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STUDY OF BRAYTON CYCLE POWER GENERATION SYSTEM USING SNAP-8 NUCLEAR REACTOR AS AN ENERGY SOURCE

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Cleveland, Ohio

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ERRATA

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Page 1, line 12: The first temperature should be 667° R (370 K).

Page 13, figure 9: The system loss pressure ratio should be 0.94.

Page 14, figure 10: The abscissa scale should be 10, 20, 40, 60, 80, 100×10^3.

Page 15, line 31: The reference number should be 9.

Page 15, lines 31 and 32: The phrase "the alternator electromagnetic efficiency" should be deleted.

Page 22, figure 16: The ordinate scale for the top curve should be 110, 140.

Page 25: The definition for A'_R should read prime radiator area (area of radiator with fin effectiveness of 1.0), ft^2; m^2.

Page 29: Equation (B5) should be

$$C_d = 0.46 \left[\frac{c}{R} \left(1 + \frac{c}{R}\right)\right]^{1/4} N_R^{-0.5}$$

Page 29: Equation (B6) should be

$$C_d = 0.073 \left[\frac{c}{R} \left(1 + \frac{c}{R}\right)\right]^{1/4} N_R^{-0.3}$$

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16. Abstract

Estimates of system characteristics were obtained for a SNAP-8 reactor-powered Brayton system. At 1710° R (950 K) turbine inlet and 1760° R (978 K) reactor outlet temperatures and a 600-kW reactor power, a net unconditioned power output between 97 and 135 kW, including a 10-percent design margin, can be obtained at specific radiator areas between 53 and 71 ft^2/kW_e (4.9 and 6.6 m^2/kW_e). Unshielded system specific weight is estimated to be about 150 lb/kW_e (68 kg/kW_e). At 1610° R (894 K) turbine inlet and 1660° R (922 K) reactor outlet temperatures, power output is 108 kW_e at a specific radiator area of 77 ft^2/kW_e (7.2 m^2/kW_e). Selection of cycle design conditions and working fluid molecular weight is discussed.

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SUMMARY

A study was made to examine in a first-order fashion the characteristics of a SNAP-8 reactor-powered Brayton cycle power conversion system over a range of output power up to a maximum set by the 600-kilowatt reactor thermal rating. At a reactor outlet temperature and turbine inlet temperature of 1760° and 1710° R (978 and 950 K), respectively, a net unconditioned power output of between 97 and 135 kilowatts, including a 10-percent design margin, can be obtained at a reactor power level of 600 kilowatts, over a range of compressor inlet temperatures from 667 ~~616~~° R (370 K ~~342 K~~) to 564° R (313 K). The corresponding radiator area requirements range from 53 square feet per kilowatt electric (4.9 m^2/kW_e) to 71 square feet per kilowatt electric (6.6 m^2/kW_e). The unshielded system specific weight was estimated to be about 150 pounds per kilowatt electric (68 kg/kW_e).

The weight of the shielded system was strongly dependent on the mission configuration and allowable radiation dosage since the shield weight might vary from nothing for a lunar base with a buried reactor to over 100 000 pounds (45 000 kg) for a manned space station with a full 4π shield.

If the reactor outlet temperature and turbine inlet temperatures are decreased to 1660° R (922 K) and 1610° R (894 K), respectively, in order to increase reactor life, a net unconditioned power output of about 108 kilowatts, including a 10-percent design margin, can be obtained at a specific radiator area of 77 square feet per kilowatt electric (7.2 m^2/kW_e) and an unshielded system specific weight of about 170 pounds per kilowatt electric (77 kg/kW_e). Here also, a trade-off between radiator area and power output can be made through an appropriate selection of the compressor inlet temperature.

INTRODUCTION

There has been considerable interest in the use of the SNAP-8 reactor as an energy source for several types of power conversion systems for application in Earth orbital and lunar surface missions (refs. 1 to 3). This interest is a result of the advanced development status of the SNAP-8 reactor as well as certain attractive characteristics of a reactor energy source. These include the characteristic high-energy density associated with the fission process and, when compared with a solar-powered system, the lack of any requirement for a solar collector, orientation, or energy storage. The primary disadvantage is the safety requirement for heavy shielding, particularly for manned missions.

Reference 1, among others, considers a 10- to 30-kilowatt SNAP-8 reactor-powered Brayton cycle power conversion system for application to a manned Earth orbiting space station. References 2 and 3 include a study of the application of a 25-kilowatt SNAP-8 reactor-powered Brayton cycle conversion system to a lunar base mission. Both are detailed studies and indicate that the Brayton cycle conversion system is of interest even at the relatively low turbine inlet temperature (1710° R or 950 K) set by the maximum SNAP-8 reactor coolant temperature.

In these studies, the power output was limited by vehicle surface area available for an integral radiator rather than by the maximum thermal power of the SNAP-8 reactor. The purpose of this study is to examine in a first-order fashion the characteristics of a SNAP-8 reactor-powered Brayton cycle power conversion system with the radiator area restriction removed. Reactor thermal power up to the maximum 600 kilowatts was considered available.

CYCLE ANALYSIS

The temperature-entropy diagram for a closed Brayton cycle is presented in figure 1. Hot gas expands through the turbine from point 1 to point 2. It then passes through the recuperator, where it is cooled to point 3. At point 3, the gas enters the waste heat exchanger, where heat is rejected to a liquid-filled radiator loop (points 3 and 4), which cools the gas down to the compressor inlet conditions at point 4. The gas is compressed to point 5, heated in the recuperator to point 6, and is then heated to the turbine inlet temperature from point 6 to point 1 in the heat source heat exchanger. The excess power of the turbine over that required to drive the compressor is utilized to drive the alternator.

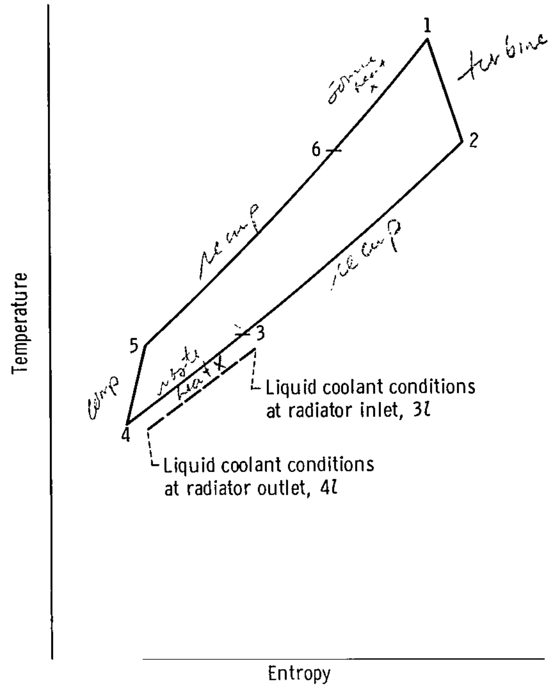


Figure 1. - Closed Brayton cycle.

Assumed Cycle Design Parameters

A reference set of cycle design parameters that past experience indicated would yield favorable cycle efficiency and radiator area was first assumed. Variations of individual design parameters were made independently of the others in the reference set, and performance curves in the form of radiator area plotted against cycle efficiency were compared with that for the reference case.

The assumed cycle design parameters are listed in table I. The turbine inlet tem-

TABLE I. - ASSUMED CYCLE DESIGN PARAMETERS

Turbine inlet temperature, T_1, ^oR (K)	1710 (950)
Compressor polytropic efficiency, $\eta_{C,p}$	0.85
Turbine polytropic efficiency, $\eta_{T,p}$	0.89
System loss pressure ratio, L	0.94
Recuperator effectiveness, E	0.925
Heat sink heat exchanger effectiveness	0.95
Heat sink heat exchanger capacity rate ratio, $\left(wC_p\right)_{gas}/\left(wC_p\right)_{liquid}$	0.90
Radiator surface emissivity, ϵ_{th}	0.88
Equivalent sink temperature, T_s, ^oR (K)	450 (250)

perature of 1710° R (950 K) allows for a 50° R (28 K) temperature differential between a nominal reactor coolant discharge temperature of 1760° R (978 K) and the cycle working fluid. (Actual reactor coolant discharge temperature will vary within the reactor "dead band" limits of 1740° and 1790° R (967 and 994 K).) The assumed compressor and turbine polytropic efficiencies of 0.85 and 0.89 are estimates based on current test data of similar turbomachinery. The selection of a system loss pressure ratio (L = 0.94) and a recuperator effectiveness (E = 0.925) was influenced by the desire to favor cycle efficiency and reduce specific radiator areas at the expense of conversion system size and weight. This is particularly desirable for a relatively low-temperature reactor system where reactor and shield weight are known to constitute a major portion of the total system weight and the relatively low turbine inlet temperature penalizes cycle efficiency and specific radiator area. The use of heat source and heat sink loops that are separate from the conversion loop is favorable to the attainment of low system pressure drop or high values of loss pressure ratio. The selection of a heat sink heat exchanger effectiveness of 0.95 and a capacity rate ratio of 0.9 was based on the results of unpublished studies that included weight optimization aspects. The radiator surface emissivity of 0.88 is representative of the value achievable through the use of a zinc oxide - potassium silicate coating (Z-93, ref. 4). The choice of a 450° R (250 K) equivalent sink temperature for the radiator is reasonable for either an Earth orbital or a lunar surface application using the methods of reference 5.

The radiator area - cycle efficiency characteristics for the reference set of assumed cycle design parameters are shown in figure 2. Curves of specific prime radiator area A_R'/P_{sh} as a function of cycle efficiency are shown for three values of equivalent sink temperature. Each curve is the envelope of individual curves of the constant cycle temperature ratio T_4/T_1 with the compressor pressure ratio p_5/p_4 as a variable. Three such curves for values of T_4/T_1 of 0.33, 0.36, and 0.39 are shown at the reference sink temperature of 450° R (250 K). The minimum specific prime radiator area ranges from a value of 23.7 square feet per kilowatt (2.2 m^2/kW) at a sink temperature of 400° R (222 K), a cycle temperature ratio of 0.415, and a cycle efficiency of 20 percent to 26.4 square feet per kilowatt (2.5 m^2/kW) at a sink temperature of 500° R (278 K), a cycle temperature ratio of 0.425, and a cycle efficiency of a little over 18 percent. As cycle efficiency is increased through a decrease in the cycle temperature ratio (decrease in compressor inlet temperature), the specific radiator area increases. The selection of a design point cycle temperature ratio thus involves a tradeoff between radiator area and efficiency and is strongly influenced by mission requirements.

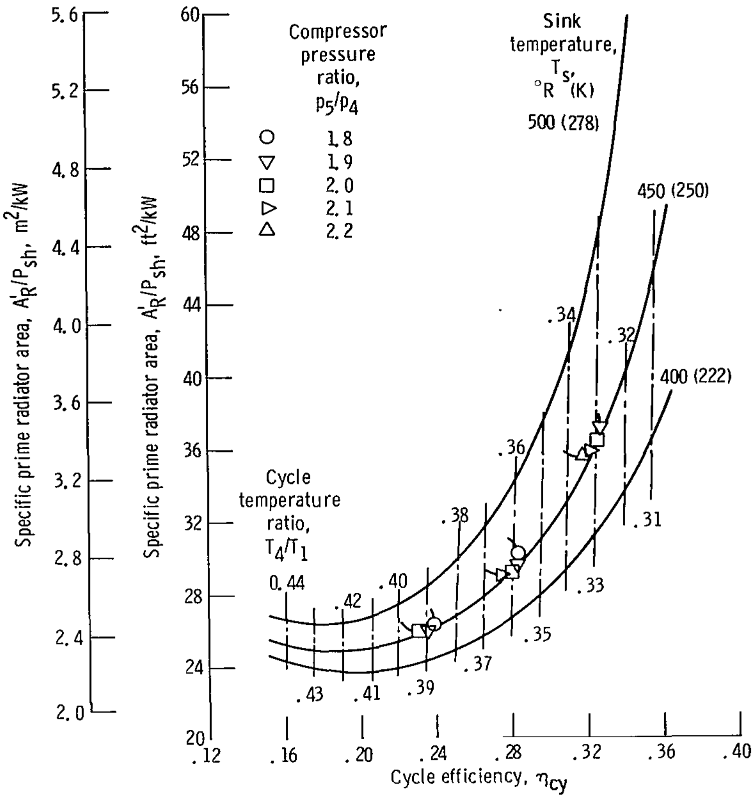


Figure 2. - Variation of specific prime radiator area with cycle efficiency.

Cycle Sensitivity Effects

The selection of final cycle design parameters was aided by determining the sensitivity of the cycle performance to changes in turbomachinery efficiency, system loss pressure ratio, recuperator effectiveness, and turbine inlet temperature.

Effect of turbomachinery efficiency. - The effect of turbomachinery efficiency on the system performance can be seen by comparing the two radiator area - cycle efficiency curves of figure 3. The upper curve is for a compressor polytropic efficiency equal to the reference value of 0.85 and is representative of the efficiency that can be obtained from a centrifugal compressor with backswept rotor blading. The lower curve assumes the use of a multistage axial-flow compressor with a polytropic efficiency of 0.88. The specific prime radiator area at the minimum point is about 15 percent lower using the 0.88 compressor efficiency. Note, also, that the cycle temperature ratio at the minimum specific radiator area point increases as the compressor efficiency is increased, maintaining an approximately constant cycle efficiency at the minimum point. The as-

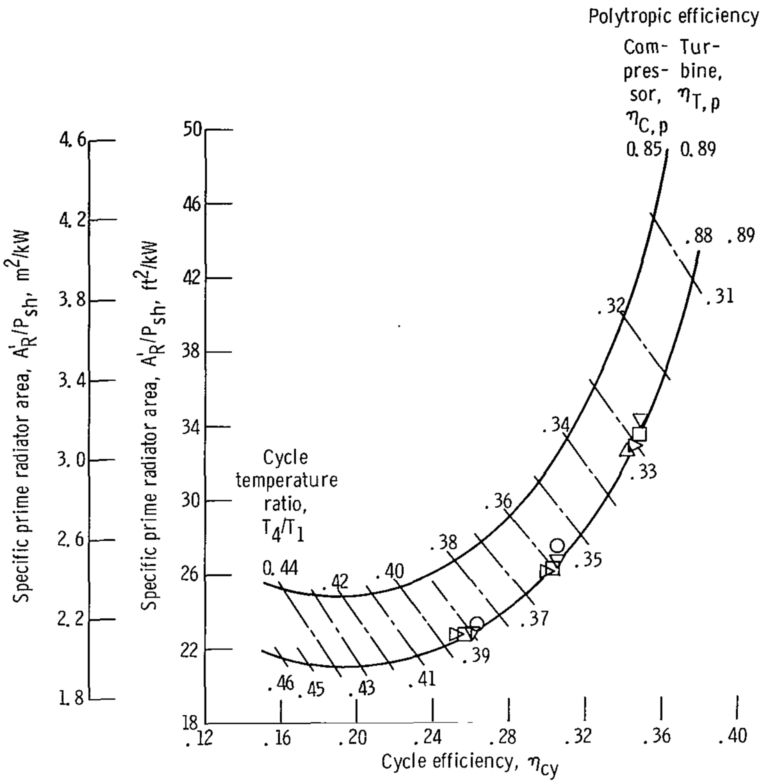


Figure 3. - Effect of turbomachinery efficiency on system performance.

sumed turbomachinery efficiencies are thus influencing factors in the selection of cycle temperature ratio.

Effect of system loss pressure ratio. - Another factor influencing the choice of cycle temperature ratio is the system loss pressure ratio L. The envelope curves showing specific prime radiator area plotted against cycle efficiency for values of loss pressure ratio from 0.9 to 0.96 are presented in figure 4. Cycle temperature ratio contours are included. The curve for 0.94 is again the reference case obtained by using the values of table I (E = 0.925). System loss pressure ratio has a marked effect on specific prime radiator area. The area at the minimum point starting with L of 0.9 decreases approximately 15 percent for each two-point increase of system loss pressure ratio. Cycle efficiency at the minimum area point is almost unchanged, but the cycle temperature ratio at the minimum area point changes appreciably with a change in the value of L. As a result, the selection of the design system loss pressure ratio strongly influences the selection of the design cycle temperature ratio.

Effect of recuperator effectiveness. - The specific prime radiator area - cycle efficiency curves for variations of recuperator effectiveness are presented in figure 5. Specific prime radiator area is not very sensitive to recuperator effectiveness. The reason for this is that the effect of reduced recuperator performance is to add radiator heat load at the high-temperature end of the radiator. This is the most efficient part of the radi-

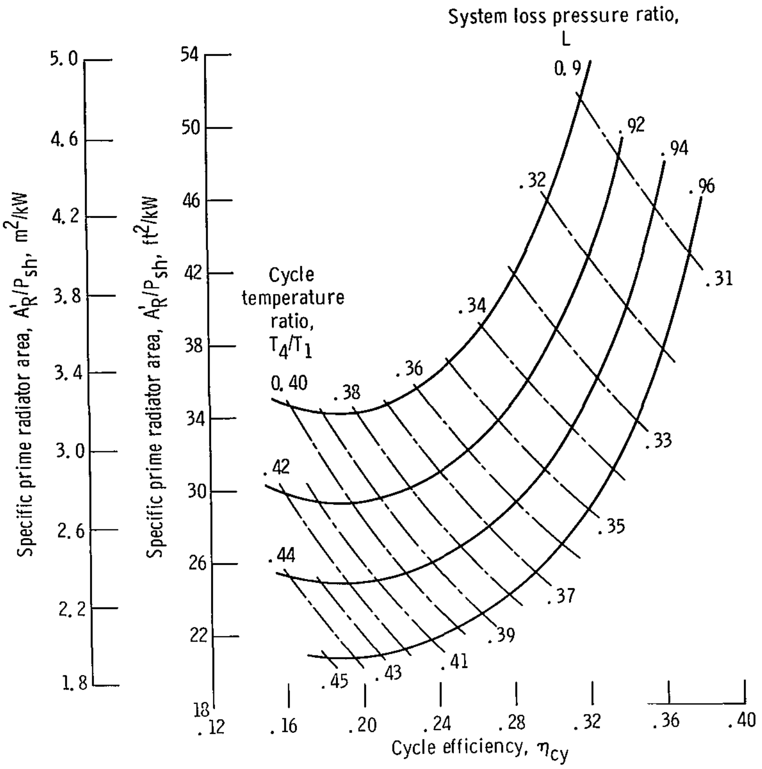


Figure 4. - Effect of loss pressure ratio.

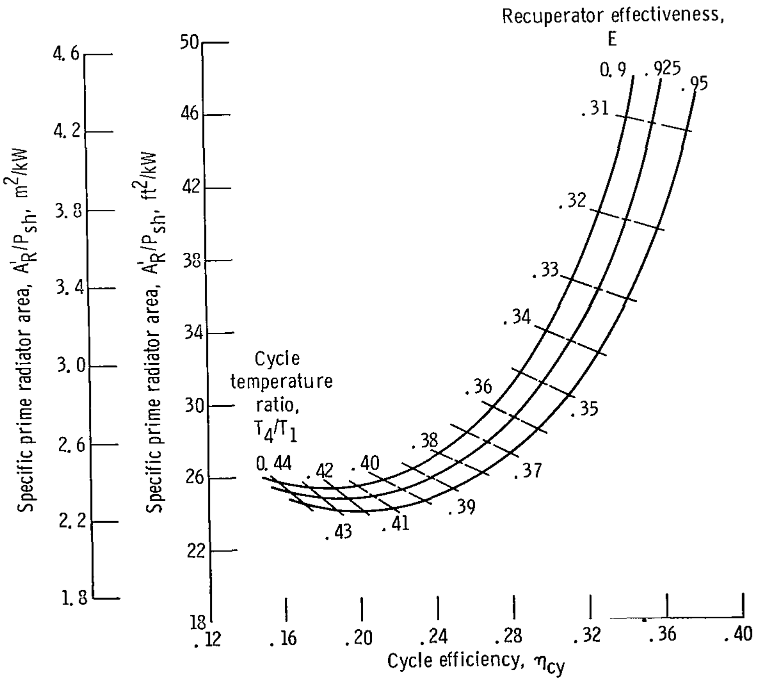


Figure 5. - Effect of recuperator effectiveness.

ator, and an increase in heat load here can be accomplished with only a small increase in area. The specific prime radiator area at the minimum point increased about 5 percent as the recuperator effectiveness was decreased from the maximum recuperator effectiveness of 0.95 to 0.9. The cycle temperature ratio and efficiency at the minimum area point are not significantly influenced by changes in effectiveness.

Effect of turbine inlet temperature. - The effect of varying the turbine inlet temperature on specific prime radiator area and cycle efficiency is shown in figure 6. Radiator

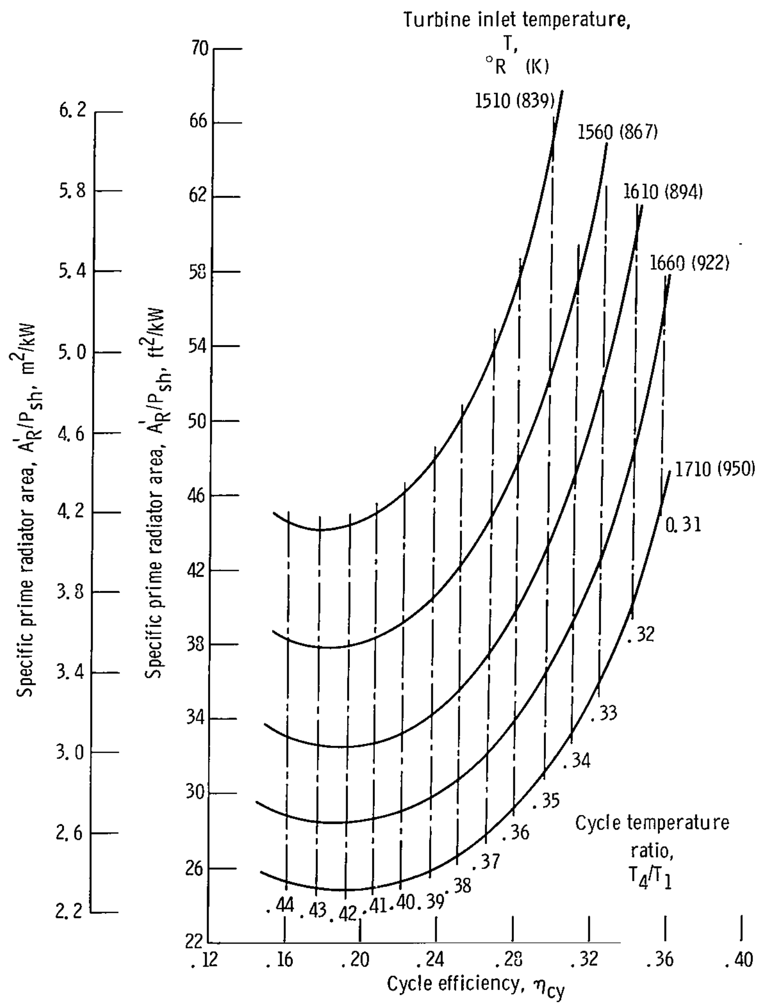


Figure 6. - Effect of turbine inlet temperature.

area - cycle efficiency envelope curves are presented for five turbine inlet temperatures ranging from 1510° to 1710° R (839 to 950 K). The specific prime area at the minimum point is increased about 15 percent for each 50° R (28 K) decrease in turbine inlet temperature. The cycle temperature ratio T_4/T_1 at which the minimum occurs is about the same for all five curves.

Selection of Final Reference System Parameters

Cycle temperature ratio and compressor pressure ratio. - The sensitivity of the cycle efficiency and prime radiator area to variations in some of the more important parameters of table I is shown in figures 2 to 6. The selection of a cycle temperature ratio of 0.36 and a compressor pressure ratio of 1.9 is a compromise between radiator area and efficiency. For the initial assumed cycle design parameters, this selection gives a specific radiator area that is about 19 percent above the minimum point of the 450° R (250 K) sink temperature curve (fig. 2).

Recuperator effectiveness and system loss pressure ratio. - The selection of design values for recuperator effectiveness and system loss pressure ratio may be affected by weight considerations as well as by cycle efficiency and radiator area. Weight calculations were made for all heat exchangers, including the waste heat radiator, for a nominal 65-kilowatt power level system for all combinations of recuperator effectiveness of 0.9, 0.925, and 0.95 and loss pressure ratios of 0.92, 0.94, and 0.96. The cycle temperature ratio and compressor pressure ratio were held constant at 0.36 and 1.9, respectively. Simplified radiator calculations were made by assuming a central fin and tube configuration with a redundant set of tubes. Fins and meteoroid armor were aluminum. The design probability that at least one set of tubes would remain unpunctured after 20 000 hours was 0.995. Structural requirements were not considered.

The results are shown in figure 7. Relative weights for the heat exchangers, radiator, and the sum of the heat exchangers and radiator are plotted as a function of cycle efficiency. All weights are normalized to a value of 1 at a recuperator effectiveness of 0.925 and a loss pressure ratio of 0.94. The heat exchanger weight includes the recuperator, the heat source heat exchangers in the primary and intermediate loops, and the gas-to-liquid-waste heat exchanger in the radiator loop. The variations in heat exchanger weight are dependent almost entirely on changes in recuperator weight that result from the changes in recuperator effectiveness. The radiator weight is dominant, being from 60 to 80 percent of the total. Because of the large influence that loss pressure ratio has on radiator area and efficiency, the system tends to optimize at high values of loss pressure ratio. The total weight is still decreasing at a loss pressure ratio of 0.96. To be conservative and because the low component pressure drops associated with a loss pressure ratio of 0.96 would result in large heat exchangers and piping, a value of $L = 0.94$ was retained. At a loss pressure ratio of 0.94, the total weights at recuperator effectiveness values of 0.925 and 0.95 are about the same. Although the volume of the 0.95-effectiveness recuperator is a little over 7 cubic feet ($0.2\ m^3$) compared with about 4.5 cubic feet ($0.13\ m^3$) for the 0.925-effectiveness recuperator, the higher efficiency and lower specific radiator area at a recuperator effectiveness of 0.95 make it a better choice. The cycle design parameters selected for the final reference system are presented in table II.

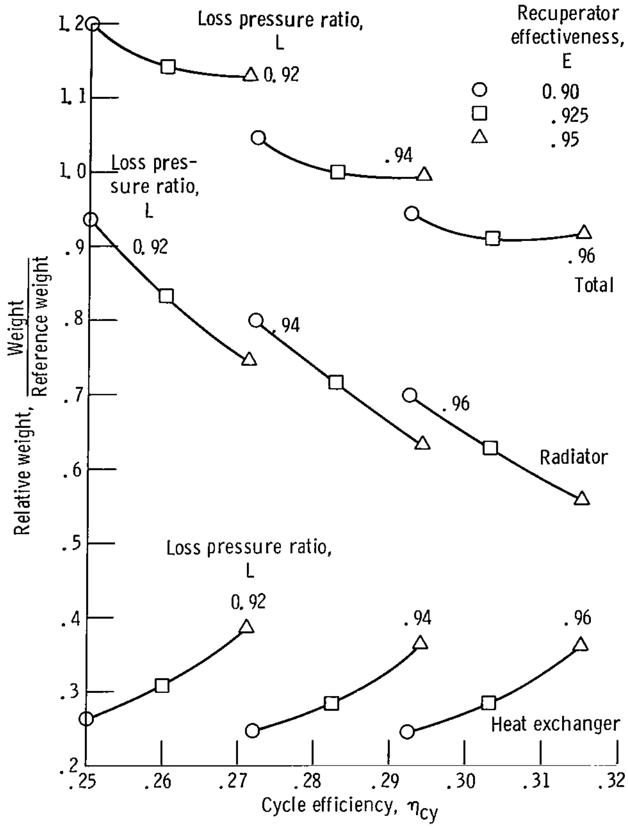


Figure 7. - Effect of recuperator effectiveness and system loss pressure ratio on heat-transfer component weights.

TABLE II. - FINAL REFERENCE SYSTEM DESIGN PARAMETERS

Turbine inlet temperature, T_1, ^oR (K)	1710 (950)
Compressor inlet temperature, T_4, ^oR (K)	616 (342)
Compressor pressure ratio, p_5/p_4	1.9
Compressor polytropic efficiency, $\eta_{C,p}$	0.85
Turbine polytropic efficiency, $\eta_{T,p}$	0.89
System loss pressure ratio, L	0.94
Recuperator effectiveness, E	0.95
Heat sink heat exchanger effectiveness	0.95
Heat sink heat exchanger capacity rate ratio, $\left(wC_p\right)_{gas}/\left(wC_p\right)_{liquid}$	0.90
Radiator surface emissivity, ϵ_{th}	0.88
Equivalent sink temperature, T_s, ^oR (K)	450 (250)

SYSTEM CHARACTERISTICS

System Arrangement

A schematic diagram of the conversion system arrangement is shown in figure 8. The system is composed of four separate loops including a eutectic NaK (sodium-potassium mixture) primary loop, a eutectic NaK intermediate loop, the Brayton cycle conversion loop, and an organic liquid-filled heat sink or radiator loop. A split shield arrangement similar to that presented in reference 1 is utilized. With this arrangement, all primary loop components except for the reactor are placed in a compartment between a primary and secondary shield. The radiation from the reactor is thus attenuated by two shield sections, while the less intense radiation from the primary loop is attenuated by a single secondary shield. The use of an intermediate loop minimizes the penetration of the secondary shield by fluid lines by substituting small liquid lines for large gas ducts. This is particularly important when multiple conversion modules are used, a desirable feature from the standpoint of reliability. Each conversion module has its own independent liquid-cooled radiator loop. The choice of a separate liquid-cooled radiator loop in preference to a direct gas radiator was made to minimize the cycle pressure loss as well as to facilitate vehicle integration.

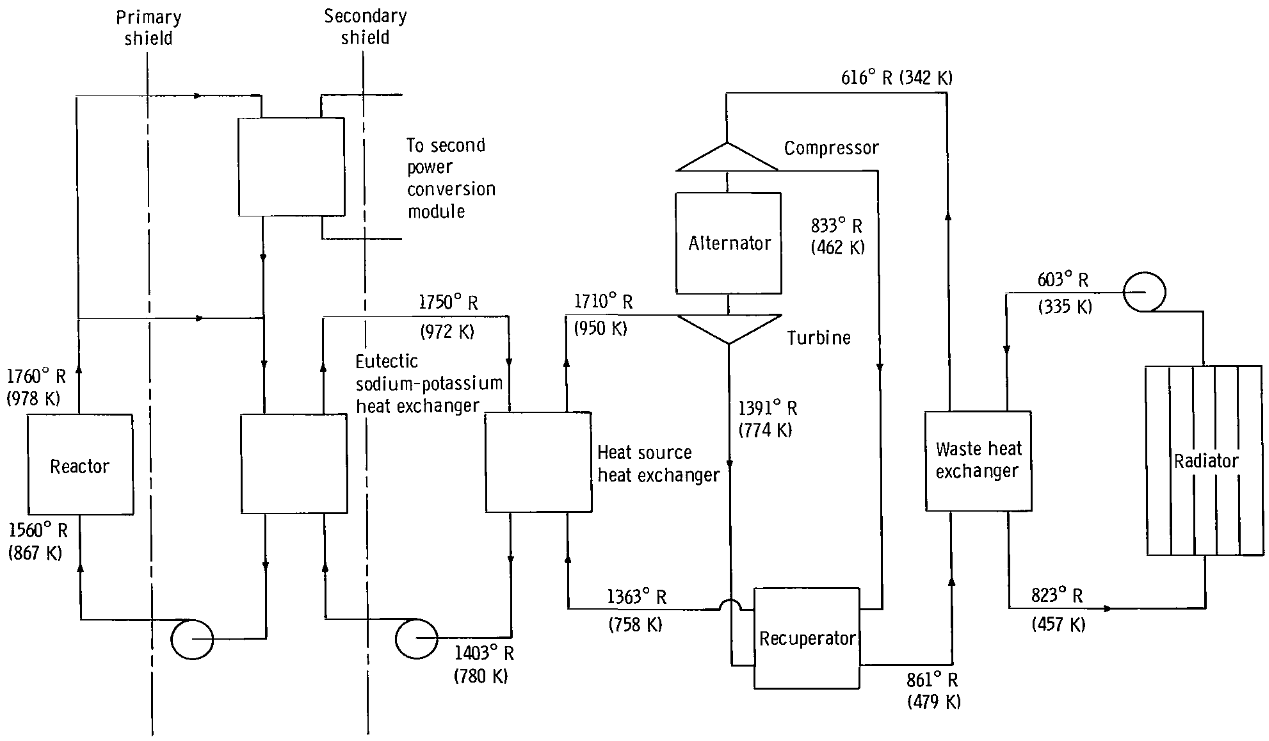


Figure 8. - SNAP-8 Brayton power generation system.

Turbomachinery Characteristics

The working fluid molecular weight and system pressure-power ratio can be selected by consideration of the turbomachinery size and rotational speed. Certain restrictions on rotational speed exist. First, the compressor and turbine must be restricted to a certain range of specific speed to maintain good efficiency. Second, the alternator windage losses that are of increasing importance as power level is increased must be kept to reasonable values. An additional constraint on rotational speed might be a requirement for a certain alternator output frequency.

In this study, power conversion modules of two different power levels were considered. One was a 65-kilowatt-net-output module capable of operating over a power range from about 30 to 65 kilowatts. Two such modules could cover the power range from 60 to 130 kilowatts. The second module was a 130-kilowatt module capable of operating over a range of 60 to 130 kilowatts with a single module.

Single-shaft rotating units were assumed in both cases. A single-stage centrifugal compressor with backswept blading and a single-stage radial-inflow turbine were assumed to be cantilevered from opposite ends of a straddle-mounted Lundell-type alternator on gas bearings. The following expressions were obtained for the compressor and turbine specific speeds by following the analysis of reference 6:

For the compressor specific speed,

$$N_{s,C} = \frac{NM^{3/4}}{\left(\dfrac{p_4}{P_{sh}}\right)^{1/2}} \left[\frac{2.92\times10^{-3}}{(\eta_{C,ad}\,\Delta T_C)^{3/4}\left(\dfrac{\Delta T_T - \Delta T_C}{T_4}\right)^{1/2}}\right] \tag{1}$$

and for the turbine specific speed,

$$N_{s,T} = \frac{NM^{3/4}L^{1/4}}{\left(\dfrac{p_4}{P_{sh}}\right)^{1/2}} \left[\frac{2.92\times10^{-3}}{\left(\dfrac{\Delta T_T}{\eta_{T,ad}}\right)^{3/4}\left(\dfrac{\Delta T_T - \Delta T_C}{T_2}\right)^{1/2}}\right] \tag{2}$$

All symbols are defined in appendix A.

The turbine rotor tip diameter is determined by the tip speed $U_{t,T}$ and the rotational speed N. For the radial turbine, theoretical maximum performance can be obtained when the tip speed is

$$U_{t,T} = 0.707 \sqrt{2g \, \Delta H_{id,T}} \tag{3}$$

The tip diameter of a centrifugal compressor with backswept blading is slightly less than that of the turbine rotor in a single-shaft machine.

In figure 9, the pressure-power ratio is plotted against the rotational speed for a

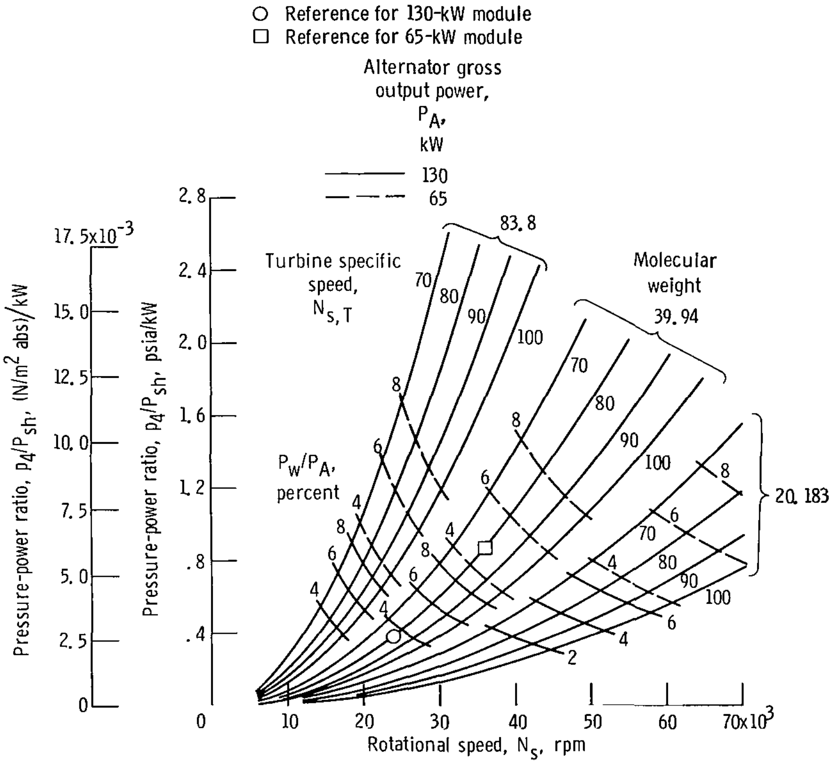
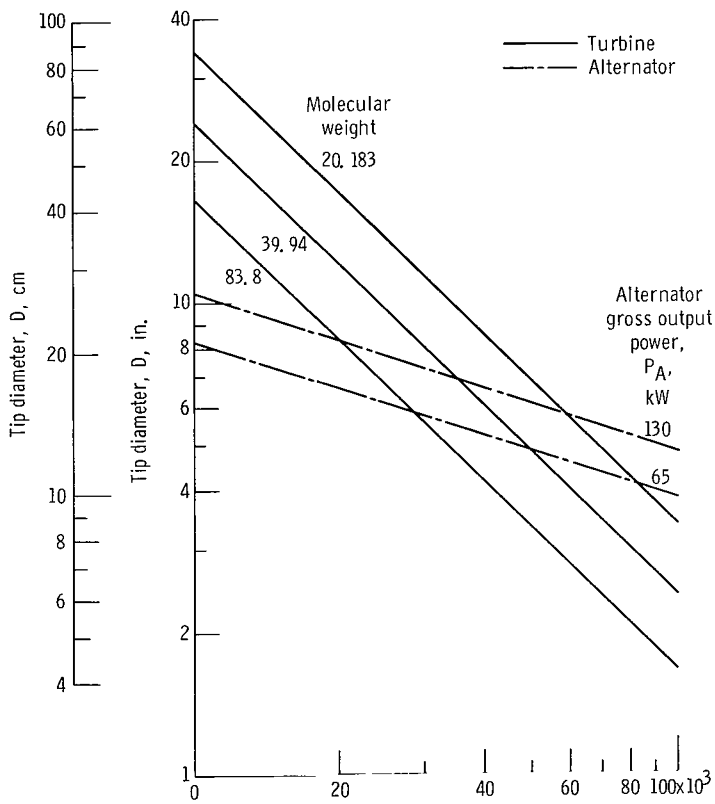


Figure 9. - Effect of rotational speed and turbine specific speed on pressure-power ratio. Turbine inlet temperature, 1710° R (950 K); cycle temperature ratio, 0.36; compressor pressure ratio, 1.9; ratio of turbine specific speed to compressor specific speed, 0.9; system loss pressure ratio, 0.94.

favorable range of turbine specific speeds (ref. 7) and for three working fluid molecular weights, 20.183, 39.94, and 83.8, which correspond to neon, argon, and xenon. The use of a mixture of helium and xenon gases is assumed, permitting variable molecular weight. Superimposed on the plot is alternator windage as a percentage of alternator gross output for the two alternator power levels.

Windage losses were calculated by using the method described in appendix B. The alternator cavity pressure was assumed to be equal to the compressor inlet pressure. The windages shown are for a Lundell-type alternator operating at design net power outputs of 65 and 130 kilowatts. Figure 9 shows that windage losses increase rapidly with rotational speed and molecular weight at any given turbine specific speed.

The tip diameter for a radial-flow turbine and the rotor diameter for the alternator as a function of speed are shown in figure 10. The tip diameter of the compressor is slightly less than that of the turbine. The turbine tip diameter is independent of power level but varies inversely with the square root of the molecular weight and inversely with rotational speed. The alternator rotor diameter shown for two power levels varies inversely as the cube root of the rotational speed for a fixed geometry and power level.



✗ Figure 10. - Tip diameters of turbine and alternator rotors as function of rotative speed. Turbine inlet temperature, 1710 R (950 K); cycle temperature ratio, 0.36; compressor pressure ratio, 1.9; system loss pressure ratio, 0.94.

Selection of Molecular Weight and Rotational Speed

The selection of rotational speed and gas molecular weight was made by picking

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combinations from figures 9 and 10 that best met the following criteria: reasonable turbomachinery tip diameter and tip speeds, alternator frequency of a multiple of 400 hertz for a two- or four-pole Lundell-type alternator, alternator windage limit of 5 percent of alternator gross output, a molecular weight offering good heat-transfer characteristics, and high pressure-power ratios to reduce volumetric flow and heat exchanger size.

Consideration of the foregoing criteria resulted in the selection of a molecular weight of argon (39.94). The high tip speed (1478 ft/sec or 450 m/sec) associated with the use of a molecular weight of neon (20.18) was considered detrimental to long life. The use of a gas with a molecular weight of krypton (83.8) resulted in low speeds and large turbomachinery diameters to keep alternator windage losses at an acceptable level. The high-molecular-weight gas also penalized the heat-transfer components because of poorer heat-transfer properties (ref. 8).

With a molecular weight of 39.94 chosen, the selection of rotational speed was made from alternator frequency requirements and the windage limitation. For the 130-kilowatt module, 24 000 rpm was selected as the highest speed that would meet alternator frequency requirements and alternator windage limitations. Referring to figure 9, a turbine specific speed $N_{s,T}$ of 80 requires a pressure-power ratio p_4/P_{sh} of 0.38 psia per kilowatt (2.6×10^3 N/(m^2 abs)(kW)) and results in an alternator windage loss at a design power of somewhat less than 4 percent of the alternator gross power output. The turbine tip diameter is about 10 inches (25.4 cm). For the 65-kilowatt module, a speed of 36 000 rpm could be used without exceeding a 5-percent windage loss limitation. At this speed, a turbine specific speed of 80 results in a pressure-power ratio of 0.87 psia per kilowatt (6×10^3 N/(m^2 abs)(kW)) and a windage loss of less than 5 percent at design power output. The turbine tip diameter is less than 7 inches (17.8 cm). Alternator windage considerations thus permit a higher rotational speed for the smaller 65-kilowatt module with a consequent reduction in turbine tip diameter and an increase in pressure-power ratio with consequent benefits in heat exchanger size.

System Efficiency

An estimate of the system efficiency was obtained through use of the loss estimates detailed in appendix B. The alternator windage loss was computed for each case by using the drag coefficients from reference ~~8~~ 9. The compressor and turbine efficiencies, ~~the alternator electromagnetic efficiency,~~ and the absolute bearing loss determined at maximum power output were assumed to remain constant as the power output of a module was varied at a constant pressure-power ratio p_4/P_{sh}. The system efficiency is shown for a range of power. In figure 11, the plot is applicable to both the 65- and 130-kilowatt nominal modules. At 50 percent of the design power, the system efficiency is 0.203 and increases to 0.217 at the design power.

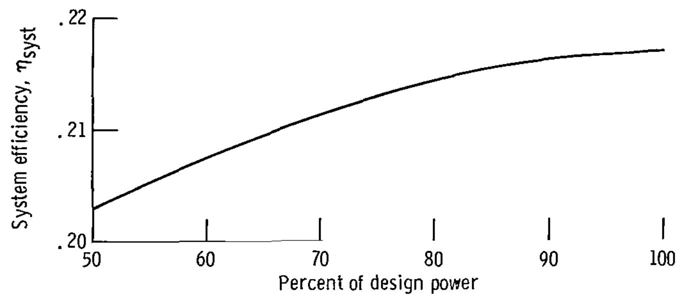


Figure 11. - Effect of power level on system efficiency.

Weight and Size Estimates

Reactor. - A reactor weight of 800 pounds (360 kg), based on the weight of the S8DR reactor, was used in the study.

Conversion system. - Two different conversion systems were considered. One used 65-kilowatt modules with turbomachinery operating at 36 000 rpm. Each module was considered to have an operating range from 30 to 65 kilowatts net output. Two such modules could then cover a power range from 60 to 130 kilowatts output. The second conversion system utilized a single 130-kilowatt module with turbomachinery operating at 24 000 rpm. This system had an operating range from 60 to 130 kilowatts net output. Each module included some fixed components that were sized for maximum power and were unchanged over the design power range. These included the rotating machinery, gas management system, control system, and structure. The heat exchangers and pumps were sized to meet the power requirements. The battery weight portion of the dc power system was also assumed to vary with power level.

The conversion system weights for the two module sizes are shown in figure 12.

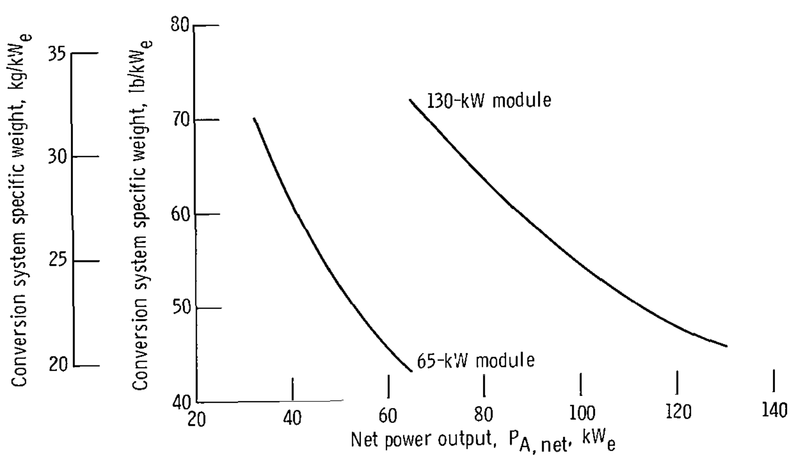
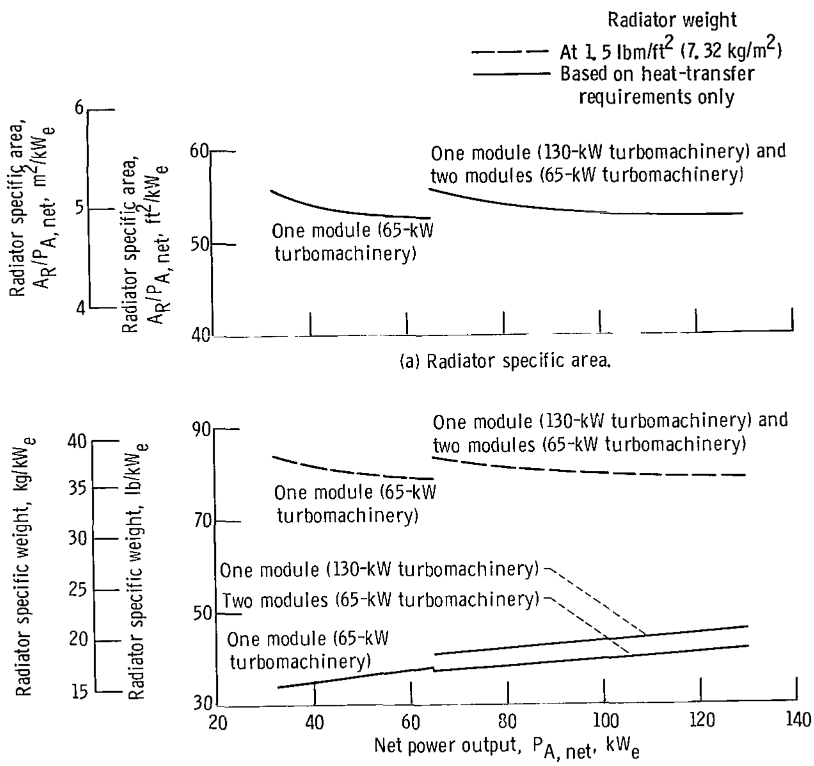


Figure 12. - Conversion system weights.

For a power range from 32 to 65 kilowatts electric using 36 000 rpm turbomachinery, the conversion system weighs about 2300 pounds (1040 kg) at 32 kilowatts electric (70 lb/kW_e or 32 kg/kW_e) and about 2800 pounds (1270 kg) at the design value of 65 kilowatts electric (43 lb/kW_e or 20 kg/kW_e). The larger module, using 24 000-rpm turbomachinery, weighs about 4700 pounds (2130 kg) at 65 kilowatts electric (72 lb/kW_e or 33 kg/kW_e) and 5850 pounds (2650 kg) at the design value of 130 kilowatts electric (45 lb/kW_e or 20 kg/kW_e).

Radiator. - The radiator area requirements are shown in figure 13(a). The radiator



(a) Radiator specific area.

(b) Radiator specific weight.

Figure 13. - Radiator characteristics.

area includes 6.5 square feet per kilowatt electric (0.60 m^2/kW_e) for auxiliary cooling in addition to that required for the cycle waste heat. This value assumes an auxiliary cooling load equal to the difference between turbine gross shaft power P_{sh} and net power output $P_{A,net}$ radiating to a 450° R (250 K) sink at an average temperature of 635° R (353 K). The radiator specific area is about 54 square feet per kilowatt electric (5.0 m^2/kW_e) at the design power level and about 57 square feet per kilowatt electric (5.3 m^2/kW_e) at half power. The increased radiator area at less than design power is brought about by the decrease in system efficiency shown in figure 11. The radiator weights with no allowance for structure are shown as the solid lines in figure 13(b). The radiator specific weight for a single module up to a net power of 65 kilowatts electric is less than 40 pounds mass per kilowatt electric (18 kg/kW_e). For one module over the power range from 65 to 130 kilowatts electric, the radiator specific weight increases from over 40 to about 46 pounds mass per kilowatt electric (21 kg/kW_e). For two modules over the power range from 65 to 130 kilowatts electric, the specific radiator weight is lower than that for one module by about 5 pounds mass per kilowatt electric (2.3 kg/kW_e). The dashed lines in figure 13(b) represent the specific weight of a radiator weighing 1.5 pounds per square foot (7.3 kg/m^2) of the actual area, which might be representative of a radiator forming the space vehicle skin with structural considerations included. The specific radiator weight with this assumption is about 80 pounds per kilowatt electric (36 kg/kW_e), which is about double the radiator weight obtained when only heat-transfer requirements were considered.

Summary of System Characteristics

Some of the important characteristics of the system are shown in figure 14. All electric power outputs are net unconditioned power. The S8DR reactor capable of a thermal power output of 600 kilowatts could supply the energy for a 130-kilowatt-electric Brayton system. At the low end of the range, 160 kilowatts thermal reactor power would be required for the 32-kilowatt-electric system. In figure 14(b), the total radiator area required is about 1800 square feet (167 m^2) at 32 kilowatts electric up to almost 7000 square feet (650 m^2) at the maximum of 130 kilowatts electric. In figure 14(c), the weight for the whole system, excluding reactor shielding, is shown for the two radiator assumptions discussed previously; that is, the radiator design that met only heat-transfer requirements and the design that is considered to weigh 1.5 pounds mass per square foot (7.3 kg/m^2) of actual area. For the first assumption, the total system weight was 4000 pounds (1810 kg) for one module at the 32-kilowatt-electric power level and 6000 pounds (2720 kg) at the 65 kilowatt-electric level. With the use of the heavier radiator, these weights were about 6000 and 8500 pounds (2720 and 3850 kg). For the power range from 65 to 130 kilowatts electric, the system weight, using the lightest radiator and one 24 000-rpm unit or two 36 000-rpm units, varies from about 8000 pounds (3630 kg) at 65 kilowatts electric to about 12 000 pounds (5440 kg) or more at 130 kilowatts electric. With the heavier radiator (1.5 lbm/ft^2 or 7.3 kg/m^2), the system weight varies from about 11 000 pounds (5000 kg) at 65 kilowatts electric to 17 000 pounds (7700 kg) at 130 kilowatts electric.

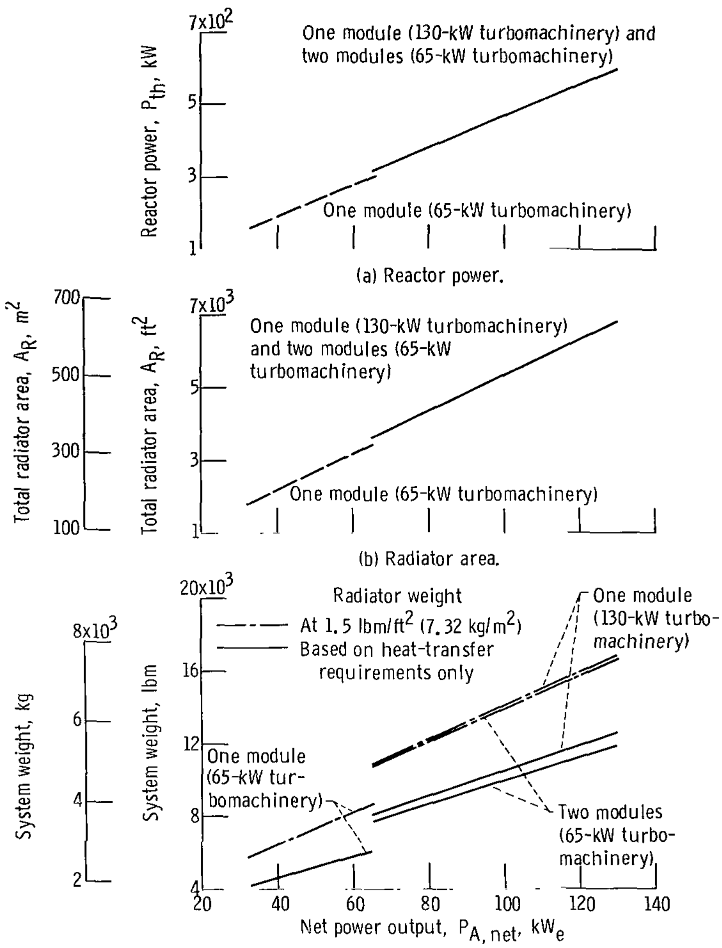


Figure 14. - Summary of system characteristics.

EFFECT OF DESIGN POINT VARIATIONS

Effect of Compressor Inlet Temperature

Varying the compressor inlet temperature by changing the radiator area makes it possible to increase power output from a Brayton system at a fixed reactor power and turbine inlet temperature at the penalty of a larger specific radiator area. Conversely, decreases in the specific radiator area can be achieved by increasing the compressor inlet temperature at the penalty of a decreased power output from a given reactor power. Thus, for missions for which radiator area is of prime importance, a high compressor inlet temperature (high cycle temperature ratio T_4/T_1) would be selected. For missions for which power output from a fixed reactor power is of prime importance and large radiator areas can be accommodated, a low compressor inlet temperature (low value of T_4/T_1) is desirable. The effect of varying the design point compressor inlet temperature on the net unconditioned power output, specific radiator area, and system specific weight is shown in figure 15. Reactor power and turbine inlet temperature are assumed to be fixed at 600 kilowatts thermal and 1710° R (950 K), respectively. Radiator weight is calculated on the basis of 1.5 pounds per square foot (7.3 kg/m^2). An increase in compressor inlet temperature from 564° R (313 K) (T_4/T_1 = 0.33) to 667° R (370 K) (T_4/T_1 = 0.39) results in a decrease in specific radiator area from 65 to 48 square feet per kilowatt electric (6.0 to 4.5 m^2/kW$_e$) and a corresponding decrease

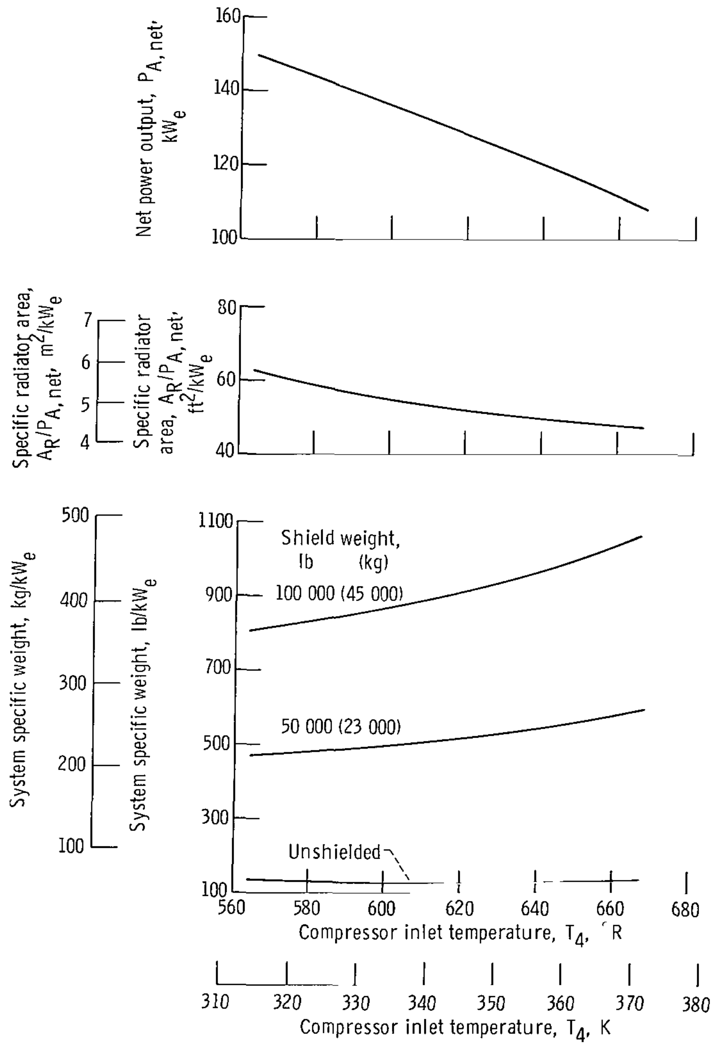


Figure 15. - Effect of compressor inlet temperature on SNAP-8 Brayton performance. Reactor power, 600 kilowatts thermal; turbine inlet temperature, 1710° R (950 K); radiator weight, 1.5 pounds mass per square foot (7.32 kg/m^2); two modules.

in power output from 150 to about 108 kilowatts electric. The specific weight of the unshielded system is almost constant over the temperature range, varying between 130 and 140 pounds per kilowatt electric (59 to 64 kg/kW_e). For manned missions utilizing 4π shields, shield weights may be of the order of 50 000 to over 100 000 pounds (23 000 to 45 000 kg). Here, the system weight is dominated by the fixed shield weight, and the higher power output at the lower compressor inlet temperatures pays off in significantly reduced system specific weight. At a compressor inlet temperature of 564° R (313 K), system specific weights are about 470 and 800 pounds per kilowatt electric (210 to 360 kg/kW_e) for shield weights of 50 000 and 100 000 pounds (23 000 and 45 000 kg), respectively. The corresponding specific weights at 667° R (370 K) are 590 and 1060 pounds per kilowatt electric (270 and 480 kg/kW_e).

Effect of Turbine Inlet Temperature

The life of the SNAP-8 reactor is strongly influenced by the operating temperature. There is, therefore, a strong incentive to reduce the reactor coolant outlet temperature and, consequently, the turbine inlet temperature of the power conversion system. The effect of reduced turbine inlet temperature on the performance of the SNAP-8 Brayton system is shown in figure 16. Reactor power is assumed constant at 600 kilowatts thermal. The compressor inlet temperature is varied along with the turbine inlet temperature so as to maintain a constant ratio of specific radiator area to the minimum specific radiator area possible at the particular turbine inlet temperature. With this condition, cycle temperature ratio T_4/T_1 varies from the reference system value of 0.36 at 1710° R (950 K) to about 0.375 at 1510° R (839 K). Radiator weight is calculated on the basis of 1.5 pounds mass per square foot (7.3 kg/m^2), and dual conversion systems were assumed. Power output is net unconditioned power.

A reduction of 100° R (56 K) in turbine inlet temperature from the reference temperature of 1710° R (950 K) results in a decrease in power output from 130 to about 120 kilowatts electric. The specific radiator area increases from about 54 to 70 square feet per kilowatt electric (5.0 to 6.5 m^2/kW_e). System specific weight is again controlled by shielding requirements. The unshielded specific weight increases from 130 to 155 pounds per kilowatt electric (59 to 70 kg/kW_e) as the turbine inlet temperature is reduced from 1710° to 1610° R (950 to 894 K). The specific weight for a system with a 50 000- and 100 000-pound (22 700- and 45 000-kg) shield increases from about 500 and 900 pounds per kilowatt electric (227 and 408 kg/kW_e), respectively, at 1710° R (950 K) to about 550 and 950 pounds per kilowatt electric (249 and 430 kg/kW_e) at 1610° R (839 K).

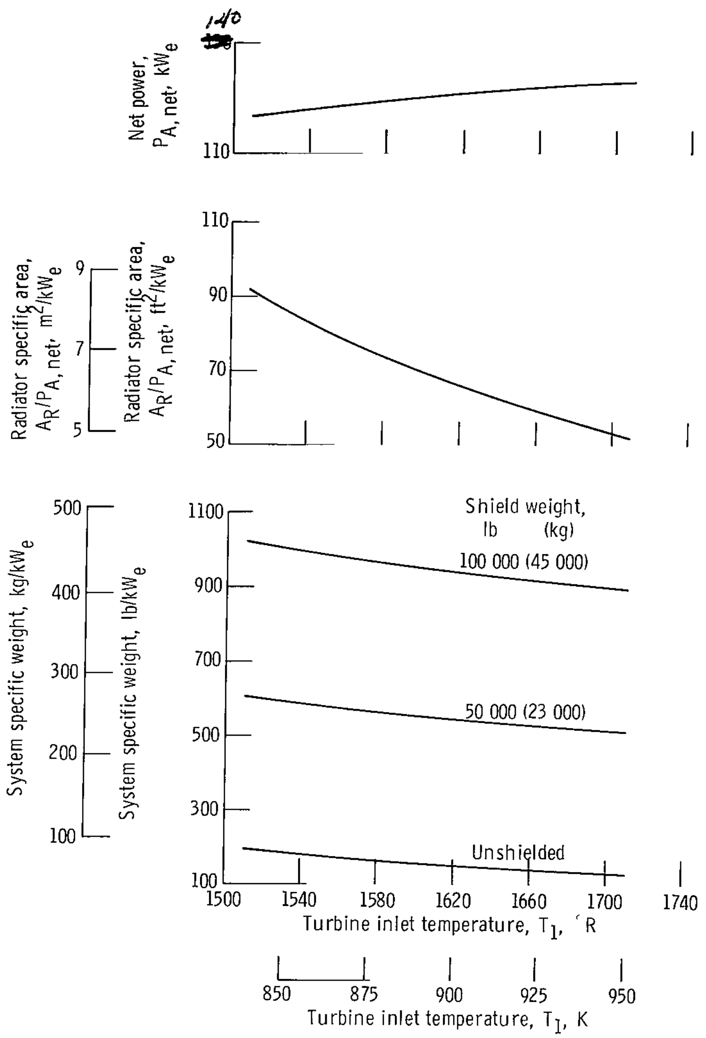


Figure 16. - Effect of turbine temperature on SNAP-8 Brayton performance. Reactor power, 600 kilowatts thermal; radiator weight, 1.5 pounds mass per square foot (7.32 kg/m^2); two modules.

CONCLUDING REMARKS

The results of the study indicated that good performance can be expected from a Brayton cycle conversion system operated with a SNAP-8 reactor as an energy source, even at the relatively low turbine inlet temperature dictated by the reactor coolant discharge temperature. A system utilizing 65-kilowatt modules can produce about 130 kilowatts of net unconditioned power at the nominal reactor design conditions of 600 kilowatts thermal at 1760° R (978 K). The use of a single 65-kilowatt module permits efficient operation in applications down to a required power level of about 30 kilowatts. Radiator area, including auxiliary cooling requirements, is about 7000 square feet

(650 m^2) or 54 square feet per kilowatt electric (5.0 m^2/kW_e) at the 130-kilowatt level. The unshielded system weight is about 17 000 pounds (7700 kg) or 130 pounds per kilowatt electric (59 kg/kW_e), if radiator weight is computed at 1.5 pounds mass per square foot (7.3 kg/m^2). Shielded system weight is strongly dependent on the mission, the configuration, and the allowable radiation dosage. Shield weights may vary from nothing for a lunar base application using the lunar soil as shielding to over 100 000 pounds (45 000 kg) for a manned space station with a full 4π shield.

By decreasing the compressor inlet temperature, power output can be increased at the expense of a larger radiator area. Conversely, radiator area can be decreased at the expense of a lower power output from a fixed reactor power. At the nominal reactor design condition, a change in compressor inlet temperature from 564° R (313 K) (T_4/T_1 = 0.33) to 667° R (370 K) (T_4/T_1 = 0.39) resulted in a decrease in calculated power output from 150 to about 108 kilowatts and a decrease in specific radiator area from 65 to 48 square feet per kilowatt electric (6.0 to 4.5 m^2/kW_e). The unshielded system specific weight remained almost constant as the compressor inlet temperature was changed.

Reactor life considerations provide a strong incentive to reduce reactor operating temperature and, consequently, the turbine inlet temperature. A reduction of 100° R (56 K) in turbine inlet temperature from the nominal design value of 1710° R (950 K) resulted in a decrease in calculated power output from 130 to about 120 kilowatts electric. The specific radiator area increased from about 54 to about 70 square feet per kilowatt electric (5.0 to 6.5 m^2/kW_e), while the unshielded system specific weight increased from 130 to 155 pounds per kilowatt electric (59 to 70 kg/kW_e).

If a 10 percent margin is incorporated into the calculated system performance characteristics to provide for the uncertainties normally encountered in a development program, the following performance characteristics may be expected. At a turbine inlet temperature of 1710° R (950 K) corresponding to a reactor outlet temperature of 1760° R (978 K), and a reactor power of 600 kilowatts, a net unconditioned power output of between 97 and 135 kilowatts can be obtained, depending on the compressor inlet temperature selected. Corresponding specific radiator areas are between 53 square feet per kilowatt electric (4.9 m^2/kW_e) and 71 square feet per kilowatt electric (6.6 m^2/kW_e), and the unshielded system specific weight is about 150 pounds per kilowatt electric (68 kg/kW_e). If the turbine inlet temperature is reduced to 1610° R (894 K), corresponding to a reactor outlet temperature of 1660° R (922 K), a net unconditioned power output of about 108 kilowatts electric can be obtained at a specific radiator area of 77 square feet per kilowatt electric (7.2 m^2/kW_e) and an unshielded system specific weight of about 170 pounds per kilowatt electric (77 kg/kW_e). A trade-off can be made between radiator area and power output through an appropriate selection of compressor inlet temperature.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 28, 1969,
120-27.

APPENDIX A

SYMBOLS

A_R	radiator area, ft^2; m^2
A'_R	prime radiator area (area of radiator with fin effectiveness of 1.0), ft^2; m^2
C_d	drag coefficient
C_p	specific heat, Btu/(lbm)(^oR); J/(kg)(K)
c	rotor gap
D	diameter, in.; cm
E	effectiveness
g	gravitational constant, 32.2 ft/sec^2; 9.8 m/sec
ΔH	specific enthalpy change, (ft)(lb)/lb; J/kg
L	system loss pressure ratio, r_T/r_C
L_A	alternator stator stack length, in.; cm
M	molecular weight, lb/lb mole; kg/kg mole
N	rotational speed, rpm
N_R	Reynolds number
N_s	specific speed, $NQ^{1/2}/\Delta H^{3/4}$
P_A	alternator gross output power, kW_e
$P_{A,net}$	alternator net output power, $P_A - P_p - P_{cont}$, kW_e
P_{cont}	control power, kW
P_p	pump power, kW
P_{sh}	gross shaft power, $P_{th}\eta_{cy}$, kW
P_{th}	thermal power into cycle, kW
P_w	alternator windage, kW
p	absolute pressure, psi; N/m^2
Q	volumetric flow rate, ft^3/sec; m^3/sec
R	radius, in.; cm
r	pressure ratio
T	absolute temperature, ^oR; K
ΔT	temperature difference, ^oR; K
U_t	tip speed, ft/sec; m/sec
W	weight, lbm; kg
w	mass flow rate, lbm/sec; kg/sec
ϵ_{th}	thermal emissivity
η	efficiency
η_A	alternator electromagnetic efficiency
η_{conv}	conversion efficiency, P_A/P_{sh}
η_{cy}	cycle efficiency, P_{sh}/P_{th}
η_{syst}	system efficiency, $P_{A,net}/P_{th}$
μ	viscosity, lb/(ft)(sec); kg/(m)(sec)
ρ	density, lb/ft^3; kg/m^3

Subscripts:

A	alternator
ad	adiabatic
C	compressor
id	ideal

p	polytropic
R	radiator
s	sink
T	turbine
1 to 6	cycle gas state points defined in fig. 1
$3l$	liquid coolant conditions at radiator inlet
$4l$	liquid coolant conditions at radiator outlet

APPENDIX B

CONVERSION SYSTEM LOSS ESTIMATES

The assumptions and procedures used in estimating the various parasitic losses in the Brayton conversion system are detailed in this section. In most cases, the estimates were based on the calculated design values of a 2- to 10-kilowatt electric Brayton power system under investigation at the NASA Lewis Research Center.

Bearing Cavity Gas Supply Loss

Two percent of the working fluid flow is assumed to be bled off at the compressor discharge to supply the bearing cavities. One-half of this fluid goes to the compressor bearing and is inserted back into the cycle at the compressor rotor tip. This portion of the flow introduces only a small loss and was disregarded. The other 1 percent of the working fluid supplies the turbine bearing cavity and is inserted back into the cycle at the turbine rotor tip. The effect of this flow on performance was determined by calculating the drop in turbine inlet temperature associated with the mixing of this flow at the compressor discharge temperature with the hot gas entering the turbine. Revised cycle temperatures reflecting changes resulting from 1 percent higher flow on the hot side of the recuperator as compared with that of the cold side and 1 percent lower flow through the heat source heat exchanger result in a cycle efficiency of about 0.955 times the cycle efficiency without bleed. This efficiency is equivalent to a loss of about 5 percent of the reactor thermal power.

Bearing Losses

Gas-lubricated bearings are assumed. Bearing friction losses were assumed to be 2 percent of the maximum gross shaft power of the module based on bearing loss calculations made on the 2- to 10-kilowatt-electric Brayton power system.

Pumping Power

The use of motor-driven rotating pumps with pump overall efficiencies of 0.35 was assumed. Other assumptions used in computing pump power were as follows:

Reactor coolant loop

Coolant . . . . Eutectic sodium-potassium mixture

Reactor outlet temperature, ^{o}R (K) 1760 (978)

Coolant temperature difference across reactor, ^{o}R (K) 200 (111)

Loop pressure drop, psi (N/m^2). . . . . 10 (6.9×10^4)

Intermediate loop

Coolant . . . . Eutectic sodium-potassium mixture

Maximum coolant temperature, ^{o}R (K) 1750 (972)

Coolant temperature difference across sodium-potassium to sodium-potassium heat exchanger, ^{o}R (K). 347 (193)

Loop pressure drop, psi (N/m^2). . . . . 10 (6.9×10^4)

Radiator loop

Coolant . . . . .Dimethyl polysiloxane (0.65 cS)

Loop pressure drop, psi (N/m^2) . . . . 25 (1.7×10^5)

Control Power

Control power requirements for the system were estimated to be 500 watts plus 3.0 percent of the alternator gross output.

Thermal Losses

Thermal losses from the system were assumed to be equal to 2 percent of the thermal input into the cycle.

Alternator Losses

An electromagnetic conversion efficiency of 0.92 was assumed for the alternator at design power. The efficiency was reduced to 0.915 and 0.90 for operation at three-fourths power and one-half power, respectively.

For purposes of windage loss computation, the dimensions of the alternator were scaled from a design for a 214-kilovolt-ampere alternator as follows: The alternator power was assumed to be proportional to the square of the rotor diameter D_A, the stator stack length L_A, and the rotational speed N:

$$P_A \propto D_A^2 L_A N \tag{B1}$$

or, in terms of L_A/D_A,

$$P_A \propto D_A^3 \frac{L_A}{D_A} N$$

The alternator rotor diameter may then be obtained as

$$D_A \propto P_A^{1/3} \left(\frac{L_A}{D_A}\right)^{-1/3} N^{-1/3} \tag{B2}$$

The term (L_A/D_A) was held constant, which left

$$D_A \propto \left(\frac{P_A}{N}\right)^{1/3} \tag{B3}$$

With the rotor diameter known, the other dimensions were obtained by assuming geometric similarity with the 214-kilovolt-ampere reference design. The rotor gap was assumed to be 1 percent of the rotor diameter, and the auxiliary gap was assumed to be 1 percent of the shaft diameter at the auxiliary gap.

The windage for rotating concentric cylinders is

$$\text{Windage} \propto \pi C_d \rho N^3 D_A^4 L_A \tag{B4}$$

The drag coefficient for the inner cylinder of concentric rotating cylinders was obtained from reference 9. For $400 < N_R < 10^4$,

0.46

$$C_d = \left[\frac{c}{R}\left(1 + \frac{c}{R}\right)\right]^{1/4} N_R^{-0.5} \tag{B5}$$

and for $10^4 < N_R < 10^5$,

0.073

$$C_d = \left[\frac{c}{R}\left(1 + \frac{c}{R}\right)\right]^{1/4} N_R^{-0.3} \tag{B6}$$

The Reynolds number is defined as

$$N_R \propto \frac{cRN\rho}{\mu} \tag{B7}$$

The windage at the rotor gap, auxiliary gap, and rotor transition section was calculated for each surface by using equations (B4) to (B6). The sum total of windage for all surfaces is the alternator windage P_w.

REFERENCES

1. Anon.: Report on the Design Requirements for Reactor Power Systems for Manned Earth Orbital Applications. Rep. DAC-57950, Douglas Aircraft Co. (NASA CR-66 254), Jan. 1967.

2. Rosenblum, M.: Design Requirements for Reactor Power Systems for Lunar Exploration. Vol. 1, Rep. LMSC-677879-I, Lockheed Missiles and Space Co. (NASA CR-90 000), Sept. 1967.

3. Rosenblum, M.: Design Requirements for Reactor Power Systems for Lunar Exploration. Vol. 2, Rep. LMSC-677879-II, Lockheed Missiles and Space Co. (NASA CR-89487), Sept. 1967.

4. Cunnington, G. R.; Grammer, J. R.; and Smith, F. J.: Emissivity Coatings for Low-Temperature Space Radiators. NASA CR-1420, 1969.

5. Bien, Darl D.; and Guentert, Donald C.: A Method for Reducing the Equivalent Sink Temperature of a Vertically Oriented Radiator on the Lunar Surface. NASA TM X-1729, 1969.

6. Glassman, Arthur J.: Thermodynamic and Turbomachinery Concepts for Radioisotope and Reactor Brayton-Cycle Space Power Systems. NASA TN D-2968, 1965.

7. Kofskey, Milton G.; and Glassman, Arthur J.: Turbomachinery Characteristics of Brayton-Cycle Space-Power-Generation Systems. Paper 64-GTP-23, ASME, Mar. 1964.

8. Vanco, Michael R.: Analytical Comparison of Relative Heat-Transfer Coefficients and Pressure Drops of Inert Gases and Their Binary Mixtures. NASA TN D-2677, 1965.

9. Wendt, F.: Turbulente Strömungen Zwischen Zwei Rotierenden Konaxialen Zylindern. Ing.-Archiv, vol. 4, no. 6, 1933, pp. 577-595.

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