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DESIGN POINT CHARACTERISTICS OF A 500 - 2500 WATT ISOTOPE-BRAYTON POWER SYSTEM

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Abstract

An analytical study was conducted to investigate the potential performance characteristics of an isotope-Brayton space power system at electric power levels from 500 - 2500 watts. Utilization of the Pu^{238} heat source, or capsule, being developed for the Multi-Hundred Watt Radioisotope Thermoelectric Generator (MHW-RTG) was assumed. A single-loop system design concept was selected. The design concept and results of first-order trade-off studies of the effects of major system parameters on system performance are presented. Results of the study indicate the potential for high system efficiency and high specific power over the entire power range.

Introduction

The NASA-Lewis Research Center has been engaged in developing technology for Brayton cycle power systems since 1963. Efforts have included technology for Brayton systems utilizing solar, isotope and nuclear-reactor heat sources over a wide range of power levels. To date these efforts have been focused on a 2-15 kWe Brayton system designed for use with an isotope heat source (ref. 1). This system has been tested with an electrical heater simulating the isotope and has met or exceeded all performance goals. More than 3200 hours of testing have recently been completed in which the characteristics of low earth orbit were simulated. In additional system testing a Brayton rotating unit has been operated for over 10 000 hours. In the past few years other developments such as the testing of smaller compressors and turbines have indicated that lower powers might be practical. Because of the success achieved in the 2-15 kWe system program and the evolving of new Brayton technology a study was undertaken to investigate the feasibility and potential performance characteristics of an isotope-Brayton system at lower power levels from approximately 500 - 2500 watts.

In the study the 2-15 kWe Brayton system was used as a point of departure. A 1600° F turbine inlet temperature was used to ensure hardware similar to that which has been successfully demonstrated in the 2-15 kWe engine program. The use of the Pu^{238} heat source, or capsule being developed for the Multi-Hundred Watt Radioisotope Thermoelectric Generator (MHW-RTG) was assumed (ref. 2); each isotope capsule generates 2400 watts of thermal power. Efforts were directed toward a system which is both simple and efficient. Results of this study are presented in Ref. 3. The results indicate that on the basis of both cost and specific power the Brayton system is superior to or competitive with either solar array-battery systems in low earth orbit or Radioisotope Thermoelectric Generators.

In the present paper attention is focused on the method and rationale for the selection of the design points as well as the performance characteristics of the Brayton system. The system and component design concepts are presented along with results of first-order trade-off studies of the effects of major system design parameters on system electric power output and system specific power. Design-point efficiency and weight estimates are presented for systems using from one to four MHW heat sources to cover the desired range of electric power output. A system configuration is shown for a system using one MHW heat source capsule.

Design Concepts

The single loop Brayton system concept selected for study is shown schematically in Fig. 1. The system consists of a heat source heat exchanger, a compressor-alternator-turbine, a recuperator, a gas radiator to reject waste heat directly to space, and auxiliary electrical equipment for system control and power conditioning. A Helium-Xenon working fluid is used. The alternator is cooled by rejecting heat directly to space by means of radiating fins or to the working fluid gas in a heat exchanger placed around the alternator. No valves are required in the loop.

Heat Source Heat Exchanger

The heat source heat exchanger concept is shown in Fig. 2. Surrounding the cylindrical MHW heat source capsule (approximately 6.5 in. diam 17 in. length) are two concentric axial flow heat exchangers. The primary heat exchanger sees the He-Xe working fluid and is used during normal system operation in space. Heat is transferred by radiation from the MHW capsule to the inner wall of the primary heat exchanger. The auxiliary heat exchanger is used for cooling the MHW heat-source when the Brayton system is not operating such as during launch countdown after the heat source has been inserted into the heat source assembly. An external source of nitrogen gas would be suitable for this purpose. The MHW heat-source is end loaded into the heat-source heat-exchanger and held in place by a retaining cap. Thermal power input to the system is increased by manifolding from two to four of these heat-source assemblies in parallel without changing the heat-source assembly design. Though not shown, a multifoil insulation system is used to reduce heat losses from the heat source assembly.

Safe operation of the heat source is an extremely important consideration. Heat-source integrity must be maintained during all conditions of launch, orbit, reentry and after impact with the Earth. Reentry and impact protection have been incorporated by the AEC into the design of the MHW heat source. The maintenance of safe heat-source temperature after installation into the heat-source heat-exchanger was considered briefly in the study. During prelaunch the heat source can be maintained at temperatures of less than 1000° F by the auxiliary cooling gas. During normal system operation the calculated heat-source surface temperature is approximately 1750° F, well below the 2000° F allowable continuous MHW heat-source surface temperature (ref. 2). Under emergency conditions, when neither the Brayton system nor auxiliary cooling system is operating, use of an insulation door to limit heat-source temperatures was briefly evaluated. This door would be opened by devices sensing heat-source temperature. In the event that all door-opening devices were to fail, a low melting-point multifoil insulation door would act as a backup (ref. 3). A more detailed investigation of the safety aspects of the heat-source heat exchanger assembly is being conducted as part of Contract NAS3-16810, "Mini-Brayton Heat-Source Assembly Design Study" (Nuclear Systems Programs Division of General Electric Corporation).

Rotating Unit

The rotating unit design concept is shown in Fig. 3. The radial flow compressor, radial flow turbine and Lundell-type alternator rotor are mounted on a common shaft. The shaft is supported by foil-type gas journal and thrust bearings. The journal bearings are placed in the gap between the alternator end bells and the shaft. This minimizes the length of the rotating group and improves rotor dynamic performance. Foil bearings do not require jacking gas during startup nor the installational complexity associated with pivoted-pad journal bearings and step-faced thrust bearings such as gimbaling, pivots, and preload devices. Potential pivot wear is also eliminated. Foil-type bearings have been demonstrated in rotating machinery of similar size, weight and rotational speed.

Figure 3 shows the alternator as cooled by passing compressor discharge gas through a finned heat exchanger attached to the alternator stator back iron. Preliminary thermal analyses of the alternator were made to determine the cooling requirements necessary to maintain an alternator hot spot temperature of 400° F or less. The analyses indicated that at the power level resulting from the use of one MHW heat-source the alternator could be cooled by radiation to space through fins attached to the stator back iron. When two MHW heat sources are used, compressor discharge gas could be employed as shown in Fig. 3. At the power levels resulting from the use of three or four MHW heat-sources, the analyses indicated that cooling with radiator discharge gas might be required. Compressor discharge cooling of the alternator for a system using one MHW heat-source could also be employed, of course, and would result in greater system packaging flexibility. With the exception of the different methods of cooling the alternator the rotating unit design remains the same over the entire power range.

Recuperator

The recuperator is a gas-to-gas plate and fin, pure counterflow heat exchanger with triangular end sections. Two different recuperator core geometries were used; one for the system employing one MHW heat source and a second for the other systems.

Radiator

System waste heat is rejected to space by a gas radiator. Use of a gas radiator results in a much simpler system compared to one employing a separate oil radiator loop. A cylindrical, bumper-fin concept was employed with stainless steel tubes and headers and aluminum armor and fins. Selection of stainless steel as the tube and header material rather than aluminum was made primarily to avoid the need for bimetallic joints in the system.

Both cylindrical bumper-fin and flat double-bumper fin radiators were investigated in the study. The weights of the flat radiators were found to be approximately half the weight of the cylindrical radiators. However, the cylindrical radiator concept was selected since it appeared to be more adaptable to varying spacecraft configurations and allowed for easy stacking of the power systems to achieve higher power output. In addition, the headers of the cylindrical radiator were felt to be good structural members which could reduce the amount of additional system support structure required. For certain applications, however, use of a flat radiator might be attractive.

Control and Electrical System

A very simple control and electrical system concept with low power consumption was utilized in the study. The concept provides for a one-button system startup, ac/dc conversion, voltage regulation, and continuous speed control regardless of user load profile. Motor starting is employed, the power system having been filled with the gas inventory required for the desired power output during manufacture. In the event that the Brayton system is shuttle launched the 400 hertz power for motor startup could be provided by the shuttle. For a remote startup in space, a battery and a 400-hertz square-wave inverter would be provided as part of the Brayton system.

System Parametric Analysis

A parametric analysis of the power system was conducted for a range of thermal input power of 2400 to 9600 watts, corresponding to the use of from one to four MHW heat-source capsules. The ranges of the major system parameters investigated are presented in table I. Table II gives values for some of the parameters which were held fixed in the analysis. The turbine inlet temperature was maintained at 1600° F. Compressor and turbine efficiencies of 0.75 and 0.85 were selected on the basis of results of testing of small turbomachinery at low Reynolds number (ref. 4). The same alternator was used to cover the range of power. In addition to the assumed losses shown in table II, calculated losses for windage and electronics requirements were included in the analysis. The effective radiator sink temperature shown is considered representative of near Earth orbit. It was assumed that heat is rejected from the outside surface of the cylindrical radiator alone. The radiator heat load included the thermodynamic cycle waste heat and the alternator cooling heat load as appropriate. Approximately 300 watts of additional radiator heat load was included to account for that fraction of the heat from the bearing and windage losses that is conducted to the alternator cooler as well as an allowance for conduction of heat from the hot turbine end of the rotating unit and scrolls to the alternator cooler.

Based on preliminary calculations a fixed fractional pressure loss allowance, $(\Delta P/P)$, of 0.02 was assumed for the heat source heat exchanger, its manifolds, and the other system ducting. An additional fractional pressure loss allowance, ($\Delta P/P$), of 0.005 was included for those cases where an alternator cooler was used. The remaining system pressure loss was distributed between the recuperator and radiator in a manner which minimized system weight. Digital computer design programs were used to calculate the weight and pressure drop for the recuperator and radiator.

Results of the parametric analysis are discussed below using the case of a one MHW heat source system as representative of those obtained. Similar procedures were used for the cases using from two to four MHW heat sources.

Pressure Ratio-Molecular Weight-Rotational Speed

Since compressor pressure ratio, working fluid molecular weight and turbomachinery rotational speed are highly interrelated especially in affecting the system turbomachinery and the system pressure level they will be discussed together. Turbine (or compressor) specific speed could also be included in the discussion. However, since the turbine specific speed was not varied but held fixed at a value of 65 which was felt to yield reasonable pressure level without significant sacrifice to turbomachinery efficiency it will not be discussed.

The selection of the compressor pressure ratio will be discussed with the aid of Fig. 4. In Fig. 4 specific prime radiator area is the radiator area required per kilowatt of system shaft power assuming a radiator fin efficiency of unity. This is the smallest radiator area possible; the actual radiator area will be somewhat larger than the prime area to yield reasonable radiator weights. The thermodynamic cycle efficiency is the difference between turbine and compressor work divided by the thermal input to the working fluid. Shown are curves for constant values of cycle temperature ratio (compressor inlet temperature ($^{\circ}$R)/turbine inlet temperature ($^{\circ}$R)). Since the turbine inlet temperature is constant, the cycle temperature ratio values of 0.25, 0.26 and 0.27 correspond to compressor inlet temperatures of approximately 55°, 76° and 96° F, respectively. Points of constant compressor pressure ratio are shown on the curve of 0.26 cycle temperature ratio. If prime radiator area and thermodynamic cycle efficiency alone were used for selection of compressor pressure ratio a value would be selected which lies on the envelope curve shown. However system pressure level must also be considered, especially at low power level. Reasonable system pressure level is necessary for good gas bearing performance. In addition, the weights of the recuperator and radiator are reduced substantially by increasing pressure level especially at low values of component fractional pressure loss ($\Delta P/P$). From these considerations a compressor pressure ratio of 1.7 was selected.

Table III shows the effects of rotational speed and working fluid molecular weight combinations on turbine and compressor size and system pressure level. At a given rotational speed, higher molecular weight increases system pressure level but results in smaller turbomachinery. Likewise, for a given molecular weight, higher rotational speed gives higher system pressure level and smaller turbomachinery size. Assuming molecular weight fixed, the heat transfer components will then be significantly smaller and lighter if the rotational speed is 60 000 rpm. The turbomachinery size and weight will also be reduced. However the bearing losses could be significantly higher. While the selection of a rotational speed of 60 000 could result in a system with higher specific power the slower rotational speed of 48 000 rpm was favored as a conservative choice. Therefore, the combination of 48 000 rotational speed - 83.8 molecular weight was selected. This selection results in acceptable system pressure levels as well as acceptable turbomachinery sizes.

Throughout the above discussions, turbomachinery performance was assumed to be constant. Actually the turbomachinery performance will be affected by the interrelationships of pressure ratio, rotational speed and molecular weight, and specific speed. The effects of these variables on the turbomachinery and system performance and weight are being investigated in more detail as part of Contract NAS3-16739, "Preliminary Design of a Mini-Brayton Compressor-Alternator-Turbine" (AiResearch Manufacturing Company).

Compressor Inlet Temperature

As indicated in Fig. 4, compressor inlet temperature has a significant effect on prime radiator area and thermodynamic cycle efficiency. Figure 5 shows how actual radiator area, system power output, and system specific power are affected by changes in compressor inlet temperature. Radiator area increases from approximately 40 ft^2 to 90 ft^2, a factor of 2.25, as compressor inlet temperature is reduced from 90° F to 30° F. At the same time the power output increases by nearly 20 percent. Trade-offs between system power output and radiator area to meet particular mission requirements can be made efficiently. This is true even with all other Brayton components fixed. This flexibility of the Brayton system has been demonstrated in the operation of the 2-15 kWe test system over a range of values of compressor inlet temperature (ref. 5). Figure 5 shows that system specific power changes by approximately 5 percent over this range of compressor inlet temperature. As compressor inlet temperature is reduced further the radiator area (and weight) will increase rapidly since the difference between the radiator exit temperature and the assumed effective sink temperature decreases. System specific power will decrease as a consequence. A compressor inlet temperature of 55° F was selected for the one MHW heat source system to obtain near optimum specific power while maintaining a very compact system (i.e., small radiator enclosed volume). For the other systems a value of 76° F resulted in higher specific power with reasonable radiator areas.

Recuperator Effectiveness

Figure 6 shows the effect of recuperator effectiveness on system power output and system specific power. As recuperator effectiveness is increased from 0.90 to 0.975, the system power output increases by more than 30 percent. The system specific power varies by approximately 10 percent over this range of recuperator effectiveness. It should be noted that the recuperator volume which varies roughly as E/1-E will increase by approximately a factor of 4.3 as effectiveness is increased from 0.90 to 0.975. A recuperator effectiveness of 0.95 was selected. This value yields optimum specific power and also results in a recuperator which can be conveniently configured with the rest of the system in the volume enclosed by the cylindrical radiator. For the higher power systems a recuperator effectiveness of 0.975 would be more nearly optimum but a value of 0.95 was selected for conservatism.

Loss Pressure Ratio

Loss pressure ratio is defined as the ratio of turbine pressure ratio to compressor pressure ratio. One minus the loss pressure ratio is a measure of the pressure drop in the system heat transfer components and ducting. The effects of loss pressure ratio on system power output and system specific weight are shown in Fig. 7. System power output increases more than 15 percent as loss pressure ratio is increased from 0.90 to 0.96. System specific power varied less than 10 percent over this range of loss pressure ratio. A value of system loss pressure ratio of 0.94 was selected.

Design Point Performance

Table IV summarizes the design selections made as a result of the system parametric analysis. System design point performance and weights were estimated using these selections and the losses previously discussed. The systems employ from one to four MHW heat sources. The turbine and compressor efficiencies were adjusted slightly to account for changes in Reynolds number over the power range. The same compressor-alternator-turbine was used except for the method of cooling the alternator. Figure 8 shows the change in alternator efficiency over the power range. A peak efficiency of 0.916 occurs at approximately 1.5 kWe. The alternator was cooled by fins rejecting heat directly to space, compressor discharge gas and radiator discharge gas for the cases using one, two, and three or four MHW heat sources respectively. Figure 9 shows the power requirements of the electrical system over the power range. The power to the electrical system provides for conversion of the alternator power to 120 volts dc and rotating unit speed control.

Tabel V is a summary of the overall system performance using one, two, three, and four MHW heat-source capsules. The resulting conditioned power to the user ranged from approximately 550 to 2670 watts with corresponding bus-bar efficiencies (conditioned power/gross thermal input) ranging from 0.23 to 0.28. The system weights and specific power (conditioned power/system weight) ranged from 570 to 1160 pounds and 1.0 to 2.3 watts per pound, respectively. Radiator area requirements were approximately 60, 90, 160, and 210 square feet for systems using 1, 2, 3 and 4 MHW isotope heat sources, respectively.

A weight breakdown for the systems using 1, 2, 3, and 4 MHW isotope heat-sources is given in table VI. The heat source weight included the Pu^{238} isotope fuel. The weights of the heat source heat exchanger(s), (including the primary and auxiliary heat exchanger(s) and manifolds), recuperator, and radiator were calculated using computer design programs. The other weights were estimated in a first order manner or scaled from similar 2-15 kWe Brayton system hardware. It should be noted that the radiator contributes approximately 30 to 33 percent of the total system weight for systems utilizing from 1 to 4 MHW heat sources. The weights presented for the 1 MHW system assume direct radiative cooling of the alternator. If compressor discharge gas cooling were employed for more convenient packaging the system weight would increase by less than 5 percent.

System Configuration

Packaging arrangements were also considered in the study. Figure 10 shows a configuration of the system using 1 MHW heat source. The cylindrical radiator is 5 feet in diameter. The heat-source heat-exchanger (isotope insertion end) looks out a slot in the radiator for isotope insertion, for isotope auxiliary cooling access, and for shutdown and emergency cooling while in orbit. Compressor discharage gas cooling of the alternator is employed in the rotating unit shown. The same general approach to system packaging was used in packaging the systems employing more than one MHW heat source. It was felt that this approach is compatible with most launch vehicles including the space shuttle.

Concluding Remarks

A study was conducted to determine the feasibility and potential performance of an isotope Brayton electric power system for the 500 to 2500 watt power range. System and component design concepts were selected which promise high reliability and long life. Conservative design assumptions and selections were made to ensure confidence of system development. The resulting system performance, though not necessarily optimum, is indicative that good Brayton system performance can be achieved at low power. For the selected system, bus bar efficiencies (conditioned power/gross thermal input) ranged from 0.23 to 0.28 over the power range. The system weights and specific power (conditioned power/system weight) ranged from 570 to 1160 pounds and 1.0 to 2.3 watts/pound, respectively. In addition to high performance the flexibility in packaging makes the Brayton system attractive for a wide variety of missions using either the space shuttle or other launch vehicles. The high efficiency, simplicity, long-life potential and ease of development of the selected power system make it an extremely attractive choice for future space missions in the 500 to 2500 watt power range.

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TABLE I. - RANGE OF MAJOR PARAMETERS INVESTIGATED

Gross thermal power, watts
2400, 4800, 7200, 9600
Compressor inlet temperature, °F
30 to 120
Recuperator effectiveness
0.90, 0.925, 0.95, 0.975
Loss pressure ratio
0.92, 0.94, 0.96
Compressor pressure ratio
1.6 to 2.4
Working fluid molecular weight
39.94, 60.0, 83.8
Rotational speed, rpm
48 000; 60 000

TABLE II. - FIXED PARAMETERS

Turbine inlet temperature, °F	1600
Compressor efficiency	0.75
Turbine efficiency	0.85
Effective radiator sink temperature, °F	-10
Probability of no radiator puncture (10 years)	0.98
System heat loss, percent	10
Cycle efficiency reduction due to bearing bleed flow, percent	5
Bearing loss, watts	100

TABLE III: - EFFECT OF ROTATIONAL SPEED AND WORKING FLUID MOLECULAR WEIGHT ON TURBOMACHINERY SIZE AND SYSTEM PRESSURE LEVEL

Rotational speed, (rpm)	Molecular weight	Turbine tip diam (in.)	Compressor tip diam (in.)	Compressor[1] inlet pressure psia
48 000	60	4.1	3.5	3.4
48 000	83.8	3.5	3.0	5.6
60 000	39.94	4.0	3.4	2.9
60 000	60	3.3	2.8	5.2
60 000	83.8	2.8	2.1	8.7

[1]Isotope thermal power = 2400 watts.

TABLE IV. - SUMMARY OF DESIGN SELECTIONS

Turbine inlet temperature, °F	1600
Compressor inlet temperature, °F	76[a]
Recuperator effectiveness	0.95
Loss pressure ratio	0.94
Compressor pressure ratio	1.7
Rotational speed, rpm	48 000
Working fluid	HeXe
Molecular weight	83.8

[a]For the one MHW system the compressor inlet temperature was 55° F.

TABLE V. - BRAYTON PERFORMANCE SUMMARY

No. of MHW Heat Sources	1	2	3	4
Gross Thermal Power (watts)	2400	4800	7200	9600
Bus Bar Efficiency $\left(\frac{\text{Conditioned Power}}{\text{Gross Thermal Input}}\right)$	0.23	0.26	0.28	0.28
Conditioned Power to User at 120 Volts DC (watts)	550	1260	1990	2670
Total System Weight (lbs)	570	710	970	1160
System Specific Power $\left(\frac{\text{watts}}{\text{lb}}\right)$	1.0	1.8	2.1	2.3

TABLE VI. - BRAYTON SYSTEM COMPONENT WEIGHTS (LBS)

No. of MHW Units	1	2	3	4
Heat Source(s)	46	92	138	184
Heat Source Heat Exchanger(s)	56	112	168	228
Recuperator	92	63	74	81
Mini-Brayton Rotating Unit	40	30	38	38
Radiator	170	225	330	390
Ducting	24	28	32	36
Structure	51	53	75	80
Insulation	66	72	80	87
Electronics	15	15	15	15
Parasitic Load Resistor	10	15	20	25
Total	570	710	970	1160

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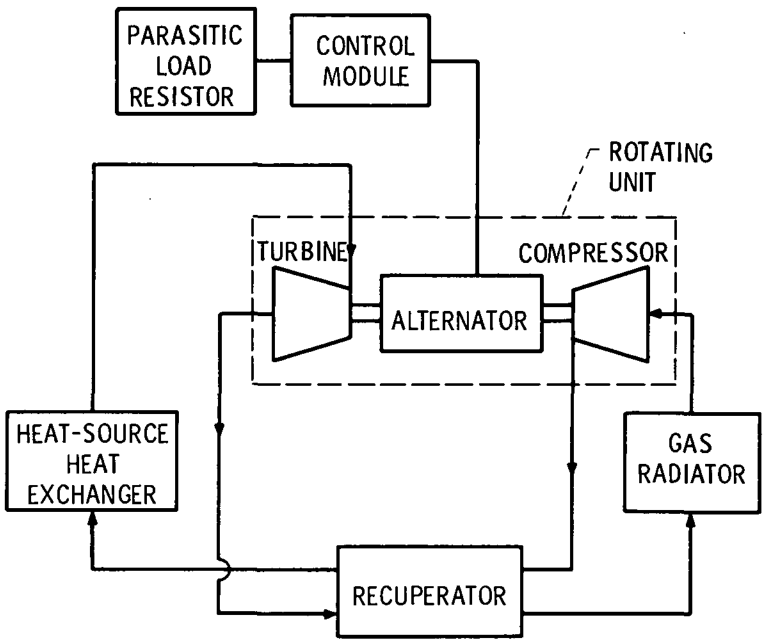

Figure 1. - System schematic.

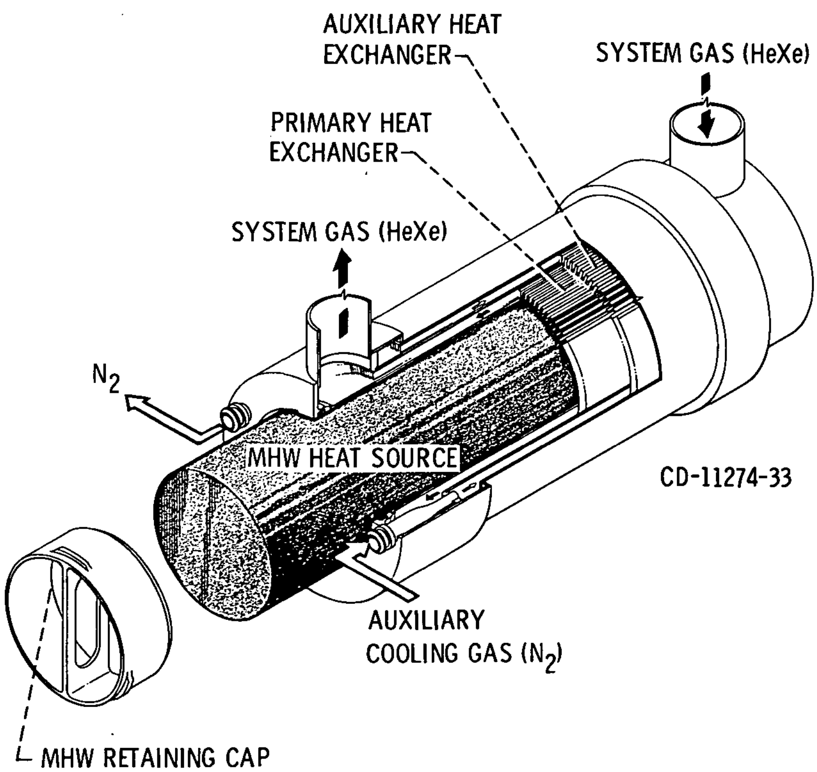

Figure 2. - Heat-source heat exchanger.

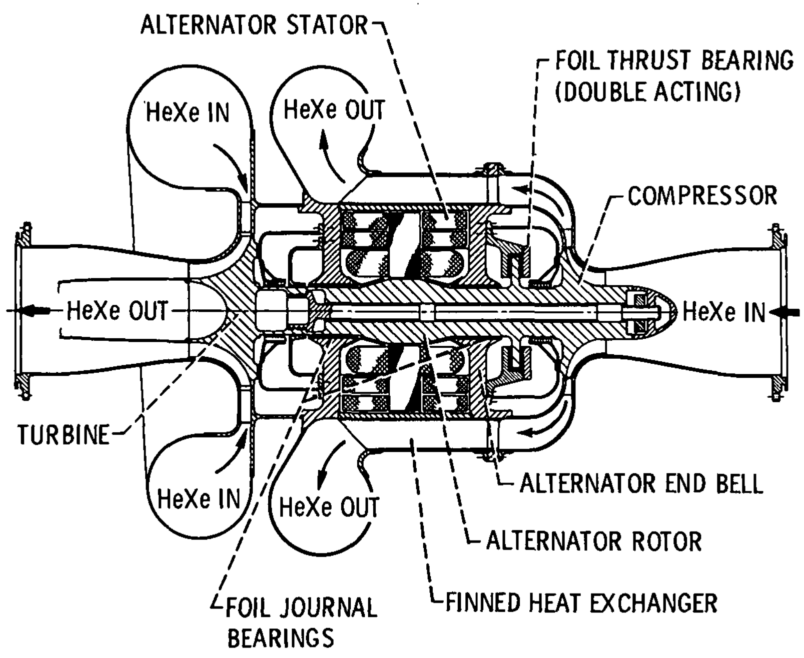

Figure 3. - Brayton Rotating unit cooled with compressor discharge gas.

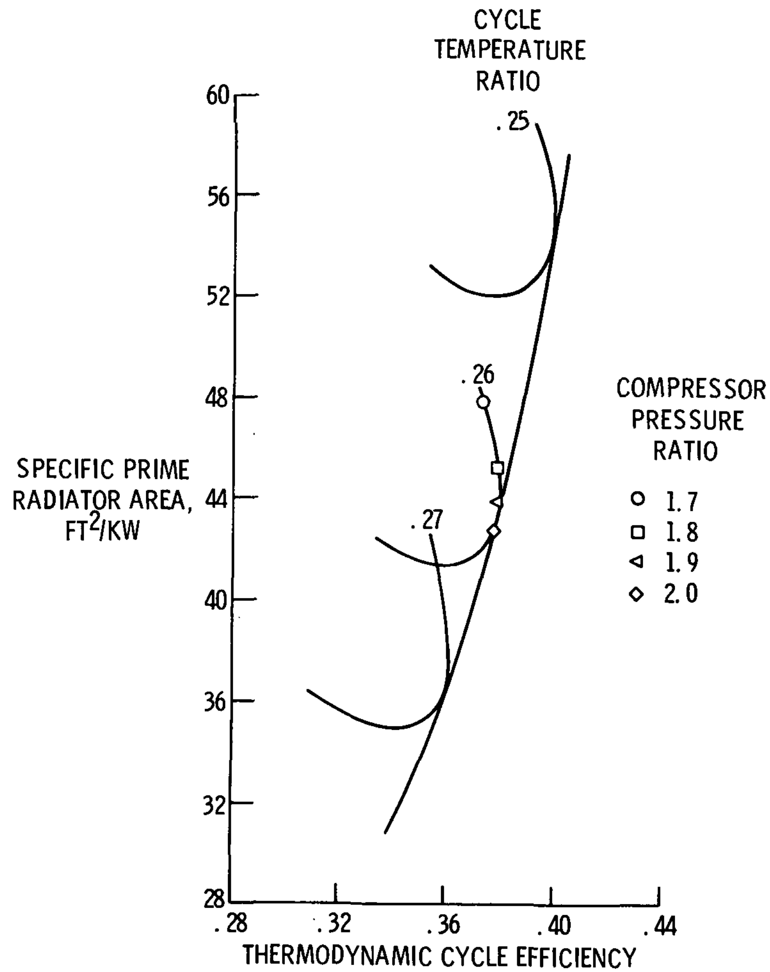

Figure 4. - Thermodynamic cycle performance characteristics.

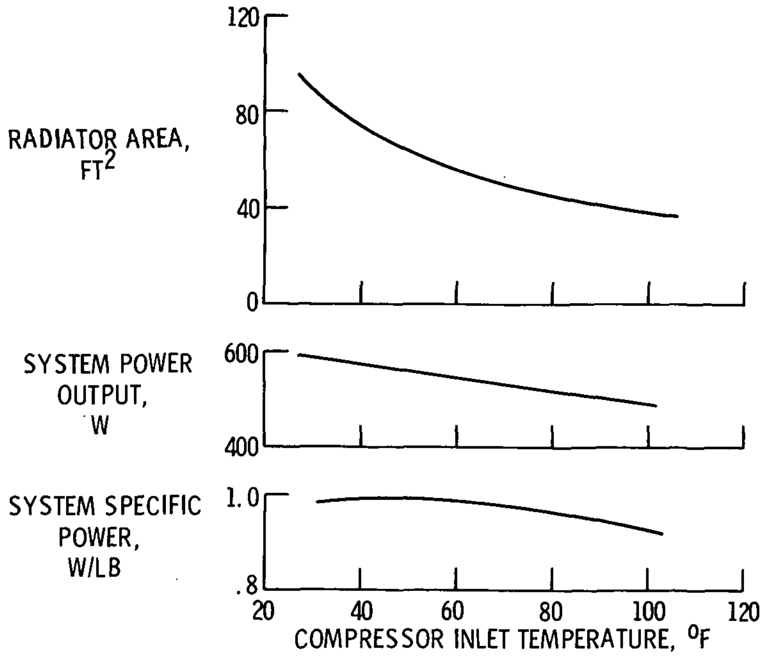

Figure 5. - Effects of compressor inlet temperature on system performance.

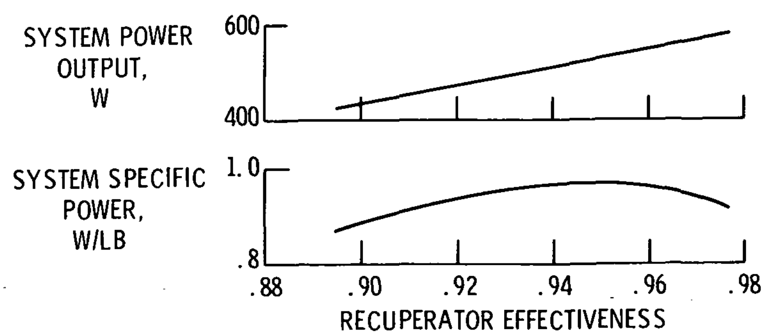

Figure 6. - Effects of recuperator effectiveness on system performance.

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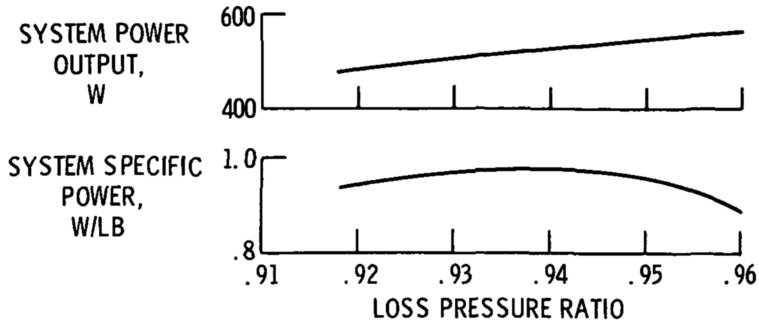

Figure 7. - Effects of loss pressure ratio on system performance.

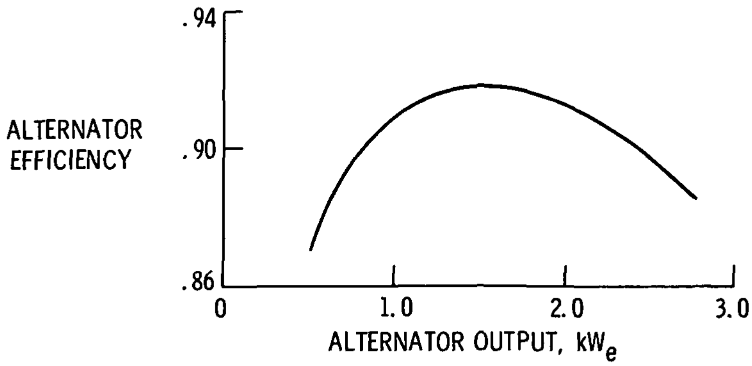

Figure 8. - Alternator efficiency versus power output.

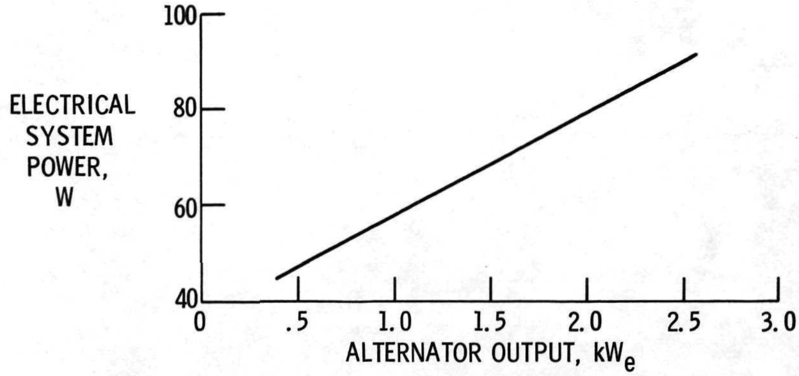

Figure 9. - Electrical system power requirements versus alternator output.

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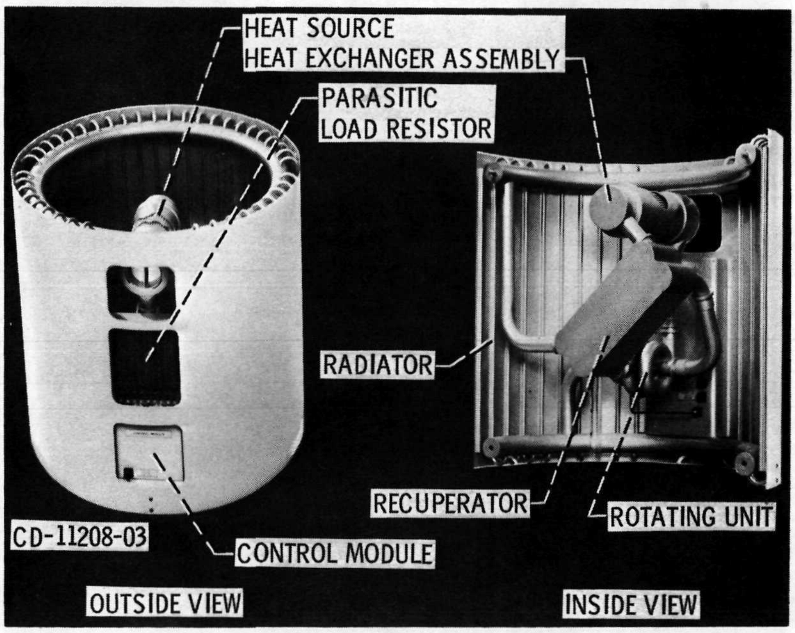

Figure 10. - Brayton power system (550 We).