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FEASIBILITY OF ISOTOPIC POWER FOR MANNED LUNAR MISSIONS

VOLUME 8--DOWTHERM A RANKINE SYSTEM

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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

This volume describes a 1.5-kilowatt (e) mercury Rankine isotope power system for application to a manned vehicle such as Apollo. To provide the required reliability, a completely redundant power system is provided for standby use. The proposed system is based on a development program utilizing the existing mercury technology to provide a flight configuration capable of demonstrating the required endurance with shutdown and startup within three years. The development program provides for the design, fabrication and test for each of the major components and one complete power conversion system (PCS). In addition, a separate boiler development program has been included.

Both the short term and growth potential for this system have been factored into the design approach, as will be described. Thus, with little modification, such as meteorite penetration and isotope selection, the design is applicable over the range of a 400- to 2400-hour mission.

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II. MERCURY RANKINE SYSTEM

A. CYCLE DESCRIPTION

The mercury Rankine thermodynamic cycle is shown on a representative T-S (temperature-entropy) diagram in Fig. 1. An ideal cycle is depicted by the solid line while the actual cycle traversed by a hardware system is shown by the dotted line.

The cycle begins at Point 1, the entrance to the alternator, which in this system acts as a preheater. The mercury is preheated to the saturated liquid line and then vaporized at almost constant temperature in the boiler. The deviation from constant temperature or the ideal situation is caused by the pressure drop in the boiler. The saturated vapor is further heated (superheated) in the upper end of the boiler to Point 2. The mercury is then expanded through the turbine to Point 3 and the saturated vapor condition. The vapor is condensed at almost constant temperature to Point 4 and again the deviation is caused by the pressure drop in the condenser tubes. The liquid mercury is subcooled to Point 5 in a subcooler. The pump then raises the mercury temperature almost isentropically to Point 1.

B. SYSTEM DESCRIPTION

The proposed system is shown in Fig. 2. It consists of two totally redundant systems. Thus, there are two completely independent boiler coils wrapped on a single heat source. These feed two separate combined rotating units (CRU) each of which contains a pump, alternator and turbine, mounted on a single shaft in a hermetically sealed housing. The CRU exhaust into separate headers and condenser tubes. The parallel sets of condenser tubes are brazed to a common fin as shown in the detail on Fig. 2.

C. SYSTEM OPERATION

The mercury working fluid is vaporized in the boiler at a temperature of 1140° F and 408 psia. It is then superheated in the same tube and subsequently expanded through the two-stage turbine to provide shaft work. The exhaust vapor at approximately 94% vapor quality is condensed to saturated liquid in the condenser-radiator at a pressure of 4.5 psia and 570° F. This liquid is then further cooled by heating the cycle flow returning to the boiler. This preheats the mercury cycle flow which is further heated by cooling the alternator before entering the boiler to complete the cycle. Before the fluid passes through the pump, it is

NOTE:

Numbers shown indicate
points in cycle schematic of
Fig. 4.
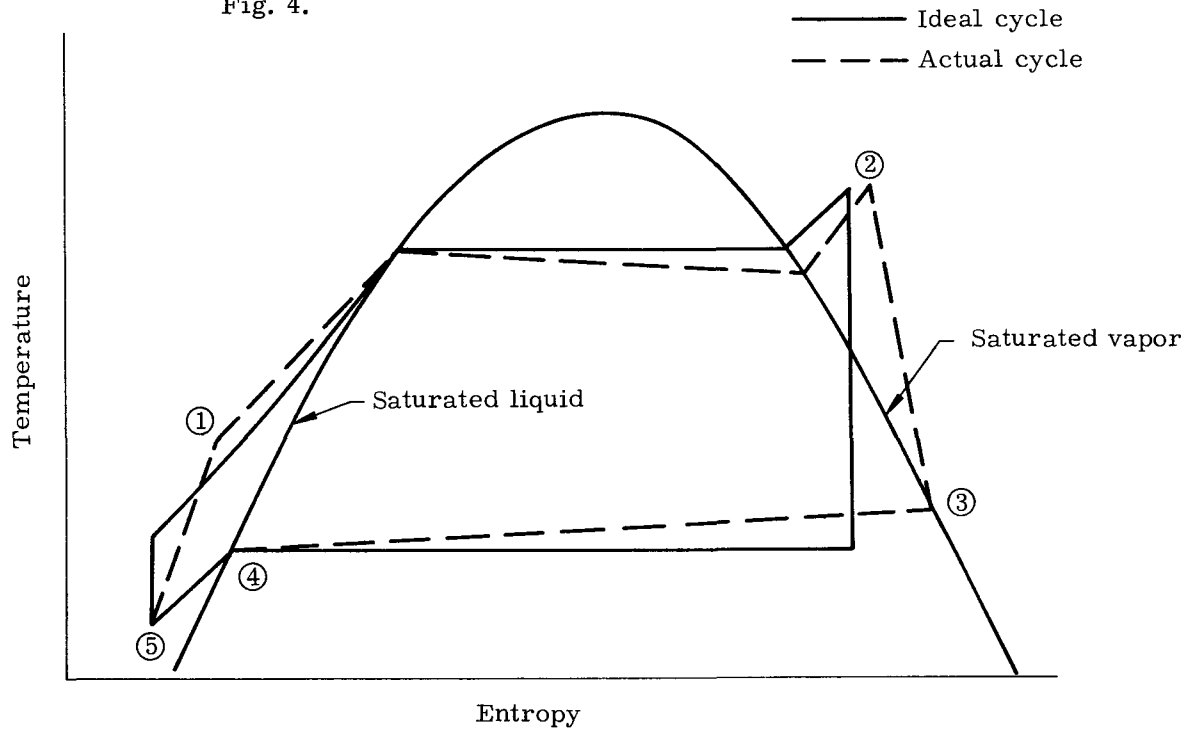
Fig. 1. Temperature-Entropy Curves--Mercury Rankine
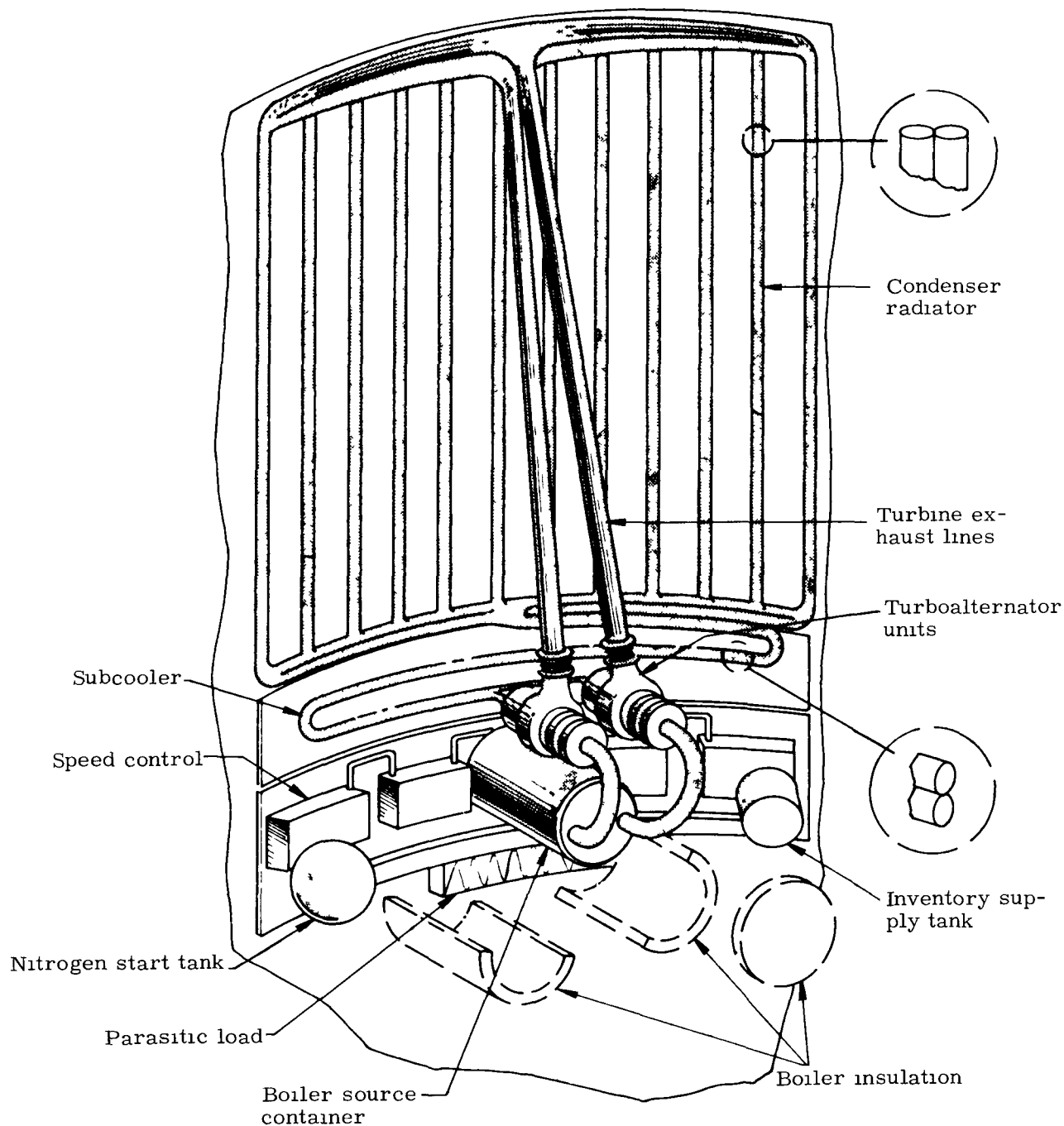
Fig. 2. 1.5-kilowatt Power System for Apollo--Mercury Rankine
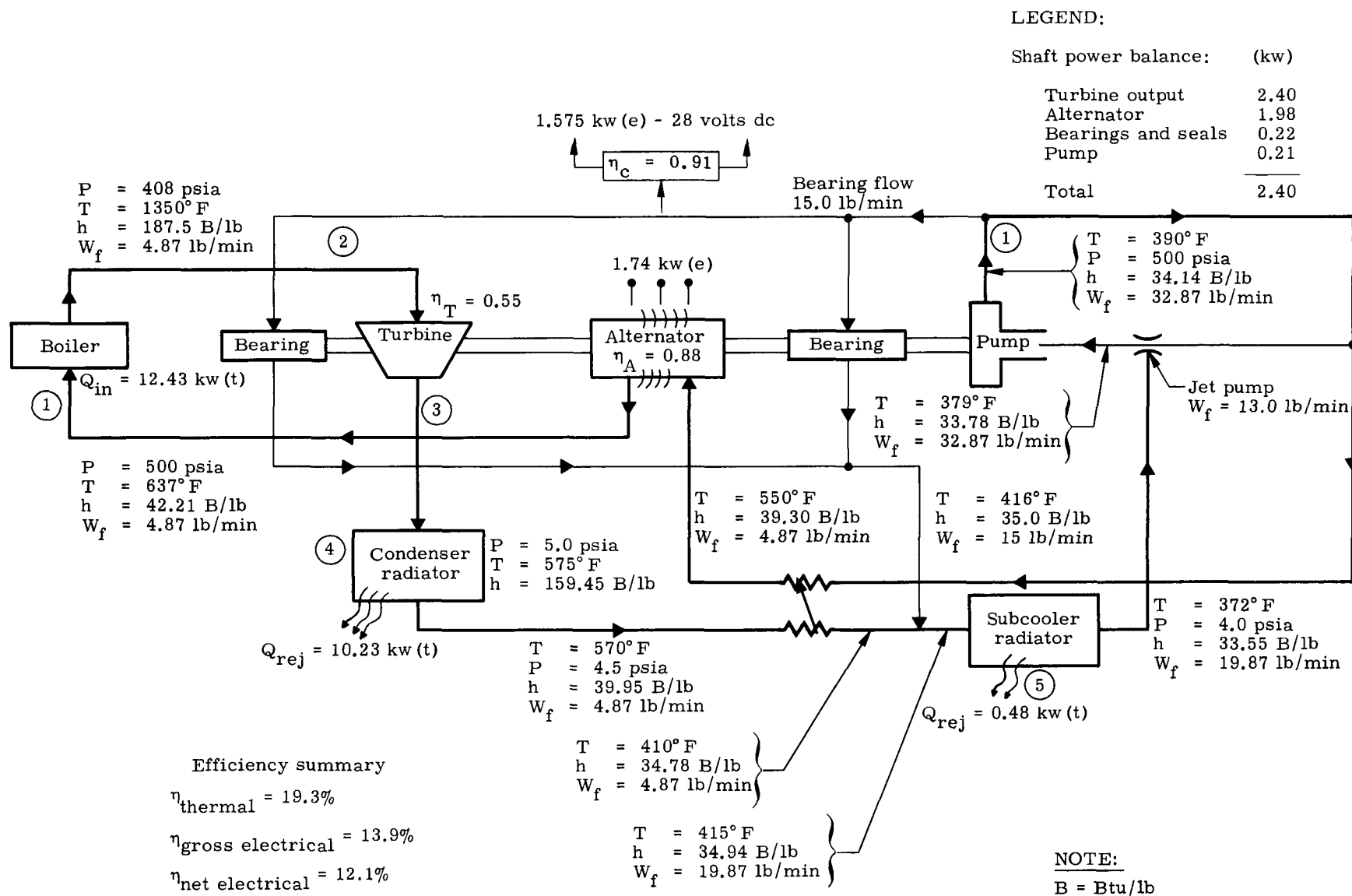
Fig. 3. Proposed 1.5-Kilowatt Mercury Rankine Schematic

LEGEND:

Shaft power balance:	(kw)
Turbine output	2.40
Alternator	1.98
Bearings and seals	0.22
Pump	0.21
Total	2.40

1.575 kw (e) - 28 volts dc

$\eta_c = 0.91$

Bearing flow 15.0 lb/min

①
T = 390° F
P = 500 psia
h = 34.14 B/lb
W_f = 32.87 lb/min

P = 408 psia
T = 1350° F
h = 187.5 B/lb
W_f = 4.87 lb/min

②

1.74 kw (e)

Boiler   Bearing   Turbine $\eta_T = 0.55$ Alternator $\eta_A = 0.88$ Bearing   Pump

Q_{in} = 12.43 kw (t)

Jet pump
W_f = 13.0 lb/min

①

③

T = 379° F
h = 33.78 B/lb
W_f = 32.87 lb/min

P = 500 psia
T = 637° F
h = 42.21 B/lb
W_f = 4.87 lb/min

T = 550° F
h = 39.30 B/lb
W_f = 4.87 lb/min

T = 416° F
h = 35.0 B/lb
W_f = 15 lb/min

④   Condenser radiator
P = 5.0 psia
T = 575° F
h = 159.45 B/lb

Subcooler radiator

T = 372° F
P = 4.0 psia
h = 33.55 B/lb
W_f = 19.87 lb/min

Q_{rej} = 10.23 kw (t)

T = 570° F
P = 4.5 psia
h = 39.95 B/lb
W_f = 4.87 lb/min

⑤

Q_{rej} = 0.48 kw (t)

Efficiency summary

$\eta_{thermal}$ = 19.3%

$\eta_{gross\ electrical}$ = 13.9%

$\eta_{net\ electrical}$ = 12.1%

T = 410° F
h = 34.78 B/lb
W_f = 4.87 lb/min

T = 415° F
h = 34.94 B/lb
W_f = 19.87 lb/min

NOTE:

B = Btu/lb

subcooled further in an additional radiator for this purpose. This lower temperature ensures that cavitation will not occur in the pump. These steps are shown schematically in Fig. 3. Also included are the bearing and jet pump flow.

The design of a dynamic mercury Rankine system must consider the source of energy and the characteristics and utilization to which power is furnished. These parameters define the envelope within which the system must operate. The state of high temperature materials technology and past experience dictate the maximum cycle temperature and pressure. Pump inlet net positive suction head requirements to avoid cavitation, condenser stability criteria and heat rejection requirements consistent with maximum cycle efficiency will dictate the minimum cycle operating temperature and pressure. Component efficiencies are determined from actual performance of similar units under test.

Control of the proposed system is achieved by providing a parasitic load which is nothing more than a resistance heater mounted to a fin and exposed to space. This unit is described in detail in the next section. Simply stated, however, the turboalternator operates at constant speed and output. If vehicle demand falls below output, a frequency sensor shunts the unused part of the output to the parasitic load. The reverse happens in the event the vehicle load is again restored. This system has the capacity to dissipate the entire alternator output if necessary and is effective in maintaining the speed of the machine within 1%. This type of parasitic speed control device has accumulated 6673 hours of operation without failure or operational difficulty of any kind.

In addition, for the extended mission capability of up to 100 days or more utilizing relatively short half-life isotopes, boiler flow can be modulated. This would permit the isotope heat source to be maintained at a constant temperature by varying turbine mass flow and, hence, power. The unit would still be maintained at constant speed. The ability of the turboalternator to operate over a range of power would thus permit the decay energy normally rejected to space to be utilized.

D. MAJOR SUBSYSTEMS

1. Heat Source

Extensive corrosion loop operating experience with mercury has demonstrated that high nickel alloys of the stainless steel type are the most resistant to mercury corrosion, penetration and mass transport phenomena. In particular, Haynes 25 has been shown to be well suited

Reflectors

6 in.

Boiler coil fin

Boiler coil 1
Boiler coil 2

Quartz lamps--1 kw

10-1/2 in.

Isotope source cavity
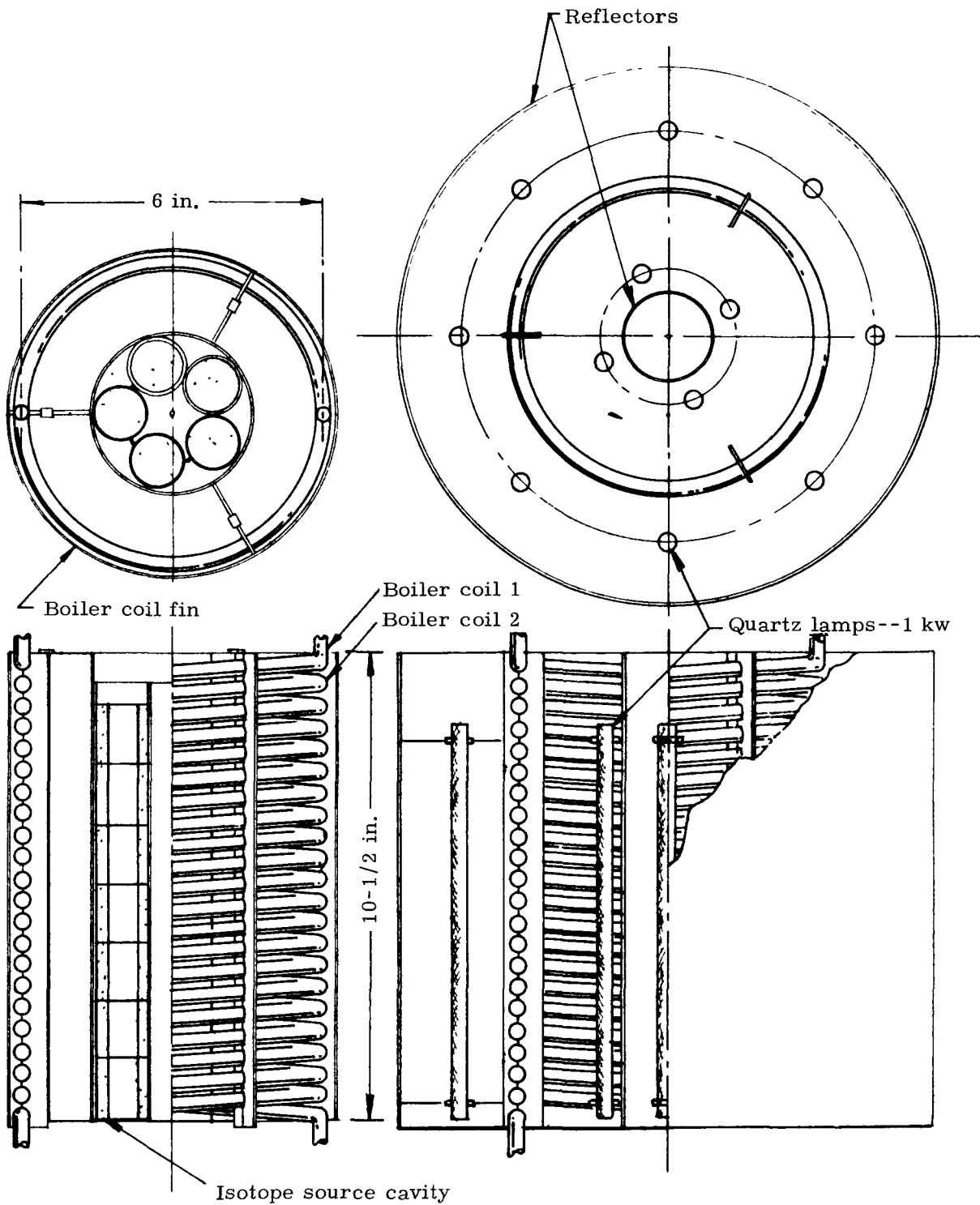
a. Radiation PCS Boiler

b. Radiation Test Boiler

Fig. 4. Radiation Test Boiler

as a material for containing boiling mercury at high temperatures and pressures. It has been determined that Haynes 25 has a wall penetration of eight mils and a corrosion product generation of 125 mg/in.^{2} for 2500 hours at 1100° F.

The boiler tubing will be fabricated from Haynes 25; it will be 17 feet long and have a 5/16-inch outside diameter with a 35-mil wall. The tubing will be wrapped in a helical coil that has a six-inch mean diameter and is 10.5 inches long. The tubing interior will contain a second wire-wrapped helix similar to that used in SNAP 1, thus causing the mercury to spiral in two directions while traversing the boiler. This creates a high g load on the mercury particles, ensuring little or no boiling difficulties in a zero g field.

Two types of boilers were considered for this study: a radiation type which would have to be developed and a conduction type such as is used on SNAP 2. The radiation type is lower weight but causes the heat source to operate at higher temperature than the conduction type.

a. Radiation-type boiler

The heat source is contained in an inner cylinder three inches in diameter and 10 inches long. There are three fin-type supports attached parallel to the axis. The isotope fin support and the coil strap support provide a rail-type track and flange to accommodate loading of the isotope. A positioning stop is provided to ensure accurate location of the isotope with respect to the coil. The outside of the coil is finned with a thin metal cylinder to enhance radiative heat transfer by increasing the incident area. The radiation boiler is shown in Fig. 4a.

The test version of this boiler will use 12 quartz glass heaters as the heat source. The heaters have an outside diameter of 3/8 inch and are 9.5 inches long, each has a power output of one kilowatt and are capable of operating temperatures of 3000° to 4000° F. These heaters have been previously used. Performance has been successful and highly satisfactory at power levels up to 120 kilowatts for 200 hours. Four heaters are positioned inside the coil and eight outside. Inner and outer reflecting surfaces are provided to direct the maximum heat flux from the heaters toward the boiler coil. The test boiler is shown in Fig. 4b.

Total boiler weight during flight is estimated to be approximately seven pounds, exclusive of the heat source.

b. Conduction-type boiler

In this configuration, the boiler tubes are simultaneously wound and seated in spiral grooves machined on the outside diameter of a thick-
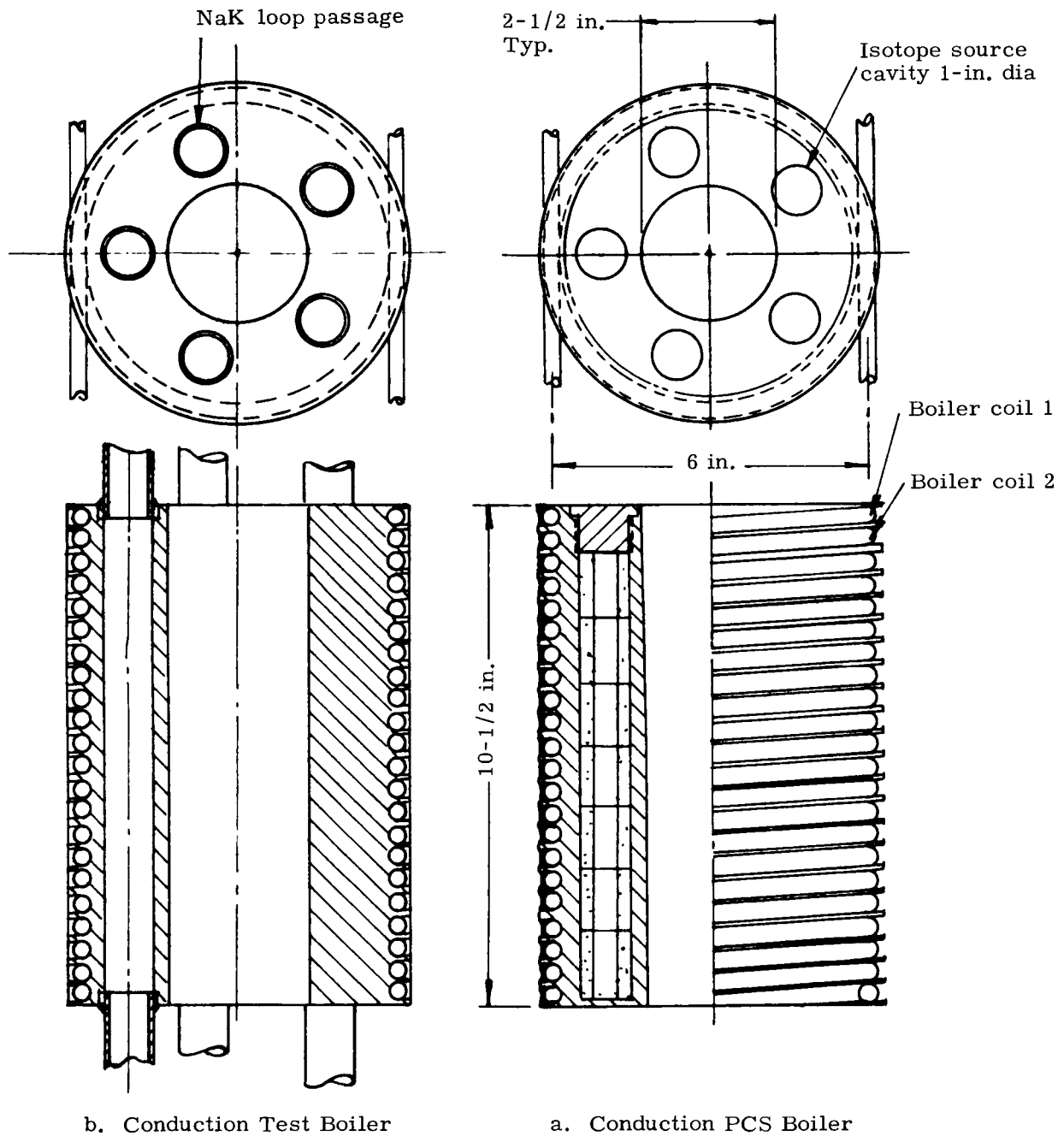
b. Conduction Test Boiler   a. Conduction PCS Boiler

Fig. 5. Conduction Test Boiler

walled nickel cylinder. The cylinder has a 6-3/8-inch outside diameter, 2-1/2-inch inside diameter and is 10.5 inches long. Five heat source rods, each with a one-inch outside diameter, are inserted into equally spaced holes bored through the length of the cylinder on a four-inch diameter bolt circle. Once formed, the coil-cylinder assembly is annealed for stress relief and brazed to provide a heat conduction path from the cylinder to the tube. Tests on this type unit have shown the contact resistance to be negligible. A conservative value of a contact coefficient in the proposed design expressed as h = 1000 Btu/hr-ft^2-°F has been assumed. This boiler configuration is shown in Fig. 5a. Estimated weight, exclusive of heat source, is 65 pounds.

It is proposed to circulate a coolant through the standby tube coil to dissipate heat while the five sources are individually loaded into the cylinder. An alternate loading scheme is to divide the thick-walled cylinder into two concentric parts. The sources would be loaded into the inner cylinder and the subassembly inserted into the outer cylinder. This system eliminates the need for the auxiliary coolant while loading.

The test version of this boiler has tube extensions provided at the inlet and outlet of each source hole through which NaK will be circulated to simulate the heat source. The test configuration is shown in Fig. 5b.

Almost any isotope fuel possessing a melting point in excess of 1800° F and power density above 5 watts/cm^3 will suffice for either type of boiler. Therefore, the proposed fuel, GdPo, is more than adequate in both instances. However, the fuel capsule considered is not adequate for intact re-entry or abort integrity.

2. Energy Conversion

a. Combined rotating unit (CRU)

The CRU (combined rotating unit), depicted in Fig. 6, consists of a pump, alternator and turbine mounted on a single shaft supported by one set of mercury-lubricated bearings. Contact seals are eliminated by vapor-liquid interfaces controlled by screw pump-type seals. A mercury vapor atmosphere is maintained in the alternator and turbine sections, thereby minimizing windage losses. The mercury vapor environment is established by heating the alternator coolant to a temperature that ensures a mercury vapor temperature in the rotor cavity higher than the saturation temperature. The CRU is hermetically sealed.

This packaging concept has been used in both the Sunflower and SNAP 2 programs. The proposed package design will be similar to the Sunflower unit. The Sunflower rotating unit has demonstrated over 3556 hours of continuous operation at 40,000 rpm.

Journal bearing

Thrust bearing

Centrifugal pump

Jet pump

Turbine

Turbine inlet scroll

High pressure labyrinth seal

Journal bearing

Screw pump seal

Screw pump seal

Alternator

Exhaust scroll
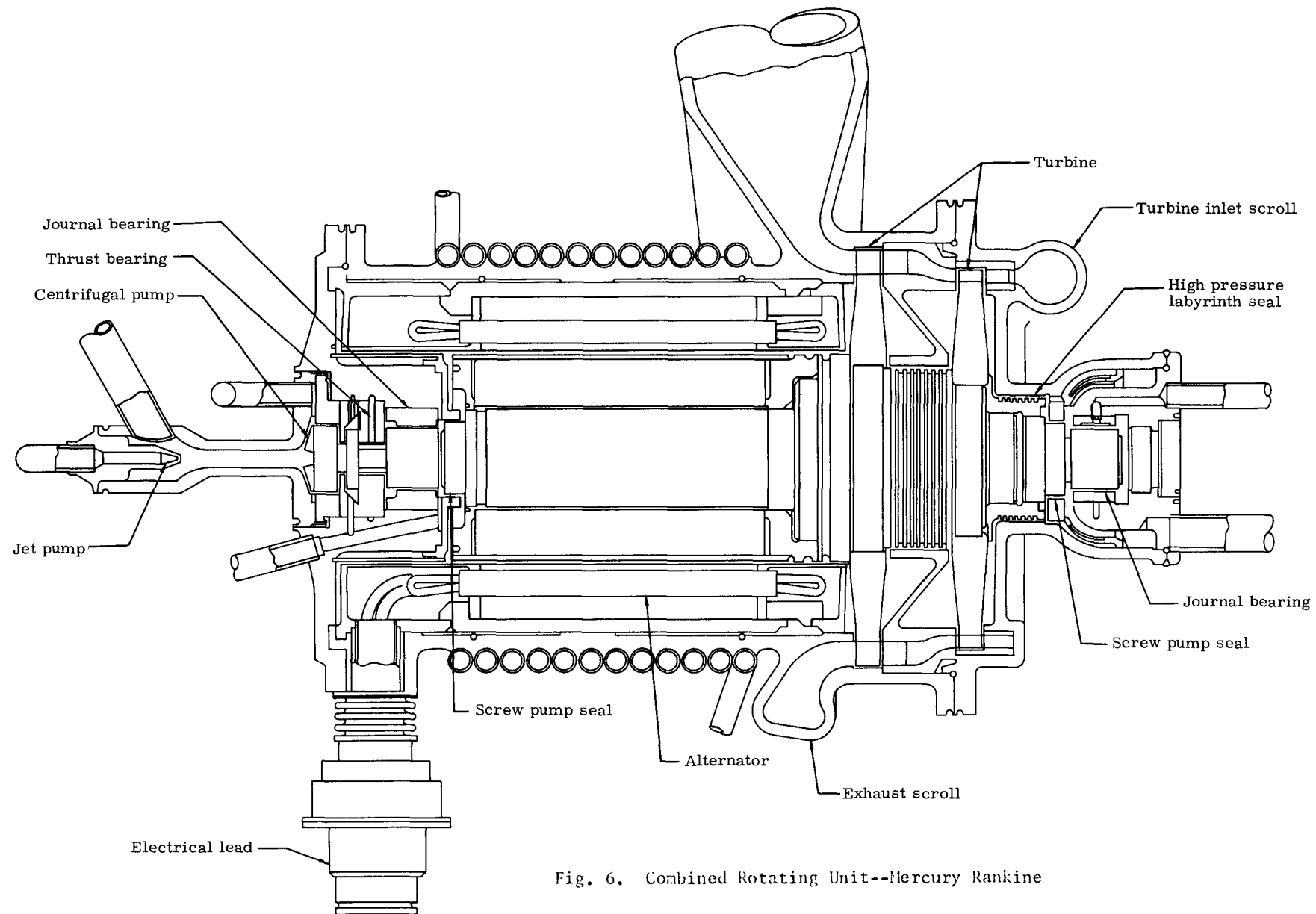
Electrical lead

Fig. 6. Combined Rotating Unit--Mercury Rankine

The proposed CRU has a design speed of 24,000 rpm. The alternator is designed to provide electrical power at 1200 cps. The estimated flight configuration weight is 35 pounds.

b. Bearings

The bearings selected are one double-acting spiral groove thrust bearing to absorb axial thrust loads and two three-sector journal bearings to absorb launch and normal operational rotating and radial loads. The bearing loads are supported by a hydrodynamic film of the working fluid, liquid mercury. This bearing combination has demonstrated long life and high reliability in over 6500 hours of accumulated operation with no apparent wear. This time includes the 3556-hour continuous run mentioned previously. The bearings selected for the proposed design are geometrically similar to the Sunflower bearings. Parametric test data show that the capacity at 24,000 rpm is adequate to carry the shaft through typical launch environmental loads and consume only 210 watts of power. The bearing flow of 15 lb/min is divided equally between the three bearings.

c. Turbine

The turbine selected is a two-stage partial admission axial flow impulse type. An optimization study of this low specific speed turbine resulted in the requirement that both stages be partial admission; 6% admission in the first stage and 10% in the second stage.

Pertinent design data are:

Rotational speed (rpm)	24,000
Wheel tip diameter (in.)	
First stage	3.8
Second stage	4.26
Blade number	
First stage	107
Second stage	111
Efficiency (%)	55

A separable high precision nozzle blading will be used to obtain maximum nozzle efficiency. The rotors will be one-piece construction with the blades cut by electrical discharge machining.

Fig. 7. Alternator Efficiency--0.95 Power Factor

d. Alternator

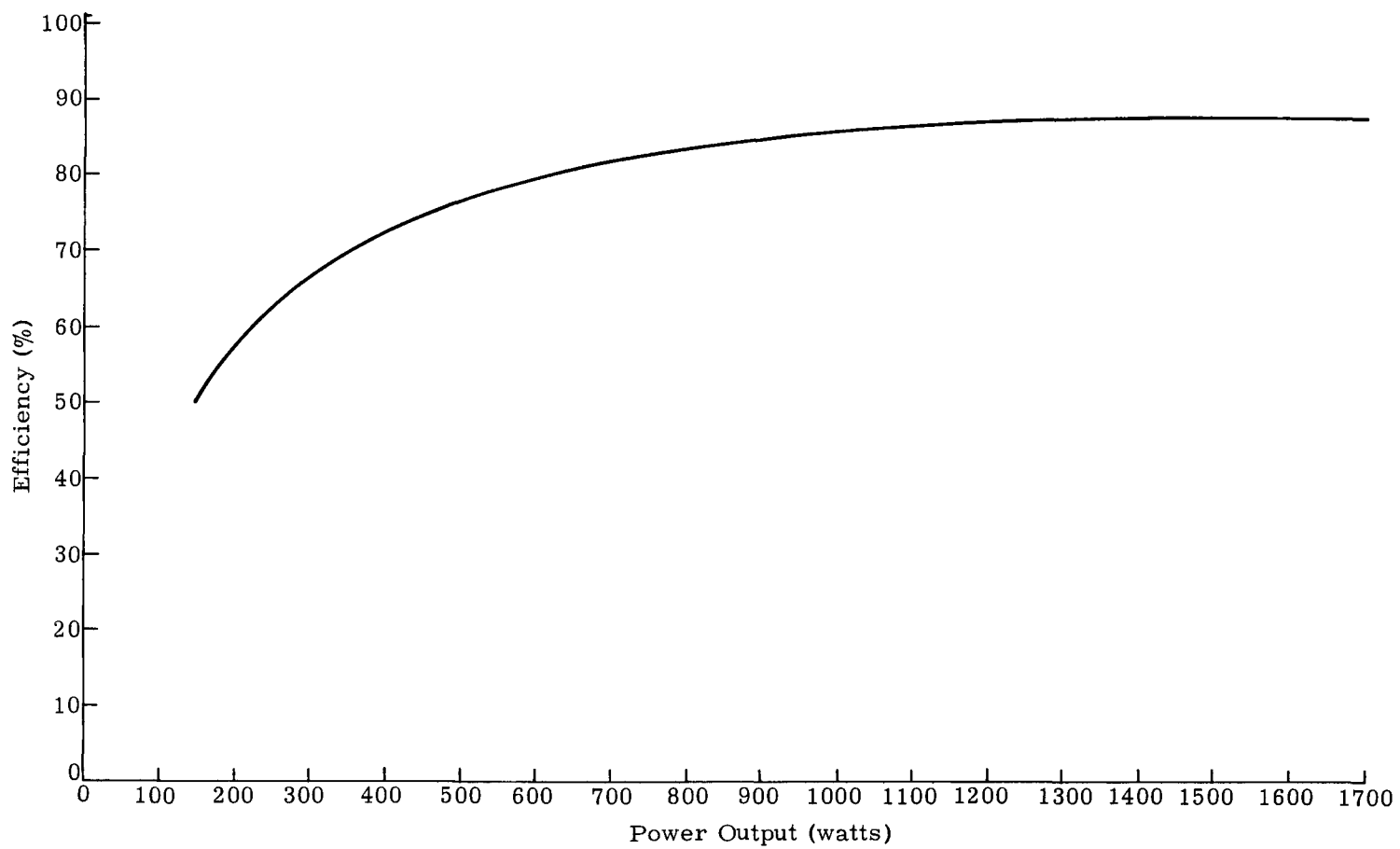The alternator will be designed at the required vehicle load voltage level to provide 28 volts direct current at the output terminals of the rectifier system. With this approach, the need for a transformer between the alternator and rectifier will be eliminated, resulting in a weight savings of approximately 15 pounds. Since the quality of power for the a-c requirements has not been defined, it has been assumed that 400-cps aircraft-quality power will be needed. As a result of this assumption, the total alternator output will be rectified to direct current and power conditioned as required. However, if 1200-cps power can be used directly for some of the a-c requirements (such as heaters, etc.), then the total system power requirements can be decreased. The voltage regulation requirements of ±5% will be attained through the following system design. The speed control will hold the CRU within a speed range of ±1% by maintaining a constant load on the alternator output. Therefore, even though the power requirements vary during the mission, the alternator will be generating constant output which will be divided by the speed control between useful and parasitic load. Since the speed and power are maintained constant, the voltage will vary due to power factor and rectifier system regulation. The regulation of the rectifier system is virtually constant over a wide load range, so the power factor range is the controlling requirement factor. Since the system load will be entirely direct current, the power factor range of 0.95 per unit is realistic. Thus, the regulation limits of ±5% can be obtained.

The alternator will be the same basic design that has been proven and previously operated for over 4329 hours. The stator will be wound with a conventional three-phase winding. Materials and design technology that has been accumulated over the last seven years on such space power programs as Sunflower, SNAP 1 and 2 will be utilized. The winding will consist of two conductors per slot to provide the terminal voltage requirements. The present Sunflower alternator design will be modified by reducing the number of stator slots to 27. The change will provide short-circuit stabilized 22 volts line to line, 1200 cps at 24,000 rpm, with the alternator operating at 700° F. Since all or nearly all of the load will utilize rectified direct current, the power factor of the load has been assumed to be 0.95 lag minimum. The efficiency of the alternator at 1700 watts and 0.95 will be 88%. A curve of alternator efficiency is presented in Fig. 7.

The rotor will consist of a six-pole Alnico V permanent magnet enclosed in a magnetic shroud ring that will keep the magnet in compression during the operating life. The permanent magnet rotor is self-exciting, eliminating requirements for field-flashing or exciter-voltage regulator circuitry. The rotor will be short-circuit stabilized and thereby eliminate the need for short-circuit stabilizing capacitors.

Power load   Load

Speed
control

Parasitic
load

75 watts

Frequency
sensing

D-C
utilization
equipment

Turbine
2400 rpm

Alternator
6 pole

22 v 1-1
1200 cps
1740 watts

3-phase
full wave
rectifier

28 v dc
5% ripple
1500 watts

A-C power
conditions

Optional-
13/22 v ac
1200 cps

A-C
utilization
equipment
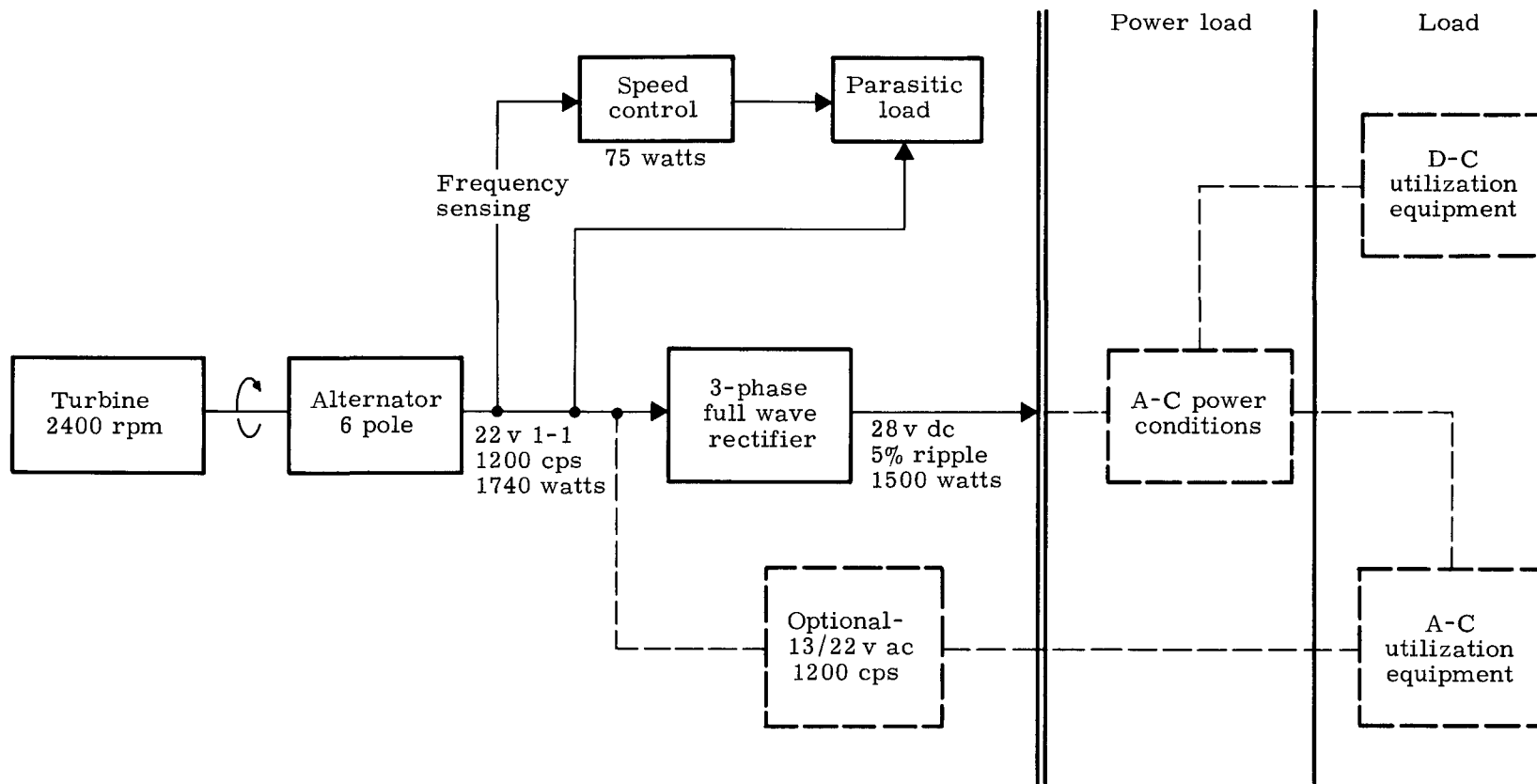
Fig. 8. Electrical Layout for 1.5-Kilowatt Power Supply

Figure 8 is a block diagram showing schematically the configuration of the control and conditioning circuitry.

e. Mercury pump

A low specific speed (N_s = 400), centrifugal impeller pump supplied with adequate NPSH (approximately four feet) by a jet pump has been selected. This pump combination has been used in systems demonstrating successful operation in excess of 4000 hours.

The jet pump is required to raise the NPSH available from the system (0.65 foot) to the impeller requirement (4.0 feet). The impeller will be mounted directly on the CRU shaft which sets the pump speed at 24,000 rpm. The direct shaft mounting eliminates the need for extra seals and bearings, and their associated losses are avoided.

The pertinent pump design parameters are presented in Table 1.

TABLE 1

Mercury Pump Parameters

Fluid	Liquid mercury ᶜ
Pressure (psia)	
Output	500
Inlet	4.0
Inlet temperature (°F)	390
System flow (lb/min)	20
Jet pump driving flow (lb/min)	13
Impeller--outside diameter (in.)	0.71
Impeller efficiency (%)	30
Pump brake power (watts)	220
Life (hr)	2500
N_s--specific speed	400
S_s--impeller	4300
Impeller--net positive suction head (ft)	4

3. Heat Rejection

The condenser-radiator configuration consists of a rectangular flat plate fin with 12 tapered tubes (see Fig. 2). The tubes are brazed to the undersurface so that the flat plate fin, which actually faces the outside, affords added meteorite protection. Statistical analysis of meteoroid penetration probability has been brought into the tube wall thickness calculation so that the total metal thickness of tube and fin affords adequate meteoroid protection. Meteoroid protection has been considered for a 0.9999% probability of zero penetration in 400 hours. The tube material is Haynes 25 and the fin is aluminum 0.050 inch thick (service module skin). The condenser-subcooler design is sized so that the vapor-liquid interface is carried entirely in the subcooler.

Experience with this type of configuration indicates that stable operation will be achieved in a zero-gravity environment. This multitube condenser-subcooler configuration has demonstrated stable operation in the Sunflower program under a stringent requirement of a negative one g environment.

A summary of the thermal characteristics is as follows:

	Condenser	Subcooler
Area (ft^2)	21.8	3.9
Temperatures (°F)		
Inlet	575	415
Exit	570	372
Heat load (Btu/hr)	34,126	2750

The assumed sink temperature was 400° R.

4. Startup and Control

The operation sequence begins with the loading of the isotope. When the boiler has come up to operating temperature, the start system is energized. This startup system would be capable of ground or orbital start and would employ a technique presently utilized for the SNAP 2 system where the mercury inventory is contained in a nitrogen pressurized reservoir. Each system would have a separate start unit. At the start signal, the mercury is injected into the system to provide pressure at the inlet to the boiler and to the bearings of the turboalternator.

Within 30 seconds, boiler pressure at the inlet to the turbine is suffi-cient to rotate the unit at design speed and electrical output. Should failure of any part of the system occur, automatic startup of the second unit would be accomplished.

The speed control employs the concept of shaft speed control by electrically loading the alternator. The controller, as shown in Fig. 9, consists of a frequency discriminator, a magnetic preamplifier, and three power amplifiers with individual parasitic loads.

The discriminator senses the frequency of the alternator voltage and gives a d-c output proportional to the frequency error from the desired nominal. This signal is amplified and used to control three power amplifiers which in turn control the current permitted to flow in the parasitic load resistors. The single phase, full-wave magnetic amplifiers have individual loads which permit current control over the full half cycle of voltage.

The components will be mounted on the vehicle skin and thus radiate their internal heat losses to space.

Operating the controller from 22 volts line to line at 1200 cps, the weight to control 1.5 kilowatts would be approximately 14 pounds.

The use of silicon-controlled rectifiers for the power amplifiers would decrease the package weight but would also decrease the re-liability and possibly increase the harmonic content of the a-c voltage. The silicon-controlled rectifiers weigh approximately 10 pounds at 1.5 kilowatts.

The weight established for a three-phase, full-wave rectifier bridge to give 1000 watts at 28 volts direct current would be approximately one pound, not including the heat sink (vehicle skin). This will result in 4.2% ripple.

Frequent load demand peaks exceed the 1500-watt nominal capabil-ity of the system. Those excesses can more advantageously be satis-fied 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 over-load. It should be assumed, however, that this battery cannot be dis-charged, and it will therefore be necessary to supplement it with an additional 25 pounds of silver-zinc cells* located in the service module.

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*Survey of Electrical Batteries, Electro-Technology, June 1963.
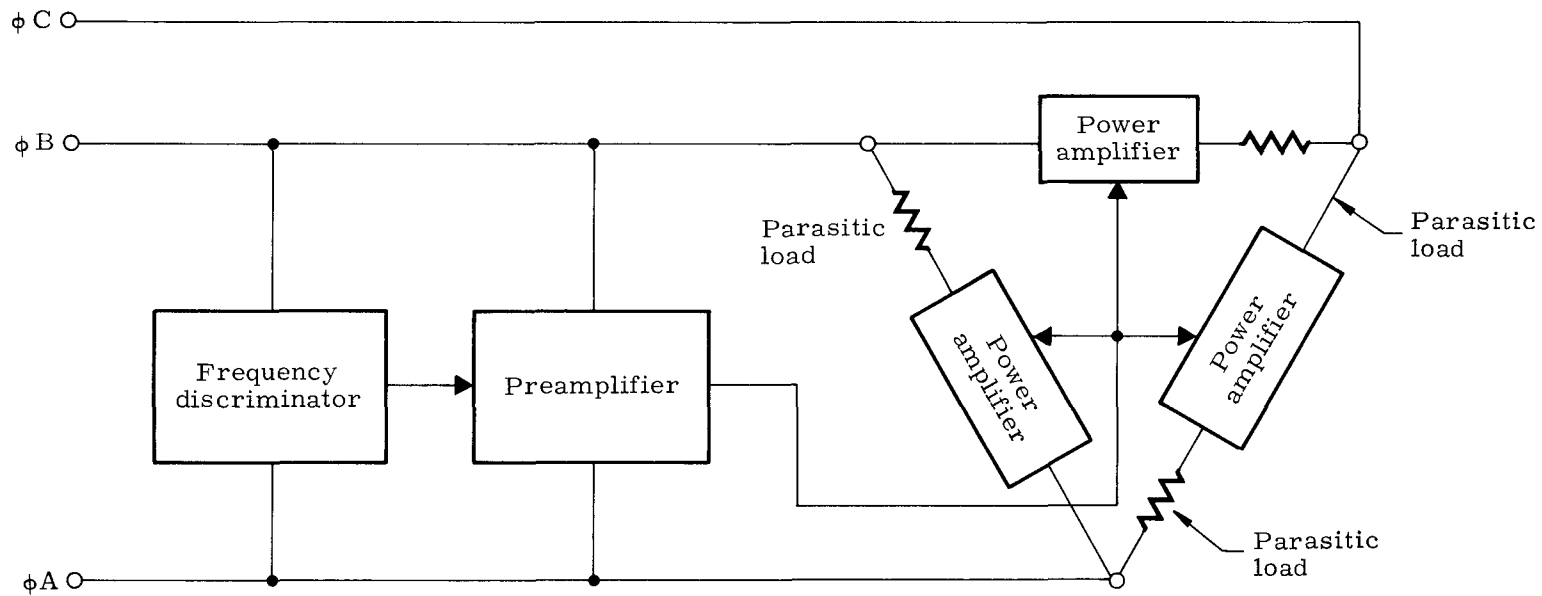
Fig. 9. Speed Control Block Diagram

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 to meet the minimum dose is 73 pounds.

E. SUMMARY OF SYSTEM CHARACTERISTICS

A summary of the system characteristics is shown in Table 2. The proposed hardware for this system takes advantage of thousands of hours of development and hardware test experience provided on the SNAP 1, 2 and Sunflower programs. The turboalternator, for example, has demonstrated a 3556-hour run without interruption or power decay. Similarly, the start system proposed has been tested by 20 consecutive start-stop cycles without damage or performance degradation to any of the prototype hardware.

A summary of the system weight is given in Table 3.

TABLE 2

System Performance Summary

Boiling temperature (°F)	1140
Boiling pressure (psia)	408
Turbine inlet temperature (°F)	1350
Turbine inlet pressure (psia)	408
Condensing temperature (°F)	575
Condensing pressure (psia)	5
Subcooler exit temperature (°F)	372
Flow rate (lb/min)	
System cycle	4.87
Bearing	15.00
Jet boost pump	13.00
Efficiency (%)	
Cycle thermal	19.3
Cycle gross electrical	13.9
Cycle net electrical	12.1
Turbine	55
Turbine type	Axial flow, partial admission impulse
Stages (No.)	2
Alternator type and efficiency	Permanent magnet, brushless--88%
Bearings	Three sector journals and spiral thrust
Pump type and efficiency	Jet boosted, low specific speed, centrifugal impeller--30%
Speed (rpm)	24,000

TABLE 2 (continued)

Output power	Three phase
Frequency (cycles)	1200
Voltage (volts)	28
Speed control	Parasitic load
Control power required (watts)	75
Gross electrical output (kw (e))	1.74
Power conditioning efficiency	0.91
Net electrical output (kw (e))	1.575 dc
Areas (ft^2)	
Condenser	21.8
Subcooler	3.86
Parasitic radiator	0.456

TABLE 3

System Weight Summary

Component	Weight (lb)	Total Weight-- Redundant System (lb)
Turboalternator	35	70
Condenser-radiator	3.8	7.6
Speed control*	14	14
Subcooler	1	2
Power conditioning*	15	15
Mercury inventory	20	20
Boiler (radiation)	5	7
Insulation and miscellaneous plumbing	40	80

*It is assumed that redundancy can be built into these units and only one of each would be required.

TABLE 3 (continued)

Component	Weight (lb)	Total Weight-- Redundant System (lb)
Startup auxiliaries	50	100
Batteries (if needed)	25	25
Total--unshielded		340.6
Shield weight--to 3 rem		73
Total--shielded		413.6

III. SYSTEM ADAPTABILITY

The design criteria used in the 1.5-kilowatt (e) mercury Rankine isotope power system for Apollo has incorporated a growth potential from the 14-day lunar mission to the extended 100-day Apollo mission.

The design is effected in two basic areas when considering even longer mission times. These are corrosion and meteoroid penetration.

The corrosion aspect was given principal consideration in selecting the boiling temperature. As previously stated, the corrosion rate can be predicted on the basis of existing data. On the basis of actual experience in test rigs, the mission would have to be extended over 100,000 hours to duplicate this corrosion product accumulation. This corrosion limitation is well within reason when compared with the 116,000 hours accumulated on the South Meadow Mercury Turbine without maintenance. This was a binary central station power generating system. In addition, the same design techniques applicable to the SNAP 2 and Sunflower systems have been used here. The design life objective of these systems is 10,000 hours.

As the length of time of exposure for the system to the space environment is increased, meteoroid protection requirements will increase if the same penetration probability is to be maintained.

Since the design is affected only on tube armor, no system change is required and performance remains the same. The relatively high radiator temperature keeps vulnerable and total areas to a minimum which simplifies integration, minimizes weight and, in the case of an existing vehicle design, provides for the least interface disturbance.

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IV. DEVELOPMENT PROGRAM

A. PROBLEM AREAS

In discussing mercury Rankine cycles, several areas of concern frequently are cited by potential system users. These are mercury corrosion, erosion and two-phase flow. These are discussed in turn.

1. Mercury Corrosion

Over 2.4 million hours of mercury materials evaluation testing have been conducted and the performance of materials in operation with mercury systems can now be confidently predicted. As a result of this work, many iron- and cobalt-based materials are being used in mercury systems. Perhaps the most convincing argument regarding operation of mercury systems relative to corrosion products is the long endurance tests which have been conducted under continuous operating conditions on the Sunflower program. In these tests, a single test rig constructed of Type 316 stainless steel has operated for a period of nine months at a nominal mercury boiling temperature of 1050° F. For the 14-day application under consideration, several corrosion factors can be applied to this operating experience. The time factor is 0.05. The temperature factor based on corrosion rate data is 2.0 because of the higher boiling temperature compared to that used in the endurance test. By selecting Haynes 25 instead of Type 316 stainless steel, corrosion rates can be reduced to a factor of one tenth those of Type 316. The size factor in this small system compared to a test rig is 0.3. Multiplying the various factors results in a 0.003 to 1 overall factor for the Apollo application compared to actual operating experience in the test rig. From such numbers, a high degree of confidence that materials corrosion is an insignificant problem is achieved.

2. Erosion

Utilizing experience from commercial mercury power plant installations, erosion can be confidently predicted as an insignificant problem. In one instance, 116,000 hours of operation was accumulated in a mercury turbine without maintenance. This time period represents 16 years at an availability of 82%. The cycle conditions in this installation were somewhat more severe than those to be encountered in the proposed system since saturated rather than superheated vapor was used at the turbine inlet.

Furthermore, in the commercial systems, the mercury was condensed at approximately 1 psia which results in greater moisture content at the last stage rotor. Sunflower-type turbines which have been tested for six months also confirm that erosion of turbine blading is not a problem in the mercury Rankine system.

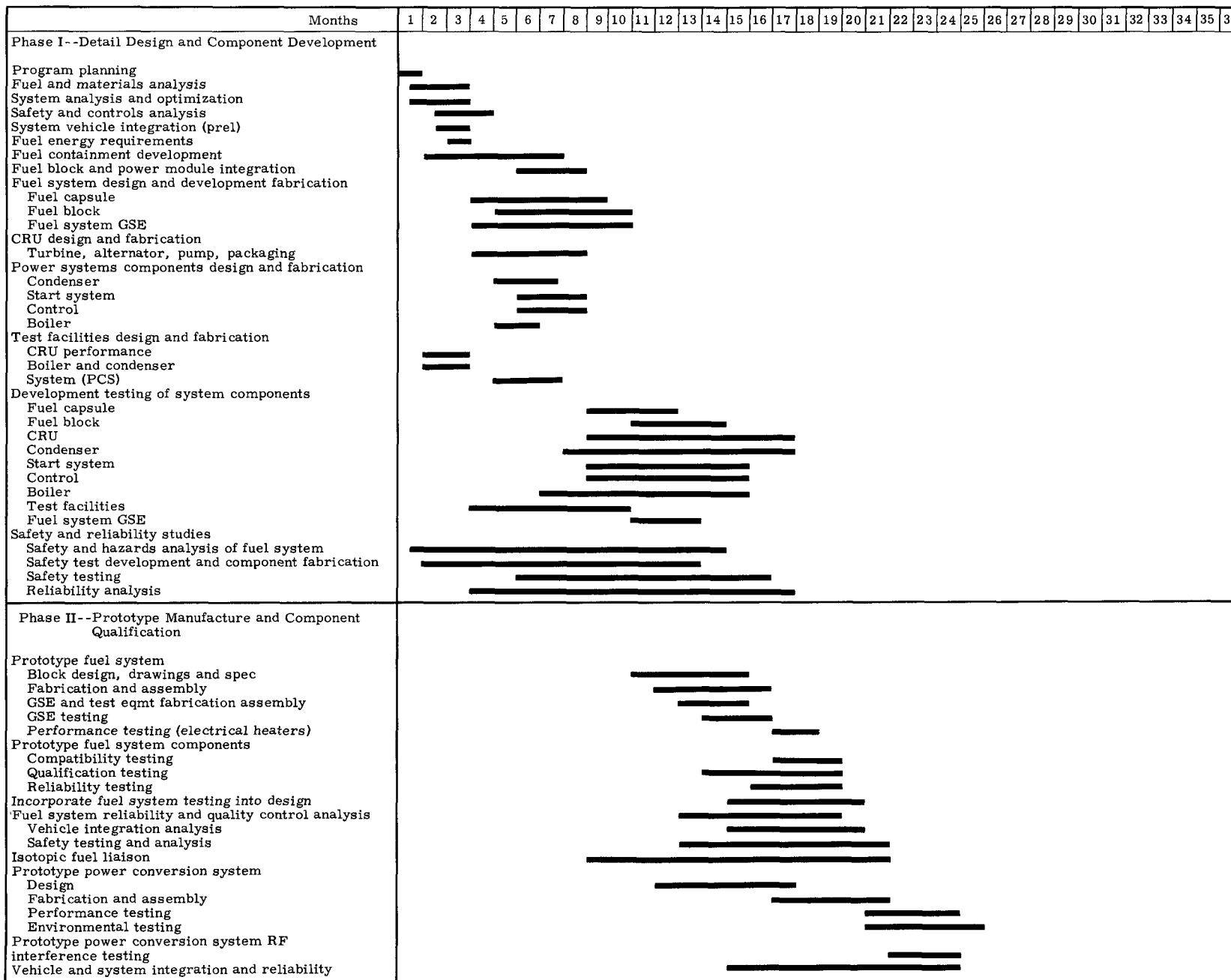| Months	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36

Phase I--Detail Design and Component Development

Program planning
Fuel and materials analysis
System analysis and optimization
Safety and controls analysis
System vehicle integration (prel)
Fuel energy requirements
Fuel containment development
Fuel block and power module integration
Fuel system design and development fabrication
    Fuel capsule
    Fuel block
    Fuel system GSE
CRU design and fabrication
    Turbine, alternator, pump, packaging
Power systems components design and fabrication
    Condenser
    Start system
    Control
    Boiler
Test facilities design and fabrication
    CRU performance
    Boiler and condenser
    System (PCS)
Development testing of system components
    Fuel capsule
    Fuel block
    CRU
    Condenser
    Start system
    Control
    Boiler
    Test facilities
    Fuel system GSE
Safety and reliability studies
    Safety and hazards analysis of fuel system
    Safety test development and component fabrication
    Safety testing
    Reliability analysis

Phase II--Prototype Manufacture and Component Qualification

Prototype fuel system
    Block design, drawings and spec
    Fabrication and assembly
    GSE and test eqmt fabrication assembly
    GSE testing
    Performance testing (electrical heaters)
Prototype fuel system components
    Compatibility testing
    Qualification testing
    Reliability testing
Incorporate fuel system testing into design
Fuel system reliability and quality control analysis
    Vehicle integration analysis
    Safety testing and analysis
Isotopic fuel liaison
Prototype power conversion system
    Design
    Fabrication and assembly
    Performance testing
    Environmental testing
Prototype power conversion system RF interference testing
Vehicle and system integration and reliability

Fig. 10. Program Schedule

Months	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36

Phase III--Unmanned Flight Operational Systems Manufacturing and Flight Rating Tests

Bismuth canning

Bismuth irradiation

Fuel processing and encapsulation

Flight operational fuel system
  Design, drawings and spec
  Components fabrication and assembly
  Performance testing
  Flight rating testing (electrical heating)
  Reliability testing (electrical and isotopic)

Fuel system reliability and quality control analysis

Safety and Hazards analysis

Flight operational power system
  Design, drawings and spec
  Components fabrication and assembly
  Performance testing
  Endurance testing
  Start-stop testing

Delivery of flight systems △

Phase IV--Manned Flight Operational System Manufacture and Flight Rating Tests

Manned flight operational fuel system
  Design, drawings and spec
  Fabrication and assembly
  Performance and qualification testing
  Reliability and quality analysis
  Safety and hazards analysis

Fuel processing and encapsulation

Manned flight operational power system
  Design, drawings and spec
  Fabrication and assembly
  Performance and qualification testing

Delivery of flight system △
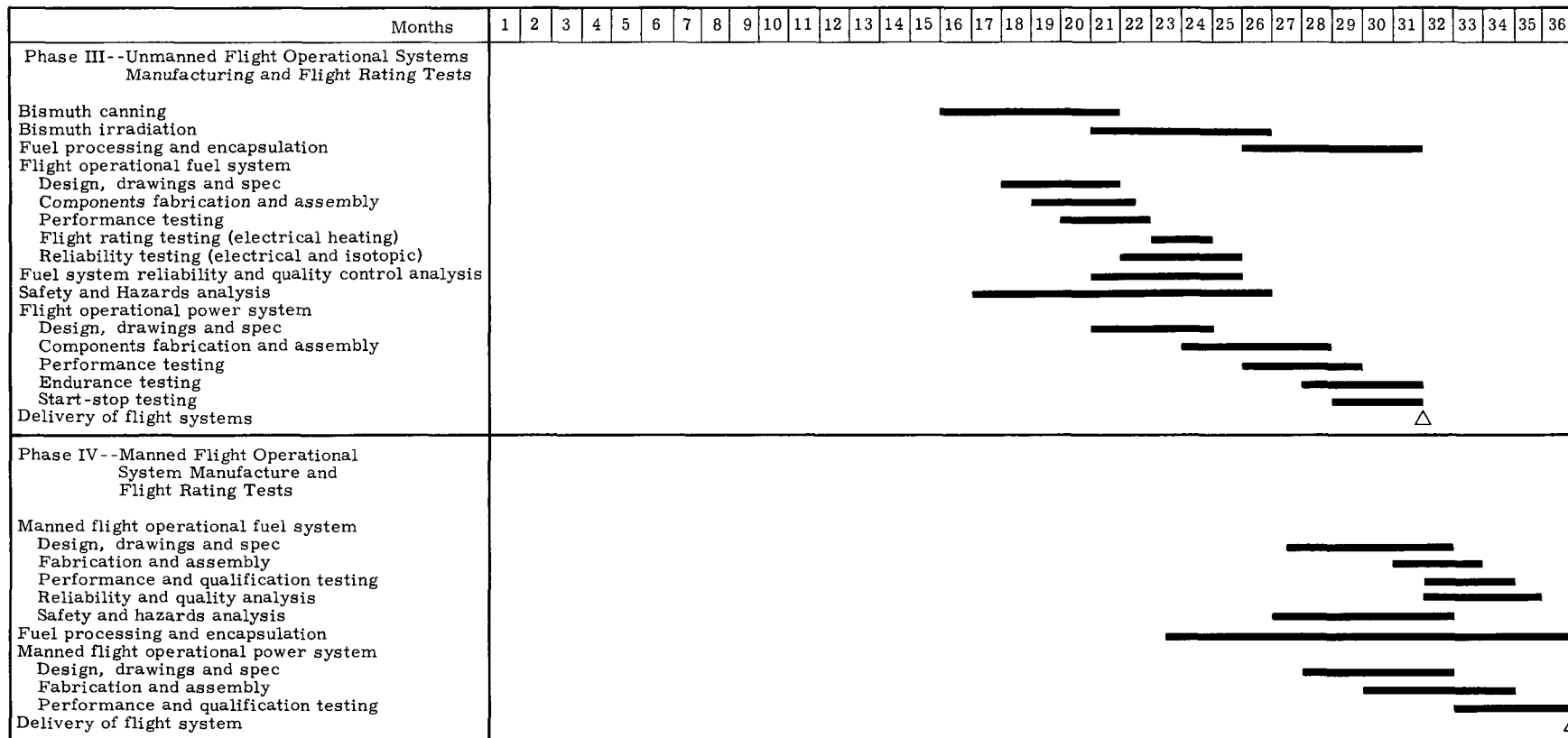
Fig. 10. (continued)

B. DEVELOPMENT PLAN

Attached are bar charts which graphically display the proposed development plan. This program provides for demonstration of a completely integrated system in flight configuration capable of shutdown and startup of the standby unit and suitable for demonstrating the mission endurance requirements of up to 2500 hours. This proposed program would take 2.5 years. The first phase consists of two basic parts--program definition and boiler design study.

Before any significant component work can begin, the basic problems of integrating the power subsystem into the primary vehicle must be examined in detail to establish the design envelope. From this study will come the overall system configuration, cycle conditions and power requirements. A concept is selected; a design agreed on; and component designs then initiated. It is anticipated that this phase will require the combined efforts of those responsible for vehicle, power subsystem and isotope heat source. The system definition phase would be accomplished under Item 3 of the development plan. This function would work closely with that of safety and hazards to coordinate with the vehicle and isotope heat source contractors to ensure that the power subsystem is properly integrated to the vehicle, consistent with changes in designs and requirements.

The boiler is considered to be the only area requiring significant component development. Because its physical configuration will be dependent on safety, handling, hazards and thermodynamic considerations, a design study would be initiated during the program definition phase above to determine the best geometry for development. It is for this reason that several concepts of boiler design have been described for the proposed system, all of which use essentially similar boiler coil designs.

It is contemplated that 2 CRU will be built during Phase II. One will be utilized for component tests of up to 2500 hours; the other will be for final system environmental and RF interference testing.

A prototype condenser and subcooler would be constructed for component test and a second flight weight condenser for systems test.

At the conclusion of the boiler study noted, a developmental boiler would be built and tested. Subsequently, a prototype boiler would be built, simulating as closely as possible, the final isotope flight unit. During Phase III, two final flight-type boiler units that are fully capable of operation with an isotope would be built for systems testing. The second of these two units would be available for isotope fuel development for handling, loading and safety tests or for actual operation with an isotope.

Prototype start systems would be constructed for test demonstrations with one or both of the CRU component tests and the final PCS demonstration.

Speed controls would be constructed for all component tests and the PCS.

During Phase III, six PCS will be built. Two will be utilized for endurance tests, one will be for start-stop tests, one spare and the final two will be for the unmanned flight tests. Two PCS will be built and tested during Phase IV. These will be the manned flight systems.

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V. COSTS

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Radioisotope Fueled Mercury Rankine Cycle Power System

Phase	Definition	Isotopic Fuel Block Cost	Power Conversion System Cost	Total
I	Design and Component Development	$1,757,000	$3,410,000	$ 5,167,000
II	Prototype Manufacture and Component Qualification	1,953,000	1,646,000	3,599,000
III	Unmanned Flight Operational Systems Manufacture and Flight Rating Tests (two systems)*	1,150,000	3,646,000	4,796,000
IV	Manned Flight Operational Systems Manufacture and Flight Rating Tests (two systems)*	605,000	882,000	1,487,000
	Total Development Costs	$5,465,000	$9,584,000	$15,049,000

Estimated per Vehicle Set of Hardware

Isotopic Fuel Processing Cost at $0.45/curie	$ 241,000
Isotopic Fuel Block Cost	60,000
Power Conversion System Cost*	175,000
Total Cost**	$ 476,000

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

**Based on quantities of 10.

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NOMENCLATURE FOR APPENDICES

A	area
C	constant
D	tube diameter
D_{avg}	average tube diameter
E	Young's modulus of elasticity
f_o	friction factor for fluid flow
g	gravitational acceleration
g_c	standard acceleration of gravity at sea level
G	mass velocity (ρu)
h_c	heat transfer coefficient for contact boiling
h_{ic}	heat transfer coefficient for intermittent contact boiling
h_f	heat transfer coefficient for film boiling
h_{fv}	latent heat of vaporization
K	thermal conductivity
K_f	thermal conductivity of liquid
K_v	thermal conductivity of vapor
L	length
$\dot{m}$	mass flow rate
Nu_δ	Nusselt number for drop
ΔP	pressure drop due to flow
P_o	probability of no meteoroid penetration
q	rate of heat flow

Re	Reynold's number, GD/μ
t	thickness
T_{base}	temperature of base of condenser fin
T_{sink}	temperature of heat sink
ΔT	temperature difference
V	velocity
W_{fin}	weight of condenser fins
W_{tube}	weight of condenser tubes
X	quality, vapor weight/total weight
α	swirl wire angle
β	$\dfrac{Nu\ (actual)}{Nu\ (film-spheres)}$
δ	drop diameter
ϵ	emissivity
η_f	condenser fin efficiency
$\eta_{isothermal}$	isothermal fin efficiency
μ_v	viscosity of vapor
ρ	density
ρ_f	density of liquid
ρ_v	density of vapor
σ	constant in Stefan-Boltzmann's law
τ	time of exposure to meteoroid hazard
ϕ	$\dfrac{\Delta P_{tp}\ (friction)}{\Delta P_v}$
tp	two-phase

APPENDIX A

CONDENSER DESIGN

The condenser design is based on the following requirements and conditions:

Radiator geometry	Flat
Tube geometry	Linear taper
Coating	TiO_2/ZrO_2
Maximum vapor inlet velocity (fps)	100
Condensing temperature (°F)	
In	575
Out	570
Maximum allowable condenser ΔP (psi)	0.5 to 1.0
Weight flow (lb/min)	4.87
Tube spacing (in.)	4
Heat rejection (Btu/hr)	34,126
Tube material	Haynes 25
Fin material	Al

From Fig. A-1 for a fin efficiency η_f of 0.9.

$$kt = 0.44$$

$$t = \frac{0.44}{128} = 0.00344 \, \text{foot}$$

$$t = 0.0413 \, \text{inch}$$

where

$$k = 128$$
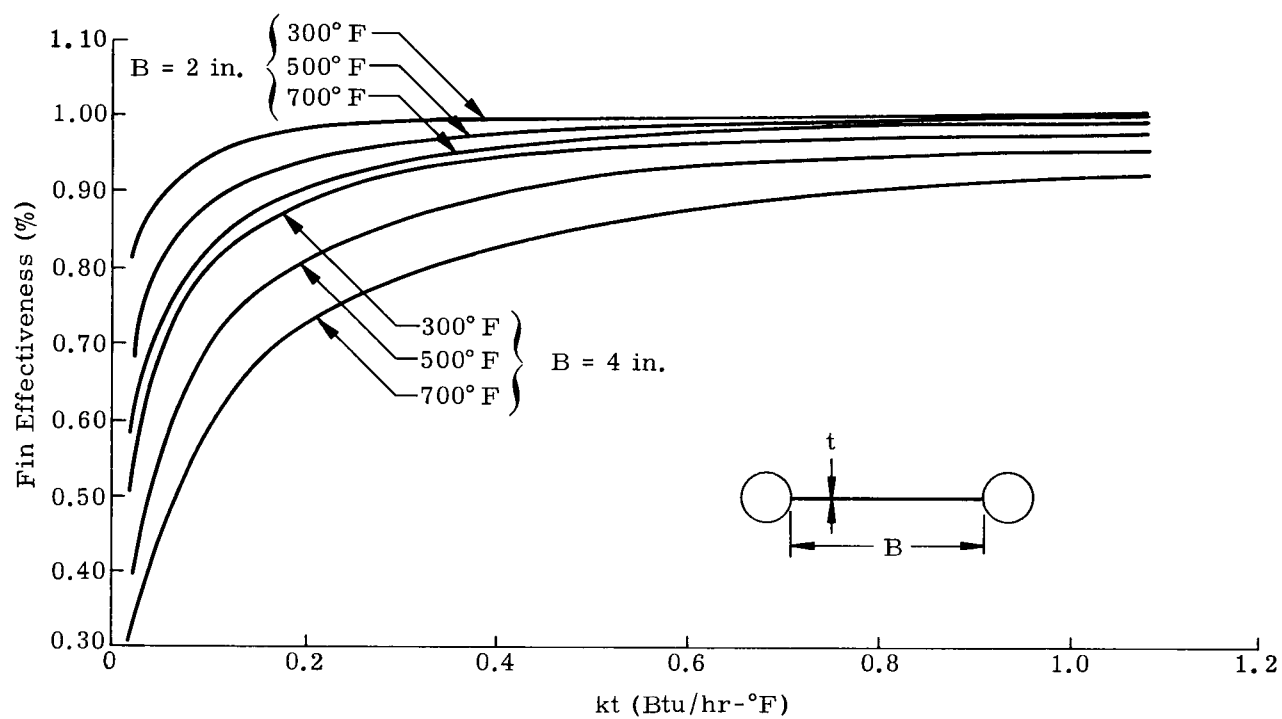
Fig. A-1. Fin Efficiency

The heat transfer by radiation from the fin is given as

$$q = \sigma A \epsilon \; \eta_f (T_{base}^4 - T_{sink}^4)$$

or

$$A = \frac{q}{\sigma \epsilon \; \eta_f (T_{base}^4 - T_{sink}^4)}$$

$$= \frac{34126}{0.173 \times 10^{-8} \times 0.9 \times 0.9 \,(1032.5^4 - 400^4)}$$

$$A = 21.8 \text{ square feet}$$

Assuming the condenser to be four feet wide, then its length is

$$\frac{21.8}{4} = 5.46 \text{ feet}$$

This provides space for 12 tubes and a flow/tube of

$$\frac{4.87}{12} = 0.406 \text{ lb/min.}$$

For a maximum vapor velocity of 100 fps, the inside diameter of the tube is

$$\dot{m} = A \rho V$$

$$A = \frac{in.}{\rho V}$$

$$= \frac{0.406 \times 144}{60 \times 0.085 \times 100}$$

$$A = 0.115 \text{ square inch}$$

$$A = \frac{\pi}{4} D^2$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times 0.115}{\pi}}$$

$$D = 0.383 \text{ inch}$$

Now the wall thickness required for meteoroid protection can be determined.

Since the tubes are brazed to the fin on the inside, the protective capability of the fin shall be considered.
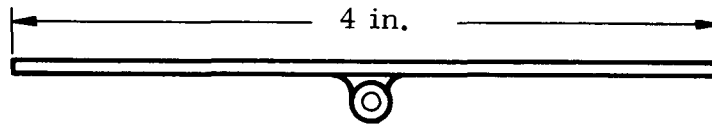
From Ref. 1 the wall thickness for meteoroid protection as seen by NASA is as follows:

$$t = 0.56 \left(\frac{A\tau}{-\ln(P_o)} \right)^{0.249} \times \left(\rho_t E_t^2 \right)^{-1/6}$$

where A is in square feet, assumed to be the projected area of tubes and headers

$$\tau = \text{days}$$

$$\rho = \text{lb/ft}^3$$

$$E = \text{lb/in.}^2$$

$$A = \frac{(12 + 4) \times 0.403 \times 5.46}{12} = 2.92 \text{ square feet}$$

where 12 + 4 is the number of tubes plus the effect of the headers and P_o = 99.99% probability of no penetration during the mission assumed to be for 400 hours or 100 days.

For the selected fin material

$$t = 0.56 \left(\frac{2.92 \times 16.7}{-\ln 0.999} \right)^{0.249} \left(169 \times \left[10 \times 10^6 \right]^2 \right)^{-1/6}$$

$$= 0.032 \text{ inch}$$

The fin is 0.50 inch thick. Thus, the skin is sufficient meteoroid protection in itself.

Now the weight of the condenser can be computed on the basis of minimum practical SS extruded tubing thickness (0.010). Assuming the header contributes 20% to the total weight, the tubing and header results in

$$W_T = 1.2 \times 12 \times 5.46 \, \frac{\pi}{4} \left(\frac{0.0403^2 - 0.383^2}{144} \right) \times 570$$

$$= 3.67 \text{ lb/condenser}$$

Finally, the pressure drop through the condenser is obtained by considering the resistance to the two-phase flow in the tubes and for an average tube size at 50% quality

$$\Delta P = \phi \, f_v \, \frac{L}{D} \times \frac{V_v^2}{2g} \, \rho$$

$$\Delta P = \frac{1.5 \times 0.03 \times 5.46 \times 12 \times 10 \times 0.085}{0.27 \times 64.4 \times 144} = 1.0 \text{ psi}$$

However, this pressure drop is not obtained due to the momentum recovery of the condensing vapor so that the actual pressure drop through the headers and tubes should be below 1.0 psi.

Subcooler Design

For the subcooler, the same width is assumed as for the condenser, namely four feet. The following conditions are given from the cycle consideration.

$Q_{rejection}$	2750 Btu/hr
T_{sink}	400° R
T_{in}	415° F = 875° R
T_{out}	372° F = 832° R
ΔT	40° F
$\dot{m}$	33 lb/mm

Assuming further

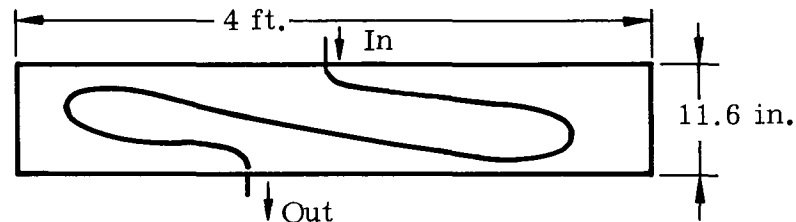|	
η_{fin}	0.90
$\eta_{isothermal}$	0.91
ϵ	0.9

$$q = \sigma A \epsilon \, \eta_f \, \eta_i \, (T_{base}^4 - T_{sink}^4)$$

$$A = \frac{2750}{0.173 \times 0.9 \times 0.9 \times 0.91 \, (8.75^4 - 4^4)}$$

$$A = 3.86 \text{ square feet}$$

and its length is $L = \dfrac{3.86 \times 12}{4} = 11.6$ inches

If the arrangement is as shown below

This arrangement gives a 2.4-inch tube spacing and from Fig. A-1.

$$kL = 0.18$$

$$t = \frac{0.18 \times 12}{128}$$

$$t = 0.017 \text{ inch}$$

PARASITIC LOAD

During part load or no load conditions, the electrical power shall be applied to a resistance heater where temperature cannot exceed 1200° F. Dissipation of the excess heat shall be improved by an additional fin.

Assuming again

$$\epsilon = 0.9$$

$$\eta_f = 0.1$$

$$T_{sink} = 400° \text{ R}$$

$$A = \frac{q}{\sigma \epsilon (T^4_{heater} - T^4_{sink})}$$

$$= \frac{1.575 \times 3410}{0.173 \times 0.9 (16.60^4 - 4^4)}$$

$$A = 0.456 \text{ square feet.}$$

For a four-foot width, the length of this fin is 1.4 inches.

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APPENDIX B

MERCURY SIDE HEAT TRANSFER

The boiler design will be based on past successful TRW experience in the mercury boiler field. The boiler tubes will have swirl wire throughout and in addition, insert devices will be provided in the low quality mercury region to establish vortex flow. The purpose of the swirl wire is to force the mercury against the tube walls, provide turbulent flow and thereby improve heat transfer in addition to providing gravity insensitivity. Different flow patterns affecting heat transfer phenomena will be checked during the analytical phase.

Reference 2 will be used to provide design information. Experience gained on SNAP 2, Sunflower, SPUD, and SCAP mercury boilers will also be drawn upon during the design phase.

Several different boiling phases will be investigated as evidenced by the following dropwise vortex boiling equations:

1. Contact Boiling

$$h_c = \frac{K_v}{D} \frac{1}{32} Nu_\delta^2 \left(\frac{K_f}{K_v}\right)^2 \left(\frac{D}{\delta}\right)^2 \frac{(1-x)^{1/3}}{t} \frac{\rho_v}{\rho_f}$$

2. Intermittent Contact Boiling

$$h_{ic} = \frac{K_v}{D} C^4 234 \left(\frac{K_v}{K_f}\right)^4 (1-x)^{4/3} x \frac{\delta}{D} \left(\tan \alpha \frac{DG\, h_{fv}\, \mu_v}{\mu_v\, \Delta T\, K_v}\right)$$

3. Film Boiling

$$h_f = \frac{K_v}{D} \frac{G^{1/3}}{4} \frac{(1-x)^{2/3}}{x^{1/3}} \frac{D}{\delta} \beta^{4/3} \left(\frac{\tan \rho_v}{\rho_f}\right)\left(\frac{hf_v\, \mu_v}{\Delta T\, K_v}\right)^{2/3}$$

4. Preheat

$$Nu = 7 + 0.025\, (P_c)^{0.8}$$

5. Superheat

$$Nu = 0.023\, (Re)^{0.8}(P_c)^{0.4}$$

6. Two-Phase Pressure Drop

$$\Delta P = \phi f \frac{L}{D} \frac{(xG)^2}{2 g_c \rho_v}$$

where f is obtained from Ref. 3.

Equations 1, 2 and 3 were expressed in terms to yield heat fluxes ($q = h \times \Delta T$). They were solved graphically (q versus ΔT) so that transition points were obtained between the boiling mechanisms as functions of ΔT with quality (x) as a parameter.

It has been determined that a pressure drop on the order of 65 at the 400 to 500 psia level will have negligible effect on the mercury properties. Therefore, the heat transfer and pressure drop phenomena may be analyzed separately.

Analysis of heat transfer Eqs 1 through 3 has shown that heat fluxes on the order of 100,000 to 700,000 Btu/hr-ft^2-°F can be driven across the mercury film. Therefore the design will not be dictated by mercury side conditions. An overall conservative heat flux, consistent with experimental data of 50,000 Btu/hr-ft^2-°F will be assumed, and a mercury side mass velocity of approximately 300 lb/sec-ft^2 will also be a design objective. Commercially available tubing of standard size will be selected consistent with a mass velocity of approximately 300 and a pressure drop of not more than 65 psi.

7. Isotope

The isotope to be used has a power density of 200 watts/cc maximum not including the void fraction.

8. Design Limitation

The final design will be tested with electric heaters rather than an isotope heat source. For this reason, an isotope power density greater than that attainable with electric heaters will not be specified.

9. Design Considerations

The final size and shape of the boiler will be determined by consideration of the following basic assumptions.

    (1) Radiation shielding will be ignored.

    (2) The isotope heat source is to be removable.

(3) A low ballistic coefficient will be maintained.

(4) A two-boiler system is required to operate on an either/or basis from a common source.

(5) The configurations to be considered will be

    (a) Radiation type.

    (b) Conduction type through a metal block.

    (c) Immersion type--liquid metal bath.

BOILER DESIGN

A heat flux of 50,000 Btu/hr-ft^2 is readily attainable and will be assumed for purposes of initial calculation. This heat flux is possible with any of the three types of configurations mentioned. Boiler tube sizes will not vary among the configurations except as affected by mechanical design considerations. Following are the deriviations of the basic tube dimensions and parameters.

1. Design Parameters

$$W = 4.87 \text{ lb m/min}$$

$$P_{out} = 408 \text{ psia}$$

$$P_{in} = \text{to suit (500 psia max)}$$

$$T_{in} = 637° \text{ F}$$

$$T_{out} = 1350° \text{ F}$$

2. Tube Length

Overall heat flux - q = 50,000 Btu/hr-ft^2 based on ID surface.

Select 5/16 in. OD x 0.035 in. wall

Surface area $= \dfrac{\pi \times 0.2425}{12} = 0.0635 \text{ ft}^2/\text{ft}$

Duty $= W (h_{out} - h_{in}) = 4.87 (187.5 - 42.21) \times 60 = 42,500 \text{ Btu/hr}$

Surface required $= \dfrac{\text{duty}}{\text{flux}} = \dfrac{42,500}{50,000} = 0.85 \text{ square foot}$

Length required $= \dfrac{0.85}{0.0635} = 13.4 \text{ feet}$

Add 3.6 feet of excess superheat length to ensure dry vapor delivery to the turbine and permit off-design operation.

Total length = 13.4 + 3.6 = 17.0 feet

Length distribution = 12.0 feet boiling and preheating + 5.0 feet superheat

Flow area = $\frac{\pi}{4}$ $(0.2425)^2$ = 0.0462 square inch

Mass velocity, G = $\dfrac{4.8 \times 144}{0.0462 \times 60}$ = 250 $\dfrac{lb}{sec\text{-}ft^2}$

This is sufficiently close to G = 300, used in the graphical solution of the boiling heat transfer equation, to assure the mercury side coefficient is not controlling.

Pressure drop = $\Delta P_{superheat}$ + $\Delta P_{boiling}$

Re = $\dfrac{GD}{\mu_v}$ = $\dfrac{250 \times 0.2425}{0.70 \times 10^{-4} \times 12}$ = 7.2×10^4

f = 0.058 (Ref. 6)

$\Delta P_{superheat}$ = $\dfrac{1 \times 0.058 \times 5.0 \times (250)^2}{0.2425 \times 2 \times 32.2 \times 4.23 \times 12}$ = 23.0 psi

(ϕ = 1, x = 1 in superheat section)

$\Delta P_{boiling}$ = $\dfrac{4 \times 0.058 \times 6.0 \times (250 \times 0.25)^2}{0.2425 \times 2 \times 32.2 \times 4.23 \times 12}$ = 6.9 psi

from 0 to 50% quality, i.e., $\overline{x}$ = 0.25 (ϕ = 4.0 Ref. 2)

$\Delta P_{boiling}$ = $\dfrac{1.25 \times 0.058 \times 6.0 \times (250 \times 0.75)^2}{0.2425 \times 2 \times 32.2 \times 4.23 \times 12}$ = 17.4 psi

ΔP_{total} = 23.0 + 6.9 + 19.4 = 49.3 psi

which is acceptable compared to 65 psi allowable.

Length of coil assuming 7/8 inch pitch which allows 1/8 inch spacing between tubes

L = (N + 1) x pitch = 12 x $\frac{7}{8}$ = 10.5 inches

Both tubes are coiled to the same diameter on a double-threaded screw mandrel.

RADIATION BOILER

Estimate the surface temperature of the source in the radiation boiler. A temperature in the range of 2000° to 2500° F is assumed acceptable. A source power density of 200 watts/cc results in a temperature considerably in excess of 2500° F. Using the expression

$$\dot{q} = \sigma A_1 \frac{1}{\frac{1}{\epsilon_1} + \frac{A_1}{A_2}(\frac{1}{\epsilon_2} - 1)} (T_1^4 - T_2^4) \quad \text{Ref. 4}$$

for concentric cylinders where subscript 1 refers to the inner cylinder and

$$T_2 = 1350 + 50 = 1400° \text{ F}$$

$$E_1 = E_2 = 0.9$$

$$A_2 = 3.14 \times 6 \times 10.5 = 198 \text{ square inches}$$

T_2 was computed to be 2250° F for an $A_1 = 94$ in.^{2}. This corresponds to an inner cylinder of 3.0-inch OD by 10.0-inch length having a uniform power density of 8.1 watts/cc. There are many possibilities for using a more concentrated source. One such scheme is for a 20 watt/cc source distributed in five 1.0 inch OD by 0.25-inch long rods. If desired, optimization of the coil and isotope diameter and length could be expected to produce a source skin temperature as low as 2000° F.

CONDUCTION BOILER

Estimate the source surface temperature. Assume (1) the heat conducted from the five sources is distributed uniformly at 5.0 inch diameter; and (2) the heat received by the coil is distributed uniformly at 6.0 inch diameter

$$q = \frac{2 \pi K \ell}{\ell n \, r_o / r_i} \Delta T_M$$

$$\Delta T_M = \frac{42.5 \times \ell n \, 6/5 \times 12}{2 \pi \times 31 \times 10} = 47° \text{ F}$$

A thermal contact conductance between the source and its cavity in the block of $h = 1000$ Btu/hr-ft^2-°F is assumed (Ref. 2).

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$$\Delta T = \frac{q}{h} = \frac{42,500}{1000} = 42.5° \text{ F}$$

The source temperature will be approximately 100° F above the tube wall or

$$T_2 = 1400 + 100 = 1500° \text{ F}$$

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