRIGERATOR STATION

$$\lambda_{RS_{SYS}} = \left[\underset{\text{From Part IIB}}{MTBF_{\text{COLD BOX ARR}}} \right]^{-1} + \left[\underset{\text{From Part IC}}{MTBF_{\text{COMP ARR}}} \right]^{-1} + \lambda_{\text{SWITCH}}$$

FIGURE B-10: ASSESSMENT OF OPTIMIZED BASELINE CABLE CIRCUIT REFRIGERATION SYSTEM MTBF

I. RELIABILITY LOGIC DIAGRAM AND MATH MODEL

A CABLE CIRCUIT SEGMENT WILL BE WITHOUT ACTIVE REFRIGERATION IF ANY ONE OF THE ABOVE ITEMS FAIL.

$$MTBF = [6\lambda_{RS_CKT} + 24\lambda_{EXP}]^{-1}$$

AND FROM FIGURE 13 PART IC

$$MTBF = \left\{ 6 \left[\left(\frac{8\lambda_{CB} + \mu_{CB}}{16\lambda_{CB}} \right) + 4\lambda_{COMP} + \lambda_{SW} \right] + 24\lambda_{EXP} \right\}^{-1}$$

II. SUBSYSTEM FAILURE RATES AND REPAIR RATES

Assume that a "fix as soon as practical" corrective maintenance policy is used. Therefore the failure and repair rates will be taken from Table 5-4.

	All Failures		Failures Requiring > 8 Hours For Repair	
	$\lambda \times 10^6$	$\mu = 1/MTTR$	$\lambda \times 10^6$	$\mu = 1/MTTR$
Comp S.S. (λ_{comp})	149	--	0.98	--
Cold Box S.S. (λ_{CB}, μ_{CB})	104.44	0.068	15.96	0.015
Remote Expander S.S. (λ_{EXP})	87.14	--	5.81	--
Switch-Over S.S. (λ_{SW}) *	0.5	--	--	--

* Reliability allocation of λ_{SW} . It is assumed that this S.S. can be repaired in < 8 hours and that it is mechanized by manual shutoff valves.

ATTACHMENT TO FIGURE B-10

III CALCULATION OF MTBF₁, MEAN TIME BETWEEN SHUTDOWN OF A CABLE CIRCUIT BECAUSE OF A LOSS OF ACTIVE REFRIGERATION FOR MORE THAN EIGHT HOURS IN ANY OF ITS SEGMENTS

MTBF₁ = 2 x MTBF computed from model given in PART I.

$$MTBF_1 = 2 \left\{ 6 \left[\left(\frac{8(15.96/10^6) + .015}{16(15.96/10^6)^2} \right)^{-1} + 4(0.98/10^6) \right] + 24(5.81/10^6) \right\}^{-1}$$

$$MTBF_1 = 2 \left\{ 6 \left[(2.694/10^7) + 3.92/10^6 \right] + 139.44/10^6 \right\}^{-1}$$

$$MTBF_1 = 2 (164.576/10^6)^{-1} = 2(6076) = 12,152 \text{ Hours}$$

IV CALCULATION OF MTBF₂, MEAN TIME BETWEEN LOSS OF ACTIVE REFRIGERATION IN ANY SEGMENT OF A CABLE CIRCUIT.

MTBF₂ = 2 x MTBF computed from model given in PART I

$$MTBF_2 = 2 \left\{ 6 \left[\left(\frac{8(104.44/10^6) + .068}{16(104.44/10^6)^2} \right)^{-1} + 4(149/10^6) + 0.5/10^6 \right] + 24(82.14/10^6) \right\}^{-1}$$

$$MTBF_2 = 2 \left\{ 6 \left[2.535/10^6 + 596/10^6 + 0.5/10^6 \right] + 1971.36/10^6 \right\}^{-1}$$

$$MTBF_2 = 2 \{ 5565.57/10^6 \}^{-1} = 359 \text{ Hours}$$

FIGURE B-II. ASSESSMENT OF OPTIMIZED BASELINE CABLE SYSTEM REFRIGERATION SYSTEM MTBF

I. RELIABILITY LOGIC DIAGRAM AND MATH MODEL

At least one of the two remote expander paths must be functional. All of the six refrigerator stations must be scheduled to provide active refrigeration for this path. This diagram can be further simplified.

ATTACHMENT TO FIGURE B-11

The model for $\lambda_{RS_{\text{sys}}}$ is derived in Figure B-9 PART II D

$$\lambda_{RS_{\text{sys}}} = \left[\text{MTBF}_{\text{COLD BOY ARR}} \right]^{-1} + \left[\text{MTBF}_{\text{COMP ARR}} \right]^{-1} + \lambda_{\text{sw}}$$

$$\lambda_{RS_{\text{sys}}} = \left[\frac{128 \lambda_{\text{cb}}^3 + 30 \lambda_{\text{cb}}^2 \mu_{\text{cb}} + 6 \lambda_{\text{cb}} \mu_{\text{cb}}^2 + \mu_{\text{cb}}^3}{96 \lambda_{\text{cb}}^4} \right]^{-1} + \left[\frac{26 \lambda_{\text{comp}}^2 + 6 \lambda_{\text{comp}} \mu_{\text{comp}} + \mu_{\text{comp}}^2}{24 \lambda_{\text{comp}}^3} \right]^{-1} + \lambda_{\text{sw}}$$

The model for the failure rate of the expander paths is derived in Figure B-3 where

$$\lambda_{\text{exp paths}} = \left[\frac{3\lambda + \mu}{27^2} \right]^{-1} = \left[\frac{3(12\lambda_{\text{exp}}) + \mu_{\text{exp}}}{2(12\lambda_{\text{exp}})^2} \right]^{-1}$$

Therefore the model to be evaluated is

$$\text{MTBF}_{\text{CABLE SYSTEM}} = \left\{ 6 \left[\frac{128 \lambda_1^3 + 30 \lambda_1^2 \mu_1 + 6 \lambda_1 \mu_1^2 + \mu_1^3}{96 \lambda_1^4} \right]^{-1} + 6 \left[\frac{26 \lambda_2^2 + 6 \lambda_2 \mu_2 + \mu_2^2}{24 \lambda_2^3} \right]^{-1} + 6 \lambda_{\text{sw}} + \left[\frac{36 \lambda_3 + \mu_3}{2(12 \lambda_3)^2} \right]^{-1} \right\}^{-1}$$

where the subscripts 1=c.b, 2=comp, 3=exp have been added for compactness.

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ATTACHMENT TO FIGURE B-11: CONTINUED

II SUBSYSTEM FAILURE RATES AND REPAIR RATES

Assume that a "fix as soon as practical" corrective maintenance policy is used. Therefore the failure and repair rates will be taken from Table 5-4

	All Failures		Failures Requiring > 8 Hours for Repair	
	$\lambda \times 10^6$	$\mu = 1/\text{MTR}$	$\lambda \times 10^6$	$\mu = 1/\text{MTR}$
Comp S.S. (λ_1, μ_1)	149	0.260	0.98	0.080
Cold Box SS (λ_2, μ_2)	104.44	0.068	15.96	0.015
Remote Expander SS. (λ_3, μ_3)	82.14	0.135	5.81	0.033
Switch-Over SS (λ_{sw})	0.5	--	--	--

III CALCULATION OF MTBF₃, MEAN TIME BETWEEN SHUTDOWN OF CABLE SYSTEM BECAUSE OF THE SIMULTANEOUS LOSS OF ACTIVE REFRIGERATION FOR MORE THAN EIGHT HOURS IN ONE OR MORE SEGMENTS IN BOTH CABLE CIRCUITS

$$\begin{aligned}
 \text{MTBF}_3 = & \left\{ \frac{6 \left[128 (15.96/10^6)^3 + 30 (15.96/10^6)^2 (0.015) + 6 (15.96/10^6) (0.015)^2 + (0.015)^3 \right]}{96 (15.96/10^6)^4} \right\}^{-1} \\
 & + 6 \left[\frac{26 (0.98/10^6)^2 + 6 (0.98/10^6) (0.08) + (0.08)^2}{24 (0.98/10^6)^3} \right]^{-1} \\
 & + \left[\frac{36 (5.81/10^6) + (0.033)}{2 [12 (5.81/10^6)]^2} \right]^{-1} \Bigg\}^{-1}
 \end{aligned}$$

ATTACHMENT TO FIGURE B-11: CONTINUED

$$MTBF_3 = \left\{ 6(1.8/10^{12}) + 6(3.5/10^{15}) + 2.927/10^7 \right\}^{-1}$$

$$MTBF_3 = (2.927/10^7)^{-1} = 3,416,341 \text{ Hours}$$

IV CALCULATION OF $MTBF_4$, MEAN TIME BETWEEN SIMULTANEOUS LOSS OF ACTIVE REFRIGERATION TO ONE OR MORE CABLE SEGMENTS IN BOTH CABLE CIRCUITS

$$MTBF_4 = \left\{ 6 \left[\frac{128(104.44/10^6)^3 + 30(104.44/10^6)^2(.068) + 6(104.44/10^6)(.068)^2 + (.068)^3}{96(104.44/10^6)^4} \right]^{-1} \right.$$

$$+ 6 \left[\frac{26(149/10^6)^2 + 6(149/10^6)(.26) + (.26)^2}{24(149/10^6)^3} \right]^{-1} + 6(0.5/10^6)$$

$$\left. + \left[\frac{36(82.14/10^6) + .135}{2[12(82.14/10^6)]^2} \right]^{-1} \right\}^{-1}$$

$$MTBF_4 = \left\{ 6(3.6/10^6) + 6(1.1703/10^9) + 3/10^6 + 1.408/10^5 \right\}^{-1}$$

$$MTBF_4 = (17.087/10^6)^{-1} = 58,524 \text{ Hours}$$

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SUBSYSTEM COUNT

- 12 COMPRESSOR S.S.
- 12 PRIMARY COLD BOX S.S.
- 6 STAND-BY COLD BOX S.S.
- 6 INTRA-STATION SWITCHOVER S.S.
- 24 REMOTE EXPANDER S.S.

FIGURE B-12. ASSESSMENT OF DOUBLE-SIZE REFRIGERATOR CONCEPT WITH STATION LEVEL COLD BOX SUB-SYSTEM STAND-BY REDUNDANCY AND SWITCH-OVER

ATTACHMENT TO FIGURE B-12

I CABLE CIRCUIT RELIABILITY

A. RELIABILITY LOGIC DIAGRAM FOR A STATION AND ASSOC. REMOTE EXPANDERS

$$\lambda = \left[\frac{4\lambda_{cb} + \lambda_{cb}}{4\lambda_{cb}^2} \right]^{-1}$$

FROM FIGURE B-4

B. MODEL FOR STATION AND ASSOC EXPANDERS

$$\lambda_{\text{STATION}} = \left[\frac{4\lambda_{cb} + \lambda_{cb}}{4\lambda_{cb}^2} \right]^{-1} + 2\lambda_{\text{comp}} + \lambda_{\text{sw}} + 4\lambda_{\text{ex}}$$

C. SUBSYSTEM FAILURE AND REPAIR RATES

Assuming 1:1 repair rate and failure rate scaling factor in doubling the size of the compressor and cold box subsystems and

ATTACHMENT TO FIGURE B-12: CONTINUED

that a "fix as soon as practical" corrective maintenance policy is used. Then these data can be taken directly from Figure B-10, PART II

	All Failures		Failures Requiring > 8 Hours For Repair	
	$\lambda \times 10^6$	$\mu = 1/\text{MTTR}$	$\lambda \times 10^6$	$\mu = 1/\text{MTTR}$
Comp S.S. (λ_{comp})	1.49	--	0.98	--
Cold Box S.S. ($\lambda_{\text{cb}}, \mu_{\text{cb}}$)	104.44	0.068	15.96	0.015
Remote Expander S.S. (λ_{exp})	87.14	--	5.81	--
Switch-Over S.S. (λ_{sw}) *	0.5	--	--	--

* Reliability allocation of λ_{sw} . It is assumed that this S.S. can be repaired in < 8 hours and that it is mechanized by manual shutoff valves.

D. MATH MODEL FOR EVALUATING CABLE CIRCUIT MTBF

$$\text{MTBF} = 2 (6 \lambda_{\text{STATION}})^{-1}$$

E. CALCULATION OF MTBF₁

$$\text{MTBF}_1 = 2 \left\{ 6 \left[\frac{4(15.96/10^6) + .015}{4(15.96/10^6)^2} \right]^{-1} + 2(.98/10^6) + 4(5.81/10^6) \right\}^{-1}$$

$$\text{MTBF}_1 = 2 \left\{ 6 \left[6.76/10^8 + 1.96/10^6 + 23.24/10^6 \right] \right\}^{-1} = 2 (151.606/10^6)^{-1}$$

$$\text{MTBF}_1 = 13,192 \text{ Hours}$$

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ATTACHMENT TO FIGURE B-12: CONTINUED

F. CALCULATION OF MTBF₂

$$MTBF_2 = 2 \left\{ 6 \left\{ \left[\frac{4(104.44/10^6) + .068}{4(104.44/10^6)^2} \right]^{-1} + 2(149/10^6) + 0.5/10^6 + 4(82.14/10^6) \right\} \right\}^{-1}$$

$$MTBF_2 = 2 \left\{ 6 \left[6.38/10^7 + 298/10^6 + 0.5/10^6 + 328.56/10^6 \right] \right\}^{-1}$$

$$MTBF_2 = 2 \left[6(627.698/10^6) \right]^{-1} = 2(3766.188/10^6)^{-1}$$

$$MTBF_2 = 531 \text{ Hours}$$

ATTACHMENT TO FIGURE B-12: CONTINUED

II CABLE SYSTEM RELIABILITY

A. RELIABILITY LOGIC BLOCK DIAGRAM FOR A SUB-STATION

$$\lambda^* = \left[\frac{8\lambda_{cb}^2 + 3\lambda_{cb}\lambda_{cs} + \lambda_{cs}^2}{4\lambda^3} \right]^{-1}$$

* Model Development: Config can be in 4 states: 2 op; 2 op; 1 failed, 1 op + 2 failed, 3 failed.

$$\therefore P = \begin{bmatrix} 1-2\lambda & 2\lambda & 0 & 0 \\ \mu & 1-(2\lambda+\mu) & 2\lambda & 0 \\ 0 & \mu & 1-(\lambda+\mu) & \lambda \\ 0 & 0 & \mu & 1-\mu \end{bmatrix}$$

$$Q = \begin{bmatrix} 1-2\lambda & 2\lambda & 0 \\ \mu & 1-(2\lambda+\mu) & 2\lambda \\ 0 & \mu & 1-(\lambda+\mu) \end{bmatrix}$$

ATTACHMENT TO FIGURE B-12: CONTINUED

$$I-Q = \begin{bmatrix} 2\lambda & -2\lambda & 0 \\ -\mu & 2\lambda + \mu & -2\lambda \\ 0 & -\mu & \lambda + \mu \end{bmatrix}$$

$$(I-Q)^{-1} = \begin{bmatrix} \frac{2\lambda^2 + \lambda\mu + \mu^2}{4\lambda^3} & \frac{\lambda + \mu}{2\lambda^2} & \frac{1}{\lambda} \\ \frac{\lambda\mu + \mu^2}{4\lambda^3} & \frac{\lambda + \mu}{2\lambda^2} & \frac{1}{\lambda} \\ \frac{\mu^2}{4\lambda^3} & \frac{\mu}{2\lambda^2} & \frac{1}{\lambda} \end{bmatrix}$$

$$\therefore \lambda = \left[\frac{2\lambda^2 + \lambda\mu + \mu^2}{4\lambda^3} + \frac{\lambda + \mu}{2\lambda^2} + \frac{1}{\lambda} \right]^{-1} = \left[\frac{8\lambda^2 + 3\lambda\mu + \mu^2}{4\lambda^3} \right]^{-1}$$

B. MATH MODEL FOR STATION

$$\lambda_{\text{STATION}} = \left[\frac{8\lambda_1^2 + 3\lambda_1\mu_1 + \mu_1^2}{4\lambda_1^3} \right]^{-1} + \left[\frac{3\lambda_2 + \mu_2}{2\lambda_2^2} \right]^{-1} + \lambda_{\text{sw}}$$

where the subscripts 1 = CB, 2 = comp have been added for compactness

ATTACHMENT TO FIGURE B-12: CONTINUED

C. RELIABILITY LOGIC DIAGRAM FOR CABLE SYSTEM RELIABILITY

At least one of the two remote expander paths must be functional. All of the six refrigerator stations must be scheduled to provide active refrigeration for this path. This diagram can be further simplified.

$$\lambda = \left[\frac{3\lambda_3 + \lambda_3}{2\lambda_3^2} \right]^{-1} \text{ From Figure B-3}$$

ATTACHMENT TO FIGURE B-12: CONTINUED

D. MATH MODEL FOR EVALUATING CABLE SYSTEM
MTBF

$$MTBF = \left[6\lambda_{RS_{SYST}} + \frac{3\lambda_3 + \mu_3}{2\lambda_3^2} \right]^{-1}$$

$$MTBF = \left\{ 6 \left[\left(\frac{8\lambda_1^2 + 3\lambda_1\mu_1 + \mu_1^2}{4\lambda_1^3} \right)^{-1} + \left(\frac{3\lambda_2 + \mu_2}{2\lambda_2^2} \right)^{-1} + \lambda_{sw} \right] + \left(\frac{3(12\lambda_{exp}) + \mu_{exp}}{2(12\lambda_{exp})^2} \right)^{-1} \right\}^{-1}$$

E. SUBSYSTEM FAILURE RATES AND REPAIR RATES

Assuming a 1:1 repair rate and failure rate scaling factor in doubling the size of the compressor and cold box subsystems and that a "fix as soon as practical" corrective maintenance policy is used. Then these data can be taken directly from Figure B-11, PART II

	All Failures		Failures Requiring > 8 Hours for Repair	
	$\lambda \times 10^6$	$\mu = 1/MTTR$	$\lambda \times 10^6$	$\mu = 1/MTTR$
Comp S.S. (λ_2, μ_2)	149	0.260	0.98	0.080
Cold Box SS (λ_1, μ_1)	104.44	0.065	15.96	0.015
Remote Expander SS. (λ_3, μ_3)	82.14	0.135	5.81	0.033
Switch-Over SS (λ_{sw})	0.5	--	--	--

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ATTACHMENT TO FIGURE B-12: CONTINUED

F. CALCULATION OF $MTBF_3$

$$MTBF_3 = \left\{ 6 \left[\frac{8(15.96/10^6)^2 + 3(15.96/10^6)(.015) + (.015)^2}{4(15.96/10^6)^3} \right]^{-1} \right. \\ \left. + \left(\frac{3(.98/10^6) + (.08)}{2(.98/10^6)^2} \right)^{-1} \right. \\ \left. + \left(\frac{36(5.81/10^6) + .033}{2(12(5.81/10^6))^2} \right)^{-1} \right\}^{-1}$$

$$MTBF_3 = \left\{ 6 \left[7/10'' + 2/10'' \right] + 2.927/10^7 \right\}^{-1} = \left\{ 2.932/10^7 \right\}^{-1}$$

$$MTBF_3 = 3,410,176 \text{ Hours}$$

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ATTACHMENT TO FIGURE B-12: CONTINUED

G. CALCULATION OF MTBF₄

$$MTBF_4 = \left\{ 6 \left[\frac{(8(104.44/10^6))^2 + 3(104.44/10^6)(.068) + (.068)^2}{4(104.44/10^6)^3} \right]^{-1} \right.$$

$$+ \left(\frac{3(149/10^6) + (0.26)}{2(149/10^6)^2} \right)^{-1} + 0.5/10^6 \Bigg]$$

$$+ \left(\frac{36(82.14/10^6) + .135}{2[12(82.14/10^6)]^2} \right)^{-1} \Bigg\}^{-1}$$

$$MTBF_4 = \left\{ 6 \left[9.81/10^{10} + 1.705/10^7 + .5/10^6 \right] + 1.408/10^5 \right\}^{-1}$$

$$MTBF_4 = \left\{ 6(6.715/10^7) + 1.408/10^5 \right\}^{-1} = (18.12/10^6)^{-1}$$

$$MTBF_4 = 55,222 \text{ Hours}$$

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