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FEASIBILITY OF ISOTOPIC POWER FOR
MANNED LUNAR MISSIONS

VOLUME 5--BRAYTON CYCLE SYSTEM

May 1964

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FOREWORD

The results of a study on the application of radioisotopic power to the Apollo mission are provided in this series of reports. The study was performed for the U. S. Atomic Energy Commission by the Nuclear Division of the Martin Company under Contract AT(30-1)-3296. The work was sponsored by the Division of Reactor Development, Isotopic Power Branch in keeping with its responsibility for the development of isotopic power systems for space missions of established or possible interest to the National Aeronautics and Space Administration (NASA) and the Department of Defense (DOD).

The complete results of the study are presented in eight volumes, as follows:

Volume 1--Summary

Volume 2--Mission Requirements, Fuel Availability, and
Nuclear Safety and Radiation

Volume 3--Thermoelectric System

Volume 4--Thermionic System

Volume 5--Brayton Cycle System

Volume 6--Stirling Cycle System

Volume 7--Dowtherm-A Rankine System

Volume 8--Mercury Rankine System

The assistance of the following companies who are recognized leaders in their respective fields and who generously provided state-of-the-art information is gratefully acknowledged: Brayton Cycle--AiResearch Manufacturing Company of the Garrett Corporation; Dowtherm-A Rankine--Sundstrand Corporation; Mercury Rankine--Thompson-Ramo Wooldridge Corporation; and Stirling--Allison Division of the General Motors Corporation.

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I. INTRODUCTION

The ideal Brayton cycle is composed of two reversible constant pressure processes and two isentropic processes. This is shown in the diagrams of Fig. 1. The working fluid is compressed isentropically from Points 1 to 2, heat is added in the constant pressure process, 2 to 3, isentropic expansion occurs from 3 to 4, and heat is rejected at constant pressure from 4 to 1. The mean effective pressure of the Brayton cycle is relatively low. This is not a serious handicap, however, since turbomachines can handle very large volumes of fluid per unit time.

It is readily seen that if the energy rejected from the cycle can be conserved, cycle efficiency can be increased. In the system discussed here, this is accomplished by regeneration in the system. Thermal energy is transferred from the turbine exhaust to the compressor discharge flow, thereby retaining much of this energy in the system as shown in the temperature entropy diagram of Fig. 2. This diagram shows the recuperator heat transfer and the deviation from the isentropic processes of the ideal cycle.

The closed Brayton cycle, as the mode of energy conversion in radioisotope fueled space vehicle power supplies in the kilowatt range, has been studied previously by Martin and AiResearch (Refs. 1 and 2).

At the start of this study, it was felt that the space radiator area available to the power supply would be limited to 90 square feet. It later developed that this may not be a constraint and thus a larger area could be used if desirable. Therefore, the basic design for this study considered a 90-square foot radiator and one that is close to an optimum with respect to weight and fuel inventory. The basic energy conversion system in both cases results from the efforts of AiResearch. This company has had extensive experience in the use of small, high speed, turbomachinery and in compact heat exchange equipment.

An electrically heated closed Brayton cycle system was demonstrated in late 1963 but no component development was carried out for this system. It was operated at shaft speeds up to 56,900 rpm. Measured efficiency was 15.2% at an output of 1.8 kilowatts.

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Fig. 1. Ideal Closed Brayton Cycle

Fig. 2. Brayton Cycle T-S Diagram

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II. BRAYTON CYCLE SYSTEM

A. SYSTEM DESCRIPTION

A schematic diagram of the proposed radioisotope powered Brayton cycle power system is shown in Fig. 3. As illustrated, the system is composed of two independent closed Brayton cycle power loops. Each loop is comprised of: an energy conversion subsystem, a heat rejection subsystem, and a controls and power conditioning subsystem. The radioisotope heat source is common to both power loops, and the electrical power conditioning equipment and control subsystem can be used by either power conversion loop. The complete system is supplied as a welded, hermetically sealed assembly. An installation outline of the proposed system is shown in Fig. 4.

Heat can be supplied to either power loop through a common heat exchanger using polonium-210 fuel as the energy source. It is intended that one power loop only will be operating while the other is on standby. Each closed Brayton cycle power loop is designed to provide a net electrical output power of 1.5 kilowatts of direct current at 28 volts, for the mission duration of 14 days--with the heat source designed for an additional 30-day prelaunch standby period.

The combined rotating unit (CRU), the heart of each Brayton cycle power loop, consists of a high frequency, permanent-magnet alternator, a single-stage centrifugal compressor and a single-stage radial inward-flow turbine. These items are mounted on a single shaft and supported by two hydrodynamic gas bearings. The alternator is located outboard from the compressor and turbine. One bearing is located outboard from the alternator and the other is located between the compressor and turbine wheels. The CRU operates at a controlled rotational speed of 45,000 rpm. The alternator generates high frequency 1500-cps power which is then rectified to 28 volts d-c output power and supplied to the bus bar.

In operating the system, the working fluid, argon, enters the compressor and is compressed to the selected pressure. The compressed gas flows to the recuperator where it absorbs waste heat from the turbine exhaust, then to the heat source heat exchanger where it absorbs heat from the isotope fuel. Heat addition occurs at nearly constant pressure. The gas then expands through the turbine, doing work on the rotor, and flows to the recuperator where it rejects heat to the compressor outlet gas. It then flows through the radiator heat exchanger, where it rejects heat to a separate liquid coolant circuit and which in turn, rejects through the radiator to space. Heat rejection in the argon also occurs at nearly constant pressure. From the radiator heat exchanger, the gas flows back to the compressor inlet to complete the cycle. The working fluid is also used to provide alternator cooling.

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Fig. 3. Schematic Diagram Radioisotope Fueled Closed Brayton Cycle Power System for Project Apollo

Fig. 4. Installation Outline of Brayton System

Fig. 5. System Pressure and Temperature Summary (at end of mission)

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A more detailed summary of the operating pressures and temperatures within the cycle for a system net electrical output power of 1.5 kilowatts is shown in Fig. 5.

The energy conversion subsystem includes the CRU, heat source heat exchanger (with the isotope fuel block) and the recuperator. The radiator heat exchanger, radiator and its coolant circuit and the evaporator comprise the heat rejection subsystem. This subsystem uses Monsanto OS-139 (Coolanol-25, silicate ester) as a coolant fluid, selected because of its ability to endure the high temperature in the operating circuit without freezing in the stand-by circuit. The working fluid is pumped around the circuit by a single centrifugal pump which is mounted on the cold side of the circuit to simplify material problems. Each circuit includes a redundant stand-by pump. Check valves are used to prevent reverse flow in the inoperative unit.

The evaporator with water evaporant is used together with the radiator during prelaunch operation. It is also used during the ascent phase of flight where the radiator is not fully effective due to high sink temperatures and/or aerodynamic heating.

The primary control element of the power system is a parasitic load control. The control system senses the output voltage of the rectifier and frequency of the alternator and compares these to a reference, and then varies the resistance of a parasitic load to maintain the alternator frequency (CRU speed) nearly constant. To maintain the turbine inlet temperature within a nominal operating band, as a result of the isotope decay characteristics, a recuperator by-pass valve is included in the system which controls the amount of working fluid by-passing the recuperator. All controls function automatically; in the event of a control element failure, a stand-by component takes over immediately.

B. SYSTEM OPERATION

1. Startup and Ground Operation

Either power system may be started by positioning of the power system selector switch. The CRU is accelerated to a rotational speed of about 60% of design speed by a small d-c starter motor. This motor will draw 500 watts from the system storage batteries for three minutes per start. Then with the addition and removal of heat to and from the closed cycle, the unit will become self-sustaining and can accelerate to the design speed of 45,000 rpm. Because of the physical mass and thermal lag of the system, the isotope fuel block must be placed in the heat source exchanger several hours before starting to allow for warm-up. Since the radiator does not provide adequate heat rejection on the

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ground, waste heat from the cycle is rejected to boiling water in the evaporator and is then dumped overboard.

During ground operation, water must be supplied continuously to the evaporator.

2. Prelaunch and Ascent

During the prelaunch and ascent phases of flight, the radiator is still ineffective due to high sink temperatures and aerodynamic heating. During this phase, the system performs as in ground operation, using the evaporant as the heat sink. The only difference from ground operation is that instead of being supplied continuously from a ground source, the evaporator uses stored evaporant. The evaporant is forced through the evaporator by high pressure nitrogen during this period. After leaving the evaporator, the evaporant is passed over the heat source radioisotope fuel block to reduce the operating temperature of the fuel and to provide an additional design margin to assure heat source containment in the event of abort. Under these conditions, the turbine inlet temperature of the CRU is approximately 1470° R for no-load, full-speed, idle operation. The amount of evaporant stored in the reservoir is sufficient, once the ground source umbilical connection is released at launch, to sustain system operation for six minutes or until the radiator has begun to operate effectively.

3. Orbital and Translunar Operation

During orbital and translunar operation, the power system operates at nominal design conditions. To accommodate the variations of real power requirements and total system output power as a result of effective sink temperature variation, a parasitic load control is used. Excess power is dissipated through the parasitic load bank that is located on the vehicle outer surface so as to radiate the waste heat to space. If direct radiation to space is impractical, the parasitic load can be cold plated to the liquid entering the radiator.

The recuperator by-pass valve is incorporated into the system to maintain the turbine inlet temperature at the nominal value of 2000° R. Whenever the operating turbine inlet temperature falls below the allowable operating band, the control system automatically adjusts the by-pass valve position to reduce the by-pass flow, which in turn increases the turbine inlet temperature to within the normal operating band. This by-pass valve primarily compensates for the isotope decay characteristics.

4. Shutdown and/or Changeover

Changeover from one power system to the other is accomplished automatically through the power system selector switch. Switchover

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to the stand-by CRU is accomplished by closing the shutoff valve in the compressor discharge duct and by breaking the control circuits to the operating CRU. This, in turn, stops the coolant pump. When this unit reaches a certain minimum speed, the stand-by unit is started and accelerated to governed speed. The unit will then start to warm up and can accept the full load within 2 to 10 minutes. This time depends on the detailed configuration of the system.

5. Overspeed Safety

The same power conditioning equipment serves either operating CRU and is completely redundant. In the event of failure of either the parasitic load circuit or the power conditioning equipment which could lead to system overspeed, the redundant control set takes over the control function as it is on the line at all times.

Trouble warning lights can be used to indicate the presence of a failed electronic component. These components are placed in the control console of the command module.

C. MAJOR SUBSYSTEMS

1. Heat Source

a. Fuel block

Thermal energy is supplied to the system by the radioisotope source. Twenty-two capsules fueled with gadolinium polonide are mounted within cylindrical holes drilled into a beryllium block. The capsule is shown in Fig. 6.

Each capsule contains four grams of Po-210, giving a total inventory of 88 grams, 252 grams of GdPo with a volume of 84 cubic centimeters (90-square foot radiator). This will provide the required thermal input of 9.52 kilowatts after 44 days. With a turbine inlet temperature of 1540° F, the fuel block surface temperature will be approximately 1950° F with a capsule centerline temperature of approximately 2062° F. The maximum fuel capsule temperature should be kept below 2000° F. This can be accomplished by reducing the turbine inlet temperature and consequently reducing the overall cycle efficiency. (A 1140° F turbine inlet temperature has approximately 12% efficiency.) The fuel block temperature can be further decreased, if necessary, by increasing the area of the fuel block. This can be done by finning the block to reduce the maximum fuel temperature by about 100° F. The capsules are composed of four concentric cylinders--gadolinium polonide fuel, tantalum, molybdenum and Haynes 25. Each capsule is designed with sufficient void volume to contain the helium gas buildup.

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Fig. 6. Fuel Capsule Brayton Cycle

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The fuel block is shown in Fig. 7. The sides of the block are clad with Udimet 700 to prevent sublimation of the beryllium in the vacuum environment.

One-inch thick graphite strips surround the edges of the block to protect it during re-entry aeroheating.

b. Heat source heat exchanger

The heat source heat exchanger, Fig. 8, transfers heat from the radioisotope fuel block to the argon working fluid in either of two completely independent gas circuits. Each circuit is divided into individual passages which are separated from each other by a passage of the other circuit. The heat exchanger has a slot in the center into which the fuel block is inserted. Each circuit carries gas across both sides of the block, providing uniform heat removal. No thermal contact is required between the fuel block and the heat exchanger, as heat transfer from the block is entirely by radiation.

Each argon passage has plain rectangular fins extending to the outer surface of the heat exchanger. The fins are copper that is clad with a corrosion resistant high temperature alloy, such as stainless steel or Hastelloy X. All joints that must be gas tight, including those separating the two gas circuits, are welded. The clad copper fins inside the individual passages are brazed to the container. The side of the heat exchanger nearest the vehicle skin is covered with an insulating plate.

In the event that coolant flow through both argon circuits is lost, an explosive charge or actuator system will remove the insulation panel, and the heat will be dissipated by conduction through one side of the heat exchanger to the side plate, as shown in Fig. 9.

An alternate method for emergency heat removal is discussed in the Stirling cycle system. Both methods are applicable to each system.

2. Energy Conversion

a. Combined rotating unit (CRU)

The combined rotating unit consists of a high frequency, permanent-magnet alternator, a single-stage centrifugal compressor and a single-stage radial inward-flow turbine. These items are mounted on a single shaft and are supported by two hydrodynamic gas bearings. One bearing is located outboard from the alternator and the other is located between the compressor and turbine wheels. The CRU operates at a nominal rotational speed of 45,000 rpm.

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Fig. 7. Brayton Cycle Fuel Block

Fig. 8. Heat Source Heat Exchanger with Fuel Block

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Fig. 9. Fuel Block Emergency Heat Rejection--Brayton Cycle

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Extensive design consideration has been given to minimizing the heat transfer between the hot and cold ends of the machine. Since the external heat load will be negligible, the cooling load is restricted to the heat rejection of the alternator and the turbine-end bearing. To reduce heat transfer from section to section, heat transfer paths have been extended and are designed for high thermal resistance.

The most direct path for heat transfer is through the shaft from the turbine wheel to the compressor wheel. To reduce the thermal transmission of this part, the rotating group will be assembled from a number of parts having poor thermal contact with each other and, where possible, having poor thermal conductivity. The turbine wheel and the impeller are pressed onto an interrupted surface so that the contact area between the shaft and impeller is small.

The thin shaft is made of Inco 718, a low thermal conductivity material. The hollow shaft is integral with the alternator rotor, and interrupted resistors are used to maintain concentricity of the individual parts of the assembly, and also to increase thermal resistance.

Provisions are built into the housing for cooling the alternator and the turbine bearing with the argon working fluid. Rotor cooling is largely by radiation and convection to the stator.

To minimize the radiation heat loss from the turbine scroll and other hot components, the entire Brayton cycle power system is enclosed, and the area filled with a granular insulation, such as Pearlite. To reduce heat loss from the turbine scroll along the outer casing to the compressor, the outer casing is made of Inconel, a material with a very low thermal conductivity, and a pad of insulation is placed behind the turbine rotor.

A permanent-magnet alternator has been selected because the complexity and additional losses imposed by an electromagnetic field excitation system are undesirable. High efficiency and reliable simplicity are essential. A permanent-magnet alternator can operate reliably in environments up to 500° F because electrical losses in the rotor are small; there is no field winding or excitation circuit to be cooled; and the stator iron losses are inherently low. Because the permanent-magnet alternator intrinsically is a high efficiency machine, the temperature rise of its components will generally be less than for other types of alternators. Since gas bearings are used, low temperature rise of the rotor is desirable to ensure dimensional stability of the bearings.

A discussion of the bearing selection for the CRU is given in Appendix C. Bearings are further discussed in Refs. 4, 5, 6, 7 and 8.

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A four-pole, 1500-cps configuration has been selected for optimum weight and size. A three-phase winding will provide power which will be rectified to direct current by a six-phase, half-wave rectifier.

Current manufacturing capabilities can produce turbine wheels down to 3.5 inches in diameter without undue sacrifice in efficiency.

A summary of the pertinent design parameters of the CRU is given in Table 1. The values listed are for a 90-square foot radiator. A complete analysis has been performed by AiResearch for this size area. These values would not change appreciably for a 120-square foot radiator. For example, as with a 120-square foot radiator, the compressor tip diameter is 4.15 inches and the compressor inlet temperature is 545° R. A sketch of the unit with overall dimensions is shown in Fig. 10.

TABLE 1
Summary of CRU Design Parameters

	<u>Compressor</u>	<u>Turbine</u>
Tip diameter (in.)	4.26	5.04
Inlet pressure (psia)	5.47	8.92
Inlet temperature (°R)	600	2000
Pressure ratio	1.744	1.535
Efficiency (%)	78.3	85.8
Tip speed (fps)	837	991
	<u>Alternator</u>	
Rating (kw)	1.5	
Volts (v ac)*	23	
Efficiency (%)	91.0	
Rotor (in.)		
Diameter	2.4	
Length	0.9	
Radial gap (in.)	0.012	
Stator diameter (in.)	3.6	

*See Appendix B for power conditioning and rectification to 28 volts dc.

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Fig. 10. Combined Rotating Unit

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b. Recuperator

The recuperator, Fig. 11, consists of a single-pass counterflow plate-fin heat exchanger of brazed and welded stainless steel and nickel construction. Type 347 stainless steel plates and nickel fins will be used. Pure counterflow heat transfer occurs in the central section of the core. The triangular end sections serve to bring the fluids into and out of the central section.

The plate-fin heat transfer surface was selected and the core designed by the use of a digital computer heat exchanger design program. Heat transfer design data for the fins were obtained from tests run at Stanford University on an AiResearch test section with fins made from production tooling. The fins are rectangular in shape with offsets in the flow direction to increase the heat transfer coefficient and to provide a strong core structure.

The strips forming the sides of the hot and cold gas passages in the core are 0.10 inch thick. The two ends of the core are so designed that two manifolds can be welded to the brazed core at each side.

The tube plates separating the passages of the two fluids are 0.006 inch thick. AiResearch has considerable experience in the fabrication of 0.006-inch brazed assemblies and anticipate no difficulties in the manufacture of this component.

Type 347 stainless steel was selected as the corrosion resistant high temperature material for the heat exchanger because this type of stainless steel is stabilized to prevent the formation of carbides at the grain boundaries at elevated operating temperatures. This phenomenon would cause the deterioration of other austenitic chromium-nickel stainless steel alloys in service. The Type 347 alloy is relatively easy to braze and weld and is readily available in sheets or plates.

The core will be brazed with a commercial nickel-based brazing alloy which provides oxidation resistance and strength equal to or better than the base material. The components of the core are coated with the brazing alloy, assembled in a special brazing fixture, and then brazed in a vacuum brazing furnace.

The recuperator weight could be reduced by the use of a more compact surface which would reduce the length and increase the face area. However, more pieces would be required and more braze joints would result. In addition, the weight saving would be small compared to the overall system weight. A much less restrictive fin would sharply increase the recuperator weight and volume.

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Fig. 11. Sketch of the Recuperator

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The design parameters of the recuperator with a 90-square foot radiator are:

Inlet temperature	
Turbine outlet gas (°R)	1731
Compressor outlet gas (°R)	799
Effectiveness	0.90
Turbine outlet gas	
Inlet pressure (psia)	5.78
Pressure drop (psi)	0.16
Compressor outlet gas	
Inlet pressure (psia)	9.48
Pressure drop (psi)	0.28
Outlet mass flow (lb/hr)	698

c. Valves and ducts

The recuperator by-pass valve is a hermetically sealed, conventional, low pressure drop, butterfly-type design, mounted in a recuperator by-pass duct. The position of the butterfly is controlled by means of an electromechanical actuator incorporating dual motors for redundancy. Activation of the valve is by application of electrical pulses of constant magnitude and length whose frequency and polarity are proportional to the input signal. Mechanical stops on the butterfly and a restriction in the valve, limit the minimum and maximum flows that can pass through the valve.

The shut-down valve is a simple, electromechanically actuated, guillotine-type valve located in the compressor discharge duct. In a manner similar to the recuperator by-pass valve, the valve design incorporates dual electrical motors for redundancy. It is a hermetically sealed assembly. The valve is normally open, and with the valve blade withdrawn, presents only a small pressure loss in the loop. Application of an electrical signal to the valve shuts off the flow from the compressor which, in turn, immediately causes the system to stop operating. Limit switches on the actuator provide full-open or full-closed signals on valve position.

The heat transfer components and the CRU will be connected by ducts of stainless steel approximately 0.031 inch thick. The ducts will be welded or brazed to the components to ensure a leakproof joint.

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Fig. 12. Sketch of Radiator Heat Exchanger

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3. Heat Rejection

a. Radiator heat exchanger

The transfer of waste heat from the argon working fluid in the energy conversion subsystem to the coolant fluid of the heat rejection subsystem is accomplished through the radiator heat exchanger. This heat exchanger is of a conventional cross-counter flow design, using four passes for the coolant fluid and a single argon pass.

The radiator heat exchanger core is fabricated from stainless steel and is of plate and fin, brazed and welded construction, and incorporates a compact staggered heat transfer surface which combines high performance with light weight and reliability. A sketch of the configuration is shown in Fig. 12.

The design parameters for the radiator heat exchanger are:

Argon side

Mass flow (lb/hr) 698

Temperature (°R)

Inlet 893

Outlet 600

Pressure (psia)

Inlet 5.62

Outlet 5.47

Effectiveness 0.90

Coolant side

Mass flow (lb/hr) 167

Temperature (°R)

Inlet 580

Outlet 853

Pressure drop (psi) 1.5

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b. Radiator

Waste heat is dissipated from the system by a conventional cylindrical fin and tube-type radiator. Aluminum radiator tubes are brazed to the outer clad of the honeycomb structure, forming the skin of the service module. The tubes are oriented parallel to the longitudinal axis of the vehicle. Headers running perpendicular to these tubes are attached at each end of the tubes to both distribute and collect the coolant. Since radiating area is limited, the surface of the vehicle (which supplied the required radiator fin area) is shared by both systems. The radiator is designed such that adjacent tubes are redundant, with alternate tubes forming each radiator. Thus, two headers--one for each system--are mounted to each end of the radiator to provide the required flow direction.

The service module skin has sufficient armor to ensure the 0.9999 probability of no meteoroid puncture for the mission lifetime of 14 days. The meteoroid armor thickness required is achieved with a skin thickness of 0.050-inch aluminum combined with a 0.200-inch ID aluminum tube having a wall thickness of approximately 0.015 inch (manufacturing requirements). The meteoroid armor thickness required was calculated using Whipple data and the penetration model in Ref. 9.

The incremental radiator weight, excluding the vehicle skin, for the 90-square foot radiator is approximately 4.6 pounds for each system or 9.2 pounds total. These values, which are six pounds per system, do not change appreciably for a 120-square foot radiator.

To obtain the desired radiating surface characteristics of low absorptivity at short, ultraviolet (solar) wave lengths and high emissivity at longer, infrared wave lengths, the radiator surface will be coated with the titanium oxide material, Titanox. This coating has an absorptivity of 0.1 to 0.15 for wave lengths up to 1.0 micron and for longer wave lengths, the emissivity increases to 0.90. A second volatile coating for protection during ground handling and transport will be applied--such as Lucite which evaporates at a pressure of about 10^{-4} mm Hg. Although this coating decreases the effectiveness of the Titanox coating, it evaporates quickly, leaving the undercoat completely effective.

c. Evaporator

The evaporator is used as a heat sink to reject heat from the power system when the radiator is not fully effective; that is, during periods of system checkout, ground operation and throughout the launch and ascent phases of flight. The evaporator forms a series section of the heat rejection loop between the radiator heat exchanger outlet and the radiator inlet, and no controls are needed on the coolant side for its operation.

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The evaporator circuit consists of two evaporator loops, one for each of the two heat rejection circuits. The evaporator circuit is supplied by a single evaporant storage and expulsion system. Water was selected as the evaporant because of its high latent heat of vaporization, favorable vapor-pressure temperature characteristics, nontoxic qualities and low cost. Pressurization and expulsion of the evaporant is achieved by regulating 1000-psia nitrogen to 20 psia. Release of nitrogen pressure is achieved by an electrically actuated shutoff valve which positively seals the bottle until the seal is broken upon valve actuation.

The evaporator has two basic modes of operation; during prelaunch, and during the first six minutes of flight. During prelaunch operation, the evaporant is supplied from an external source through a ground connection, switching over to the expendable evaporant expulsion system after ground disconnect. The evaporant flow rate is 2.0 lb/min at a regulated pressure of 15 psia.

Storage of the nitrogen and the water evaporant is in corrosion resistant stainless steel spheres, with a combined weight of approximately 15 pounds (wet). Shutoff valves are provided in each of the evaporator circuits to permit individual operation as desired.

The evaporator is a plate-fin heat transfer matrix constructed of stainless steel with nickel fins, and is fabricated entirely by brazing and welding. It is a cross-counterflow configuration with two passes on the liquid coolant side and one pass on the evaporant (steam) side.

A water distribution pad at the bottom of the heat exchanger assures even water distribution. A sketch of the evaporator configuration is shown in Fig. 13.

The design working parameters are summarized.

Coolant circuit

Coolant	Monsanto fluid OS-139
Mass flow (lb/hr)	167
Temperature (°R)	
Inlet	853
Outlet	580
Pressure drop (psi)	0.5

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Fig. 13. Sketch of the Evaporator

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Evaporant circuit

Inlet temperature (°R)	530
Evaporant flow rate (lb/hr)	120

d. Coolant pumping unit

The coolant pumping unit is an integral unit, packaged in a single housing, and is functionally and structurally identical with the pumping unit currently supplied for the glycol-water cooling system of the Apollo environmental control system. This unit consists of a filter, a reservoir, a reservoir isolation valve and two magnetically coupled, electric motor-driven centrifugal pumps in parallel. A check valve is located in each pump outlet to prevent reverse flow through the nonoperating unit. All components are mounted in a single A356 aluminum housing.

The reservoir, a bellows assembly, assures a nominal inlet pressure of 10 psig, provides for temperature expansion or contraction, and supplies a nominal amount of fluid makeup for leakage. Fluid capacity of the reservoir is 35 cubic inches. In the event of excessive reservoir leakage, the shutoff valve can be operated to isolate the reservoir. This isolation valve employs a bellows to effect isolation without external leakage. Both reservoir and isolation valve bellows are manufactured from corrosion resistant stainless steel. The filter is installed upstream of the pumping unit.

Redundant pumping is provided for reliability, requiring separate checking to prevent back flow through one pump when the other is operating. Each pump is magnetically coupled, providing a drive which is completely isolated from the working fluid, thereby reducing the probability of leakage and permitting motor replacement without opening the system. The pumps are a Barski-type centrifugal design in which pumping is achieved through high speed rotation (23,000 rpm) of a straight-bladed aluminum impeller. Pump operation requires 35 watts of power to provide a 0.362-gpm fluid flow at a 30-psi minimum differential pressure. In sizing the electric motor pump combination, ample provisions have been made to provide for contingency power requirements.

4. Power Conditioning and Controls

In the design of the electrical equipment and controls, reliability was a prime consideration. Wherever practicable, redundant circuits have been employed, and where this has not been feasible, the individual elements comprising the circuits have been greatly derated. In addition, the system arrangement has been made with the objective of pro-

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viding the least possible interaction between circuits. Construction of the system components would be accomplished on a modular basis. Wherever possible, the latest techniques of welded circuitry and miniaturized packaging would be employed to provide a highly reliable, lightweight system.

Referring to the schematic diagram of the power system in Fig. 3, the equipment and components have been designed to provide a relatively simple, yet highly efficient and reliable, power conversion and control system. The method for developing the d-c output power is by direct rectification of the high frequency, three-phase voltage developed by the alternator. The method for turbine speed control of the system is by varying a parasitic load on the alternator such that the overall electrical load remains constant. A separator temperature control limits the turbine gas inlet temperature to 2000° R.

Each of the primary power conversion and control components has been duplicated to provide an alternate, or backup component in case of failure. The functions which have been duplicated are:

- (1) Frequency and voltage sensing.
- (2) Parasitic load.
- (3) Power rectification.
- (4) Overvoltage and overcurrent protection.

The changeover from a primary to a secondary component in the event of a failure is automatic and will provide only a minimum disturbance to preceding or following circuits. Redundant circuits are employed within the separate turbine temperature controls.

A description of these components and circuits is presented in Appendix B.

Frequent load demand peaks exceed the 1500-watt nominal capability of the system. Those excesses can more advantageously be satisfied by a battery system. The worst single overload occurs during earth orbit prior to translunar injection. This overload requires 1270 watt-hours augmentation from batteries. A battery capability in excess of 1500 watt-hours exists in the command module for use during re-entry. It may be possible to use these batteries to support this overload. It should be assumed, however, that this battery cannot be discharged, and it will therefore be necessary to supplement it with an additional 25 pounds of silver-zinc cells (Ref. 10) located in the service module.

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5. Shielding

The system design studied yields a dose of 13.7 rem and meets the maximum radiation design dose of 30 rem without the use of radiation shielding. In Volume 2 of this report, the various dose rate contributions are tabulated. The shielding requirements for this system to meet the minimum practical radiation design dose of 3 rem are also discussed. The total shield weight required to meet the minimum dose is 66 pounds.

6. Reliability

Reliability was a prime consideration in the design of the radioisotope-fueled closed Brayton cycle power system. Emphasis was placed on designing the system and system components with a large tolerance for off-design operation, using the minimum number of moving parts, avoiding series and pyramiding functions, utilizing simple, proved and developed hardware design concepts, and finally, operating at moderate stress levels. The basic design finally adopted is believed to be about as simple as the system can be made and still provide the required reliability. The temptation to become sophisticated and incorporate features which are otherwise desirable, but which would increase complication, has been scrupulously avoided. For example, in the process of selecting the proposed design, various possible configurations of the power system gas loop, the coolant loop and the control systems were evaluated. This included: dual CRU per gas loop, a common recuperator between gas loops, configurations where the radiator coolant loops were common with dual power system gas loops, and configurations where coolant loops could be interconnected by valves. As can be seen in Fig. 3, only the radioisotope heat source and the evaporant supply and delivery circuit are common to the two, otherwise entirely independent power generating and heat rejection loops are used.

The high reliability of this system is derived, in part, from the particular system configuration selected. This is further enhanced by the basic nature of the Brayton cycle. In this regard, the Brayton cycle power system offers the following inherent reliability advantages, largely resulting from the use of the inert, all gas phase working fluid.

- (1) All component materials exposed to the working fluid are operating in the inert argon gas environment, allowing the use of standard materials in all components. The use of argon renders the system free of corrosion, oxidation and mass transfer problems, and all heat transfer surfaces remain free of contamination. In addition, problems are eliminated which might arise from bearing or turbine erosion.

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- (2) Reliability can be demonstrated by ground testing. Since the system utilizes a single-phase gas cycle, a ground test is essentially equivalent to a zero g test environment. No special equipment is required for the gas loop start; thus, the basic loop is easily tested. Operational test of the system on the ground accurately simulates the conditions that will be encountered in space.
- (3) All equipment required for producing the proposed components is currently being used to produce similar hardware designs for numerous aircraft and manned spacecraft applications. Therefore, the fabrication of the proposed equipment fully utilizes currently available materials, tools, techniques and machinery, thus assuring a high degree of process and quality control.

The logic reflected in the power generating system design is presented herein for mission success and for safe abort (crew safety). For crew safety, it is necessary that no component failure occur anywhere in the dual loop system for the first 147 hours of the mission, and the calculation considers both power generating loops to be in series (not parallel) for this phase. The value of 147 hours is that employed by North American Aviation Space and Information Division's Apollo Project for that initial period during which an abort mission decision is made if a component failure occurs in any essential subsystem.

The failure rates employed to obtain probabilities for crew safety and mission success are all derived from other space or missile programs. Consequently, many of the failure rates employed are known to be conservative. The sole exception is the isotope heat source; this failure rate has been arbitrarily assigned as a reasonable value, but it probably is conservative too.

The failure rates assigned to the principal dynamic elements of the system, i.e., the CRU and the coolant pump packages (motor, pump, reservoir, check valve and filter), are based upon factual field and laboratory experience on delivered hardware. Specifically, the CRU failure rate is based upon a high speed continuously operated, electric motor driven, helium compressor for the Minuteman ICBM weapon system. The unit employs gas bearings and has a three-year life (schedule time between overhaul). Many such units are in service, and the presently observed mean time between failures is 485,000 hours. This experience is considered applicable to the CRU, since the latter component's primary and secondary modes of failure also are associated with gas bearings, and a comparable electrical machine, the permanent magnet alternator. It is known, from very extensive

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data, that the compressor, shaft, and turbine wheel of gas turbines contribute negligibly to the failure rates of such machines, whereas bearings account for more than 95% of observed failures.

The failure rate assigned to the coolant pump packages is compatible with that assigned to the essentially identical equipment currently being supplied by AiResearch to North American Aviation for the Apollo environmental control system.

Utilizing the failure rates for the system components and elements as tabulated in Table 2, and based upon no spares being available for any components, the initial reliability analysis for mission success and crew safety yields probabilities of 0.99844 and 0.99959, respectively. It is important to note that those elements of the system most prone to failure (electronic equipments) are also most readily removed and replaced (plug-in and pull-out). Such spares would represent standby redundancy and when reflected in the analysis, increase the probabilities for mission completion and crew safety to 0.99851 and 0.99996, respectively. Undoubtedly, the only meaningful fashion to consider the increase in these probabilities is to consider the reduction in their complements, the probabilities for failure. The full significance of carrying spares can then be adequately evaluated. Thus, the probability of mission failure is reduced from 0.00156 to 0.001487 (a negligible difference), and the probability for unsafe abort has been reduced from 0.00041 to 0.00004, an order of magnitude change which may well be considered worthwhile.

TABLE 2
Radioisotope Powered Closed Brayton Cycle Electrical
Power System Component Failure Rates

<u>Component</u>	<u>Failure Rate (λ)</u> <u>(10^{-6}/hr)</u>
Radioisotope heat source	0.02
Radioisotope heat exchanger	0.1
Recuperator	0.2
Recuperator by-pass valve	0.1
Argon ducts and fittings	0.1
Starter motor and clutch	0.1
CRU	4.0

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TABLE 2 (continued)

<u>Component</u>	<u>Failure Rate (λ)</u> <u>(10^{-6}/hr)</u>
Radiator heat exchanger	0.1
Debris trap and filter	0.1
Coolant ducts and fittings	0.1
Coolant pump and motor	25.0
Check valve	0.1
Evaporator	0.1
Radiator	0.1
Nitrogen storage tank	0.1
Pressure regulator and shutoff valve	10.0
Evaporant storage	0.1
Evaporant shutoff valve	0.1
Overvoltage sensor	25.0
Rectifier	25.0
Circuit breaker	1.0
Voltage and frequency sensor	25.0
Power system selector	5.0

7. Evaluation of the Reliability Analysis

The reliability analysis of the Brayton cycle system considers the major areas of reliability which may arise due to the operating environment of the Apollo mission. However, a great deal of additional work is required to establish actual system reliability. It is difficult to duplicate the zero g, thermal sink and space radiation characteristics, in a ground environment. Further duplication of the mechanical shock, strain and vibration forces of the launch maneuver and landing must be simulated on operational equipment. Other points which should be raised in connection with this reliability analysis are as follows:

- (1) Failure rates are based on similar equipment in other applications. There is no way to substantiate the data for a space application.

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- (2) The startup operation requires 500 watts for three minutes. This is to be supplied from the system storage batteries. Reliability of these batteries has not been factored into the calculation.
- (3) Another potential performance failure might result during shutdown or changeover of CRU. Closing the shutoff valve seems to create a large overpressure on the exhaust side of the compressor. Further reliability analysis should consider the probabilities associated with turbine reversal and strength of "plumbing" subject to overpressure, and the consequences of failure in such instances.
- (4) Another performance reliability area requiring further work is the humidity problem. Problems associated with how the evaporant coolant is disposed of other than into the vicinity of the Brayton generator must be solved. This coolant system must operate in a high g environment which presents nonuniform stress patterns on the surface of the heat exchanger. The problems associated with the reliable performance of the coolant system should be considered.
- (5) Another performance reliability problem is sealing the closed loop argon system where the entire loop is subject to shock and vibration, and to an external vacuum environment. No quantitative estimate has been made concerning the magnitude of this reliability problem.
- (6) With regard to reliability safety, the catastrophic effects of the 23,000-rpm pump when a puncture occurs in the piping or radiator should be evaluated.
- (7) The turbine in this Brayton cycle system operates at twice the rpm of the turbine in the Dowtherm-A system. The Brayton cycle estimate of CRU failure rate is about an order of magnitude less than the Dowtherm-A rate. This seems inconsistent.
- (8) While modular replacement is emphasized, there is a question of how much maintenance can really be considered during system operation. Plug in-pull out components do not appear reasonable.
- (9) A detailed part reliability analysis would be desirable. The ability of vibration and pressure sensitive equipment such as check valves, coolant reservoir bellows, pumps, circuit breakers, and relays to operate in both high g environment of boost and low g environment of space should be substantiated.

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- (10) The CRU gyroscopic effects must be studied.
- (11) Catastrophic effects may be associated with crew safety if the 45,000-rpm turbine shears a blade. The tolerances on this turbine must be extremely close. The ability of the CRU to perform adequately under launch conditions must be studied more carefully.
- (12) The problem areas mentioned are risk areas. In the event of development failures, some tradeoff analysis of substitutes necessary to compensate for deficiencies is desirable from a reliability standpoint.

It is obvious that the reliability analysis contained herein is a first attempt and, hence, should only be used for gross evaluation. A large amount of additional work must be performed to establish the actual system reliability.

D. SUMMARY OF SYSTEM CHARACTERISTICS

As compared to the conventional open Brayton cycle, the closed cycle has the advantage of allowing the turbomachinery design to be quite simple--single-stage compressor and turbine--and still achieve high performance. This is true for two reasons: (1) the use of the inert monatomic gases improves the attainable compressor and turbine efficiencies, and (2) the closed cycle operates at much lower pressure ratios than an open Brayton cycle, which results in lower aerodynamic losses and allows the use of lower tip speeds to attain high efficiencies. High cycle efficiency results from the combination of high compressor and turbine efficiencies plus the use of recuperation. Recuperation permits recovery of a large portion of the cycle waste heat, correspondingly reducing the required isotope inventory. Also, recuperation reduces the optimum compressor and turbine pressure ratio and increases the cycle through flow, leading to further increases in turbomachinery efficiency. However, since the cycle uses a low temperature radiator, adequate radiator area must be available to achieve high overall cycle efficiency. The system design parameters are detailed in Table 3 and estimated weights are given in Table 4.

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TABLE 3
Summary of System Design Parameters

	<u>Radiator Areas (ft²)</u>	
	<u>90</u>	<u>120</u>
Net electrical output power (kw)	1.5	1.5
Working fluid	Argon	Argon
Inlet temperature (°R)		
Turbine	2000	2000
Compressor	600	545
Shaft speed (rpm)	45,000	45,000
Recuperator effectiveness	0.90	0.90
Total pressure loss factor (r_t/r_c)	0.88	0.88
Compressor		
Inlet pressure (psia)	5.47	4.51
Pressure ratio	1.744	1.773
Rotor diameter (in.)	4.26	4.15
Efficiency (%)	78.3	77.5
Turbine		
Inlet pressure (psia)	8.92	7.49
Pressure ratio	1.535	1.56
Rotor diameter (in.)	5.04	5.12
Efficiency (%)	85.5	85.1
Alternator type	Permanent magnet	
Alternator rotor diameter (in.)	2.4	2.4
Alternator-rectifier efficiency (%)	88.5	88.5
System heat loss (watts)	300	260
Windage and bearing losses (watts)	200	160
Auxiliary and control power (watts)	100	100
Gross shaft power output (kw)	2.01	2.01
Gas flow rate (lb/hr)	698	575

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	<u>Radiator Areas (ft²)</u>	
	<u>90</u>	<u>120</u>
Net cycle heat input rate, end of mission (kw)	9.22	7.88
Cycle efficiency (%)		
Gas	21.78	25.5
Overall	15.75	18.43

TABLE 4

Brayton System Weight Estimate

	<u>Radiator Areas (ft²)</u>			
	<u>90</u>		<u>120</u>	
	<u>Unit (lb)</u>	<u>Redundancy (lb)</u>	<u>Unit (lb)</u>	<u>Redundancy (lb)</u>
Combined rotating unit	30		30	
Recuperator	70		56	
Recuperator by-pass valve	5		5	
Starter motor and clutch	5		5	
Ducts and miscellaneous	20		20	
Structures	<u>12.5</u>		<u>12.5</u>	
Total energy conversion system	142.5	285	128.5	257
Radiator heat exchanger (wet)	50		54.5	
Cooling pump unit (wet)	5		5	
Evaporator (wet)	17		17	

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TABLE 4 (continued)

	<u>Radiator Areas (ft²)</u>			
	<u>Unit</u> <u>(lb)</u>	<u>90</u> <u>Redundancy</u> <u>(lb)</u>	<u>Unit</u> <u>(lb)</u>	<u>120</u> <u>Redundancy</u> <u>(lb)</u>
Radiator tubes and headers (wet)	10.9		14	
Ducts and miscellaneous	<u>2.5</u>		<u>2.5</u>	
Total radiator system	85.4	170.8	93	186
Total expendable evaporant system	14	14	14	14
Frequency voltage sensor	2		2	
Parasitic load	10		10	
Circuit breaker and overvoltage sensor	0.9		0.9	
Power rectifier	2.1		2.1	
Turbine temperature control	<u>1.25</u>		<u>1.25</u>	
Total controls section	16.25	32.5	16.25	32.5
Power system selector	1.5		1.5	
Batteries	25		25	
Cables and miscellaneous	<u>5</u>		<u>5</u>	
Total power system	31.5	31.5	31.5	31.5
Capsules	14.9		14.9	
Graphite	14.8		14.8	
Beryllium	28.8		28.8	

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TABLE 4 (continued)

	<u>Radiator Areas (ft²)</u>			
	<u>90</u>		<u>120</u>	
	<u>Unit (lb)</u>	<u>Redundancy (lb)</u>	<u>Unit (lb)</u>	<u>Redundancy (lb)</u>
Heat exchanger and insulation	<u>42</u>		<u>39</u>	
Total heat source	100.5	<u>100.5</u>	97.5	<u>97.5</u>
Total unshielded weight		634.3		618.5
Total shielded weight (3 rem at command module)		700.3		684.5

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III. BRAYTON SYSTEM ADAPTABILITY

A. POWER LEVEL

The output power of the Brayton system may be decreased with the aid of the parasitic load or by allowing the working fluid to bypass the recuperator. Due to turbomachinery component matching requirements, increasing the power of a given system significantly above its rated output is not possible unless the system is overdesigned.

B. FUEL

Heat is supplied to the Brayton system by fuel capsules enclosed in a beryllium block. The capsules for the present design are fueled with polonium-210, a radioisotope with a high power density and a favorably low radiation yield. If the fuel polonium is replaced by other available radioisotopes, two design changes are apparent. Since polonium-210 has the highest power density of available isotopes, the fuel loading required by other isotopes would be greater in size and weight. This increase in fuel necessitates increasing the number or enlarging the diameter of the fuel capsules and, consequently, enlarging the dimensions of the fuel block. Increasing the fuel capsules and/or fuel block dimensions will increase the weight of the unit somewhat; however, not significantly. The overall performance and design of the system would remain the same.

A second effect produced by fuel replacement is that of shielding. If strontium-90 or cerium-144 were considered as the fuel replacement, the shielding required would be prohibitive. Plutonium-238 or promethium-147, however, would produce only small changes in the shielding requirements.

Based upon this discussion, it is apparent that strontium-90 and cerium-144 are not desirable fuel replacements, while promethium-147 and plutonium-238 are acceptable. Since the power densities of these fuels are lower than polonium-210, the fuel capsules and fuel block dimensions would be increased. These changes, however, are minor.

C. VEHICLE

The Brayton system package is compact and relatively light in weight as indicated by the conceptual design presented here and may be readily incorporated into a spacecraft with little difficulty.

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IV. DEVELOPMENT PROGRAM

A. DEVELOPMENT PLAN

The design, development, manufacture and qualification of a radioisotope-fueled closed Brayton cycle power system to meet the Apollo mission requirements can be accomplished within 32 calendar months after receipt of order as shown in the attached program schedule, Fig. 14. As illustrated, the program is broken into four basic phases of work which overlap in calendar time, but which in themselves terminate with a logical development milestone. The objectives and/or end products of these four phases of work are discussed.

1. Phase 1--Detail Design and Component Development

The objective of Phase 1 is to produce, at the end of 13 months, development-tested rotating machinery, controls, power conditioning, and heat transfer equipment designs. This phase allows sufficient time for the iterative process of making up for any deficiencies discovered in any component design. Redesign may be accomplished either by corrective action within the component itself, or by redesign elsewhere within the system, whichever is deemed most practicable.

2. Phase 2--Prototype Manufacture and Component Qualification

The intent of Phase 2 is to qualify the major elements of the system as components and/or subsystems, and to operate a single-gas loop system made up of prototype components. Operation of this prototype system will provide the necessary test data and information required to verify predicted loop performance and to establish control system compatibility.

3. Phase 3--Flight Operational System for Unmanned Missions

During Phase 3, preliminary flight rated tests will be performed on complete flight-type systems, and a prototype system will be operated on the radioisotope fuel. Several systems will be manufactured for installation and flight testing in the Apollo vehicles. These systems will be delivered in the required number of sets for unmanned vehicle launches.

4. Phase 4--Flight Operational Systems for Manned Flight

During Phase 4, support will be provided for the installation and launch of the unmanned systems. Concurrently, additional qualification and reliability testing, required above and beyond the preliminary flight rating tests, will be completed to produce a fully qualified system design. Phase 4 ends during the 32nd month, with the delivery of a system qualified for manned flight.

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*Includes incorporating development test information into design

Fig. 14. Program Plan

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*Includes fabrication and assembly of components redesigned to incorporate development test information

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Fig. 14. (continued)

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Referring to the program plan in Fig. 14, the detailed work to be accomplished in Phase 1 starts with a resolution of the statement of work. This statement of work, which is mutually agreed upon, results in inputs which define the system performance, orbit and environment. With these restraints, the system design optimization, very similar to the one conducted for this study, can be made to establish the individual component requirements. After the required component performances have been defined by this optimization study, the detailed designs may then proceed on the components. At this point in time, parallel efforts can be initiated on the turbine and compressor aerodynamic designs, and with the general sizes of these components known, installation studies can be made on gas lubricated and ball bearings to support the rotating assembly of the CRU.

Design and fabrication lead times on the system components are such that compressor and turbine maps will be established, and a match point for these components selected before fabrication of any heat transfer equipment is begun. By this technique, minor differences in the aerodynamic performance of the turbine or compressor components can be compensated for in the design of the heat transfer equipment. The program will encounter no delay, since the heat transfer components can be fabricated and delivered on a much shorter schedule than that required for the turbomachinery.

By the ninth month of Phase 1, sufficient component design and test data will be available to permit release of prototype designs for fabrication. From the 13th through the 19th month, these prototype components will undergo extensive performance and qualification tests such that prior to the start of Phase 4, all components will be fully qualified. As soon as practical during Phase 2, starting about the 16th month, laboratory tests will be started on a complete prototype system to confirm that the system components have been properly matched and will meet the performance specifications. The second objective of Phase 2 will be to operate with the flight-type system controls which, to this point, will have been tested only as subsystems or on analog simulators. All component and system tests of a prototype nature will be monitored from the reliability standpoint, such that potential reliability problems will be recognized early in the development phase, and corrective action can be taken.

It is intended that all testing through Phase 2 will be conducted using electrical heaters to simulate the isotope power input. Similarly, early testing on the system will be conducted with a laboratory cooling system to simulate the function of the space radiator. However, while not shown on the schedule, it may prove desirable to fabricate prototype radiator panels of the area and materials equivalent to that which will be available on the Apollo vehicle, but of a diameter which can be tested in a vacuum tank simulating the space thermal environment.

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During Phase 4, the remainder of the system qualification and reliability tests will be completed, and the results of the flight tests on the unmanned system will become available for evaluation of the system suitability for manned flight. A period of time is allowed after the unmanned flight tests and after the qualification tests to incorporate any changes dictated as a result of the flight program. Delivery of the manned-flight rated systems will commence during the 32nd month and will continue thereafter as dictated by the needs of the program.

Referring to the general category of anticipated development areas, it should be emphasized that nothing that is being proposed herein represents an advancement in the state of the art, and that there is no doubt that the system proposed can be successfully built. However, a certain amount of development effort is anticipated, and consequently, time has been allotted in the schedule to go through an iterative design and test process in certain areas to establish components which will meet the system requirements. The areas in which there will be significant design and/or development effort expended are:

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A brief discussion of these development areas is presented in the following paragraphs.

1. Gas Bearings

The primary design proposed for this application will be a hydrodynamic gas bearing of which three different design concepts are under test and evaluation at AiResearch. Since gas bearing design is a relatively new art, and limited operation has been accomplished under severe loads (or at zero g), very probably more than one bearing design approach will be pursued until a satisfactory solution has been achieved. To avoid any possibility of program delay, a parallel ball bearing installation will be designed for the combined rotating unit assembly. As reported in Ref. 15, a Brayton cycle combined rotating unit has been successfully operated on wick-lubricated ball bearings. Also, experience on a wide range of turbomachinery has shown that the life of such a bearing system is entirely adequate for the proposed 14-day mission, but care must be taken that there is no oil contamination of the argon working fluid.

As the primary line of approach, it is considered most profitable to spend time and development money solving the problems associated with gas bearings, since this will ultimately simplify and provide duration and growth potential in the system.

2. Aerodynamic Performance

Independent tests will be conducted on the turbine and compressor impellers to establish their performance and operational characteristics. Undoubtedly, based on these results, modifications will have to be made to these components to satisfy the system requirements. To reach the desired match point, which ensures both good operating point performance and desirable startup characteristics, these modifications will be accomplished only after the first performance maps are produced.

3. CRU Development

Care must be taken, and undoubtedly, time will be spent, in developing the combined rotating unit design so that the bearings, alternator and aerodynamic components are properly integrated to yield the most rugged, highly reliable design practicable. This involves: (1) developing the CRU assembly and balancing techniques, (2) conducting critical speed, lubrication system stress and heat transfer analyses, and (3) testing the CRU to establish the acceptable startup and shutdown procedures. It should be pointed out, however, that many of the CRU integrity problems have already been solved to a degree for a machine of this type, as was reported in Ref. 15.

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4. Heat Source Heat Exchanger Design and Fabrication

Considerations in the design and development of the heat source heat exchanger include the optimization of geometry to produce a minimum weight design. Current programs are yielding heat transfer and friction data for the range of low Reynolds numbers that will be encountered in this analytical design. However, apart from these factors, some material and fabrication problems have to be answered. They include:

- (1) Possible deterioration of the thermal conductivity of copper after 1000 to 10,000 hours at 1600° F.
- (2) Unknown change in ductility and strength characteristics of the container and braze materials after long exposure at high temperature.
- (3) Unknown creep and stress rupture properties of the welded and brazed joints at high temperatures.
- (4) Different thermal expansion coefficients of the structural materials at the expected operating temperatures.

5. Radiator Performance Testing

Since the radiator is an integral part of the Apollo space vehicle, the critical area with respect to the radiator will be to establish acceptable methods and equipments for testing. To accomplish this, it is proposed that system tests be conducted on smaller cylindrical radiators, or radiator panels, of equivalent area to that which is available on the vehicle skin. Also, it is assumed that testing in the vehicle will be conducted in conjunction with other tests on the Apollo vehicle.

6. Controls

Historically, control systems have proved to be a vexing problem in many power system developments. This is due largely to the fact that the data around which these controls must be designed does not become available until very late in the program. The rotating equipment with its high inertia-to-torque ratio should be extremely easy to control, but undoubtedly, some development effort will be required to establish startup and switchover techniques which are satisfactory for space operation.

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7. System Packaging and Vehicle Integration

The problem of system packaging and vehicle integration involves a large number of interfaces with other systems and structures. These problems are considered straightforward and not difficult to overcome. The integration problems involved in this area should be minimized by the generous amount of space available for this system.

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V. COSTS

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Radioisotope Fueled Brayton Cycle Power System

<u>Phase</u>	<u>Definition</u>	<u>Isotopic Fuel Block Cost</u>	<u>Power Conversion System Cost</u>	<u>Total</u>
I	Design and Component Development	\$1,757,000	\$ 2,773,000	\$ 4,530,000
II	Prototype Manufacture and Component Qualification	1,953,000	1,731,000	3,684,000
III	Unmanned Flight Operational Systems Manufacture and Flight Rating Tests (two systems)*	1,150,000	3,989,000	5,139,000
IV	Manned Flight Operational Systems Manufacture and Flight Rating Tests (two systems)*	<u>605,000</u>	<u>1,756,000</u>	<u>2,361,000</u>
	Total Development Costs	<u>\$5,665,000</u>	<u>\$10,249,000</u>	<u>\$15,714,006</u>

Estimated per Vehicle Set of Hardware

Isotopic Fuel Processing Cost at \$0.45/curie	\$ 189,000
Isotopic Fuel Block Cost	50,000
Power Conversion System Cost*	<u>340,000</u>
Total Cost**	<u>\$ 579,000</u>

*System includes dual conversion units with the exception of the fuel block.

**Based on quantities of 10.

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APPENDIX A

BRAYTON SYSTEM OPTIMIZATION STUDY

To realize the full potentialities of the system, an extensive optimization study was performed. The purpose of this investigation was to define the cycle operating conditions yielding minimum system weight, minimum isotope inventory or some compromise between these objectives.

A complete analysis of the system includes consideration of the effects of the following variables:

- (1) Maximum cycle temperature.
- (2) Cycle temperature ratio (compressor inlet to turbine inlet).
- (3) Radiator area and effective sink temperature.
- (4) Working fluid.
- (5) Shaft rotational speed.
- (6) Component efficiencies (compressor, turbine, alternator-rectifier).
- (7) Recuperator effectiveness.
- (8) System pressure level.
- (9) System pressure losses and distribution thereof.
- (10) Parastic auxiliary losses.

Digital computer programs were used for system optimization to calculate the turbomachinery and heat transfer component characteristics and the cycle thermodynamics. One program was used to determine compressor and turbine efficiency, cycle pressure level, optimum compressor pressure ratio, CRU dimensions, mass flow rate, cycle efficiencies, and other variables as a function of power level, gas molecular weight, recuperator effectiveness, turbine-to-compressor pressure ratio and compressor inlet temperature. The program matches empirical turbomachinery performance data with cycle thermodynamic characteristics by an iterative procedure. For each set of input variables, the optimum compressor pressure ratio corresponding to maximum cycle efficiency was determined. Thus, each cycle performance point

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Fig. A-1. Brayton Cycle System Performance

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represents a unique turbomachinery design. After defining the optimized cycle, the computer generates detailed design data for the heat transfer equipment in the system. Other individual programs are used to determine the design characteristics of radiator, radiator heat exchangers, and recuperator. Here again, the design of the heat exchangers is based on considerable empirical data. At present, there is no program available for the heat source heat exchanger.

Inert gases are favored as working fluids because of their chemical inertness and because of their favorable thermodynamic characteristics as monatomic gases. A gas with a low molecular weight, high specific heat, operating at a high pressure, is desirable for the design of the heat transfer components. For turbomachinery design, the desirable properties are high molecular weight and low specific heat, and operating pressures which result in satisfactory specific speed and Reynold's number values. Based on these divergent considerations, argon was selected as the working fluid for the cycle, giving the best overall compromise between turbomachinery performance and heat transfer component size.

One parameter which was maintained constant throughout the design optimization was the turbine inlet temperature of 2000° R. The turbine inlet temperature of 2000° R is considered a very conservative design value for the Apollo mission, and as such would provide one means of growth margin in the system design, or could compensate for uncertain design variables.

The results of the cycle optimization studies are shown in Fig. A-1. These curves show how compressor inlet temperature affects cycle efficiency, through flow rate, and optimum compressor pressure ratio for a recuperator effectiveness of 0.85 and 0.90. These are gas cycle efficiencies and do not include allowance for bearing and alternator losses or control power requirements. As shown, increased cycle efficiency requires higher recuperator effectiveness and/or lower compressor inlet temperature. Final selection of the compressor inlet temperature depends on radiator size and weight characteristics.

Figure A-2 shows system weight, radiator area and system thermal input power requirements (at the end of the mission and including heat losses from the isotope heat source and system), as a function of compressor inlet temperature. As shown, an available radiator area of 90 square feet will allow system operation at a design compressor inlet temperature of 600° R. Although, as indicated on the curve, a recuperator effectiveness of 0.85 results in a slightly lower system weight than using a recuperator with 0.90 effectiveness, the reduction in isotope power requirements, 11.7 to 9.52 kilowatts(t) was considered to be a greater advantage and, consequently, the higher effectiveness was selected for system design.

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Fig. A-2. Brayton Cycle System Performance

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If a radiator area greater than 90 square feet is available, the system weight and/or isotope power requirements can be reduced as indicated on Fig. A-2. The pertinent cycle design parameters are listed in Table 3.

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APPENDIX B

POWER CONDITIONING AND CONTROLS BRAYTON CYCLE POWER SYSTEM

1. Rectification Circuit

The function of converting the high frequency generated a-c power to 28 volts d-c is accomplished by the circuit shown in Fig. B-1. The circuit is composed of a conventional 6-phase, half-wave rectifier with an inductor input filter. Protecting the rectifiers from overload are circuit breakers and an over-voltage sensor.

a. Rectifier

The multiple-phase source was chosen to provide a d-c output with a minimum amount of a-c ripple voltage. The 6-phase, half-wave silicon rectifier configuration was selected for its high efficiency and quality of output. Voltage regulation for the no load to full load conditions will not exceed 2%. During normal operation, the load current passes through only one rectifier, which thereby increases its operating efficiency. This is an advantage. The power handling efficiency for this application has been determined to be in excess of 98% up to 100% overload condition.

To provide a high quality d-c source, the a-c ripple voltage at the output has been reduced to approximately 2% by a single-section, L-C filter. Due to the relatively high ripple frequency (9 kc), the filter components may be made extremely small and lightweight. The resistor shunting the d-c output is included to maintain the ripple at 2% even at zero load.

b. Over-voltage sensor

Over-voltage protection is provided by sensor. This sensor consists simply of an avalanche-type rectifier sensing the output voltage (dual circuits are employed for redundancy). A sustained current through this device activates the circuit breakers which, in turn, shut off the electrical power to the rectifiers.

c. Circuit breakers

The circuit breakers will be activated either by being subjected to an over-current condition or upon receipt of an over-voltage signal. These devices are conventional aerospace types which can be reset on demand.

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Fig. B-1. Rectification Circuit

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2. Parasitic Load Control Circuit

The function of controlling the turbine to constant speed is accomplished by the subsystem shown in block diagram in Fig. B-2. Referring to the diagram, the amount of power to be dissipated by the parasitic load is determined by the comparison of two input voltage signals, one signal inversely proportional to alternator output frequency, and the other directly proportional to the system rectified d-c output voltage. With the conditions of nominal d-c voltage output and generated frequency, there will be only a small amount of power dissipated in the parasitic load. When the output voltage or generator frequency increase beyond the control setting, the amount of load power to be dissipated will be increased, increasing the load on the turbine and thereby reducing the frequency and voltage to the nominal values. Conversely, when the alternator frequency or output voltage drop below their control settings, the parasitic load is reduced. For a range of power outputs from zero to 1.5 kilowatts, the turbine speed will be controlled to +1% of the design value.

A brief description of the circuits comprising the functional elements of this subsystem are as follows:

a. Frequency sensor

The simple L-C network and amplifier circuit shown in Fig. B-3 converts the generated a-c voltage into a d-c signal voltage inversely proportional to alternator output frequency.

b. Voltage sensor

The voltage sensor circuit shown in Fig. B-4 provides a d-c signal voltage to the comparison amplifier proportional to the voltage difference of the power rectifiers and that of an avalanche diode.

c. Comparison amplifier

The circuit shown in Fig. B-4 also includes the comparison amplifier. This is a conventional differential amplifier with an emitter bias supplied by a constant current source. The purpose of this amplifier is to generate an output d-c signal for the pulse generator proportional to the differential inputs from the frequency sensor and the output voltage sensor circuits.

d. Pulse generator

The d-c output signal of the comparison amplifier is converted into a variable-frequency, constant-voltage, pulsed, d-c output signal, by the circuit shown in Fig. B-5. Basically, this generator is an asym-

Fig. B-2. Parasitic Load Control System Block Diagram

Fig. B-3. Frequency Sensor

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Fig. B-4. Comparison Amplifier and Voltage Sensor

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Fig. B-5. Pulse Generator

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metrical multivibrator whose pulse lengths are controlled by a variable voltage charging source for the timing capacitors. This circuit provides gate pulse signals from 100 cps to 10 kilocycles to the CR of the parasitic power switching circuit.

e. Parasitic power switching circuit

The function of applying the parasitic load on the alternator is accomplished by the circuit shown in Fig. B-6. This consists basically of two circuits, the first being a rectification circuit essentially identical to that of the main power rectifier, and the second, an SCR switching circuit that regulates the amount of rectified power to be dissipated by the parasitic load resistor. Referring to the schematic in Fig. B-6, the circuit operation in a given time period is as follows:

- (1) SCR 1 is turned on by a gate pulse received at Gate 1 from the pulse generator (shown in Fig. B-5).
- (2) The load current, I_L , is conducted through SCR 1.
- (3) The load current, I_L , induces a voltage across inductor, L_1 .
- (4) Capacitor C charges to the voltage of Inductor, L_1 .
- (5) The receipt of a trigger pulse at Gate 2 from the pulse generator turns on SCR 2.
- (6) The charge stored in Capacitor C reverse biases SCR 1, thereby turning it off.

By variance of the time intervals between Gate 1 and Gate 1 pulses, which is determined by the combined variation in d-c output and alternator frequency, the average voltage across the parasitic load resistor, R, is varied to increase or decrease the load on the alternator and thereby maintain turbine speed control.

3. Turbine Inlet Temperature Control

The turbine inlet temperature is controlled to 2000° R by varying the amount of argon which bypasses the recuperator, which in turn varies the system thermal efficiency. Increasing the amount of recuperator bypass to reduce turbine inlet temperature below 2000° R, in turn reduces system power.

The amount of flow bypassing the recuperator is controlled by a valve located upstream of the recuperator. This valve is driven by a motor which receives its signal from a control circuit sensing turbine inlet temperature and comparing this to a reference. Any error between the sensed temperature and the reference is converted into

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Fig. B-6. Parasitic Power Switching Circuit

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pulses of constant magnitude and length whose frequency is proportional to the magnitude of the error. If the temperature exceeds 2000°R , a signal through the control causes the by-pass valve to open.

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APPENDIX C

BEARING SELECTION

The first question to be answered concerning the rotor support bearing is concerned with the type to be used, i.e., gas or ball bearings. The choice depends primarily upon the requirements of the mission and the time available for development work.

The environmental conditions of this isotope powered Brayton cycle alternator will place the turbomachine rotor, housing and sump at steady state in temperatures in the neighborhood of 340° F. Under these conditions, a wick-lubricated ball bearing with polyphenol-ether as the lubricant offers an excellent solution.

Chance Vought Aircraft Corporation has evaluated an AiResearch turbomachine, which was of nearly the same size as the proposed Brayton cycle unit, under environmental conditions very close to those of the CRU (Ref. 4). This test unit had wick-lubricated ball bearings with polyphenol-ether lubricant, bearing temperatures of 380° F, rotational speed of 63,000 rpm, and it was operated in a high nuclear radiation environment. After 128 hours of operation under these conditions, there were no measurable signs of performance deterioration. Based on this evidence, the use of wick-lubricated ball bearings with polyphenol lubrication is feasible for mission durations up to 15 days (Ref. 3).

However, since some of the current and all of the future mission requirements demand operational lifetimes greatly in excess of that which appears to be feasible with ball bearings, gas-lubricated bearings must be considered. A gas-lubricated bearing system has a reliable lifetime that is limited only by the damage done to the shaft and bushing during stopping and starting. Since, by proper material selection, thousands of stops and starts are possible with gas bearings with negligible effects, a gas-lubricated bearing system offers a higher reliability and longer life than a ball bearing support system.

The selection of the optimum gas-lubricated journal bearing design for this application must be based upon considerations of the anticipated operating conditions of the machine during launch and orbit. During the relatively short launch phase of the mission, the unit will be subjected to relatively high accelerations or g loads, while during the remainder of the flight, the unit will be free from the forces of gravity and subjected only to the gyroscopic loads associated with the possible tumbling of the spacecraft. If, during orbit, the spacecraft is stabilized so as to prevent tumbling, the bearings will be operating in a zero g environment, subjected only to the aerodynamic and magnetic loadings of the rotor supported by these bearings.

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AiResearch studies have indicated that the operation of a rotor supported on complete, self-acting, gas-lubricated journal bearings in a zero g environment would lead to bearing failure. This failure is caused by the occurrence of a self-excited whirl in the journal bearing. In a zero g environment, the self-acting forces in the lubricating film are dominant and these forces drive the journal into half-frequency self-excited whirl, which leads to bearing failure. Only journal bearings that are not vulnerable to self-excited whirl can be operated with no external load, i.e., zero g, without probable failure.

Whether a journal bearing will or will not be susceptible to self-excited whirl is dependent upon the geometrical design of the bearing.

The three basic journal bearing geometries that are inherently free from self-excited whirl are described with respect to their suitability to this application.

1. Externally Pressurized Bearings, Isolated Pad and Sixsmith Designs

Although the isolated pad and the Sixsmith externally pressurized journal bearing designs could be used to support the rotor for this application at the design speed of 45,000 rpm, they both require relatively large amounts of externally pressurized gas. The bleeding of this high pressure gas from the compressor discharge will have the same effect on the overall Brayton cycle performance as a large drop in compressor efficiency. Since the frictional power consumption of an externally pressurized journal bearing is the same order of magnitude as that of a self-acting bearing, and the self-acting bearing does not require externally pressurized gas, the externally pressurized type of journal bearing does not represent the optimum solution for this application.

2. Foil Journal Bearing

The foil-type journal bearing offers many advantages for use as the rotor support bearing for this application. The foil bearing is self-acting and requires no externally pressurized gas supply; it is free from self-excited whirl and is, therefore, suitable for use in a zero g environment. The power consumption is less than that of a similarly loaded complete journal bearing, and, in general, manufacturing difficulties are minimal. A few disadvantages, however, are associated with the foil bearing for this application. The starting torque of a foil bearing is very high, even with zero foil tension. The hydrodynamic requirement for a sinusoidal exit film thickness demands high values of tension during operation, or the sinusoidal shape will be unstable and the foil will flutter and wear. This requirement for a relatively large value of tension to prevent flutter places a loading on the bearing independent of the shaft loading and results in smaller clearances and higher power consumptions under no-load or zero g conditions. Ac-

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curate positioning of the shaft relative to stationary labyrinth, compressor and turbine seals and clearances would be very difficult under the anticipated g loads of launch, or even the relatively light loadings resulting from vehicle tumbling. Because of these disadvantages, the foil journal bearing is not considered to be suitable for this application.

3. Tilting-Pad Journal Bearing

The tilting-pad journal bearing seems to be well suited for this application. The bearing is self-acting and requires only ambient gas (at compressor inlet pressure) for lubrication. The film thickness is automatically adjusted by the pivoting pads to yield the highest possible load capacity with the lowest possible loss. The power loss is smaller than for a similar complete journal bearing. The bearing is inherently free from self-excited whirl up to rotational speeds comparable to the resonant frequency of the shoes on the gas film. Rigid, precise positioning of the shaft with respect to stationary seals is possible even under heavy shock and g loads.

Some disadvantages, however, are associated with the tilting-pad journal bearing. The first problem has to do with accurate machining and assembly of the individual shoes and pivots so that the final assembly will be representative of the theoretical design. It is believed that this problem, for this particular application, has been overcome.

The other problem usually associated with tilting-pad journal bearings is difficulty in starting. AiResearch has conducted an extensive experimental study of the effects on starting of various combinations of shaft and bushing materials, as well as coatings applied to these surfaces. As a result of this research effort, it is believed that the starting of the tilting-pad journal bearing should offer no problem.

The tilting-pad type of gas-lubricated journal bearing seems to be the best possible bearing selection for this application.

4. Methods of Analysis and Design

The hydrodynamic design and optimization of the tilting-pad journal bearings can be accomplished by means of an IBM 7090 computer analysis method by Dr. V. Castelli of the Franklin Institute (Ref. 5). Extensive experimental testing, both at AiResearch and the Franklin Institute, has proved the validity of this analysis.

The gas-lubricated thrust bearing can be designed and analyzed by means of a proper combination of two recently developed, very powerful techniques. The first (Ref. 6), which represents the analytical and experimental results of a five-year cooperative research study by seven

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companies engaged in the research and development of gas-lubricated bearings can be used to obtain the steady-state operating performance of the thrust bearings.

The second, which can be used to predict the behavior of the thrust bearing under the nonsteady-state loadings of launch, is an IBM 7094 analysis done by AiResearch as a part of the SPUR program. Extensive testing by AiResearch has proved the accuracy of these two design techniques.

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