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MASTER

ORGANIC RANKINE KILOWATT ISOTOPE POWER SYSTEM

First Annual Summary Report, August 1, 1975–August 1, 1976

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Work Performed Under Contract No. EX-76-C-16-3071

Sundstrand Energy Systems
Rockford, Illinois

**TECHNICAL INFORMATION CENTER
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FIRST ANNUAL SUMMARY REPORT
ORGANIC RANKINE
KILOWATT ISOTOPE POWER SYSTEM

(Covering the Period 1 August 1975 to 1 August 1976)

by
SUNDSTRAND ENERGY SYSTEMS
ROCKFORD, ILLINOIS

Prepared for the
**ENERGY RESEARCH AND
DEVELOPMENT ADMINISTRATION
UNDER CONTRACT E(49-15)-3071**

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SECTION 1.0 INTRODUCTION

On 1 August 1975 under the Nuclear Research and Applications Division of ERDA Contract E(49-15)-3071, Sundstrand Energy Systems commenced development of a Kilowatt Isotope Power System (KIPS) directed toward satisfying the higher power requirements of satellites of the 1980's. The KIPS is a plutonium oxide fueled organic Rankine cycle turbine power system which will provide design output power in the range of 500 - 2000 W(e) with a minimum of system changes. The current contract covers Phase I of a planned three phase program comprising:

- I Flight system conceptual design and ground demonstration
- II Flight system design and ground qualification
- III Flight system production, acceptance testing and delivery

The principal objectives of Phase I are to: (1) conceptually design the flight system, (2) based on the flight system concept, design and build the ground demonstration system (GDS), (3) conduct performance and endurance testing using electric heaters to simulate the radioisotope heat source, (4) identify and initiate long lead development efforts required to achieve the initial flight qualification hardware availability date of April 1981, and (5) finalize the flight concept design and prepare the program plan for the Phase II effort. Figures 1-1 and 1-2 show schedule details of the three phases.

Figure 1-1 KIPS Phase I Schedule

Figure 1-2 KIPS Program Schedule

This annual summary report covers the work accomplished from 1 August 1975 to 1 August 1976 under the aforementioned contract.

The energy source originally baselined for KIPS was the Low Cost High Performance (LCHP) heat source being developed under contract to ERDA by Teledyne Energy Systems. In late 1975, Sundstrand was directed to utilize the developed Multi Hundred Watt (MHW) heat source on KIPS to assure availability for a 1981 flight qualification date. The MHW heat source is baseline in this report. The early Phase I design work with the LCHP heat source is summarized to present and document all of the results of the effort during this report period.

SECTION 2.0 SUMMARY

As shown in Figure 1-1, the major milestones which occurred in this report period were: (1) the preparation and submittal to ERDA of the program plan, quality plan and reliability plan, (2) the preparation of the flight system conceptual design and (3) the first preliminary design review.

SECTION 3.0 STATEMENT OF WORK AND DESIGN OBJECTIVES

The basis for work under this contract is the Statement of Work (SOW) which is presented herein.

APPENDIX A, STATEMENT OF WORK TO SUNDSTRAND ENERGY SYSTEMS
CONTRACT E(49-15)-3071

KILOWATT ISOTOPE POWER SYSTEM

PHASE I STATEMENT OF WORK

Task 1.0 Program Management

The contractor shall provide program management and supervision which shall include but not be limited to the following:

- a. Program Direction
- b. Contract Administration
- c. Planning and Scheduling
- d. Evaluation of Progress and Performance
- e. Procurement
- h. Liaison with Other Program Participants and Potential User Organizations

The contractor shall prepare a program plan to perform the work described in this work statement. This plan shall require approval by the contracting officer, and any revisions shall require ERDA approval. The contractor shall provide program management services throughout the period of the contract.

Task 2.0 Flight System Conceptual Design

The contractor shall perform a conceptual design study of a plutonium fueled flight system meeting the objectives listed in Table I System Performance Objectives. The design shall consider power, weight, envelope, cost, efficiency, operation in zero g, safety, reliability and quality for a system to support the delivery of a flight power system in

April 1981. Of particular importance is the preparation of a FMEA and identification of long-lead development items required in pursuit of Task 9 to meet long-term objectives of flight hardware delivery in March 1981. The heat source technology being developed by Teledyne Energy Systems, Timonium, Maryland, under ERDA Contract E(49-15)-3075 (formerly SNSO-3) should be given primary consideration. The MHW heat source technology may be considered as an alternate. The applicable heat source specification is Appendix A 2.3, "Safety Heat Source Design Specification for Space Applications," Rev. 1, July 9, 1974. The technical basis shall be documented and reported in a Flight System Conceptual Design Report.

TABLE I SYSTEM PERFORMANCE OBJECTIVES

Power

1. The basic system should be capable of any power output from 500 to 2000 watt (EOM) with minimum redesign and requalification. Optimization studies will be done at 1300 Watt(e) output.
2. 28 volt DC $\pm$ 2% is the reference output voltage. A capability for higher voltage AC output may be advantageous if it can be achieved directly from the generator.
3. Power delivered to the spacecraft shall be variable from 0 to 100% of the power system output with a response time of the order of milliseconds. This may be accomplished by providing a parasitic load.
4. Partial power during launch to parking orbit is desirable but not mandatory. Partial power during orbital transfer to final orbit or flight path is even more desirable. Ability to withstand spin stabilized orbital transfer is desirable but not mandatory.
5. Most potential missions for the power system will be on non-spinning spacecraft but the ability to operate throughout lifetime on a spin-stabilized spacecraft would be advantageous.

Lifetime

Lifetime requirements $L = 7$ years.

MMD (mean mission duration) = 5 years where:

$$MMD = \int_0^L R dt$$

and $R =$ probability of producing design power at time t

Failure modes in the prelaunch, launch and operation phases should be identified.

Weight and Cost

1. Cost objective for recurring costs at 1300 watt(e) output is less than \$5000 per watt(e) assuming a cost of \$1000 per watt(t) for the MHW-type heat source, and \$800 per watt(t) for the 800°C heat source.
2. Weight objective at 1300 watt(e) output is equal or less than 450 lbs.

Interface Requirements

1. Capable of launch on Titan IIIC or the Space Shuttle.
2. On-pad temperature limit of exposed surfaces based on explosive vapor environment is 380°F.
3. The sink temperature is that of geosynchronous orbit. The design must be capable of stable operation with unbalanced solar input.
4. Thermal input to the spacecraft must be controlled. In some cases there may be advantages to the spacecraft from a radiator that operates in the vicinity of 100°F.
5. Magnetic field at 1 meter shall be (TBD).
6. On-pad biological dose at 1 meter is ≤ 0.20 REM per hour.
7. The power system shall be resistant to natural and manmade radiation.
8. Design flexibility in the size and shape of the power system is desirable. This might include the ability to adapt the radiator to the launch vehicle shroud shape, to utilize radiator interior space for other components, or otherwise integrate the power system with specific spacecraft requirements.
9. On-pad and in-orbit fueling capability should be provided.

Task 3.0 Preliminary Ground Demonstration System (GDS) Design and Planning

3.1 Preliminary GDS Design

The contractor shall identify, develop, and recommend the design

approach to be used for the ground demonstration system (GDS). The contractor shall draw upon the Flight System Conceptual Design as a basis for the GDS design to the extent that so doing increases the effectiveness of the GDS in simulating a flight system without incurring substantial penalties in program duration or cost. The design effort shall include failure mode analysis and definition of reliability objectives.

The contractor shall perform trade studies leading to selection of an optimum control system for the flight design. The contractor shall perform system design and component development for subsequent improvements to the initial flight concept and the GDS to increase efficiency and reduce weight. These improvements are to be determined acceptable by the end of Phase I with the growth versions proven prior to power system operational use in the 1980's.

The contractor shall perform layout studies of packaging alternatives for the GDS which permits preliminary design of the arrangement of the major components, structural requirements and electrical and mechanical interconnections. The contractor shall prepare concept drawings describing the recommended design approach to the entire GDS including control and instrumentation system, and identify the required facilities. This Preliminary Design will be submitted for ERDA approval in a design review meeting.

3.2 GDS Test Plan

The contractor shall prepare a GDS test plan defining detailed objectives, the functional requirements for the test facility, power system components, controls and instrumentation, and special test equipment and the data to be obtained. Performance and endurance testing of the GDS shall be performed in a vacuum environment, if germane. It is suggested that the endurance test be approximately 2000 hours duration. However, the contractor shall recommend the type and duration of the endurance test based on the basic objectives of substantiating the FMEA, and determining the life-limiting components and the potential system lifetime. The test plan will identify and provide whatever demonstration test that may be feasible on earth and desirable to confirm the ability of the power system to function in

a zero g environment. The test plans will be approved by the ERDA prior to their implementation in Task 6. The test plan will be updated and revised as required to accommodate program changes and contributions from other task activities.

Task 4.0 Detailed Analysis and Design of the GDS

4.1 Layout

The contractor shall determine by analysis the size and arrangement of interconnecting components of the GDS to minimize heat loss and other performance-degrading effects consistent with the conceptual design developed in Task 2.0.

4.2 Power Conversion Components

The contractor shall prepare detailed designs of all the major elements of the power conversion system including component specifications, manufacturing and assembly drawings, and parts lists. The design will be supported by sufficient transient and steady state performance analyses to assure proper and reliable performance throughout all phases of the component and system lifetime. The analysis shall include FMEA studies to assist in system reliability and design option.

4.3 Heat Exchanger

The contractor shall conduct a trade-off study of heat exchanger designs for regeneration and heat rejection as appropriate to the selected power system concept. Technology status and costs of fabrication and testing along with performance factors shall be considered.

4.4 Control System

The contractor shall prepare the detail design of the GDS controls including circuit schematics, circuit diagrams, detail part lists, assembly drawings and wiring diagrams. High reliability shall be designed into the control system by employing the methods of component derating (voltage, current and temperature) and critical circuit redundancy.

4.5 Structure

The contractor shall perform as necessary the detail design and prepare the manufacturing drawings and specifications for supporting structures, thermal insulation, ducting, piping, bellows, valves, test instrumentation and other components as required in the GDS. A suitable mock-up of the system may be constructed to work out assembly and interface details.

4.6 Materials Stability/Compatibility

The contractor shall prepare and implement a development program plan as required to support the materials selections and fabrication approaches associated with the GDS. This will include measurements of properties and thermal and radiolytic stability of working fluids, lubricants, coatings and structural materials, and include compatibility, joining studies and such other materials investigations deemed necessary. If thermal or radiolytic instabilities exist, the GDS test plan shall be modified to include evaluation of the effect of such instabilities on power system performance.

4.7 Heat Source Assembly

The contractor shall perform design and analysis including detailed component specifications and manufacturing and assembly drawings to characterize and fabricate the heat source assembly to be used in the GDS. This heat source assembly shall consist of an electrical heat source mechanically and thermally simulating an isotope heat source, heat exchange components to provide heat to a working fluid or heat transfer components, auxiliary cooling provisions if appropriate to dissipate the heat in the event the power conversion or normal heat removal system is inoperative for any reason, and emergency cooling provisions to prevent excessive temperatures to the heat source in the event of any postulated system malfunction. The design shall be consistent with isotope heat source technology and with eventual application to a flight system. The contractor shall also include design of the electric heater power supply(s) and system control interfacing.

4.8 Test Facilities

The contractor shall design the test facilities consistent with the GDS design to accomplish the required testing. The design shall include test system support services, GDS interfacing structures, instrumentation hook-ups, readout equipment, and appropriate data acquisition and reduction provisions.

4.9 Design Review

The contractor shall present the completed detail design in a

design review meeting for ERDA approval prior to fabrication. Earlier review of specific components may be requested where long-lead items are involved. The contractor shall issue a topical report covering the final approved design of the GDS.

Task 5.0 Fabrication and Assembly

The contractor shall procure, fabricate, and assemble, based on ERDA approved detail designs, the GDS and test facilities. The completed, assembled, and instrumented test system will be placed into the test facility and the hook-up of all support systems, data acquisition system and controls will be accomplished in accordance with the results of Task 4.8.

Task 6.0 Test and Evaluation

6.1 Performance Test

The contractor shall conduct the approved test program of Task 3.2 to determine the actual performance capability of the system, provide data to correlate with predictions, and identify problem areas requiring further design or development. The GDS should be optimized to operate at 1300 watt(e) output and a complete performance mapping should be performed at that power level. In addition, it is highly desirable that the testing program demonstrate, if possible, the capability of the system technology to provide flexibility of power output in the range 500 to 2000 watt(e) with minimum redesign and reconfiguration. A series of start-up and shutdown tests will be performed to assess the transient characteristics and behavior of the system, if germane. This series of

tests will be performed in sufficient depth to permit comprehensive cycle evaluation and loss analysis. Measurements will be taken at all of the key state points as well as of selected metal and insulation system temperatures. The contractor shall issue a topical report on the results of performance testing within 30 days after completion of testing.

6.2 Endurance Test

The contractor shall conduct an endurance test program in a vacuum environment according to the GDS Test Plan prepared in Task 3.2 at a set of approved state points. The contractor shall provide a lifetime assessment of the potential flight system based on the combined results of component development tests, the performance test and the endurance test.

Task 7.0 Quality, Reliability and Safety

7.1 Quality and Reliability

The contractor shall provide the necessary quality and reliability assurance associated with the design, fabrication, and testing of the GDS. A quality assurance and reliability plan, consistent with the general objectives of the ERDA QA program requirements "SNS-1" (Rev. 1, 2/28/75 and reliability program requirements "SNS-2" dated 6/74, but specifically modified for the GDS with the intent of establishing background data for the subsequent Phase II effort, shall be prepared and submitted for approval by the ERDA prior to implementation.

7.2 Safety

The contractor shall consider the ultimate requirements of nuclear flight safety as it relates to the proposed power conversion system components in the design of the GDS. Packaging arrangements, materials selection and failure modes shall be considered to assure that the design reflects minimum ultimate nuclear risks in a flight system based upon the GDS design.

Task 8.0 Reports and Data

The contractor shall provide monthly top summary reports and financial reports, monthly technical and quality status reports, annual technical reports, and appropriate special reports and documentation associated with safety, design, testing, materials properties specifications and operating procedures.

Task 9.0 Advanced Development

The contractor shall, to the extent he believes necessary, prepare a detailed program for long-lead development tasks, oriented toward the flight system but requiring initiation during Phase I to be compatible with the initial flight hardware delivery objective of March 1981. These development efforts may be proposed in any areas of the system where timely upgrading, testing or evaluations are deemed necessary in advance of the completion of the GDS program. Each development task will require approval by the ERDA prior to implementation.

Task 10.1 Flight Design and Engineering Program Plan

The contractor shall submit recommended design and program plans for the Phase II Flight Development and Qualification Program based upon assessments of findings of previous task activities. The contractor shall consider and make recommendations concerning the following areas

and any other he deems appropriate.

- a. Design to accommodate operation through launch and the nominal space mission
- b. Advanced power conversion system components
- c. Packaging and thermal insulation
- d. Component environmental testing
- e. Safety and reliability assessment
- f. Advanced flight design heat source assembly
- g. Compatibility and stability of materials
- h. Schedule and cost estimates

However, all recommendations for advanced development efforts should be made in consideration of the schedule requirement of completing qualification tests by March 1980 and delivering a flight unit by March 1981.

4.1 FLIGHT SYSTEM CONCEPTUAL DESIGN REQUIREMENTS

The KIPS requirements are represented diagrammatically in Figure 4-1. There are three principal design objectives for the KIPS: Safety, Cost, and Weight, and the design is constrained by environmental factors and the 1981 flight date time factor.

To meet the principal objectives, several considerations occur. These are: 1) the preferable use of the New LCHP Heat Source Technology, 2) a high thermodynamic efficiency of the PCS, 3) capability to easily adapt the KIPS to various power levels and launch vehicles, and 4) a long mission duration requiring high reliability.

The use of the New (LCHP) Heat Source Technology, while desired to minimize system weight and isotope costs, has questionable availability for a 1981 flight. Therefore, the MHW heat source is baseline. KIPS is compatible with either heat source. The required high thermodynamic performance affects the same principal objectives of safety and cost by reducing isotope inventory. The requirements of power level and integration flexibility are intended to minimize overall costs by developing one power conversion system that needs only minimal changes (hence, minimal requalification costs) to meet any power requirement in the 500 to 2000 watt(e) range. The need for high reliability and long life is significant in reducing the overall cost of satisfying a continuous power system requirement such as AFSATCOM by making the time between replacement, with the high costs associated with system replacement, as long as possible.

This design must also be sensitive to the constraints placed on the system by the various environments, i.e., Launch Pad, During Launch, In Orbit, and Shuttle Launch, as well as the 1981 flight date which requires well developed technology (not using the LCHP heat source nor a topping cycle at this time reflects this logic; however, this growth capability exists for KIPS to improve safety and reduce cost in the future).

Following are a number of considerations that may affect the ultimate flight system design configuration that is not defined at this time; such as:

- General integration with the spacecraft
- Orientations and magnitudes of angular momentum vectors desired for minimal impact on spacecraft attitude control requirements
- Waste heat radiator design to reject heat at a low temperature (100°F) to enable the radiator surfaces to be used as stable temperature mounting surfaces for electronics and other payload devices in a fully integrated PCS/spacecraft approach

Also, in the absence of a defined mission, several mission scenarios were considered. These involve two different launch vehicles (Titan IIIC and Space Shuttle), and two different types of orbit transfer vehicles, one fully stabilized and the other spin-stabilized during the orbit transfer maneuvers. They also involve two different modes of operation, one where the system is loaded with fuel and started on the launch pad, and the other where it is loaded and started in orbit.

Since the KIPS is to be flexible with respect to vehicle integration, the FSCD is configured to be compatible with any combination of launch vehicle or orbit transfer vehicle with either a ground start or orbit load and start.

Figure 4-1 KIPS Requirements

Since the details of the spacecraft and operational modes are unknown at this time, a detailed integration of the KIPS and spacecraft cannot be made. The basic components which comprise the KIPS are designed to operate with the high acceleration loads and vibration environments that will occur during launch on the Titan IIIC and during transfer orbit maneuver on a spinning spacecraft. It has been estimated that with probable packing arrangements on a spin-stabilized spacecraft during orbit transfer, the KIPS rotating equipment and jet condenser could experience axial accelerations of ~ 9 "g"s and lateral accelerations of ~ 5 "g"s. These two vehicles have approximately the same maximum "g" levels which are much more severe than seen in the other types of launch and orbit transfer vehicles. Since the KIPS is designed to operate in these high "g" fields as well as in zero "g", it will operate satisfactorily with all types of vehicles considered feasible for the potential applications.

4.2 SYSTEM OPERATION

The PCS configuration selected to provide the necessary flexibility for the various modes of start up and operation required for different missions is shown in block diagram form in Figure 4-2. The major components are the three heat source/boiler modules, the power conversion rotating machinery, regenerator, jet condenser, and the radiator.

During operation of the system, an accumulator maintains system reference pressure of 36 psia at the pump inlet. The turbo-pump boosts this pressure to 90 psia and directs the liquid working fluid (Dowtherm-A) through two parallel loops, the cooling loop, and the power loop.

Eighty percent of the total pump flow passes through the cooling loop. Waste heat rejected by the radiator is controlled by means of a thermal sensor and radiator bypass valve which maintains radiator outlet temperature at $170 \pm 5^\circ\text{F}$. Cool liquid from the radiator passes through a filter which will trap any meteoroid generated spallation particles from the radiator tubing. This cool liquid then passes to the jet condenser liquid injector.

The remaining 20% of the pump outlet flow passes through a filter to a fluidic flow control valve. The fluidic valve diverts approximately half of this flow through a hydraulic resistance element back to the pump inlet while the other half passes through the alternator cooling passages to the regenerator. The liquid passing through the regenerator picks up heat from the turbine exhaust vapor flow. This hot liquid from the regenerator then passes through the boiler where it is preheated, vaporized, and superheated. The superheated vapor then passes through a centrifugal separator (simple swirl chamber) which prevents any non-volatile matter from entering the turbine nozzles. A temperature sensor element of the fluidic flow control valve senses turbine inlet temperature and allows the valve to adjust the flow to keep turbine inlet temperature constant.

Mechanical power is generated by the turbine as the high pressure superheated vapor is expanded to low pressure. This turbine power supplies the alternator, feed pump, and bearing shaft power requirements.

The low pressure, highly superheated turbine exhaust vapor then enters the vapor side of the regenerator where it rejects heat to the incoming liquid. The slightly superheated vapor leaving the regenerator enters the jet condenser vapor funnel and is condensed on the subcooled liquid streams which are being injected at high velocity into the low pressure vapor funnel. These high velocity liquid streams, which have condensed the low pressure vapor, pass through a throat at the apex of the vapor funnel into a diffuser where sufficient pressure is recovered allowing the flow to enter the accumulator where it is filtered and supplied to the pump.

Figure 4-2 Flight System Block Diagram Without Support Modules

The fluid film bearings are lubricated with liquid working fluid taken from the pump inlet. The bearing drains vent into the high velocity vapor stream near the liquid jet injector in the jet condenser. The bearing liquid flow rate is controlled by high pressure drop viscojet orifices.

The alternator electrical output is rectified to DC. A voltage regulator controls the alternator field to produce constant 28 volt DC power.

Although the basic KIPS-PCS as described is capable of operating satisfactorily in a speed wild load-matching mode, a constant speed mode of operation has been baselined since the changing angular momentum associated with a variable speed operating mode may have some negative impact on a 3 axis stabilized spacecraft attitude control system. While the potential impact is not known with certainty because the spacecraft is not defined, the constant speed mode of operation can be deleted in favor of speed wild with a subsequent PCS weight savings if later found acceptable.

Constant speed control is provided by an electronic speed control system which senses alternator frequency and applies electrical load to a dissipative load bank to maintain constant turbo-alternator speed at all customer electrical load demands. If the speed control system is switched off, CRU speed will increase as load is decreased. At no load, CRU speed will be approximately twice rated speed. At high CRU speed, the pump output pressure increases to provide higher velocity fluid streams in the jet condenser. This makes the liquid streams less susceptible to deflection due to vibrations and high "g" levels normal to the jets which can occur during launch and orbit transfer maneuver. The fluid film bearings are capable of withstanding considerably higher loads at these higher rotative speeds. Thus, by defeating the speed control system during launch and orbital transfer, the PCS can operate through periods of high shock and vibration loads and yet have minimal bearing losses during normal operation in space.

The turbine inlet temperature control continually adjusts turbine loop flow rate to compensate for isotope decay, changes in radiator outlet temperature, and changes in applied load when the speed control system is switched off for launch. During the system start sequence, the turbine inlet will be cold so the temperature sensor will adjust the turbine loop flow rate to its minimum setting to allow more rapid system warm up.

4.3 SYSTEM START UP MODES

A more detailed FSCD block diagram is shown in Figure 4-3. This shows three locations where support equipment modules can be attached to the PCS. These support modules provide system start up, heater temperature control, and heat rejection control. The type of support equipment required depends on the launch mode and launch vehicle. None of the support modules are required after the system is placed in orbit.

Reliability of the system is enhanced and isotope handling is minimized if the entire system is loaded and started on the ground. Accordingly, the baseline approach is to start the system on the ground, verify proper operation of the system, and then launch the fueled and running system with the minimum amount of support equipment.

When the system is launched into orbit using the Space Shuttle as the launch vehicle, the PCS is located in the Space Shuttle bay and attached to the Space Shuttle Freon coolant supply. Freon coolant is circulated through a cooling coil on the radiator loop to remove system waste heat and through the heater temperature control support module. This module supplies liquid Dowtherm to auxiliary cooling tubes in the boiler modules. The liquid Dowtherm is, in turn, cooled by Space Shuttle-supplied Freon.

Figure 4-3 Flight System Block Diagram

After the cooling lines are attached, the isotope can be loaded into the boiler modules and access doors closed. Boiler tubing is permitted to rise to working temperature but is prevented from exceeding this temperature by the temperature control module.

The starting system is then attached and working fluid pumped into the system to establish the jet condenser liquid jet flow and begin circulating fluid through the boiler loop.

After the system has warmed up and is running at rated speed and load, the start module is detached and removed. The heater temperature control module is shut off but left attached to provide emergency boiler cooling if required.

To accommodate high launch shock and vibration loads, the electrical load bank/speed control system is switched off, permitting CRU speed to rise. After launch and orbital insertion, the speed control system is switched on and the PCS is operating in its normal mode.

When the Titan IIIC launch vehicle is used and no cooling fluid is available after the launch, the PCS is installed in the launch vehicle and started on the ground. A ground cooling loop is used to subcool the fluid entering the radiator.

The radiator bypass valve opens to mix warm fluid with the subcooled liquid from the radiator to maintain a jet condenser liquid inlet temperature of 165°F. The cold radiator can then act as a heat sink for system heat rejection during the launch phase when the radiator shroud prevents heat rejection to the atmosphere.

After launch, the protective radiator shrouds are ejected, permitting the radiator to resume its heat rejection function. In this mode, none of the support modules are launched with the PCS.

It may be desirable to transport a non-fueled, non-running system into space. This situation could occur if a PCS system fails in space and it is deemed desirable to salvage the fuel modules and install them in another PCS.

In this case, the start support equipment module is transported in the Shuttle bay with the new PCS and used to start the system after the isotope has been installed. Space Shuttle coolant is used for heater temperature control during start up. If the Shuttle bay doors are open during the fueling and start up interval, the radiator can reject heat to space and the auxiliary heat rejection cooling loop will not be required.

The support module attachment points can also be used for ground fueling and ground checkout of the PCS. The heater temperature control and overtemperature safety loop can use facility nitrogen instead of a liquid loop to simplify the support system if desired.

The start system is being designed as a flyable system since it may be needed for space starts of a PCS. A less expensive, heavier, non-flyable version can be used for system checkout and all launch modes requiring ground starts.

During ground operation with the isotope heat source, the PCS output power capability is reduced somewhat due to higher insulation heat losses. Full load operation can be ground checked when running the ground checkout run with electric heaters since the heaters can be designed with excess heat output to balance the higher insulation losses that occur during ground operation. An alternative is to ground checkout in a vacuum chamber.

Upon completion of the system checkout, the electric heaters can be sequentially replaced with isotope heat modules while the system is operating.

SECTION 5.0 FLIGHT SYSTEM CONCEPTUAL DESIGN

The KIPS baseline Flight System Conceptual Design (FSCD) is illustrated in Figure 5-1 and 5-2.

The heat source assembly is located symmetrically around the power conversion system and at the base of the radiator assembly which is mounted to the spacecraft bulkhead as shown in Figure 5-1.

The major subsystems are:

- Power Conversion System (PCS)
  - Combined Rotating Unit (CRU) - turbine, alternator, pump and bearings
  - Electrical and fluid controls
  - Regenerator and jet condenser
  - Filters, check valve, orifices
- Heat Source Assembly (HSA)
  - Heat source - MHW (plutonium oxide)
  - Boiler and auxiliary cooler
  - Emergency Heat Dump System (EHDS)
  - Structure/Insulation system
- Radiator Assembly (RA)
  - Radiator
  - Auxiliary cooler
  - Bypass valve

Key features of the KIPS design are a supersonic axial impulse turbine, homopolar inductor alternator, centrifugal pump, working fluid (Dowtherm-A; as used in this report to mean the high purity eutectic of biphenyl and phenyl ether) lubricated fluid film bearings, once-through radiantly heated boiler, jet condenser, and liquid working fluid pumped radiator.

Figure 5-3 is a block diagram of the KIPS and the three support modules. The start module is used to start the power system either for a ground start or a space start. The preferred mode of operation is a ground start. In this case, the start module would be disconnected prior to launch and the KIPS would be operating and producing partial power as required during launch and orbit transfer. The heater temperature control support module is used for cooling during the period between heat source assembly and power system startup. It is also used as emergency cooling for the heat source while the system is in the Space Shuttle. This cooling is supplied by a ground module for the Titan III and the launch vehicle for the Shuttle. The heat rejection support module is used for on the pad cooling prior to launch for Titan III applications and continuously while in the Shuttle bay. These support modules are decoupled after performing their function and are not part of the KIPS/Spacecraft system that is injected into operational orbit.

Figure 5-4 is a schematic of the system illustrating typical operating conditions. The liquid outlet from the pump is split into two loops. The power loop passes liquid through the regenerator to extract energy from the turbine exhaust vapor. At the energy source, additional heat is transferred to the working fluid which is vaporized to drive the turbo- pump alternator. The cooling loop passes

Figure 5-1 KIPS Baseline Flight System

Figure 5-2 KIPS Power Conversion System

(Identical to Figure 4-3 but repeated for ease of review.)

Figure 5-3 Flight System Block Diagram

Figure 5-4 Typical Operating Conditions

liquid flow through the radiator to reject waste heat and then to the jet condenser to condense the vapor emanating from the regenerator. The combined liquid is returned through the accumulator to the pump.

Table 5-A is a summary of performance indicating the overall characteristics of the KIPS. Failure rate is calculated to be 0.0616 failures/year which is lower than the 0.1022 failures/year required by the 7 year life and 5 year mean mission duration. Total wet weight is 448.2 pounds which is within the 450 pound requirement. For 1300 watts of electrical power at beginning of mission, the overall system efficiency of unrectified 30 or 100V AC output is 19.6 percent; including rectification to 28VDC $\pm$ 2% system efficiency is 18.2% (reference Appendix II). Overall dimensions are 48" diameter by 127" long which is the result of optimizing the radiator only as a function of liquid side ΔP and weight. These dimensions can be changed to accommodate booster or spacecraft constraints. The KIPS is designed to be operating and generating power during launch, parking orbit, and orbital transfer to final orbit.

Table 5-B is a summary of efficiency and weight for several alternative system configurations. The baseline KIPS produces 1300 W(e) output (BOM) rectified to 28 VDC, operates at a constant speed by using a parasitic load resistor (PLR), and uses 3 MHW heat source assemblies. Due to the higher component reliability, a fluidic valve is the baseline flow controller. However, a mechanical flow control is being considered as a backup. While having somewhat less component reliability than the fluidic valve, it does give higher system efficiency. Topping with a thermoelectric generator would produce the most significant efficiency gain (reference Appendix III). Other potential future options include: (1) eliminating the PLR at a weight savings if the spacecraft can accommodate the variable angular momentum that may result, (2) using unrectified power, if it is preferable to the spacecraft designer, will result in both a weight savings and significant efficiency gain, and (3) incorporating the LCHP heat source assembly at an 83.8 lb. weight savings (Figure 5-5 illustrates the KIPS using the LCHP-HSA). The following illustrates the weight breakdown at 7200 thermal watts for the baseline KIPS:

	<u>MHW</u>	<u>LCHP</u>	<u>Difference</u>
LCHP; Modules (30 x 2.95 lb.) + Support (17 lb.)		102.2	
			38.2
MHW: Helipac (3 x 42 lb.) + Support (3 x 4.8 lb.)	140.4		
Remaining structure, EHDS, insulation	<u>102.6</u>	<u>57.0</u>	<u>45.6</u>
Total	243 lb.	159.2 lb.	83.8 lb.

The lower weights for the LCHP heat source and support structure are a result of the design flexibility results from freedom to use materials to satisfy KIPS system requirements. Although the basic building block of the MHW heat source, the 100 w fuel sphere assembly, could be arranged to yield a total heat source weight closer to that of the LCHP, the fixed 2400 watt helipack assembly imposes a large penalty.

Tables 5-C and 5-D list the component failure rate data and weight summary respectively for the baseline KIPS. Figures 5-6 and 5-7 illustrate the efficiency summaries with the fluidic and mechanical flow controls at 1300W(e), 2000W(e), and 500W(e) power levels. Figure 5-8 illustrates the electrical loss distribution at 1300W(e) output.

Table 5-A KIPS Characteristics

Rated Output Power	1300W(e) BOM
Optional Ratings	500 to 2000W(e) in ~18W(e) increments
Input Power @ 1300W(e) BOM	7146W(t) ⁽¹⁾
Overall Thermal Efficiency ⁽¹⁾	18.2% rectified to 28VDC 19.6% @ 30 VAC 19.6% @ 100 VAC
Peak Working Fluid Temperature	650°F
Total Weight	431.5 Dry 448.2 Wet
Envelope Dimensions for 1300W(e) BOM ⁽²⁾	48 inches Diameter 127 inches Length
Design Point Output Voltage	28 VDC ± 2%
Response - Variable Output Power - 0 to 100%	Milliseconds
Partial Output Power Capability	On Pad Launch to Parking Orbit Orbital Transfer to Final Orbit
Output Power Capability - 0 to 100%	Spin Stabilized Spacecraft
Reliability at a 7 year Lifetime & 5 year MMD	0.0616 failures/year ⁽³⁾
Capable of Stable Operation with Unbalanced Solar Input	
Resistant to Natural & Induced Radiation	
Capability for On-Pad to In-Orbit Fueling	

(1) For the MHW-HSA thermal incremental loading is limited to 100W(t); therefore, the thermal input power would be 7200W(t) rather than the required 7146W(t). The efficiencies listed are based upon 7146W(t).

(2) Flexible in design for both size and shape to fit the specific spacecraft.

(3) The design objective reliability required by the Statement of Work for the 7-year lifetime and 5-year mean mission duration is ≤ 0.1022 failures/year for constant failure rate.

Table 5-B KIPS Optional Configurations Efficiency and Weight Summary
1300 W(e) BOM Design Point

	<u>30 or 100 VAC Electrical Power</u>	<u>Rectified 28 VDC Electrical Power</u>
EFFICIENCIES (Percent)		
KIPS with Fluidic Flow Control (0.22 failures/million hours) ⁽²⁾	19.6	18.2 ⁽¹⁾
KIPS with Mechanical Flow Control (0.34 failures/million hours) ⁽²⁾	20.2	18.8
KIPS – RTG Topping ($\eta = 7\%$) ⁽³⁾ with Fluidic Flow Control	25.2	23.9
KIPS – RTG Topping ($\eta = 7\%$) ⁽³⁾ with Mechanical Flow Control	25.8	24.5
WEIGHTS (Pounds)		
KIPS with MHW Heat Source Assembly		
Parasitic Load	436.4	448.2 ⁽¹⁾
Speed Wild	428.2	440.0
KIPS with LCHP Heat Source Assembly		
Parasitic Load	352.6	364.4
Speed Wild	344.4	356.2
 (1) Baseline KIPS		
(2) Estimate Valve Failure Rate		
(3) Selinide RTG at 900°C (1652°F) Hot Side and 400°C (752°F) Cold Side Results in Thermoelectric Thermal Efficiency of 7%		

Figure 5-5 Optional KIPS Flight System with LCHP-HSA

Table 5-C Preliminary Component Failure Rate Data

COMPONENT NAME	FAILURES PER MILLION HOURS		AVERAGE K_F FACTORS			DATA
	OPERATING	START	GROUND	LAUNCH	SPACE	SOURCE
System Accumulator	0.35		8	800	1	1
System Filter (Accumulator)	0.011		8	800	1	1
Check Valve (Accumulator Outlet)	0.112		▲	▲	▲	1
Check Valve (System Pump Outlet)	0.112					1
In Line Filter	0.01					1
Flow Limiter & Temperature Sensor	0.22					3
Regenerator	0.5					4
Boiler	0.5					4
Centrifugal Separator	0.01					4
Combined Rotating Unit (CRU)						
Turbine Assembly	1.0					4
Alternator	0.7					1
System Pump	0.1					4
Jet Condenser	0.5					4
Visco Jets (2)	0.02					1
Radiator Auxiliary Cooler	0.156					1
Radiator	1.0					4
Radiator By-Pass Valve	0.24					1
Radiator Filter	0.011					1
Start Module						
Start Accumulator	--	0.35				1
Start Valve (Solenoid)	--	1.87	▼	▼		1
Start Pump/Motor	--	2.25	8	800		1
Disconnects (2)	--	1.12	8	800		1
Inverter	--	0.02	See Note	See Note		2
Electronic Controls						2
Voltage Regulator	0.043					2
Rectifier	0.148					2
Speed Controller	0.051					2
Tuning Capacitor Bank	0.0002				▼	2
Electrical Connectors	0.24				1	2
Parasitic Load Resistors (PLR)	1.0		▼	▼	1	4
TOTAL	7.034					

Data Sources: (1) AVCO Corp., "Reliability Engineering Data Series - Failure Rates"

(2) MIL-HDBK-217B

(3) Harry Diamond Laboratories

(4) Engineering Estimate

Note: K_F factor for ground and launch phases vary with electronic device type.

Table 5-D KIPS Weight Summary

<u>Component</u>	<u>Weight (lb.)</u>
Load Radiator*	6.0
Radiator*	80.6
Isotopes*	126.0
Heat Box Assembly*	117.0
Alternator*	18.9
Alternator Housing*	8.1
Turbine Input Housing	4.6
Turbine Exhaust Housing	3.0
Turbine Wheel Assembly	3.7
Regenerator Tubes*	14.0
Regenerator Inner Housing	2.7
Regenerator Outer Housing	3.2
Center Housing	4.3
Alternator End Housing	2.5
Jet Condenser*	3.4
Accumulator	2.3
Electronics*	8.1
Electrical Harness*	6.0
Valves & Orifices	3.3
Centrifugal Separator	0.5
Tubes & Fittings	4.0
Shock Mounts	1.2
Support Struts	3.1
Insulation Blanket	1.5
Filters & Miscellaneous	3.5
 Weight of Unit (Dry)	 431.5
Weight of Dowtherm-A*	16.7
 TOTAL UNIT WEIGHT (Wet)	 448.2
* — Calculated; others estimated	

Figure 5-6 KIPS Efficiency Summary (Fluidic Flow Control)

Figure 5-7 KIPS Efficiency Summary (Mechanical Flow Control)

Figure 5-8 Electrical Loss Distribution (1300 Watts)

SECTION 6.0 FIRST PRELIMINARY DESIGN REVIEW

On 25, 26 and 27 May 1976 the first preliminary design review was held at Sundstrand, Rockford, Illinois. The items covered in this review were:

- Combined Rotating Unit
- Jet Condenser/Accumulator

A presentation and discussion of the Flight System Conceptual Design were also made at this review.

6.1 COMBINED ROTATING UNIT

Turbine

The turbine design includes the nozzles, the turbine blades and wheel, and the diffuser between the turbine discharge and the entrance to the regenerator.

The turbine drives the alternator and feed pump which includes the mechanical, electrical, and friction power losses in the bearings and alternator.

The vapor discharged from the exit of the turbine is diffused to a low Mach number by an annular diffuser. The diffuser exit then discharges the gas vapor into a larger annulus which is the inlet to the regenerator.

The turbine speed selected is 33,683 rpm for nominal total inlet conditions of 57.0 psia and 650°F for superheated Dowtherm A vapor. The inlet total enthalpy is 396.266 BTU/lb for these conditions. The total to static pressure ratio and pitch diameter were specified as 558.3 and 5.95 inches respectively. The ratio of the rotor speed (based on pitch diameter) to the spouting velocity v_p/C_0 is 0.447 based on performance studies for maximum efficiency of an impulse rotor design.

The large pressure ratio results in supersonic Mach numbers at the nozzle exit and supersonic relative Mach numbers at the inlet to the rotor blading. A single-stage, partial admission, axial impulse turbine is designed to satisfy the power requirements of the KIPS power conversion system. The 1.3 KW flight condition is the design condition. The aerodynamic design for the Ground Demonstration System (GDS) is the same as that used for the Flight System Conceptual Design (FSCD).

Table 6-A illustrates the turbine nozzle design.

The rotor blade profile is shown in Figure 6-1 based on the pitch line analysis. The hollow blade is being considered to minimize the mass moment of inertia of the turbine wheel. The 0.6 inch axial chord is used to enhance the transition to turbulent flow and minimize the possibility of separation on the blade suction surface in the vicinity of the trailing edge. The Reynolds number based on the exit conditions and axial chord is 8732. The geometry of the turbine blades and the wheel are summarized in Table 6-B. A constant blade section is used at all radii between hub and tip of the rotor. The hub and fin diameters were determined by prescribing a 5% rotor lap of the hub and a 15% rotor lap at the tip for a total rotor lap of 20%. The rotor lap in percent refers to percent of nozzle height. A 0.020 inch shroud is attached to the blade tip with two 0.030 inch lugs that extend

Table 6-A Axisymmetric Nozzle Geometry (Sized for 1.3 KW)

<u>QUANTITY</u>	<u>0.5 KW</u>	<u>1.3 KW</u>	<u>2.0 KW</u>
Nozzle Throat Area (Sq. In.)	.007547	.015093	.02264
Throat Diameter (In.)	.056594	.056594	.056594
Nozzle Exit Area (Sq. In.)	0.69197	1.38394	2.07590
Nozzle Exit Dia. (In.)	0.54192	0.54192	0.54192
Number of Nozzles	3	6	9
Nozzle Angle (Deg.)	15.0	15.0	15.0
Nozzle Total Pressure (psia)	53.2	57.0	61.0
Mass Flow (lbm/sec)	.0140	.030	.048
Arc of Admission	.320	.640	.960

Figure 6-1 KIPS Turbine Blade Profile

Table 6-B KIPS Turbine Blade and Wheel Design

Blade chord	0.5995 inch
Blade height	0.650 inch
Blade solidity	3.24
Number of Blades	101
Blade blockage at pitch line	11.17%
Inlet metal angle	27.352 deg
Exit metal angle	31.790 deg
Minimum blade thickness	.0095 inch
Starting margin	5.52 deg
Blade area	0.0465 sq. in.
Hub diameter	5.354
Pitch diameter	5.95 inch
Tip diameter (not including shroud)	6.654 inch

into honeycomb material above the shroud. This construction reduces gas leakage and allows for wheel growth due to thermal and centrifugal expansion.

Turbine Off-Design Performance: The turbine must produce the necessary output power to provide available power levels from 0.5KW to 2.0KW. An off-design analysis was used to obtain the expected turbine performance. The inlet Dowtherm A vapor temperature was maintained at a constant value of 650°F. The exit or exhaust static pressure was also held constant at 0.1021 psia. The inlet total pressure and the number of nozzles were allowed to vary to provide the necessary vapor mass flow rate for a given power requirement. The variation of predicted turbine efficiency with velocity ratio is shown in Figure 6-2.

Diffuser: The diffuser has an exit area to inlet area of 2.2 and a length to inlet height of 5.8. The diffuser design was based on the experimentally determined optimum geometries presented by Sovran and Klomp and turbomachine diffuser experience. A sketch of data is shown in Table 6-C. Flow distribution to the regenerator has been maintained by providing sufficient distance between diffuser exit and regenerator core.

Turbine Mechanical Design: A layout drawing of the turbine portion of the Combined Rotating Unit (CRU) is shown in Figure 6-3. The rotors of the turbine and alternator are supported radially and axially by Dowtherm A fluid lubricated, hydrodynamic, tilting pad bearing.

Figure 6-2 Turbine Performance at Design Speed

Table 6-C Annular Diffuser

Pressure	0.1286 psia
Temperature	524.5°F
Enthalpy	341.46 Btu/lb
Specific Volume	495.6 ft ³ /lb
Absolute Velocity	123.7 ft/sec
Mach Number	0.224
Total Pressure	0.132 psia
Total Temperature	525.17°F
Total Enthalpy	341.77 Btu/lb
Pressure Coefficient	0.569

Figure 6-3 Turbine Layout

The turbine wheel will be made from Inconel 718. The thermal and dynamic tip expansion is 0.015 and 0.022 inches at rotor speeds of 33783 and 60000 rpm. The stresses are below design limits. The mass moment of inertia of the wheel disk is 0.0485 in-lb-sec². A thin metallic sheet is placed inside the turbine housing to reduce heat transfer from the vapor between the wheel and housing. This vapor is heated by disk friction dissipation.

The turbine shroud is made of hastelloy X and brazed to the turbine wheel. The shroud reduces tip leakage as well as prevents fatigue cracking at the root (hub) of the blades. Two integral annular rings, one at each end of the shroud, are used as a labyrinth seal to prevent tip leakage.

Alternator

The alternator used in the KIPS Ground Demonstration System (GDS) is a four pole homopolar inductor alternator. This type of machine was selected for its simple and rugged rotor construction, making it well suited for high speed applications. A homopolar inductor rotor generally contains no electrical windings, cages, magnets, slip rings, etc., and can often be machined from a solid single piece of steel. All electrical windings are contained in the stator and are not subjected to rotational stresses and vibrations.

The KIPS alternator is designed to operate with a rectifier bridge package over an output power range of 500 to 2000 watts at 28 volts DC. Optimum performance is at approximately 1300 watts. Optimum point performance calculations are made with the basic assumption that sufficient capacitance will be connected line-to-line at the alternator terminals to ensure that the power factor at the alternator terminals is not less than 0.93. Alternator performance calculations at loads other than 1300 watts (1.0 per unit) are slightly in error because in all calculations the power factor is assumed to remain constant with load.

Table 6-D shows the calculated effect of improved power factor on alternator output, efficiency and system efficiency.

Table 6-D Alternator - Rectifier System Performance

	<u>Without Cap.</u>	<u>With Cap.</u>
Electrical Output	1300 w.	1300 w.
Alt. KVA	1.977	1.56
Vph.	16.8	12.92
Iac.	39.22	40.26
P.F.	.73	.93
Alt. Losses	122.5 w.	86.0 w.
Alt. Efficiency	90.3%	93.1%
Control Losses (approx.)	25 w.	20 w.
Rectifier Losses	85 w.	106 w.
Electrical Efficiency	84.8%	86.0%

The rectifier losses were obtained by conducting tests on 3 phase rectifier bridge composed of six IN3289 diodes selected for the KIPS application. The 106 watts are used in the system efficiency calculations on the basis of the estimated capacitance requirement of 60 MFD per phase.

The alternator stator assembly consists of a magnetic yoke containing the field coil sandwiched between two stator cores, which contain one common stator winding.

The yoke is made from Armco magnetic ingot iron. This material was chosen for its very high permeability and excellent machinability. It is commonly used in DC motor and generator frames, electro-magnets and other applications where high permeability is desired without any particular requirements for physical strength or core losses. In the KIPS alternator the yoke carries only unidirectional (DC) magnetic flux which does not induce core losses.

Each stator core is a stack of laminations containing 24 slots on its inner periphery. The laminations are made of Allegheny Ludlum electrical steel 4750, 0.006" thick. This material is a 48% nickel 52% iron alloy. It was chosen for its very high permeability at low induction levels and extremely low core loss characteristics. To obtain these characteristics, the material must be subjected to a very critical heat treat cycle.

The laminations are made using a photo-chemical etch process to approximately $\pm 0.003''$ tolerances. Approximately 250 laminations are stacked on an appropriate stacking fixture and are bonded together with Scotchcart 265 epoxy adhesive to form a tightly stacked laminated core. The O.D. and I.D. of the core are then ground to close tolerances to provide a good fit with the magnetic yoke and a uniform airgap with the rotor surface.

The stator winding is a conventional 3 phase, WYE connected, 60 degree phase belt winding. Because of the stringent efficiency requirements, the winding was designed for minimum losses. Current density was kept at a minimum (155 amps/in² at rated load), and measures were taken to minimize eddy current losses. These can be considerable when the stator slot is relatively deep and the alternator output frequency is high. To reduce eddy current losses the stator conductors are made up of 31 strands of AWG No. 20 insulated wire each, and the strands are transposed (twisted) in the end turn area such that those located at the bottom of one slot will be at the top of the next slot. In this fashion all strands will have nearly equal reactances and the eddy current producing voltages will be minimized. Because no particular output voltage waveform requirements exist in the KIPS application, the stator winding was designed with a full pitch and minimum distribution to obtain maximum possible output with minimum field excitation.

In addition to the usual electrical, thermal and mechanical requirements, the KIPS alternator insulation materials were selected with added attention to their long term stability in Dowtherm A and their outgassing characteristics when operating at elevated temperatures in a near vacuum (0.1 PSIA) over a 7 year period.

Table 6-E defines the tentative KIPS GDS alternator insulation system.

The alternator rotor is an integral part of the combined rotating unit (CRU) located between the turbine wheel and system pump. Rotor configuration is a typical four pole homopolar inductor design with two poles of like polarity at each end, the ends displaced 90 degrees (180 electrical degrees) with each other and connected by means of a large magnetic shaft. Often such rotors are machined from a single solid piece of steel. Solid rotors, however, have inherently high pole face losses and alternator efficiency suffers.

Table 6-E KIPS GDS Alternator Insulation System

<u>COMPONENT</u>	<u>TRADE NAME</u>	<u>MATERIAL</u>
Magnet Wire	DuPont Heavy ML Film	Double Coat Polyimide Varnish
Slot Cells	DuPont Kapton Sheet .005" thick	Polyimide Sheet (flexible)
Core end ins.	DuPont Kapton Sheet .005" thick	Polyimide Sheet (flexible)
Field coil ins.	DuPont Kapton Sheet .005" thick	Polyimide Sheet (flexible)
Tie cord	DuPont Kapton Tubing	Polyimide Sheet (flexible)
Slot Wedges	Dixon Corp. Meldin P.I. .025" thick	Polyimide Sheet (rigid)
Impregnating varnish	DuPont Pyre ML Varnish	Polyimide Varnish
End leads	Teflon	

The most effective way to reduce pole face losses is the use of a laminated pole structure. The KIPS rotor is designed with such a pole structure. The rotor laminations are made from AISI grade M-19 electrical steel sheet 0.014" thick, fully annealed. The laminations are made using a photo-chemical etch process and are then oxide coated. A 1.45 inch long stack of laminations is then electron beam welded to each end of a shaft machined from 8620 steel. The Electron Beam (E.B.) welding is done in four steps because of the penetration limitations of the E.B.W. equipment.

After all the laminations are welded to the shaft they are welded together axially by means of 8 TIG welds. The function of these welds is to provide support to any portion of a lamination which may not have been welded to the shaft because of an imperfect weld.

The 8620 steel shaft is mildly hardened and tempered to obtain the necessary strength and rigidity. Hardening also increases the material's magnetic retentivity which enables the alternator to generate sufficient residual voltage to start bootstrapping action during system startup. The alternator residual no load voltage at 33,600 rpm is estimated at 3.3 volts.

Table 6-F is a summary of the alternator design.

Outgassing: For the system to operate efficiently over a 7 year period, contaminants must be kept out of the working fluid. The alternator insulation system may be a potential source of contamination. The source of contaminants can be foreign substances and gases absorbed by the materials through normal handling and brought into the system, volatile substances contained within the insulation materials such as varnish solvents, and gases given off by the material itself due to normal degradation with age and temperature. The former two can be readily eliminated through a prolonged vacuum bake process at elevated temperatures prior to sealing of the system. The latter can not because it tends to be constant over very long periods of time — years. It is for this reason that materials possessing the longest life at the highest temperatures were chosen for this application. These materials should be the most stable over long periods of time and should possess the lowest outgassing rates.

Table 6-F Alternator Design Summary

<u>TYPE</u>	<u>HOMOPOLAR INDUCTOR</u>
Nominal Rating: KVA	1.56
V	12.56 volts
I	40.26 Amp.
P.F.	.93
f	1122 Hz
RPM	33680
OVERSPEED RPM	58000
Poles	4
Phases	3
Connection	WYE
Slots	24
Conductors/Slot	2
Strand/Conductor	31
Strand Size	AWG #20
Parallel Circuits	1
Phase Belt Angle	60°
Coil Pitch	(1-7) 6
End Leads	AWG #6 Teflon
Air Gap	.025
Field Turns	375
Wire Size	AWG #19
Fld. End Leads	AWG #18 Teflon

At this time, the outgassing rate of the alternator insulation system is not known, nor are the acceptable limits. Future phases of this development program should include extensive outgassing tests on the alternator stator assembly.

Test Rig: An alternator test rig is being constructed for the purpose of verifying the alternator design and performance prior to the release of the GDS drawings. Electro-magnetically, the test rig is nearly identical with the GDS unit; it has, however, its own bearings, is designed for mounting on a reaction torque table, is instrumented with thermocouples and is driven by means of an external prime mover.

Bearings

The bearings for the Combined Rotating Unit (CRU) support both axial and radial loads under a variety of different operating conditions. The baseline operating point for which losses need to be minimized and hence performance maximized is the zero "g" condition when bearing loads are at minimum and comprise only turbine, pump, alternator and imbalance loads. The most stringent operating condition will be during launch when accelerations in the axial and radial directions reach 9 and 5 "g"s respectively, and superimposed upon this are shock and vibration loads. In addition, ground operation in the vertical direction will have 1 "g" axial loads and horizontal operation likewise will have 1 "g" radial loads.

The type of bearing selected for this application is dictated by the lubricating fluid available and the mission duration, but obviously working fluid lubrication leads to great simplification of the system. Long life data is not available for rolling element bearings operating in Dowtherm; hence, liquid lubricated ball bearings have been rejected from a reliability standpoint.

Ball bearings have run for long times in high vacuum with dry film lubrication only but loads, size and speeds were significantly different from those to be experienced in KIPS; hence this type of lubrication was not considered.

Since the working fluid exists in both liquid and vapor phases, the latter has been considered as the lubricating medium, in order to simplify the fluid management scheme. However, other problems arise when vapor bearings are considered. First, for self-acting vapor (or gas) bearings unit loads of even 10 psi may be considered high (though failure loads of over 20 psi have been reported for some foil bearings). In order to withstand launch loads this limitation would lead to excessively large bearings; for example, the thrust bearing would need to be larger than 3" in diameter. In addition, low temperature vapor might condense within the bearing causing potential starvation problems while the use of hot vapor to overcome condensation would have a significant effect on cycle performance.

If hydrostatic vapor bearings were considered, high temperatures would again be needed and might result in sealing problems because of the low back pressure turbine operation.

The above reasoning resulted in the selection of the liquid phase for the lubricant. For zero "g" operation, radial bearing stability becomes a major selection criterion. Any tendency of the bearing to exhibit fractional frequency whirl will ultimately lead to problems, indeed, if the whirl is poorly damped, then bearing failure can occur. Several rigid bearing configurations were investigated for stability; these configurations included lobed, tilted lobed and herringbone bearings. In each case the rotor mass is too high for stable operation.

The tilting pad bearing is somewhat more complex in structure than the above rigid bearings but is inherently stable due to the fact that the shaft movement is essentially colinear with the applied load. The tilting pad bearing consists of a series of equally spaced shoes arranged circumferentially around the shaft. A pivot in the back side of the shoe allows it to tilt in the pitch direction in order to form a converging wedge for hydrodynamic pressure generation. Typically, instability only arises when pad pitch inertias are high enough to prevent adequate shaft tracking. In addition, unloaded pads can have a tendency to 'flutter' which is undesirable from the pivot standpoint. Both of these potential instabilities can be substantially reduced by 'preloading' the bearing, a technique which sets the bearing pads closer to the shaft than would be dictated by the machined clearance. This has the same effect as applying an external load.

This preload technique can best be seen from Figure 6-4. Although preloading has a stabilizing effect and can be used to increase the stiffness of the bearing (since the bearing stiffness increases with load) the power loss increases, minimum film thickness decreases so these have to be weighed against the advantages.

An additional advantage of the tilting pad bearing is the self-aligning characteristic which eliminates alignment problems common with rigid bearings. It is for this reason that tilting pad thrust bearings have been selected. In combination with the thrust bearing pads, load equalizers distribute the load evenly across the whole bearing set, helping to minimize the size; hence power loss. A double acting

4 Pad Bearing Zero Preload
(a)

4 Pad Bearing Preload = a
(Preload Coefficient = $\frac{a}{R_P - R_S}$)
(b)

Figure 6-4 Geometrical Arrangement of Pads

gimbal arrangement was originally investigated for this but discarded in favor of a double row of equalizers to reduce pivot stresses at maximum load.

The selection of the bearing configuration has the following factors in mind:

- Minimum film thickness during launch loads
- Power loss at design point
- Critical speed map
- Pivot stresses
- Pad stability
- Maximum allowable shaft movement

Radial bearing loads include the sum of the following components which vary according to the operating condition:

- imbalance load
- alternator magnetic load
- pump load
- turbine load
- rotor weight
- shock and vibration loads

The rotor weight has been calculated to be 9.50 lb; this translates into a minimum balance capability of 0.000114 in.oz.

Since the rotor will not be balanced in the tilting pad bearings in the CRU, certain components will need to be removed from and replaced on the shaft after balancing. These components are the thrust runner and lock nut, the pump impeller and seals. To eliminate assembly difficulties these parts will have small clearances (0.0001-0.0003 in) on the shaft; this can lead to additional imbalance if the parts move after balancing. Computing the weights for each component and allowing for the clearance, results in approximately .0005 in.oz. maximum imbalance in each balance plane.

The bearing reactions due to this imbalance are a function of bearing damping and speed ratio which results in an appropriate amplification factor. The resulting bearing dynamic load has to be used for final design, noting however that this is a worst case condition, since the imbalance could be less than this maximum figure.

The alternator magnetic loading has been calculated to be 1.2 lb/mil deflection. Using pinned housings and line-boring techniques the alternator will be held concentric within 0.002 inches which gives 1.2 lb maximum load on each bearing. The load direction is indeterminant.

The pump load is given by the following equation:

$$W = .36 \left(1 - \left(\frac{Q}{Q_{des}} \right)^2 \right) * D_{imp} * w * \Delta P$$

where   w   =   load (lbs)
           Q   =   flow (gpm)
      D_{imp}   =   impeller tip dia (in)
           w   =   impeller width (in)
      ΔP   =   pump pressure rise (psid)

The turbine load results from the asymmetry of the partial admission turbine. At the design point speed it is calculated to be 0.655 lb and is inversely proportional to speed.

The static load due to the rotor weight is a function of local gravitational acceleration and rotor orientation. Vertical shaft orientation, on ground and operation in space, leads to zero static bearing loading from rotor weight. Horizontal ground operation gives a total load of 9.5 lb while maximum radial launch loads of 5 "g"s result in an equivalent rotor weight of 47.5 lb.

The input shock load to the unit is 775 "g"s peak half cycle sine with 0.2 millisecs pulse width. The support structure reduces this value to approximately 10 "g"s. The final bearing geometry selected is given in Table 6-G.

Table 6-G Radial Bearing Geometry

Shaft diameter (in)	.5500 +0000 -0002
Pad ID (in)	.5525 +0002 -0000
Machined clearance (in)	.0025 ± 0002
Assembled clearance (in)	.00175
Preload coefficient	0.3
Pad thickness at pivot (in)	0.150
(tapered towards leading and trailing edges)	
Pad arc	80°
Pivot point	44° from leading edge
Leading edge radius (in)	.05
Pad material	M-50 with silver flash
Shaft material	M-50 at turbine end AISI 8620 at pump end

Thrust bearing loads include the sum of the following:

- Pump load
- Turbine load
- Rotor weight
- Shock and vibration load

In zero "g" operation the only thrust loading includes the turbine and pump loads. Therefore, to minimize bearing losses it becomes conceivable to use only one set of thrust bearings. This is also predicated on the preferred launch orientation with the rotation axis vertical. To this end the pump load is arranged to be in the same direction as the turbine load by appropriate selection of seal diameters. At the design point this load is 1.9 lb and varies approximately proportional to the square of the speed, since the seal pressures vary with the tip speed.

The rotor weight is calculated to be 9.5 lb. For normal GDS operation the bearings will see just this 9.5 lb but launch operation for the flight system will apply a 9 "g" acceleration load. Table 6-H gives a summary of the expected loads for different operating conditions.

Table 6-H Thrust Bearing Loads (Total)

<u>RPM</u>	<u>ZERO 'G' OPERATION (LBS.)</u>	<u>MAXIMUM LAUNCH (LBS.)</u>	<u>GROUND OPERATION (LBS.)</u>
20000	.73	86.2	10.2
33700	2.1	87.6	11.6
40000	3.3	88.3	12.8
50000	5.6	91.1	15.1
60000	7.7	93.2	17.2

Different rigid bearing configurations were initially investigated, such as a step bearing, but were rejected because satisfactory methods of load equalization were not possible and the large variation in loading presents problems in selecting an optimum size, while any wear can result in loss of load capacity. The large variation in loading leads to the requirement that potential non-perpendicularity of the thrust runner and housing should not reduce the load capacity of the bearing.

The load equalized tilting pad bearing allows each pad to share the load equally and thus allows the size to be minimized. The simplest form of load equalizer incorporates the two pad bearing supported on a single gimbal ring as shown in Figure 6-5(a).

When more than two pads are employed the method of load equalization becomes more complex. The usual equalizer system employs two sets of equalizing pads distributing the load as shown in Figure 6-5(b) for a large commercial bearing or a double gimbal ring arrangement as shown in Figure 6-5(c) in which one ring gimbals against the other. Tilting pad bearings used by Sundstrand in the large Sundyne pumps use the former arrangement.

a) SINGLE GIMBAL – TWO PADS

b) LOAD EQUALIZERS – MULTIPLE PADS (DEVELOPED DIAGRAM)

c) DOUBLE GIMBAL – MULTIPLE PADS

Figure 6-5 Thrust Bearing Load Equalizers

When the performance of a two pad bearing is compared to multipad configurations, the losses for the two pad bearing are less than the multipad bearing for a given film thickness, not accounting for losses between pads (assuming that the bearings are not flooded).

However, for high loads associated with launch the effect of increasing the number of pads is to decrease the pivot loading. For this reason six pads were chosen and for the same reason the multipad equalizer system was chosen over the double gimbal.

Bearing Lubrication and Scavenge

In order to maximize bearing size for a given load capacity the pump inlet flow is used for the source of lubricant which is at a higher temperature than radiator outlet temperature. Individual Lee Visco jets are used to supply the correct bearing flows at each end of the rotor, allowing small flow rates to be metered without using small orifices. Bearing flow is filtered by the main filter in the system accumulator and measured by a single flow meter on the GDS.

Simplification of the lubrication system is possible if the lubricant is introduced from the inside of the shaft. The method has been tested successfully at Sundstrand and is especially beneficial at the turbine end where housing temperatures are relatively hot compared to saturation conditions. However, introduction of the fluid into the shaft becomes more difficult at the pump end than the turbine end. The fluid would have to pass through the eye of the impeller and filtration of this flow would be impossible before reaching the bearing.

The lubrication scheme, therefore, employs external lubrication at the pump end bearings and internal lubrication at the turbine end. The external lubrication is provided by an annular groove cut in the outside of the bearing module feeding four small feed tubes which carry the cool lubricant right to each pad leading edge at the shaft. The annulus in turn is fed by an external tube. This method has been used successfully on the Sundstrand 1.0 Kw(e) Remcom system. The KIPS internal lubrication scheme uses a small stationary feed tube which passes into the center of the rotating shaft. Orifices in the end and side inject the required flow split into two separate internal chambers, one for the thrust bearings and one for the radial bearings. The use of separate cavities prevents the lubricant from being removed by one bearing only or from leaking out of the end of the shaft. The centrifugal field pumps the liquid up through radial feed holes to the thrust disc face and to the radial bearing pads. The effects of launch loads on the distribution of the lubricant is minimized by keeping the feed orifices in the tube and the receiving chambers as close together axially as possible.

Once the bearings have been lubricated with sufficient flow to remove the heat generated by liquid shear, the lubricant has to be removed from the cavity and returned to the system for heat rejection. For ground operation this presents no problem because the gravitational field can be used to drive the liquid to the required location. In space, however, these forces are no longer available and alternate schemes have to be employed.

Different schemes have been considered by Sundstrand for solving this problem, at the same time minimizing any power loss which might be associated with the scavenge scheme. The proposal scheme employed capillary pumping, but further investigation showed that this method is not viable. The use of slingers is being considered as one method of removing the liquid and transporting it to the jet condenser. This method forces the liquid to leave the tip of a disc with relatively high velocity and depends on a recovering sufficient portion of the resultant momentum to drive the liquid down the drain lines to the jet condenser.

Three schemes of dealing with the liquid once it reaches the jet condenser area are being considered. The baseline method is to inject the liquid into the high velocity vapor stream where it will be atomized and carried through the jet condenser throat with the liquid stream. Since the bearing flow represents only 3% of the total jet condenser flow this should not present a problem. The alternate schemes are: a) to use the fluidic valve bypass flow to drive a jet pump which will collect the drainage flow and inject it into the vapor stream or b) to allow the liquid drainage to be scavenged away by the high velocity liquid flow downstream of the jet condenser throat but upstream of the jet re-attachment point.

Sundstrand has built a test rig to investigate the slinger method of drainage. Some success has been obtained in using the slingers to remove the liquid from the cavity, but at this time there appears to be insufficient remaining momentum to carry the liquid through the required length of drain tube without some gravity head, or without wetting the tip of the slinger disc with resultant relatively high power consumption. Further investigation is continuing in this direction with an attempt being made to optimize the shape of the cavity in order to reduce the liquid friction losses.

Meanwhile, alternate schemes are under consideration and test. The pump end bearing presents less of a problem than the turbine end because it has no thrust disc, which, if allowed to churn in an open cavity, could consume considerable shaft power. The present method which is being designed for the pump end and whose efficiency has been proven by test is to run the cavity flooded and to employ a low leakage noncontacting seal between the cavity and the alternator. For the required bearing flow the total pressure drop in the drainage tube is very small, and thus the pressure drop across the seal is very low. Two seal types were considered, first, a centrifugal seal which would prevent all leakage but would consume about 40 watts of power or a close clearance bushing type seal. Both types were built and tested and the latter was chosen because of its low power and low leakage at the anticipated pressure differential. With a seal clearance of 0.003 in. and a length of 0.25 inches the leakage amounts to only 12.5 cc/hr and the measured power consumption was 7 watts. The leakage flow passes through the seal and by judicious arrangement of the housing and alternator windage/balance ring, can be trapped and evaporated by the heat generated in the rotor poles. There would appear to be no problem with this latter method of disposing of the leakage flow but the thermal model of the CRU is currently being used to investigate this area. Obviously this scheme is totally independent of a gravity field.

The configuration currently presented for the turbine end will drain with no problems in the GDS, as the system is essentially being designed in the same manner as the Sundstrand 1.0 KW(e) Remcom system. A centrifugal seal prevents any liquid from passing into the turbine wheel cavity while slingers are used to throw the liquid out of the cavity to the drains. The design envelope is sufficient to allow redesign or retrofit of an alternate method if determined superior. Three alternate schemes are currently being investigated.

The bearing cavity could run totally flooded as at the pump end, with leakage rates reduced to the same values requiring only 17 BTU/hr to vaporize; hence, not impacting turbine performance. The clearances around the thrust disc would be optimized for minimum power loss, with preliminary calculations indicating an additional loss in the order of 20 watts.

The second method again employs a flooded cavity with the same seal leakage but uses a centrifugal type seal at the tip of the thrust disc to prevent the whole cavity from flooding. This seal would only run wet sufficiently to drive the liquid through the drainage lines.

The third scheme employs pitot probes to scoop the liquid out of a trough at the tip of each of the slinger discs. These would generate sufficient head to inject the drainage flow whenever required while drag losses for an 0.020 inch diameter tube would be approximately 8 watts per tube. The small leakage which might bypass the disc would be evaporated by the heat leak from the hot turbine housing.

The heat leak from the turbine will evaporate a small amount of the drainage flow after it passes through the bearings, but the thermal model indicates that this will not be a large amount.

Figure 6-6 shows a section view of the turbine end bearing housing showing radial and thrust bearings, drainage slingers, lubricant injection tube and snubber for reverse load.

System Pump

The system pump designed for the GDS is similar to the one shown in the FSCD. The main difference is that the GDS pump is larger than the FSCD pump to overcome some of the additional pressure losses caused by adding instrumentation, valves and joints. These items are required for assembly, testing and development of the GDS.

Identical floating seals that were proposed in FSCD will be used in the GDS. Because of this design feature the pump leakage and recirculation loss are kept to a minimum.

Two types of pump impellers were considered for the FSCD and manufactured per EP2605-5059 and EP2605-5065 drawings. The test data shows that EP2605-5059 vane pump impeller is more efficient than EP2605-5065 drilled port impeller. The test data comparison is shown in Figure 6-7.

Using this test data and sizing factors based on large impellers, the vane pump impeller is redesigned for the GDS flow and pressure requirements of 2.35 gpm and 76.8 psid respectively. The comparison of the FSCD and GDS impellers is shown in Table 6-1. Figure 6-8 illustrates the selected design.

6.2 TEST CONDENSER AND ACCUMULATOR

Jet Condenser

The KIPS GDS jet condenser is illustrated in Figure 6-9 and the FSCD jet condenser is shown in Figure 6-10.

The vapor inlet was sized and shaped so as to provide a smooth acceleration to Mach 0.6 (~275 ft/sec). There is no known effect of vapor inlet velocity upon jet condenser performance; however, this Mach number was successfully employed in Sundstrand's 6KW jet condenser.

The liquid injector assembly is a radial feed type. It consists of four radial support struts that also serve as liquid feed lines, three concentric distribution rings, and forty-five nozzles.

The two phase flow moves axially through the vapor funnel, with the vapor condensing upon the liquid jets.

The point at which the liquid jets converge is the throat.

Figure 6-6 Turbine Bearing Configuration

Figure 6-7 Pump Performance

Table 6-1 Impeller Comparison

	<u>FCD</u>	<u>GDS</u>
Inlet Diameter — Inches	.295	.295
Inlet Blade Thickness — Inches	.005	.005
Inlet Blade Angle	18°	18°
Inlet Blade Height — Inches	.0775	.0775
Outside Diameter — Inches	.668	.736
Exit Port Width — Inches	.0415	.040
Exit Blade Angle	22.5°	22.5°
Exit Blade Thickness — Inches	.020	.020
Material	17-4PH	17-4PH
No. of Blades	6	6
Specific Speed	1380	1050
Diffuser — Throat Area — Sq. In.	.0227	.0182
Flow — GPM	2.35	2.35
Speed — RPM	33680	33680
Pressure Rise — PSID	56	76.8
Pump Efficiency	56.4	60
Power Consumption — Watts	102.4	130.8

Figure 6-8 Vane Type Impeller

Figure 6-9 KIPS GDS Jet Condenser

Figure 6-10 KIPS FSCD Jet Condenser

The high velocity condensate stream then enters the primary diffuser and undergoes a sudden expansion to fill the space.

The stream then enters the secondary diffuser in which most of the dynamic pressure is converted to static pressure.

A check valve has been placed between the diffuser and the accumulator to assure bearing supply in the event of an emergency shutdown.

The jet condenser is coupled to the system accumulator by a thermal expansion bellows. This bellows will accommodate the axial misalignment due to manufacturing tolerances and differential thermal expansion of the housings and structural support as well as the radial misalignment due to manufacturing tolerances.

The bearing scavenge is a line that delivers the scavenged liquid to the high velocity area of the jet condenser. There the flow is atomized and swept downstream by the vapor flow.

Additional features include jacking screws for fine tuning the vapor funnel throat location to the liquid injector focal point and an assembly/protective shell to lock the jet condenser and accumulator together at assembly and provide protection for the thermal expansion bellows against damage.

The necessary instrumentation ports have been provided to determine and monitor the jet condenser performance.

Provisions have been made to incorporate additional instrumentation and/or an alternate bearing scavenge configuration should either jet condenser rig or GDS experience deem it necessary.

Design: The design of the GDS jet condenser follows the procedure outlined in the Flight System Design Report AER 132A. The critical equation is a condensing rate equation derived from Sundstrand's 6KW experience.

$$Q/A = \frac{136 (P_{mix})}{.32} \quad \text{Btu/ft}^2 \text{ hr}$$

where Q —   heat to be transferred (Btu)
           P_{mix} —   mixing chamber pressure (psia)
           A —   condensing area (ft²)

$$A = N\pi D_j L$$

N — no. of nozzles
 D_j — jet diameter
 L — jet length

The results of the design analysis are presented in Table 6-J and FMEA in Table 6-K.

Table 6-J GDS Jet Condenser Geometry

State Points:

<u>Location</u>	<u>Mass flow (lbm/sec)</u>	<u>Temperature (°F)</u>	<u>Pressure (psia)</u>
Liquid In	0.271	173	71.5
Vapor In	0.030	250	0.1
Bearing In	0.002	217	0.1
Liquid Out	0.303	217	35.8

Length	L	= 10 in.
Mixture chamber pressure	P_{mix}	= 0.083 psia
Discharge coefficient	C_d	= 0.77
Contraction coefficient	C_c	= 0.79
Velocity coefficient	C_v	= 0.98
Number of nozzles	N	= 45
Orifice diameter	D_o	= 0.0145 in.
Jet diameter	D_j	= 0.0127 in.
Effective vapor diameter at the liquid injector	D_{mix}	= 3.14 in.
Diameter at liquid injector	D_l	= 4.00 in.
Number of rings	n	= 3

<u>Distribution</u>	<u>No. of Nozzles</u>	<u>Diameter (In.)</u>	<u>Angle to Axis (degrees)</u>
Outer	21	2.808	8° 12'
Middle	15	1.932	5° 31'
Inner	9	1.014	2° 49'

Vapor velocity at liquid injector	V_v	= 278.3 ft/sec (M = .6)
Liquid velocity	V_L	= 103.0 ft/sec ($Re_D = 11,800$)
Stream deflection due to acceleration	δ_a	= 0.028 (in.) max. deflection due to acceleration of 9g axial and 5g lateral simultaneously
Stream deflection due to vibration	δ_v	= 0.054 (in.) (due to 4.2 g's @ 27 Hz)
Throat diameter	d_t	= 0.133 in.

<u>Diffuser:</u>	<u>Length (in.)</u>	<u>Half angle (degrees)</u>
Primary	0.800	1
Secondary	3.250	3

Pressure recovery	ψ	= 50%
Liquid side pressure drop		1.9 psid

Table 6-K GDS Jet Condenser Reliability

FAILURE MODE AND EFFECTS ANALYSIS

GROUND DEMONSTRATION SYSTEM

SHEET 5 OF 16

NO.	COMPONENT & FUNCTION	FAILURE MODES (FM)	POSSIBLE FAILURE CAUSES	FAILURE EFFECT (FE)	DESIGN/SAFETY CONSIDERATIONS	③ CRITICALITY RANK, ① × ② × ③ ② FE RANK ① FM LIKELIHOOD			
						①	②	③	
1.5	<u>Jet Condenser</u> Condenses turbine exhaust vapor by directing the vapor into a vapor funnel coaxially with liquid jets of subcooled liquid working fluid.	Condenser floods.	Liquid injector cracked due to material defect, manufacturing defect, fatigue due to vibration. Misdirected jet from a liquid nozzle partially blocked with contaminant particle. Diffuser misaligned due to thermal effects. Excessive concentration of noncondensable gases.	Flow stops, loss of electrical power output. Heater temperature control module controls HSA temperature.	Liquid injector has 45 nozzles. Extensive development testing of jet condenser will be performed. Orifice diameter is 0.0145.	25	100	2500	A
		Funnel cracked.	Material defect, defective weld, fatigue from vibration.	During operation in air, air in leakage results in jet condenser flood out and no start or electrical power loss. Heater temperature control module controls HSA temperature. During operation in vacuum, external leakage results in loss of working fluid and electrical power loss. Heater temperature control module controls HSA temperature.	Filtration is provided upstream of jet condensor by filter at radiator outlet. Nozzle braze joints will be penetrant inspected. Designed for 1×10^{-9} failure probability. Jet condenser will be helium leak checked.	5	100	500	A
1.6	<u>Check Valve</u> (Jet Condenser Outlet) Maintains pressure in fluid loop to ARU bearings in event of safety shutdown.	Failure to open at starting.	Valve jammed closed due to mechanical failure.	PCS fluid loop blockage. Jet condenser floods out. System will not start.	Valve has large flow passages. Design is such that flow is thru clearance so that flow will tend to pass contaminants thru valve. Check valve is not included in flight conceptual design.	10	100	1000	
		Failed open.	Valve stuck or jammed open due to contamination or mechanical failure.	No effect on system operation. Loss of bearing protection in event of safety shutdown.					

Flight System Similarity: The GDS jet condenser design is prototypic of the FSCD. The performance differences as indicated by the vapor and liquid side pressure drops are very small.

The GDS was designed to maintain similarity with the test rig and with minor rework the rig injector may be incorporated in the GDS.

<u>Configuration</u>	<u>Unit</u>	<u>Vapor ΔP</u>	<u>Liquid ΔP</u>
Center feed	FSCD	.0144 psid	1.714 psid
Radial feed	GDS	.0101 psid	1.901 psid

Jet Condenser Parameters

States: Table 6-L shows the fluid state points pertinent to the jet condenser based on a 1300 W(e) organic Rankine cycle system utilizing Dowtherm "A" as the working fluid. The state points were generated by Sundstrand Computer Program OPT for Rankine cycle system optimization.

Table 6-L State Points

<u>1300 W(e)</u>	<u>$\dot{m}$(lbm/sec)</u>	<u>h(BTU/lbm)</u>	<u>ϕ(lbm/ft³)</u>	<u>T(°F)</u>	<u>P(psia)</u>
Liquid In	.2691	47.11	63.656	173.0	73.5
Vapor In	.0299	232.79	.001927	249.3	0.1
Liquid Out	.2990	65.67		217.0	36.8
Flow Ratio	$\alpha = \frac{\dot{m}_{\text{Liquid}}}{\dot{m}_{\text{Vapor}}} = 9$		$FR = \frac{\dot{m}_{\text{total}}}{\dot{m}_v} = 10$		

α — used in system optimization program

FR — used in proposal discussion

Governing equations: The following two equations define the geometry of the jet condenser. The first is a condensing rate equation derived from Sundstrand's experience in developing a jet condenser for the 6 KW ORC system. The second equation is the standard orifice equation applied to the multi-orifice injector.

$$(1) \quad Q/N\pi D_j L = 136 (P_{\text{mix}})/0.32 \quad \text{based on 6 KW}$$

$$(2) \quad \frac{N\pi D_o^2 C_d}{4} \sqrt{\frac{2g\Delta P}{\phi}} = m_L \quad \begin{array}{ll} D_o & \text{— orifice diameter} \\ D_i & \text{— jet diameter} \end{array}$$

choose: L, P_{mix}, C_d

system optimizing program generates: $Q, \Delta P, \phi, m_L$

eqns above supply: N, D_j

The equation below spaces the N nozzles such that the distance between rings and the distance between nozzles on a ring are all equal (see figure below).

$$S = \frac{\pi n D_{mix}}{N + \pi n^2 + \pi n} \quad \text{where } n \text{ is no. of rings}$$

S is spacing of and upon rings

Multipower: Table 6-M is a presentation of parameters for a jet condenser optimized at the noted power levels.

Table 6-M Jet Condenser Optimization

	500 W	1300 W	2000 W
P mix (psia)	.1	.1	.1
Vapor Mach. No.	.6	.6	.6
Dve (in)	2.129	3.140	3.961
N	19	45	87
Dj (in)	.0140	.0125	
L (in)	10	10	

The choices for demonstrating a power level of 2000 W(e) were studied. Four reasonable choices exist: (1) Optimize the jet condenser for 2000 W(e) state points (choose $M = 0.6$, $P_{mix} = 0.1$ and size using procedure used to size the 1300 W(e) condenser); (2) build a jet condenser limited to 45 nozzles due to focusing considerations and maintain $Mach = 0.6$; (3) utilize all 1300 W(e) jet condenser hardware by reworking injector nozzles to pass the 2000 W(e) liquid flow; (4) add nozzles of the 1300 W(e) size (0.0145 in. Dia.) to pass the 2000 W(e) liquid flow. In all cases it would be necessary to rework the throat to a larger size to pass the 2000 W(e) flow. Significant parameters are presented in Table 6-N for comparison.

Table 6-N Jet Condenser Parameters

Option	N	Dj (in)	Dmix (in)	Pmix (psia)	M	Dthroat (in)
(1)	87	.0105	3.961	.1000	.600	.147
(2)	45	.0146	3.504	.1392	.600	.147
(3)	45	.0146	3.125	.1538	.744	1.47
(4)	61	.0125	3.125	.1423	.827	.147

Water Flow Tests: Figure 6-11 illustrates the results of water flow tests with various configuration nozzles. The attempt with these tests was to identify an orifice configuration which was the least susceptible to brooming with entrained non-condensable gases. This led to the selection of the sharp edged orifice as the initial injector nozzle for jet condenser rig testing.

- 1 Air bubbles trapped within stream boundary $L/D > 3$
- 2 Air bubbles cast off at inlet tend to deflect stream, $.5 < L/D < 3$. L/D of 3 is point of reattachment vena contracta at nozzle exit plane.
- 3 Sharp inlet, $L/D < .5$. Air bubbles seem to be cast off as "burp" at nozzle. Stream remains essentially unaffected.

Figure 6-11 Brooming With Various Jet Condenser Nozzles

System Accumulator

The GDS Accumulator designed is as shown in Figure 6-12. The major mechanical arrangement differences between the FSCD and GDS accumulator are shown in Figures 6-12, 6-13 and Table 6-O. Because of providing stock inside the accumulator for mounting of the bellows and filter the internal fluid volume of the accumulator at operating condition has changed from 238 cubic in. in FSCD to 207 cubic in. in the GDS.

Stress analysis has been completed on the GDS accumulator. It shows why a bellows joint between the jet condenser and accumulator is required. A diaphragm type joint in this location was proposed for the FSCD but the stress levels encountered to accommodate the differential thermal growth between the struts and the components was beyond the capability of the diaphragm material.

During the development of the GDS if jet condenser flood-out occurs it is desirable to keep the bearings lubricated while the turbine speed is reduced. In order to provide this, the bellows displacement has been maximized in the GDS accumulator and a check valve has been added at the jet condenser outlet.

Figure 6-12 Accumulator (Ground Demonstration System)

Figure 6-13 Accumulator (Flight System Conceptual Design)

Table 6-O Accumulator Changes

<u>ITEM NO.</u>	<u>CHANGE DESCRIPTION</u>	<u>GDS DESIGN IMPACT</u>	<u>REASON FOR CHANGE</u>
1	Increase wall thickness	Adds weight — affects shock vibration and acceleration	Allows the shell to be hogged from solid and eliminates M/F development in M/C and welding for GDS.
2	Added clamp and O Ring to split line	Adds weight and a non hermetic seal	To allow for disassembly of the accumulator and access to the bellows and Filter during development.
3	Changed flow direction through filter	Adds weight	New assembly for GDS. Complicated the potential sealing problems for the filter.
4.	Increased filter area	Adds weight	Provisions for development and analysis — to be able to determine real size of filter and prevent possible hardware failures during hardware cleaning development.
5	Revised filter Mounting	Adds weight and internal seals	Allows for ready replacement of the filter and possibly a more rigid design. Will aid development.
6	Increased Bellows size	Adds weight	Allows for development of correct pressure and gas volume size to insure flow to bearings during shut down.
7	Revised Bellows Mounting	Adds weight, internal seal affects shock and vibration characteristics	Allows for access to bellows for N ₂ line and hence variations in pressure during development to determine effects.
8	New Bellows Pressurization Scheme	Adds weight, and internal/external Seal.	See item 7
9	Deleted Hermetic Cap from fill QD	Delete weight — makes internal/external "O" ring seal	To aid filling and draining during development.
10	Revise outlet manifold configuration and position	Adds weight	Allows for nitrogen tap to be external and a bigger gas volume. Ease to M/F.
11	Add mechanical fitting to outlet line	Adds 2 internal/external "O" rings	Allow for assembly and disassembly during development.

Table 6-O Accumulator Changes (Cont'd)

<u>ITEM NO.</u>	<u>CHANGE DESCRIPTION</u>	<u>GDS DESIGN IMPACT</u>	<u>REASON FOR CHANGE</u>
12	Added temperature sensor	Adds weight	Instrumentation
13	Added stock	Adds weight	Ease of M/F.
14	Added stock	Adds weight	Ease of M/F.

SECTION 7.0 WORK PLANNED IN NEXT REPORTING PERIOD

Following is a listing of the work to be done in the next reporting period of 1 August 1976 to 1 August 1977.

- Completion of all design reviews
- Update of the GDS test plan
- Completion of fundamental rig tests on the jet condenser, regenerator, turbine, flow controls and electronic controls. It is expected that iterations beyond the fundamental tests will continue throughout the Phase I program.
- Completion of analysis and design of the GDS and periodic updates of the FSCD as mission and application information becomes available
- Completion of fabrication and assembly of the GDS and initiation of component and system level development and performance testing
- Continual involvement and monitoring of the design, fabrication, assembly and testing by Reliability and Quality Control
- Initiation of significant Advanced Development items

Most significant is the development of certain key components such as the jet condenser which will receive continuous rig testing until optimum operation is achieved. Other areas of the system such as turbine, generator, scavenging, seals, bearing system, accumulator, pump, and regenerator, HSA and radiator are not restrictive to the operation of the KIPS. Effort here primarily is in the area of defining, measuring and correcting as required, potentially deficient performance (specifically efficiency) in order to minimize internal power losses. The flow controls and electronic controls will be breadboard tested dynamically to ensure proper functioning, stability of operation and acceptable quality of power.

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