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FEASIBILITY OF ISOTOPIC POWER FOR MANNED LUNAR MISSIONS

VOLUME 6--STIRLING CYCLE SYSTEM

May 1964

MND-3296-6

SPECIAL REREVIEW FINAL DETERMINATION	Reviewers	Class.	Date
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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

In the early application of the Stirling cycle to practical engine designs, two serious problems arose due to the necessity of transferring heat to a gas. The difficulty of transferring heat to a gas at high rates limited engine speeds and, consequently, the power output was small for a given size engine. In addition, the metal through which the heat was transferred deteriorated rapidly since its operating temperature was much higher than the already high gas temperatures. If the metal temperatures were kept low, both the efficiency and the rate of heat transferred would be reduced, thereby reducing the power output. With the present availability of more heat-resisting metals and better heat exchanger designs, the feasibility of these engines has been greatly increased.

A Stirling engine concept has been developed by N. V. Philips Research Laboratory, Eindhoven, Netherlands. The adaptation of this concept for use in space has been undertaken by the Allison Division of General Motors Corporation.

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II. STIRLING CYCLE SYSTEM

A. CYCLE DESCRIPTION

The ideal Stirling cycle is characterized by two constant temperature and two constant volume processes as shown in Fig. 1. Process A-B is an isothermal compression; B-C is a constant volume heat addition process; C-D is an isothermal expansion; and D-A is a constant volume heat rejection process.

This cycle, in its basic form with no regeneration, is less efficient than the Carnot cycle due to the transfer of heat in the constant volume processes. (The corresponding processes in the Carnot cycle are isentropic and no heat is lost.) If a regenerative arrangement is used, where the heat transferred in Process D-A is returned reversible during Process B-C, then the Stirling cycle efficiency becomes equivalent to that of the Carnot cycle.

A schematic diagram of a Stirling engine is shown in Fig. 2. The engine is composed of a cylinder, two pistons, three heat exchangers and a rhombic drive mechanism to coordinate piston motion. The operation of the engine is governed by the movement of the power piston and the displacement piston. The power piston is used to compress and expand the working fluid within the system. The displacer piston serves two functions: it shuttles the working fluid between the expansion and compression spaces during the constant volume processes; and it maintains the working fluid in the expansion and compression spaces during the isothermal process.

The actual operation of this engine is demonstrated with the aid of Fig. 3, where the piston positions at the beginning of each process are shown. The operational cycle is started with the displacer piston against the closed end of the cylinder and the power piston at the bottom of its stroke (Fig. 3a). These positions of the pistons allow the greatest amount of gas in the "cold space" and also give the maximum cylinder volume positions. The gas is compressed within the cold portion of the engine by moving the power piston toward the closed end of the cylinder while holding the displacer piston fixed (Fig. 3b). During the process, heat is removed to maintain an isothermal compression. After this compression, the displacer piston is moved toward the power piston, displacing the relatively cool compressed gas from the cold space and causing it to be heated as it passes through the regenerator and heater (Fig. 3c). At the end of this constant volume heat addition portion of the cycle, the pressure level of the gas is considerably higher than it was at the end of the compression. By moving the work and displacer pistons down together on the work stroke, as shown in Fig. 3d,

Fig. 1. Theoretical Stirling Cycle

Fig. 2. Stirling Engine Schematic

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Fig. 3. Cycle Phases

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the working medium is caused to expand in the "hot end" of the engine and heat is added to maintain an isothermal expansion.

Cycle pressure bears on the top of the power piston at all times because the ports in the cylinder are never blocked by either of the pistons. After expansion of the gas is completed, the displacer piston is returned to the closed end of the cylinder, forcing the gas from the hot space to the cold space. The energy stored in the matrix of the regenerator is available for heating the gas during reverse flow, thus completing the cycle. This overlapping leads to a reduction in output, but not in efficiency, since ideally, the heat addition and rejection can still be carried out isothermally. The actual operation of a typical Stirling engine illustrated by a pressure volume diagram is shown in Fig. 1. An actual diagram for the temperature entropy processes cannot be shown in a meaningful manner, since the processes are not steady flow, nor does any fixed mass undergo one process at a time.

B. SYSTEM DESCRIPTION

A schematic diagram describing a radioisotope fueled Stirling powered system is shown in Fig. 4. The system includes two distinct Stirling engine units employing a common heat source and space radiator. Although each system uses a common heat source and radiator, each is designed with its separate coolant loop and none of the plumbing is shared. A radioisotope is used to supply the required thermal power. The heat source is cooled by a sodium-potassium eutectoid (NaK 77). The coolant NaK flows from the heat source to a temperature control device which prevents overheating at beginning of life. Temperature control is achieved by bypassing a small portion of the total flow and directing this flow to a high temperature radiator which dissipates the excess heat to space. Having passed through the temperature control device, the NaK is circulated through the engine head, where it is cooled by the engine working fluid. Either a canned rotor pump or an electromagnetic pump can be used to circulate the NaK through the circuit.

The heat transferred to the engine working fluid is converted to the shaft power by means of the Stirling cycle. Alternators then convert the shaft power to electrical power.

A second cooling loop is provided to absorb and reject the waste heat of the cycle. This loop is also employed to dissipate heat developed by mechanical friction, generator losses and the NaK pump. The coolant selected for this loop is propylene glycol. The coolant absorbs heat from the source mentioned and transfers this heat to a low temperature radiator where it is dissipated to space.

Fig. 4. Stirling Cycle Flow Schematic

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The output of the generators is directed to a power control device which maintains a constant load by distributing the output between the payload and the parasite load. The power control provides both frequency and power control.

As indicated in the schematic diagram, Fig. 4, redundancies of the system, radiator and coolant pump are provided to increase the overall system reliability. At the start of the mission, Engine 1 will operate. If difficulties arise during the mission due to a malfunction of Engine 1, operation of Engine 2 will be initiated and Engine 1 will be shut down. If a pump malfunction arises during the operation of the second system, the pump of System 1 may be used. In addition, if one of the tubes of Radiator 2 is punctured, the radiator from System 1 may be used to continue the operation of the system.

Both redundant systems will be mounted in one bay of the service module as illustrated in Fig. 5. It is evident that this unit is quite compact and uses only a small portion of the space available.

C. SYSTEM OPERATION

To start the Stirling engine, heat is supplied until an average engine temperature of 600° F is achieved. At this point, the coolant pump is started and the engine, unloaded, is motored by d-c power until it becomes self-sustaining at approximately 500 rpm. The starter is then de-energized, the heater temperature increased and load applied to the engine. The time required to heat the engine during startup on the launch pad is approximately 15 minutes. The remaining startup sequence time is 13 minutes, giving a total initial (on the pad) startup time of 28 minutes.

During normal operation, the NaK loop of the backup system will be maintained at 600° F, to facilitate in-orbit startup. This is accomplished by a small circulation of NaK in the standby loop. Thus, the in-orbit startup time for the secondary engine is only 13 minutes, which is the latter phase of the startup discussed, excluding system heating. Table 1 summarizes the startup events and associated times.

TABLE 1
Start Sequence

<u>Event</u>	<u>Estimated Elapsed Time (min)</u>
Energize NaK and coolant pump	0
Rotate isotope slab into heat exchanger (full power position)	0+

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Fig. 5. Stirling System Installation in Service Module of Apollo Vehicle

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TABLE 1 (continued)

<u>Event</u>	<u>Estimated Elapsed Time (min)</u>
Heat NaK system to 600° F	15.0
Energize starter and engine startup (unloaded)	2.5
De-energize starter	2.6
Assume full load	8.0
Total initial startup time	28.1
In-orbit startup time (excludes system heating)	13.1

D. MAJOR SUBSYSTEMS

1. Heat Source

The heat source is formed from a rectangular beryllium block through which 12 0.75-inch holes have been drilled parallel to the surface to house the fuel capsules as illustrated in Fig. 6. The outer surface of the block is clad with 30 mils of Udimet 700 to prevent sublimation of the beryllium in a space environment and to maintain the integrity of the beryllium block. One-inch graphite strips are attached to all the 3.75-inch edges to ensure intact re-entry, preventing damage by aeroheating.

Each hole in the beryllium matrix contains two fuel capsules. As shown in Fig. 7, each fuel capsule is composed of a series of four concentric cylinders. The radioisotope fuel, 252 grams of gadolinium-polonide (90 ft² radiator) with a volume of 71 cm³, forms the center tubular section. Tantalum, with a 10-mil thickness, surrounds the outer surface of the fuel form, providing a compatible noncorrosive diffusion barrier. A thin tantalum cylinder, perforated for helium gas release, is mounted to the inner surface of the fuel form. A 90-mil molybdenum wall surrounds the tantalum, providing helium pressure buildup resistance. Haynes 25 surrounds the molybdenum to prevent it from oxidizing when the capsule is subjected to an oxygen atmosphere.

Each capsule is designed to have sufficient void volume in the center of the capsules to contain the helium gas buildup.

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Fig. 6. Fuel Block--Stirling Cycle

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Fig. 7. Fuel Capsule

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Fig. 8. Stirling System Heat Source Heat Exchanger

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Two heat exchangers are mounted parallel to the lateral surfaces of the heat source to cool the beryllium block while providing heat required by the cycle. Figure 8 shows the heat exchanger design. Heat is radiated from the lateral surfaces of the block to the parallel heat exchanger surfaces, conducted through the exchanger walls and transferred to the fluid by convection.

As shown in Fig. 8, extruded René tubes are brazed to a thin René wall. Alternate tubes provide the flow required for each Stirling cycle system while the redundant tubes serve as fins to collect the heat from the block. Headers are provided to distribute flow through the heat exchangers and to collect flow at the heat exchanger exits. Each heat exchanger has four headers (two at each end) as illustrated in Fig. 8 to provide the required redundancy.

During ground operations prior to launch, the radiating surface area provided is sufficient to dissipate the waste heat using both the free convection and radiation modes of heat transfer. During the missile ascent where aeroheat is generated, cooling will be provided by an evaporator. During this mode of operation, the excess heat will be transferred to water stored in the evaporator, causing the water to vaporize.

During the coasting portion of the trajectory, normal system operation will begin and the system will function as previously discussed. The water vapor in the evaporator will condense by radiating heat to space. The evaporator is used again during the lunar orbit where the effective radiating background temperature is greater than 400° R.

As indicated previously, if a malfunction of the first engine occurs, the standby unit will be used. If additional problems develop with the pump or radiator of the second system, the appropriate redundant components from the first system will be used.

2. Stirling Engine

The Stirling engine is made up of three major subassemblies. These are:

- (1) Head, comprised of the cylinder, NaK manifold, heater, regenerators and cooler.
- (2) Rhombic drive mechanism, including pistons, shafts, rods, cranks, seals and gears.
- (3) Crankcase, with its bearing supports, oil mist fan, filter and generator stator mounting provisions.

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The cylinder head assembly is flange-mounted to the crankcase and is retained by a row of capscrews. The cooler section consists of two rows of tubes located annularly and brazed in place at the top and bottom. The inner wall of the cooler housing section acts as the displacer piston cylinder bore. It also retains the regenerator element in position. The complete cooler assembly is fitted into the main cylinder housing and the coolant medium is sealed by two rubber O-rings.

The annular regenerator matrix consists of many layers of fine mesh wire screen. The two end plates are retained by an outer and inner cylindrical sleeve.

The main cylinder housing contains the heater tubes which are microbrazed into position. A manifold, integral with the cylinder head, directs the flow of liquid metal over the heater tubes.

The rhombic drive mechanism was selected for use in the Stirling cycle engine because:

- (1) It provides for optimum phasing of the displacer and work piston. This can be readily accomplished by selecting the proper distance between the crankshafts and length of the connecting rods, crank throws and crossheads.
- (2) There are no side loads on the pistons; therefore, enhancing seal life.
- (3) Engines with any number of cylinders can be completely balanced.
- (4) The cool section of the engine can be isolated from the hot section by the displacer piston. This permits the location of all seals in the cool section and permits considerable latitude in the selection of seal materials.

The rhombic drive employed in this engine consists of a reflector image crank and connecting rod assemblies that are offset equal distances from a vertical centerline. Each crank carries three connecting rods. The outer rods are joined to the upper crosshead and the inner rods are joined to the lower crosshead. A hollow piston shaft, made as an integral part of the work piston, is connected to the upper crosshead. A smaller shaft, passing through the work piston and its shaft, connects the displacer to the lower crosshead. Gears, doweled and bolted to the crankshafts, maintain the cranks in proper relationship to each other. Careful dimensional control is maintained by holding very close tolerances to ensure precise linear movement of the pistons.

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The entire engine-generator unit is completely sealed, requiring only external connections for heat rejection to the cooling fluid and electrical connections to the generators. The mean pressure of the working medium is 1500 psi, with a fluctuation of 500 psi above and below this mean. The mean pressure in the buffer space below the work piston is 1500 psi. The pressure in the crankcase is slightly over 1400 psi and is essentially constant.

Sealing between the working space and the buffer space is accomplished by a close fitting work piston. The pressure balance between the two spaces is accomplished by a very fine capillary tube connecting the two. The pressure in the buffer spacer under the work piston reduces bearing loads; hence, mechanical friction. Minimum buffer space pressure in the crankcase eliminates the requirement for a pump to return the leakage to the buffer space and greatly reduces the leakage past the seal.

Both the displacer and work pistons are plain pistons which operate without lubrication. Sealing of the gas at the piston-cylinder interface is accomplished by a soft metal coating on the pistons. The piston shaft seals, which prevent oil from entering the buffer spacer and permit minimum gas leakage from the buffer space into the crankcase, are made of babbitt metal. A crown spring is used to press a babbitt ring against the shaft, developing a hydrodynamic wedge which pumps oil toward the crankcase as the shaft reciprocates. Flow around the back of the babbitt ring is prevented by an O-ring.

The drive mechanism of the engine will produce a high precision balance. This is accomplished by:

- (1) Matching the weight, length, and center of gravity of the displacer rods, and the weight, length, and center of gravity of the power connecting rods.
- (2) Matching the combined weight of the displacer-rod-yoke assembly and a portion of the displacer connecting rod weight to the combined weight of the power piston-rod-yoke assembly and a portion of the power yoke weight.
- (3) Attaching counterweights to each crankshaft to dynamically balance the combined rotating and reciprocating parts.

Balance of the engine is obtained by adhering exactly to the foregoing weight and dimensional conditions. An analysis was performed which considered the effect of tolerances for the following:

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- (1) Weight measuring devices.
- (2) Manufacturing and assembly tolerances.
- (3) Dynamic balance machines.

Results of the investigation indicated that the amplitude of unrestrained vibration of the engine due to the foregoing deviations can be held to less than 0.0003-inch double amplitude. This has been confirmed by actual engine testing.

The crankcase is a simple cylinder with domed ends to withstand the high gas pressure. Bulkheads located across the case carry the main roller bearings of the crankshafts. This type of structure utilizes a box section which produces considerable rigidity to withstand distortions which would otherwise tend to bind the drive mechanism and cause high engine friction.

The alternators are both housed within one end of the crankcase and are mounted on one of the bulkheads. The engine oil cooler manifold is attached to the inside of the crankcase. The oil-helium separator is also located inside the crankcase. Liquid coolant coils are attached to the outer surface of the case to cool the alternators.

To provide a simplified lubrication system, roller bearings are used for the crankshaft and crankpin bearings. The inside of the crankcase will be designed in such a manner as to minimize the unswept volume and to eliminate all undesirable hidden pockets where oil can collect under zero gravity conditions. Short-duration zero g flight tests, performed by WADD and analyzed and tested by Allison, have shown that the amount of oil required for lubrication should be approximately 25% of the unswept volume. In addition to the agitation provided by the mechanism, the moving parts will be further lubricated by a fan integral with the 12,000-rpm alternator drive gears. The oil throwoff from the drive mechanism flows into the eye of the centrifugal impeller. By action of the impeller, the oil mist is pressurized to approximately 10 inches of water.

The oil mist leaving the impeller flows into the collector housing which contains two orifices. These jets direct the pressurized mist onto the rotating or large end of the connecting rods, thus lubricating their roller bearings through radial slots placed near the periphery of the connecting rod.

The materials selected for the major parts of the engine are shown in Table 2.

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TABLE 2
Materials Selected for Stirling Engine Design

Head assembly

Cylinder	Waspalloy
Heater tubes	Waspalloy
Regenerator matrix	316-SS screen
Cooler tubes	AMS 5572 (310)
Cooler housing	AMS 5521 (310)

Rhombic drive

Crankshafts	AMS 6260
Connecting rods	AMS 6260
Piston shafts	AMS 6415
Main and rod roller bearings	Vacuum melted steel

Pistons

Work piston	AMS 6415
Displacer piston	310 SS and Hastelloy

Crankcase

Case	Titanium
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Seals

Pistons	White metal
Shafts	Babbitt-lined bronze

3. Radiator

Waste heat is rejected from the system by a cylindrical fin and tube-type radiator. The radiator is designed as an integral part of the shell of the service module of the Apollo vehicle. As shown in Fig. 9, the tubes are mounted within the honeycomb structure which forms the skin of the service module. The tubes, extruded into the shape shown in Fig. 9, are brazed to the outer clad of the aluminum honeycomb and are oriented parallel to the longitudinal axis of the vehicle. Both the honeycomb and the tubes are made from aluminum.

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Fig. 9. Space Radiator--Stirling Cycle

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Based upon reliability considerations, two Stirling engine systems are provided. Since the radiating area is limited, the skin of the vehicle provides radiating surfaces for both systems. Duplicate tubes, spaced equidistant, are brazed to the skin which provides the required fin area. Alternate tubes form each radiator system. While flow is passing through one set (every other tube) of tubes, the other adjacent tubes remain redundant.

The service module skin has sufficient armor to ensure the 0.9999 probability of no meteoroid punctures for the mission lifetime of 14 days. The meteoroid armor thickness required was calculated using Whipples data and the penetration model in Ref. 1.

Four 0.400-inch inside diameter and 0.015-inch thick (manufacturing requirements) aluminum tubes or headers run along the inner surface of the vehicle skin, perpendicular to the longitudinal axis. They are attached at each end of the radiator tubes (0.200-inch ID and 0.015 inch thick).

Each radiator (120 ft^2) is designed to dissipate 5.41 kilowatts of thermal energy. Propylene glycol enters the radiator at 160° F and exits at 140° F . The total fluid pressure drop through the radiator and headers is 1 psi.

4. Control System

Control of the permanent magnet alternator is achieved by indirect speed control resulting from the system voltage regulation. The uniform field and the natural regulation slopes of the permanent magnet alternator are utilized to control both output voltage and engine rpm by use of the parasite load. If the load requirement is increased or the engine rpm drops, the voltage will drop, resulting in a reduction of parasite load. If the required load is decreased or the engine rpm increases, the voltage will tend to rise, resulting in a parasite load increase.

The voltage-current characteristics of the permanent magnet alternator are given in Fig. 10. It is apparent from the curves that a change in the load or engine rpm provides an adequate voltage signal to quickly change the parasite load to ensure control stability.

The control system block diagram is shown in Fig. 11 and the logic schematic is shown in Fig. 12.

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

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Fig. 10. Estimated Characteristics of Permanent Magnet Alternator

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Fig. 11. Control System Block Diagram for Permanent Magnet Alternator-Starter Assembly

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Fig. 12. Control System Logic Schematic for Permanent Magnet Alternator

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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 located in the service module (Ref. 2).

5. Shielding

The system design studies yield a dose of 16.4 rem and meet 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 to meet the minimum dose is 69 pounds.

E. SUMMARY OF SYSTEM CHARACTERISTICS

Although the basic operating principles of the Stirling cycle have been known for about 150 years, it has been only in recent years that the theory has been developed in sufficient detail to permit the design of efficient engines. Engines of various sizes and configurations have been built which have substantiated the accuracy of the output and efficiency calculations.

One of the advantages of the Stirling representative cycle is that with ideal operation, its efficiency is equal to that of a Carnot cycle operating at the same temperatures. As in all cycles, however, irreversibilities, such as heat transfer temperature differences and fluid friction, prevent the attainment of the ideal efficiency in actual practice. These losses may be minimized by proper selection of heat exchanger and drive mechanism dimensions. The selection of other variables, such as rotational speed, mean cycle pressure and temperature, is strongly related to the application, however.

The pressure level in the engine affects the output directly. The output per unit displacement is directly proportional to the mean cycle pressure except for relatively minor effects due to the change in Reynolds number affecting heat transfer and fluid friction coefficients.

A mean pressure of 1500 psi is presently being used in the Allison Model PD-67 engine without adverse effects, and it is felt that higher pressures are feasible. Higher pressures, however, would tax the external heat transfer of the coolant and NaK. A value of 1500 psi was chosen for this application.

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Fig. 13. PD-67A Stirling Engine Performance Calibration at 441 Hours Endurance

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Rotational speed has a more complicated effect on engine performance. Increasing speed will result in higher output, except that the flow loss will increase in proportion to the cube of speed. At high speed, the flow losses are very high and can be reduced only by increasing the flow area of the heat exchangers. In addition to increasing the physical size of the heat exchangers, there is an increase in the clearance volume in the engine which reduces the cycle pressure ratio and the work output. The physical size of the engine, therefore, does not decrease appreciably as rotational speed is increased. Also, the efficiency will be reduced at high speed due to the greater loss. The performance of Allison Model PD-67A as a function of speed is shown in Fig. 13. A design speed of 2400 rpm was selected for this application as a compromise between size and efficiency. The solid curve, Fig. 13, is test data and the dashed curve shows what can be obtained with a roller bearing mechanism (also based on test data). Further improvements in performance which would increase the efficiency to about 35% at 2400 rpm were indicated by the lists to be possible.

Table 3 summarizes the system parameters of the reference design.

TABLE 3
Stirling System Performance

	<u>Radiator Areas</u>	
	<u>90 square feet</u>	<u>120 square feet</u>
Working fluid	He	He
Heater temperature (° F)	1250	1250
Mean radiator temperature (° F)	200	150
Mean pressure (psi)	1500	1500
Net electrical power (kw)	1.5	1.5
Coolant pump power (kw)	0.07	0.06
Control power (kw)	0.09	0.09
Gross electrical power (kw)	1.66	1.65
Generator efficiency (%)	83	83
Shaft power (kw)	2.00	1.98
Shaft thermal efficiency (%)	27	30
Heat into engine (kw)	7.42	6.62
Heat for T/E pump (kw)	0.330	0.292
Heat in (kw)	7.750	6.912

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~~CONFIDENTIAL~~TABLE 3 (continued)

	<u>Radiator Areas</u>	
	<u>90 square feet</u>	<u>120 square feet</u>
Heat rejected (kw)	6.25	5.412
Heat loss NaK loop (kw)	0.74	0.66
Total heat required (kw)	8.490	7.752
Net efficiency (%)	17.7	19.3
Po-210 metal fuel inventory (gm)	74	68
Maximum fuel temperature (° F)		1871
Maximum fuel block surface temperature (° F)		1765

The component weights for a 1.5-kilowatt(e) Stirling cycle system are summarized in Table 4.

TABLE 4

Stirling System Weight Estimates

	<u>Radiator Areas</u>			
	<u>90 square feet</u>		<u>120 square feet</u>	
	<u>Unit (lb)</u>	<u>Redundancy (lb)</u>	<u>Unit (lb)</u>	<u>Redundancy (lb)</u>
Engine generator	90		90	
Control and start system	20		20	
NaK pump	25		25	
Coolant pump	10		10	
Liquid inventory	10		10	
Structure and piping	30		30	
Total Stirling System	185	370	185	370
Fuel capsules	13.7		13.7	
Beryllium	24.1		24.1	
Graphite	11.9		11.9	
Total Fuel Block	49.7	50	49.7	50

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TABLE 4 (continued)

	<u>Radiator Areas</u>			
	<u>90 square feet</u>		<u>120 square feet</u>	
	<u>Unit (lb)</u>	<u>Redundancy (lb)</u>	<u>Unit (lb)</u>	<u>Redundancy (lb)</u>
Structure (support and hydraulic systems)	55	55	55	55
Heat exchanger and insulation	15	15	16	16
Radiator (tubes and headers, wet)	10.9	22	14	28
Batteries	<u>25</u>	<u>25</u>	<u>25</u>	<u>25</u>
Total unshielded weight		537		544
Total shielded weight (3 rem at command module)		606		613

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III. STIRLING SYSTEM ADAPTABILITY

A. POWER LEVEL

The power level of the existing Stirling engine design may be varied by adjusting the engine speed or the operating pressure level. The output per unit displacement is proportional to the mean cycle pressure. Thus, to increase the power developed by the engine the system pressure can be increased. These higher pressures, however, impose more stringent requirements upon engine design and decrease the reliability of the system. The present mean cycle pressure used (1500 psi), however, is conservative and may be increased. If the mean cycle pressure were increased to 2000 psi, the present engine could produce approximately two kilowatts (e), however, the thermal input power would have to be increased proportionally.

Variation of rotational speed has a complex effect upon the engine performance, as previously discussed. Increasing speed increases the engine output; however, the efficiency is decreased. It seems that adjusting the engine speed may be used to somewhat change output power, but the pressure variation method is preferable.

B. FUEL

The Stirling system fuel block supplies the thermal heat required by the system. The fuel incorporated into the present design, polonium-210, has a relatively high power density and favorable radiation characteristics. If the fuel were replaced by another radioisotope, two design changes would be probable. First, since available isotope replacements have lower power densities, larger or more fuel capsules would have to be designed to house the required increase in fuel. This would increase the fuel block dimensions. Design changes of this type are minor and would not affect the system performance.

A second and possibly more significant change resulting from a change in fuel is that of the shielding requirements. As an example, if strontium-90 were used, the shielding requirements would be prohibitive. Thus, strontium-90 could not be incorporated into this design.

Typical radioisotopes which could be easily interchanged with polonium-210, while necessitating minimum shielding changes, are plutonium-238 and promethium-147. These fuels may double the fuel block dimensions; however, this is of minor consequence. The incorporation of cerium-144, as well as strontium-90, would increase the shield weight significantly and may not be feasibly interchanged for this system.

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C. VEHICLE

The Stirling system occupies relatively little volume and on this basis it is desirable for space application. Based upon the conceptual design presented, it appears that, in general, mounting the Stirling system to available booster vehicles should present no problems.

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IV. DEVELOPMENT PLANS

The development schedule required to develop a Stirling cycle power system is presented. During Phase I, Detail Design and Component Development, emphasis is placed on solving problem areas of the Stirling system. These are:

- (1) NaK loop--The liquid metal loop required for the present concept offers problems of pumping and fluid corrosiveness.
- (2) Sliding seals--Sliding seals are present in this system at the pistons and at the rods. The propensity for wear is always present. System performance suffers by gradual degradation, although catastrophic failure is not expected.

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Fig. 14. Program Plan

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V. COSTS

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Radioisotope Fueled Stirling 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	\$ 3,058,000	\$ 4,815,000
II	Prototype Manufacture and Component Qualification	1,953,000	2,058,000	4,011,000
III	Unmanned Flight Operational Systems Manufacture and Flight Rating Tests (two systems)*	1,150,000	3,000,000	4,150,000
IV	Manned Flight Operational Systems Manufacture and Flight Rating Tests (two systems)*	<u>605,000</u>	<u>2,764,000</u>	<u>3,369,000</u>
	Total Development Costs	<u>\$5,665,000</u>	<u>\$10,880,000</u>	<u>\$16,345,000</u>

Estimated per Vehicle Set of Hardware

Isotopic Fuel Processing Cost at \$0.45/curie	\$ 159,000
Isotopic Fuel Block Cost	50,000
Power Conversion System Cost*	<u>150,000</u>
Total Cost**	<u>\$ 359,000</u>

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

**Based on quantities of 10.

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