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ANALYSIS OF A 35- TO 150-KILOWATT BRAYTON POWER-CONVERSION MODULE FOR USE WITH AN ADVANCED NUCLEAR REACTOR

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16. Abstract

Reference parameters were selected for a Brayton power-conversion module with a turbine inlet temperature of 1144 K (2060° R) to be used in a nuclear-reactor-powered system capable of producing up to 450 kilowatts, electric. Unshielded system specific weight is 41 kg/kWe (90 lb/kWe) with a specific radiator area of about 2.8 m^2/kWe (30 ft^2/kWe) at full power.

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ANALYSIS OF A 35- TO 150-KILOWATT BRAYTON POWER-CONVERSION MODULE FOR USE WITH AN ADVANCED NUCLEAR REACTOR

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SUMMARY

A study was made to examine the characteristics of a Brayton power-conversion module for use in an advanced-reactor-powered system capable of producing up to 450 kilowatts of net unconditioned power. A turbine inlet temperature of 1144 K (2060° R) was assumed, based on a reactor-coolant-outlet temperature of 1222 K (2200° R). Conversion-module maximum electric power output was 150 kilowatts, but performance down to a power level of 35 kilowatts was considered.

The relatively high module power level most significantly affected selection of system working fluid and turbomachinery rotational speed because of increased alternator stresses and windage loss. To best meet alternator windage loss and frequency requirements with relatively compact and efficient power-conversion modules, a turbomachinery rotational speed of 24 000 rpm and a helium-xenon mixture working fluid with a molecular weight of 39.94 were selected. Consideration of heat-transfer-component weight and radiator area requirements resulted in selection of a recuperator effectiveness of 0.925 and a system loss pressure ratio of 0.94 as reference values.

The overall efficiency of a power-conversion module operating at full power was 0.24 but dropped to 0.21 at 35 kWe. A system consisting of three conversion modules operating at full power can produce 450 kWe from a reactor thermal input of 1840 kWt. The unshielded system specific weight is 41 kg/kWe (90 lb/kWe), and the required radiator area is 2.8 m^2/kWe (30 ft^2/kWe).

INTRODUCTION

To give NASA the capability to perform future space missions requiring from 100 to 500 kilowatts of electric power, the development of large electric power systems for operation in space with a nuclear-reactor energy source is required. The inert-gas, closed Brayton-cycle, power-generation system is a prime candidate for use in space power applications now under consideration.

To meet space power needs from several kilowatts to hundreds of kilowatts, NASA is investigating a family of Brayton-cycle power-conversion modules. As the first step, for use in low-power-range applications of less than 100 kilowatts of electric power, NASA is investigating a 2- to 15-kWe Brayton power-conversion module which has been performance and endurance tested with good results (ref. 1) at the Space Power Facility of the Lewis Research Center. Selection of the operating conditions of this module was based on the use of a radioisotope heat source. Reference 2 presents the weight and efficiency of a Brayton power-conversion module coupled to a zirconium hydride reactor (SNAP-8) to produce approximately 100 kWe of power. Although good performance can be obtained from such a system, power output is limited by reactor thermal power capability, and radiator area could be reduced if reactor coolant outlet temperature were increased.

It was the purpose of this study to examine the performance and weight characteristics of a Brayton-cycle power-conversion module for use in a system producing approximately 100 to 500 kWe. Considerations relative to the selection of working-fluid molecular weight, pressure level, and turbomachinery diameters and rotational speed were also included. The assumed heat source is a nuclear reactor presently under study at Lewis Research Center. Thermal power capability of this reactor is approximately 2 megawatts. The reactor is designed to operate initially at a reactor outlet temperature of 1222 K (2200° R) and to have the capability to eventually operate at 1433 K (2560° R). The assumed turbine-inlet temperature of the Brayton-cycle working-fluid is 1144 K (2060° R). To meet the system power range of 100 to 500 kilowatts, the selected power range of a conversion module is 35 to 150 kilowatts of net system output power $P_{A,net}$. The corresponding gross alternator output power P_A range is 40 to 160 kilowatts. Several of these modules can be coupled to the heat source to produce a power-conversion-system output power to meet the particular mission requirement.

The results of this study were used as bases for more-detailed, contractor, conceptual-design programs for the turbomachinery and heat-transfer components (refs. 3 to 5).

METHOD OF ANALYSIS AND ASSUMPTIONS

Cycle Thermodynamic Characteristics

Figure 1 presents the temperature-entropy diagram, with state-point numbering for the closed Brayton thermodynamic cycle as used in this study. (All symbols are defined in appendix A.) Hot gas expands through the turbine from point 1 to point 2. It then

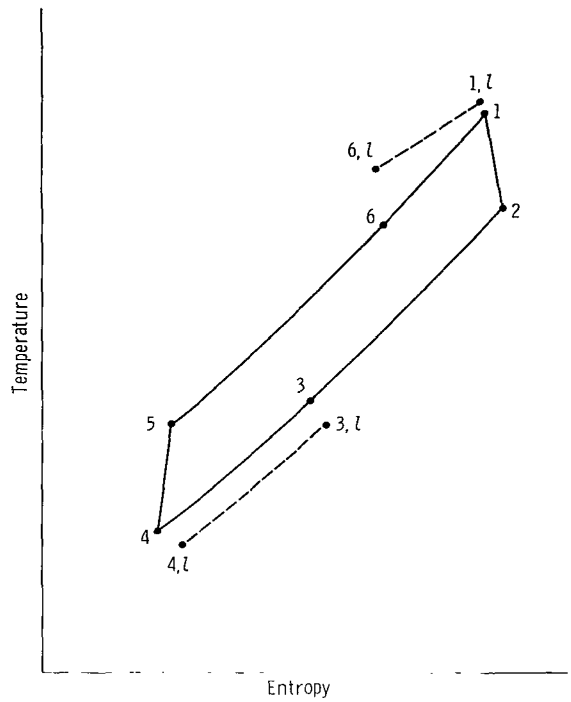

Figure 1. - Brayton-cycle temperature-entropy diagram for assumed configuration.

passes through the recuperator, where it is cooled to point 3. The gas is further cooled to compressor-inlet conditions at point 4. This cooling takes place in the waste heat exchanger, where heat is rejected to the liquid-filled radiator loop (point 3, l to 4, l). The gas is compressed to point 5 and heated in the recuperator to point 6. The gas is then further heated to turbine-inlet conditions at point 1. This heating is accomplished in the heat-source heat exchanger by an intermediate-loop (point 1, l to 6, l) fluid which separates the reactor coolant from the cycle working fluid. The excess power of the turbine over that required to drive the compressor drives the alternator and produces the useful power.

Cycle Thermodynamic Analysis and Assumptions

The Brayton-thermodynamic-cycle analysis is similar to that of reference 6. For the parametric-cycle analysis, the reference set of cycle parameters is presented in table I. The turbine-inlet temperature T_1 of 1144 K (2060° R) assumes a 77.8-K (140°-F) temperature differential between reactor-coolant-outlet temperature of 1222 K (2200° R) and the gas temperature at the turbine inlet. The assumed compressor poly-

TABLE I. - REFERENCE PERFORMANCE PARAMETERS

Turbine-inlet temperature, T_1, K (^{o}R)	1144 (2060)
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
Waste-heat-exchanger effectiveness, E_{HS}	0.95
Waste-heat-exchanger capacity-rate ratio, C_R	0.90
Radiator sink temperature, T_S, K (^{o}R)	278 (500)
Radiator-surface emittance, ϵ	0.90

tropic efficiency $\eta_{c,p}$ of 0.85 and the assumed turbine polytropic efficiency $\eta_{t,p}$ of 0.89 are estimates based on test data of similar machines. A system loss pressure ratio L of 0.94 was selected to favor cycle efficiency and reduce radiator area.

For high efficiency without large penalties in heat-exchanger size, a recuperator effectiveness E of 0.925 was selected. An organic-liquid-filled radiator loop has been assumed. To simplify radiator-area calculations, the film-temperature drop between radiator liquid and tube wall has been neglected. However, the liquid-film-temperature drop was considered in the more detailed radiator calculations which follow the cycle analysis. A waste-heat-exchanger effectiveness E_{HS} of 0.95 and a capacity-rate ratio C_R of 0.90 were assumed. An equivalent sink temperature T_S of 278 K (500^{o} R) is assumed to be representative of possible applications for lunar-surface or Earth-orbit missions. This sink temperature and a radiator-surface emittance ϵ of 0.90 are assumed achievable through the use of oxide coatings.

With this set of cycle parameters, a plot of specific prime radiator area as a function of cycle efficiency was used to select a reference cycle temperature ratio T_4/T_1 and compressor pressure ratio p_5/p_4 and the corresponding efficiency and radiator area. The sensitivity of cycle efficiency and radiator area to variations in some of these component parameters was then determined. The effect of variations in sink temperature, system loss pressure ratio, recuperator effectiveness, compressor efficiency, and turbine-inlet temperature were studied, and the results are presented in appendix B.

Power-Conversion-Module Considerations

A schematic drawing of the assumed power-system arrangement is presented in figure 2. The system consists of the power-conversion module coupled to the primary loop through an intermediate NaK loop and to the heat-rejection (radiator) loops. The primary loop includes the reactor, intermediate heat exchanger, and electromagnetic

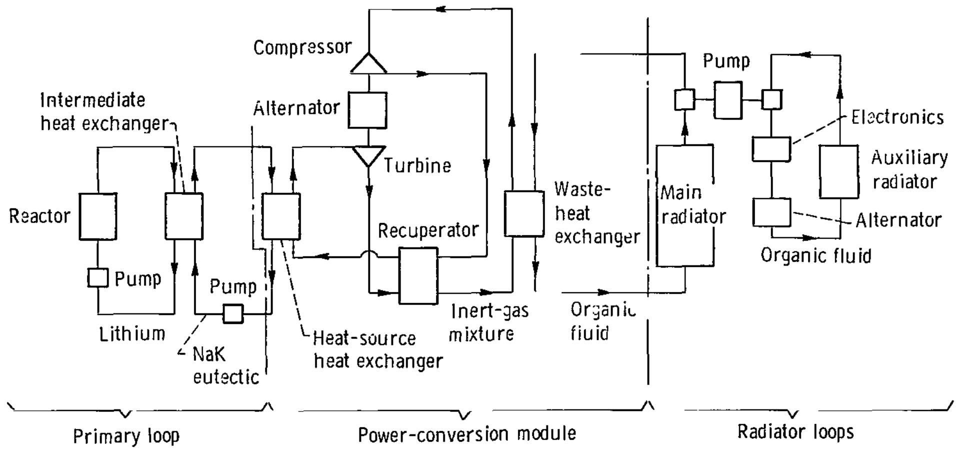

Figure 2. - Brayton-system schematic.

pumps. The power-conversion module includes all the gas-loop components. The heat-rejection loops include the main radiator for rejecting cycle waste heat, an auxiliary radiator for alternator and electronic-component cooling, and motor-driven pumps. Lithium was the assumed primary-loop fluid; NaK eutectic, the intermediate-loop fluid; and a superrefined mineral oil, the radiator-loop fluid. The Brayton-cycle working fluid was assumed to be a mixture of helium and xenon because of the improved heat-transfer properties of such mixtures over the pure gases (ref. 7). In this study, a design net system power of 150 kilowatts was assumed with off-design performance down to a power of 35 kilowatts.

The molecular weight of the cycle working fluid and the compressor inlet pressure were selected by consideration of their effects on turbomachinery size, rotational speed, performance, and heat-transfer-component size. The heat-transfer components were sized and the turbomachinery performance was determined on the basis of the following assumptions:

Turbomachinery. - A single-shaft rotating unit mounted on gas bearings was assumed. The single-stage, centrifugal compressor with backswept blading and the single-stage, radial-inflow turbine are cantilevered from opposite ends of a straddle-mounted, Lundell type alternator. A turbine specific speed of 0.620, dimensionless, $\left(80 \text{ (rpm/sec}^{1/2})(\text{ft-lbm/lbf})^{3/4}\right)$ was selected for near-optimum turbine and compressor efficiency (ref. 8). The alternator electromagnetic efficiency at design power of 150 kilowatts was assumed to be 0.92; alternator efficiency as a function of power is presented in figure 3. In order to minimize alternator cooling problems, a windage limit of 8 kilowatts, which corresponds to 5 percent of gross alternator output, was assumed. Windage losses were calculated by using the method presented in reference 2 and compare well with the Phase I results of the contractor studies presented in references 3 and 4. However,

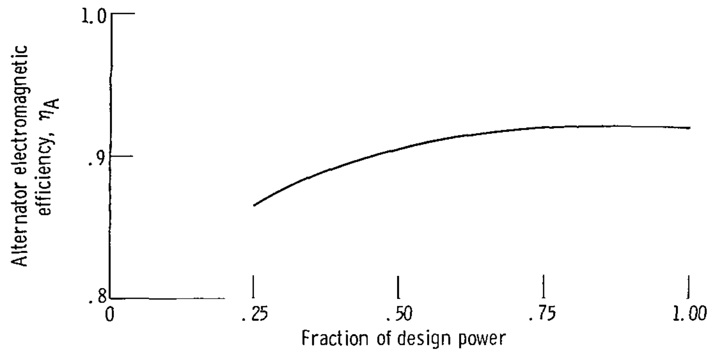

Figure 3. - Variation of alternator electromagnetic efficiency with power.

more recent data (ref. 9), which were obtained subsequent to the completion of this study and include slotted stator effects, indicate that the windage loss may be of the order of 75 percent higher. Alternator cavity pressure was made equal to compressor inlet pressure. The following additional criteria were assumed: a rotational speed compatible with an alternator frequency that is a multiple of 400 hertz; a reasonable turbomachinery tip diameter; and a reasonable pressure level, to reduce volumetric flow and heat-exchanger size.

Heat exchangers and radiator. - Weight calculations were made for all system heat exchangers by using unpublished, digital computer programs. Core models similar to those of the 2- to 15-kWe Brayton system were assumed. The recuperator and waste heat exchanger had plate-fin geometries, the NaK-to-gas heat-source heat exchanger had a disk-finned tubular geometry, and the lithium-to-NaK heat exchanger had a shell-and-tube geometry. The intermediate heat exchanger is not included in the power-conversion-module weight. Radiator weight and area calculations were made with the aid of an unpublished digital-computer program. A bumpered-finned-tube configuration with a redundant set of tubes was assumed. Fins and meteoroid armor were aluminum. The design probability that at least one set of tubes would remain unpunctured after 50 000 hours was 0.99. The radiator areas used represent a compromise between minimum area and minimum weight and are approximately 20 percent above minimum area. Radiator weights do not include any structural requirements. The auxiliary radiator is used for electronic and alternator cooling. The auxiliary radiator area was determined by assuming a heat load equal to the difference between gross shaft power P_{SH} and net alternator output $P_{A,net}$ and an average radiating temperature of 367 K (660° R).

Performance Characteristics of the Power-Conversion Module

Efficiency. - An estimate of the power-conversion-module efficiency, based on net system output, over a range of 35 to 150 kilowatts was obtained through the use of the loss estimates detailed in appendix C.

The compressor and turbine efficiencies and absolute bearing losses were determined at maximum power output for each rotational speed. They were assumed constant as module output power was varied while maintaining a constant compressor inlet pressure-to-shaft power ratio p_4/P_{SH}. System-loss pressure ratio and recuperator effectiveness were also held constant. Alternator electromagnetic conversion efficiency was varied as a function of power as shown in figure 3. The alternator windage loss was computed over the range of power by using the method described in reference 2.

Weight. - The power-conversion-module weight included some fixed components that were sized for maximum power and remained unchanged over the power range of 35 to 150 kilowatts. These fixed items included the rotating machinery, auxiliaries, and structure. An allowance of 454 kilograms (1000 lb) was made for auxiliaries which included the gas-management system, control system, and other small subsystems. The turbine-alternator-compressor assembly was assumed to weigh 363 kilograms (800 lb) at 24 000 rpm and 272 kilograms (600 lb) at 36 000 rpm. Structure, insulation, and wiring were assumed to be 15 percent of the module weight.

The heat exchangers and ducts were resized to maintain system-loss pressure ratio and effectiveness at each power level. Radiator area and weight were determined at each power level. Although required pumping power was varied with module power, a fixed pump weight of 227 kilograms (500 lb) per power-conversion module was included.

RESULTS

Cycle Considerations

The plot of specific prime radiator area as a function of cycle efficiency for the set of reference parameters is presented in figure 4. The performance curve is the envelope of individual curves of constant cycle temperature ratios, with the compressor pressure ratio as the variable. These individual curves are shown for cycle temperature ratios of 0.32, 0.34, 0.36, and 0.38. For the reference parameters, the minimum area of 1.04 m^2/kW (11.2 ft^2/kW) occurs at a cycle temperature ratio between 0.40 and 0.42 and a cycle efficiency of about 0.20

An operating cycle temperature ratio is selected by the user to give a reasonable trade-off between specific prime radiator area and cycle efficiency based on the performance requirements of his particular mission. Depending on mission requirements, a value of cycle temperature ratio in the range of 0.42 to 0.32 might be selected. This range represents radiator areas from minimum to 46 percent above minimum area. For

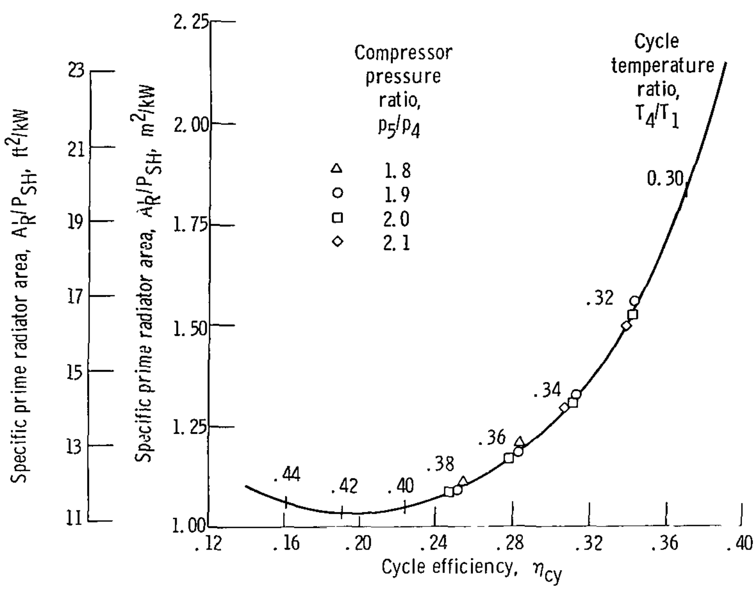

Figure 4. - Variation of specific prime radiator area with cycle efficiency. Turbine-inlet temperature, 1144 K (2060° R); system loss pressure ratio, 0.94; recuperator effectiveness, 0.925; waste-heat-exchanger effectiveness, 0.95; compressor polytropic efficiency, 0.85; turbine polytropic efficiency, 0.89; sink temperature, 278 K (500° R); radiator-surface emittance, 0.90.

this study, a cycle temperature ratio of 0.34, representing the middle of the area range, was selected as the reference value.

The flexibility of the Brayton system has been demonstrated in the operation of the 2- to 15-kWe test system over a range of cycle temperature ratios (ref. 1). To meet a particular mission requirement, a fixed conversion system can be made to operate efficiently over a range of cycle temperature ratios by varying compressor inlet temperature through changes in radiator area.

At the selected cycle temperature ratio of 0.34, a compressor pressure ratio of 2.0 corresponds to the tangency point with the performance envelope. However, because of turbomachinery design considerations, a compressor pressure ratio of 1.9, which requires a slightly larger specific prime radiator area at a higher efficiency, is selected to yield a more favorable system operating point. Therefore, the reference values of cycle temperature ratio and compressor pressure ratio are set at 0.34 and 1.9, respectively, with a corresponding cycle efficiency of 0.313 and a specific prime radiator area of 1.34 m^2/kW (14.4 ft^2/kW).

Power-Conversion-Module Considerations

The results of the analyses of the working-fluid molecular weight and the turbo-

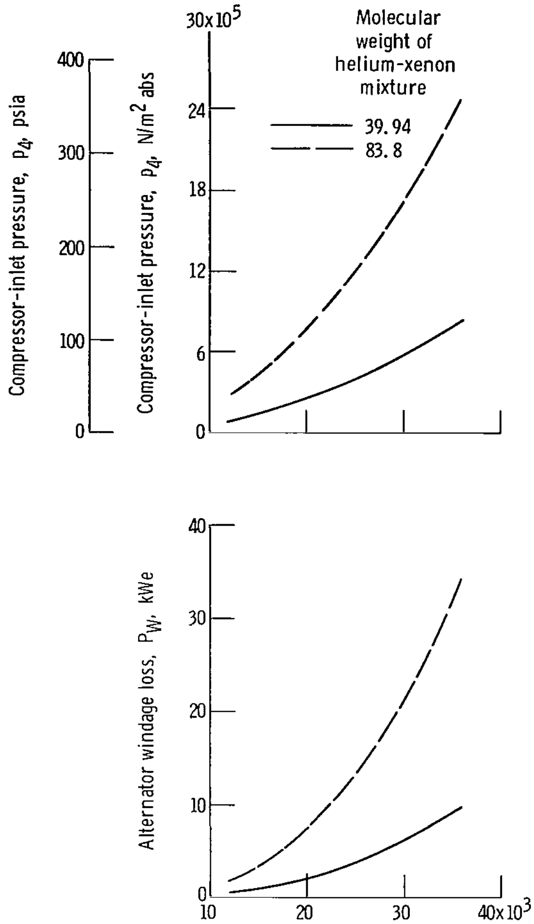

Figure 5. - Pressure-level and alternator-windage-loss characteristics of the power-conversion module. Net output power of system, 150 kWe; alternator efficiency, 0.92; compressor pressure ratio, 1.9; turbine specific speed 0.620, dimensionless $(80\ (rpm/sec^{1/2})(ft\text{-}lb/lbf)^{3/4})$.

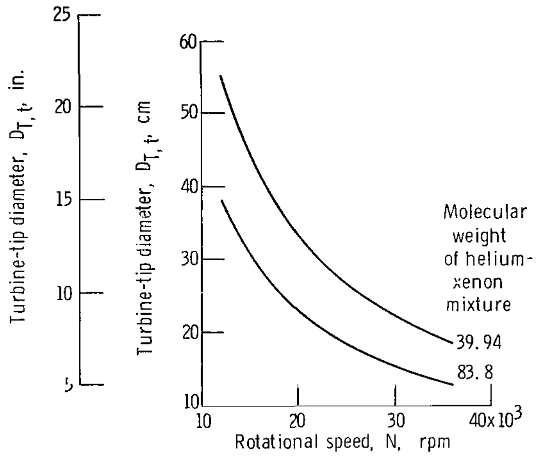

Figure 6. - Turbine-tip diameter of the power-conversion module. Net output power of system, 150 kWe; alternator efficiency, 0.92; compressor pressure ratio, 1.9; turbine specific speed, 0.620, dimensionless $(80\ (rpm/sec^{1/2})(ft\text{-}lb/lbf)^{3/4})$.

machinery rotational speed for the 150-kilowatt power-conversion module are presented in figures 5 and 6. Helium-xenon mixtures with molecular weights of 39.94 and 83.8 (corresponding to argon and krypton) were considered as possible working fluids in this study. These represent medium and high molecular weights. A low-molecular-weight mixture corresponding to neon (20.18 g/g mole) was not considered because the high turbine tip speed (approximately 500 m/sec, or 1600 ft/sec) required with a single-stage, radial-inflow turbine results in high turbine stresses detrimental to long life at the high operating temperature. The effect of rotational speed on compressor-inlet pressure and on alternator-windage loss for the two working-fluid molecular weights is shown in figure 5. The turbine tip diameter is shown as a function of rotational speed in figure 6. For a compressor with backswept blading, the compressor tip diameter is slightly less than that of the turbine.

A working fluid with a molecular weight of 39.94 was selected as best fulfilling the design criteria. The use of a gas mixture with a molecular weight of 83.8 requires low rotational speeds to keep windage losses at an acceptable level, and this results in large turbomachinery diameters. The high-molecular-weight mixture also has the disadvantage of poorer heat-transfer properties than the 39.94 mixture (ref. 7), so that the heat-transfer components are penalized (ref. 5). In a comparison of systems with equal pressure levels, the heat exchangers for the fluid with the higher molecular weight (83.8) will be larger than those sized for the same pressure drops with the 39.94-molecular-weight fluid. At the same rotational speed, the higher-molecular-weight fluid produces higher compressor inlet pressures that compensate for the poorer heat-transfer properties, but alternator windage losses are also higher because of the high pressure and high molecular weight.

For the 39.94-molecular-weight fluid, 24 000 rpm was selected as the highest speed meeting the alternator frequency requirements and the alternator windage loss limit. For example, frequencies of either 400 hertz with a two-pole alternator or 1200 hertz with a six-pole alternator are obtainable. At this speed, the turbine tip diameter was 27 centimeters (11 in.) and the windage loss was 4 kilowatts (the more recent data in ref. 9 indicate the design windage loss may be as high as 7 kW) at the compressor inlet pressure of 38 N/cm^2 abs (55 psia).

At a rotational speed of 36 000 rpm, a frequency of 1200 hertz is obtainable with a four-pole alternator. A rotational speed of 36 000 rpm provides the advantages of more compact turbomachinery, as shown in figure 6, and smaller heat exchangers and ducting because of the increased compressor inlet pressure of 83 N/cm^2 abs (120 psia). However, at this higher speed, and with a working-fluid molecular weight of 39.94, the alternator windage loss exceeded the assumed limit. Because of the advantages of system compactness and lower weight with 36 000-rpm turbomachinery, the possibility of reducing windage loss by reducing alternator cavity pressure was investigated. It was found that the use of a jet-ejector pump utilizing compressor discharge bleed flow in combination with labyrinth seals and low-leakage lift-off seals to reduce alternator cavity pressure was a reasonable approach to the problem, with little penalty in system performance.

Preliminary calculations indicated that windage losses could be reduced to 6 kilowatts with a pumping-fluid flow of approximately 1 percent of system working-fluid flow rate. Therefore, 36 000 rpm was also selected as a possible rotational speed, pending a further, more detailed analysis of the ejector pump, seal concepts, and alternator rotor stresses.

Selection of Power-Conversion-Module Parameters

To aid in the final selection of the cycle temperature ratio, system-loss pressure ratio, and recuperator effectiveness, the sensitivity of the heat-transfer-component weights of the power-conversion module to variations in these parameters was determined.

Cycle temperature ratio. - The effect of cycle temperature ratio on heat-transfer-component weight, radiator area A_R, and cycle efficiency is shown in figure 7 for a molecular weight of 39.94 and rotational speeds of 24 000 and 36 000 rpm. The curve of

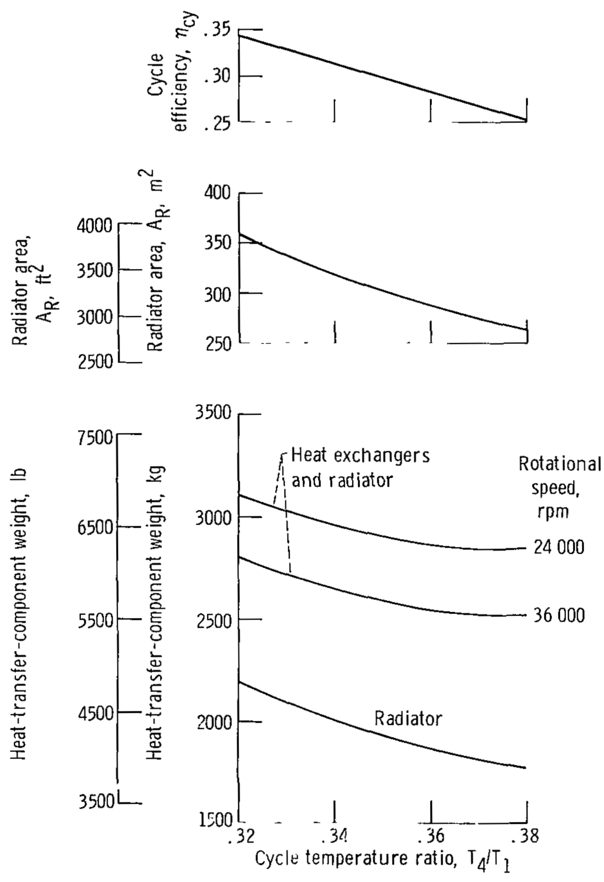

Figure 7. - Effect of cycle temperature ratio on heat-transfer-component weight, radiator area, and cycle efficiency for a 150-kWe power-conversion module with a working-fluid molecular weight of 39.94.

total heat-transfer-component weight (including radiator) minimizes at a cycle temperature ratio of 0.37 for both rotational speeds. The minimum total weights are 2840 kilograms (6250 lb) at 24 000 rpm and 2520 kilograms (5550 lb) at 36 000 rpm. At a cycle temperature ratio of 0.34, the total weights are approximately 4 percent above minimum, while radiator area and cycle efficiency have both increased by approximately 16 percent.

Recuperator effectiveness and system loss pressure ratio. - The effect of recuperator effectiveness and system loss pressure ratio on relative heat-transfer-component weight and radiator area is presented in figure 8 for effectiveness values of 0.90, 0.925, and 0.950 and loss pressure ratios of 0.92, 0.94, and 0.96. The sum of the weights of the radiator and the power-conversion-module heat exchangers is plotted as a function of cycle efficiency. All weights and areas are normalized to a value of 1 at a recuperator effectiveness of 0.925 and a loss pressure ratio of 0.94. Since the recuperator is the largest heat exchanger, the variations in total weight are due primarily to changes in recuperator and radiator weight resulting from changes in effectiveness and loss pres-

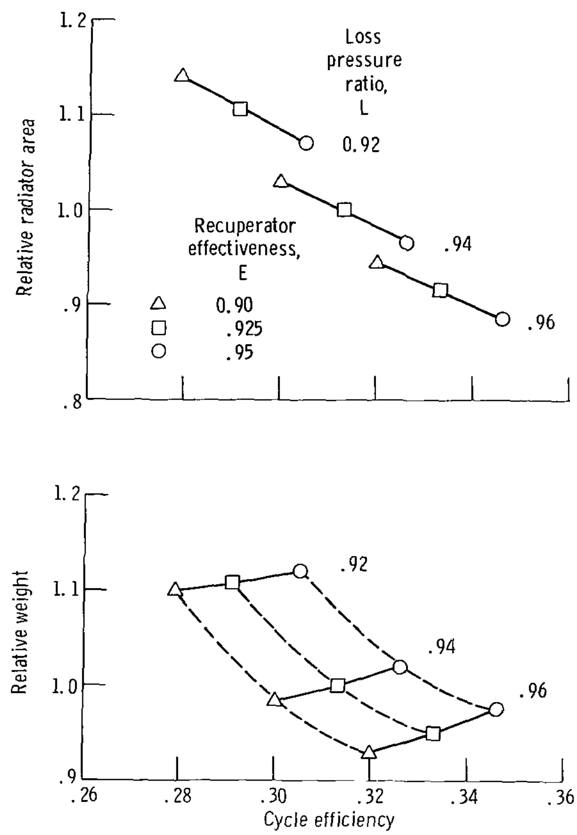

Figure 8. - Effect of recuperator effectiveness and system loss pressure ratio on heat-transfer-component weight and radiator area based on a loss pressure ratio of 0.94 anu a recuperator effectiveness of 0.925.

sure ratio. The radiator weight is dominant, being approximately three-fourths of the total weight. The total weight optimizes at high values of loss pressure ratio because of the significant effect that loss pressure ratio has on radiator area and efficiency. The total weight is still decreasing at a loss pressure ratio of 0.96. To be conservative and because of the uncertainty in achieving the low component pressure drops associated with a loss pressure ratio of 0.96 in actual hardware, a value of 0.94 was retained. Subsequent contractor studies have verified these trends of weight with loss pressure ratio and indicate that a value of 0.96 or above could be selected (ref. 5).

At a loss pressure ratio of 0.94, total weight and radiator area vary only slightly over the range of recuperator effectiveness. While the lowest weight occurs at an effectiveness of 0.90, the least radiator area is required at 0.95. The lowest weight occurs at the lowest effectiveness because the relatively high turbine-inlet temperature tends to reduce radiator weight relative to recuperator weight. Reductions in radiator weight obtained by increasing effectiveness are less than the increased weight of the recuperator. Although the heat-transfer components (including radiator) are lightest at an effectiveness of 0.90, the efficiency and radiator area penalties make it a poor choice. Selection of an effectiveness of 0.95 favors cycle performance and radiator area at the expense of heat-exchanger compactness. Based on required heat-transfer units; a recuperator with an effectiveness of 0.95 is approximately 50 percent larger and heavier than one with an effectiveness of 0.925.

Anticipating a possible increase in system loss pressure ratio to 0.96 or above, a recuperator effectiveness of 0.925 is retained as the reference value. The crossplots at constant effectiveness, shown in figure 8, indicate that at a lower effectiveness, minimum weight occurs at higher values of L.

The selected reference parameters for the power-conversion module are, therefore, identical to the initial values shown in table I.

Performance Characteristics of the Power-Conversion Module

Efficiency. - System efficiency as a function of net system output power is shown in figure 9 for rotational speeds of 24 000 and 36 000 rpm. At full power of 150 kWe, the system efficiencies are approximately 0.24 for both rotational speeds. The efficiency at 36 000 rpm is slightly lower because of the increased alternator windage loss. Although the increased windage loss had little effect on system efficiency, it does have a significant effect on the alternator-cooling load, which is reflected in increased radiator area. At low power, the efficiency drops off rapidly because of the fixed bearing losses, the fixed portion of the control power, and the decreasing alternator efficiency. At 35 kilowatts, the efficiency of a 36 000-rpm module is 0.20 compared to 0.21 for a 24 000-rpm module.

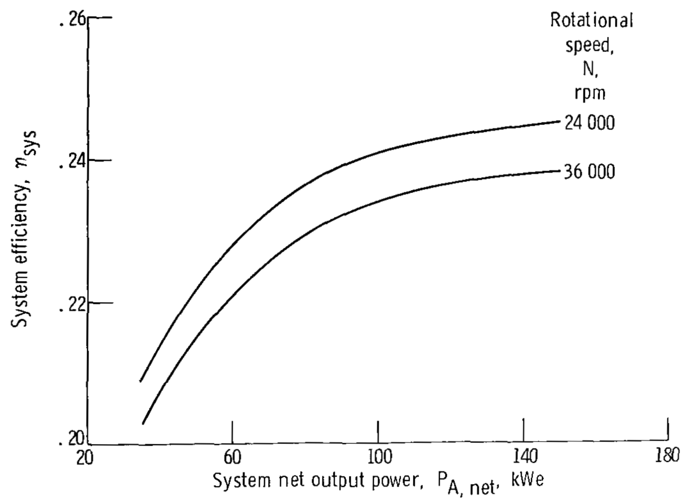

Figure 9. - Effect of module power on efficiency.

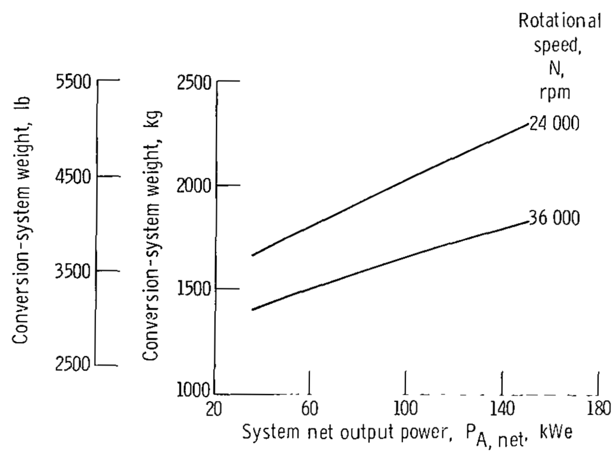

Figure 10. - Effect of power level on power-conversion-module weight.

Weight. - The power-conversion-module weights over a net system power range of 35 to 150 kilowatts are presented in figure 10. Two different power-conversion modules were considered. One used turbomachinery operating at 24 000 rpm, while the other had 36 000-rpm turbomachinery. The conversion module using 36 000-rpm turbomachinery is lighter, weighing about 1800 kilograms (4000 lb), or 12.1 kg/kWe (26.7 lb/kWe), at 150 kilowatts of electric power, and 1400 kilograms (3100 lb), or 40.1 kg/kWe (88.5 lb/kWe), at 35 kilowatts of electric power. The heavier module, using 24 000-rpm machinery, weighs about 2300 kilograms (5050 lb), or 15.3 kg/kWe (33.7 lb/kWe), at 150 kilowatts electric power, and about 1660 kilograms (3650 lb), or 47.2 kg/kWe

(104 lb/kWe), at 35 kilowatts of electric power. At full power, the use of 36 000-rpm turbomachinery yields a 21-percent saving in conversion-module weight.

Although a 36 000-rpm conversion module is attractive from the standpoint of weight and compactness, the stresses in large-diameter, Lundell type alternator rotors are of serious concern (refs. 3 and 4). For conservatism, the 24 000-rpm conversion module is used to determine overall system weight and performance.

Radiator. - The area $A_{R,tot}$ and weight requirements of the power-conversion-module radiator are shown in figure 11. The total radiator area includes the area required to reject cycle waste heat and the additional area required for cooling the alternator and electronic components. Weight and area are presented over the module power range of 35 to 150 kilowatts for turbomachinery rotational speeds of 24 000 and 36 000 rpm. At full power of 150 kilowatts for the 24 000-rpm module, the required area is 409 square meters (4400 ft^2), and the weight is 2570 kilograms (5560 lb). Because of the higher windage losses and lower efficiency of the 36 000-rpm module, required radiator area and also weight increase by approximately 5 percent over the entire power range.

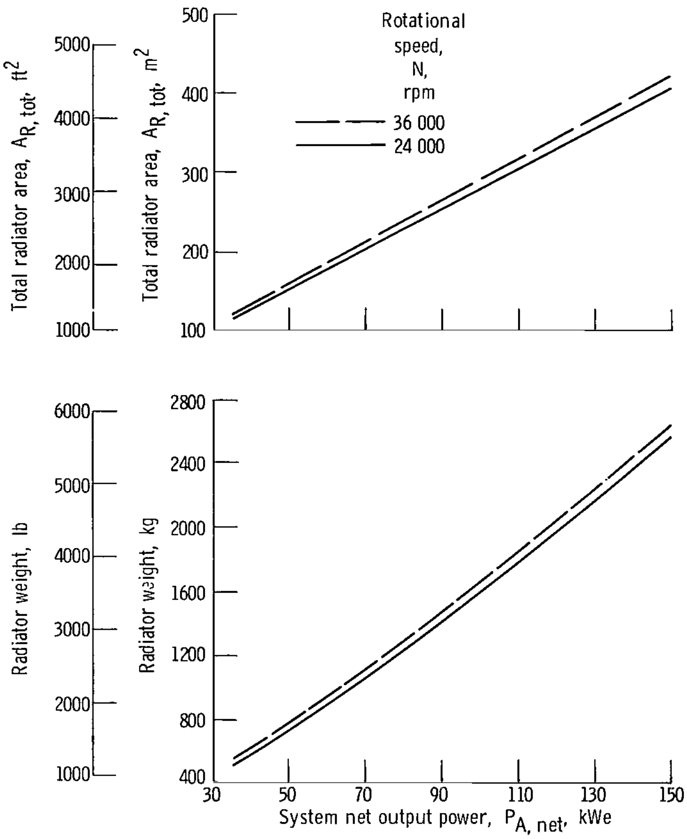

Figure 11. - Effect of module power level on radiator area and weight.

System Considerations

Some of the important characteristics of the system using the 24 000-rpm turbomachinery are presented in figure 12. The system consists of the reactor and primary loop components, one to three power conversion modules, and the waste heat rejection loop components (radiator and pumps). System weight includes an 1810-kilogram (4000-lb) reactor allowance but no shielding. The use of multiple power-conversion modules yields a net alternator output power range of 35 to 450 kilowatts (fig. 12(a)). Electric power outputs are net unconditioned power. The specific radiator area $A_{R,tot}/P_{A,net}$ is total radiator area divided by net system power, and the system specific weight $W_T/P_{A,net}$ is total system weight divided by net system power.

At the low end of the power range, 170 kilowatts of reactor thermal power are required to produce 35 kilowatts of net electric power. At this power, the specific radiator area shown in figure 12(b) is 3.25 m^2/kWe (35 ft^2/kWe), and the unshielded system specific weight shown in figure 12(c) is 113 kg/kWe (250 lb/kWe).

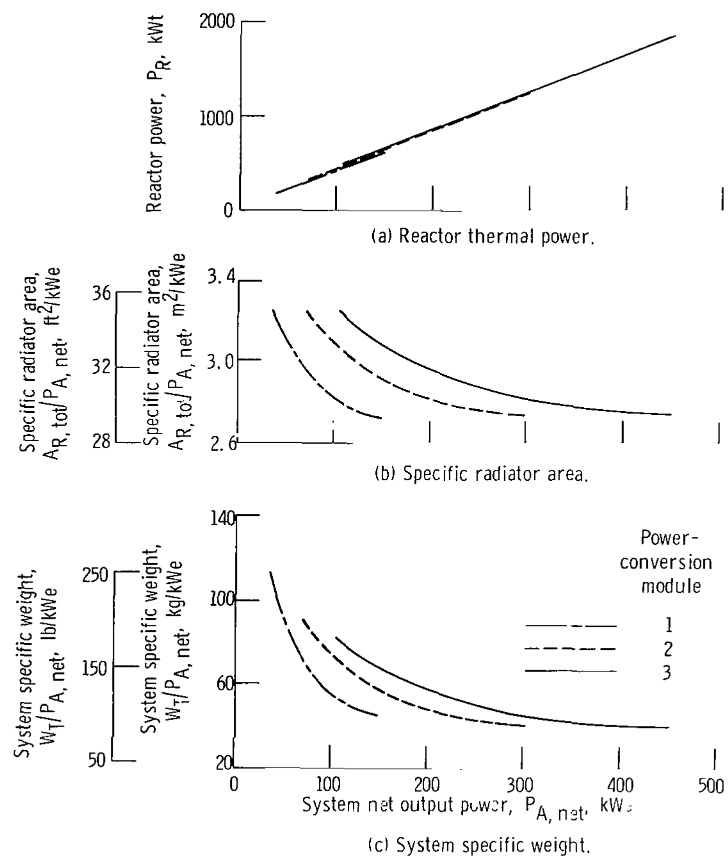

Figure 12. - Unshielded system performance summary for 24 000-rpm turbomachinery.

Three modules operating at full power produce 450 kilowatts of electric power from 1840 kilowatts of reactor thermal power. Specific radiator area is 2.74 m^2/kWe (29.5 ft^2/kWe), while unshielded system specific weight is reduced to 40.8 kg/kWe (90 lb/kWe).

CONCLUDING REMARKS

The results of the study indicated that the conversion-module power level has a significant effect on the selection of working-fluid molecular weight and turbomachinery rotational speed. A helium-xenon gas mixture with a molecular weight of 39.94 was selected as the working fluid primarily on the basis of pressure-level and windage considerations. At a power level of 150 kWe, the assumed alternator windage loss considerations and frequency requirements limit rotational speed to a maximum of 24 000 rpm. Consideration of the weight and the radiator-area requirements of the heat-transfer components resulted in a selection of a recuperator effectiveness of 0.925 and a system loss pressure ratio of 0.94 as reference values.

A system with a turbine-inlet temperature of 1144 K (2060° R) and with three 150-kilowatt modules can produce 450 kilowatts of net unconditioned power from a reactor thermal power of 1840 kilowatts. The use of a single 150-kilowatt module permits operation in applications down to a required power level of 35 kilowatts with an overall system efficiency of 0.21. At the 450-kilowatt level, radiator area is about 2.8 m^2/kWe (30 ft^2/kWe) including alternator and electronics cooling requirements. The unshielded system weight is 41 kg/kWe (90 lb/kWe).

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, July 21, 1971,
112-27.

APPENDIX A

SYMBOLS

A_R radiator area with fin, m^2; ft^2

$A_{R,tot}$ total radiator area (including auxiliary radiator) with fin, m^2; ft^2

A'_R prime radiator area, m^2; ft^2

C_R capacity-rate ratio of waste-heat exchanger, $(mc_p)_g/(mc_p)_l$

c_p specific heat at constant pressure, J/(kg)(K); Btu/(lb)(^oR)

D diameter, m; in.

E recuperator effectiveness

E_{HS} waste-heat-exchanger effectiveness

g acceleration due to gravity, 9.81 m/(sec^2)(kg/N); 32.2 ft/sec^2

ΔH specific-enthalpy change, J/kg; (ft)(lb)/lb

J mechanical equivalent of heat, 1 N-m/J; 778 (ft-lb)/Btu

L system loss pressure ratio, $(p_1/p_2)(p_4/p_5)$

m mass flow rate, kg/hr; lb/hr

N rotational speed, rpm

N_S specific speed, $\dfrac{\pi N\sqrt{Q}}{30(gJ\,\Delta H_{id})^{3/4}}$, dimensionless; $\dfrac{N\sqrt{Q}}{(\Delta H_{id})^{3/4}}$, $\left[(rpm)/(sec)^{1/2}\right]$ (ft-lbm/lbf)$^{3/4}$

P_A alternator gross output power, kWe

$P_{A,net}$ system net output power, $P_A - P_p - P_{cont}$, kWe

P_{cont} control power, kWe

P_p pump power, kWe

P_R reactor thermal power, kWt

P_{SH} gross shaft power, $\eta_{cy}(P_{TH})$, kW

P_{TH} thermal power into cycle, kWt

P_W alternator windage loss, kWe

p pressure, N/m^2; psi

Q	volumetric flow rate, m^3/sec; ft^3/sec
T	temperature, K; ^oR
T_S	effective radiator sink temperature, K; ^oR
W_T	total system weight, kg; lb
ϵ	surface hemispherical emittance
η	efficiency
η_{cy}	cycle efficiency, P_{SH}/P_{TH}
η_{sys}	system efficiency, $P_{A,net}/P_R$

Subscripts:

A	alternator
c	compressor
cy	cycle
g	gas
id	ideal
l	liquid
net	net
p	polytropic
sys	system
T	tip
t	turbine
1	turbine inlet
2	turbine outlet
3	gas-side inlet of waste-heat exchanger
3, l	liquid-side outlet of waste-heat exchanger
4	compressor inlet
4, l	liquid-side inlet of waste-heat exchanger
5	compressor outlet
6	inlet of heat-source heat exchanger

APPENDIX B

CYCLE SENSITIVITY EFFECTS

The sensitivity of cycle efficiency and specific prime radiator area to variations in several of the cycle parameters is presented in this section. The effect of variations in sink temperature, recuperator effectiveness, system loss pressure ratio, compressor efficiency, and turbine inlet temperature are shown. A single parameter is varied while the remaining parameters retain the values presented in table I.

Sink Temperature

The effect of sink temperature on specific prime radiator area and cycle efficiency is shown in figure 13. The sink temperature range from 250 to 306 K (450° to 550° R) represents a range of possible applications from Earth orbit to the lunar surface (ref. 10). For the three values of sink temperature, the performance-envelope curves are shown with cycle-temperature-ratio contours. At cycle temperature ratios of 0.32, 0.34, 0.36, and 0.38, compressor-pressure-ratio curves are included. Minimum area occurs at a cycle temperature ratio of approximately 0.41 and an efficiency of 0.20 for all three sink temperatures.

Recuperator Effectiveness

Performance envelopes of the specific prime radiator area as a function of cycle efficiency are presented in figure 14. Compressor pressure ratio is not held constant. Cycle-temperature-ratio contours are included. Minimum specific prime radiator area increases by about 5 percent as the recuperator effectiveness is reduced from 0.95 to 0.90. Specific prime radiator area is not very sensitive to changes in recuperator effectiveness because reduced recuperator performance adds heat load to the high-temperature end of the radiator. Since this is the most efficient part of the radiator, the added heat load can be rejected with only a small increase in area. The recuperator-effectiveness range shown does not significantly affect the cycle temperature ratio or efficiency at the minimum-area point.

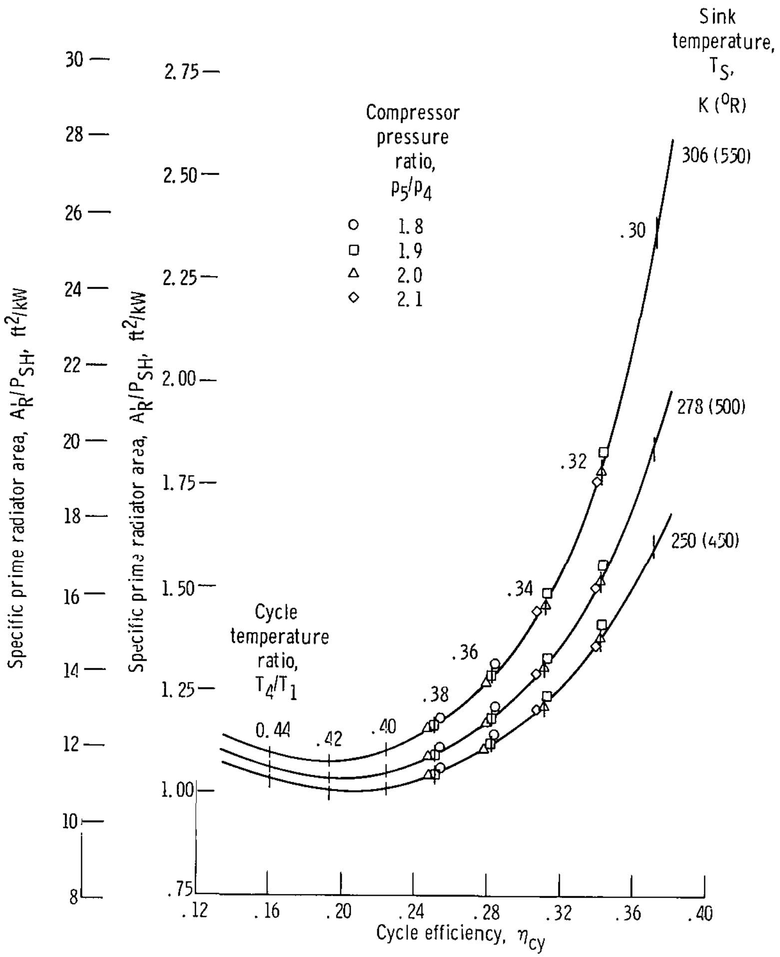

Figure 13. - Effect of sink temperature on thermodynamic-cycle performance. Turbine-inlet temperature, 1144 K (2060° R); compressor polytropic efficiency, 0.85; turbine polytropic efficiency, 0.89; system loss pressure ratio, 0.94; recuperator effectiveness, 0.925; waste-heat-exchanger effectiveness, 0.95; capacity-rate ratio of waste-heat exchanger, 0.90; surface hemispherical emittance, 0.90.

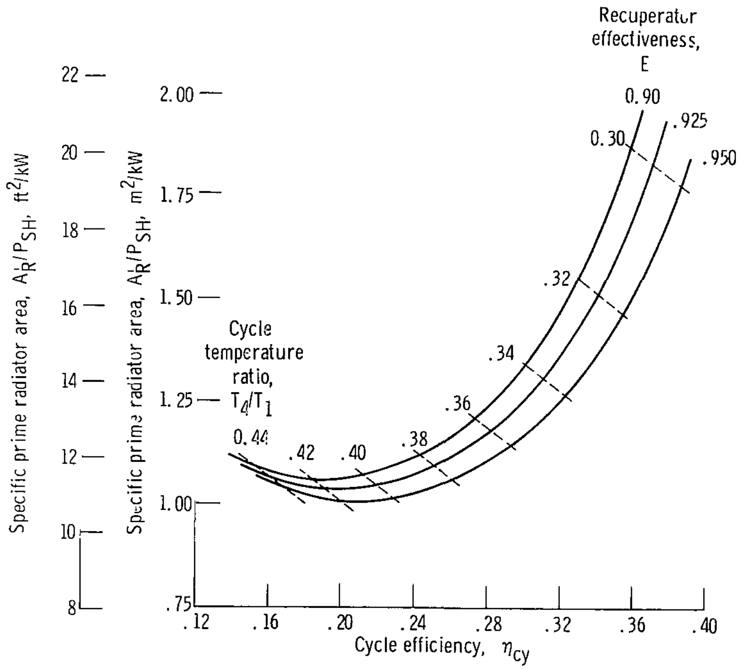

Figure 14. - Effect of recuperator effectiveness on thermodynamic-cycle performance. Turbine-inlet temperature, 1144 K (2060° R); compressor polytropic efficiency, 0.85; turbine polytropic efficiency, 0.89; system loss pressure ratio, 0.94; waste-heat-exhanger effectiveness, 0.95; capacity-rate ratio of waste-heat exchanger, 0.90; effective radiator sink temperature, 278 K (500° R); surface hemispherical emittance, 0.90.

System Loss Pressure Ratio

Envelopes of the specific prime radiator area as a function of cycle efficiency for system loss pressure ratio values of 0.92, 0.94, and 0.96 are shown in figure 15. System loss pressure ratio L has a significant effect on specific prime radiator area. Based on the reference value of 0.94, a decrease in L to 0.92 increases minimum area by 17 percent, while an increase in L to 0.96 results in a 15-percent decrease in minimum area. Cycle temperature ratio at the minimum-area point changes considerably with changes in L, but the cycle efficiency at that point is almost constant.

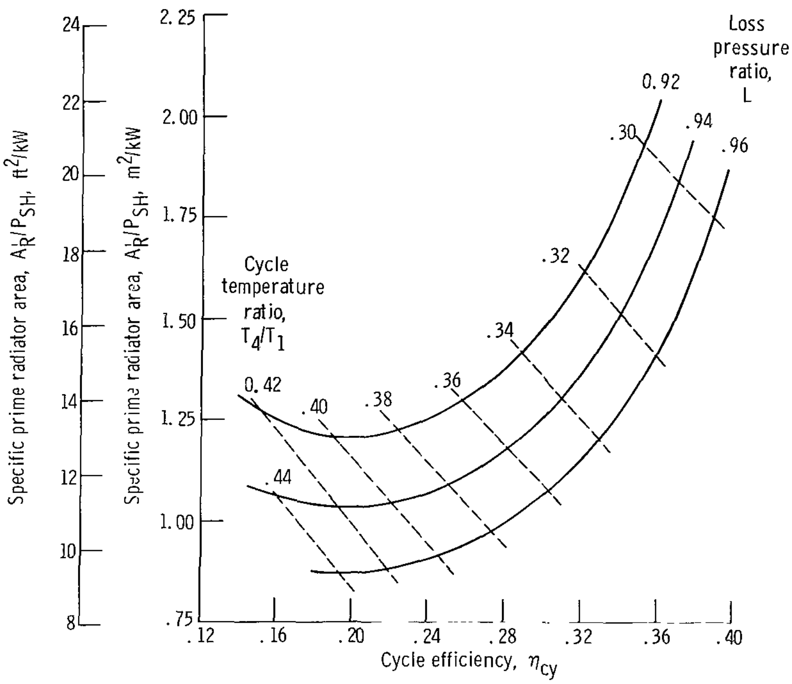

Figure 15. - Effect of system loss pressure ratio on thermodynamic-cycle performance. Turbine-inlet temperature, 1144 K (2060° R); compressor polytropic efficiency, 0.85; turbine polytropic efficiency, 0.89; recuperator effectiveness, 0.925; waste-heat-exchanger effectiveness, 0.95; capacity-rate ratio of waste-heat exchanger, 0.90; effective radiator sink temperature, 278 K (500° R); surface hemispherical emittance, 0.90.

Compressor Efficiency

The effect of improved compressor efficiency on system performance can be seen by comparing the two performance envelopes shown in figure 16. The reference performance envelope shown in the upper curve can be achieved with a compressor polytropic efficiency of 0.85 (adiabatic efficiency of 0.83 at a pressure ratio of 1.9), which is representative of centrifugal compressors with backswept rotor blading. The lower curve assumes a polytropic efficiency of 0.88 (adiabatic efficiency of 0.864 at a pressure ratio of 1.9), which may be achievable with a multistage axial compressor or by improvements in large radial machine design. The use of an axial compressor with a large number of stages for high efficiency could result in more complicated turbomachinery designs. The cycle temperature ratio at the minimum-area point increases with increasing compressor efficiency while the cycle efficiency remains approximately constant.

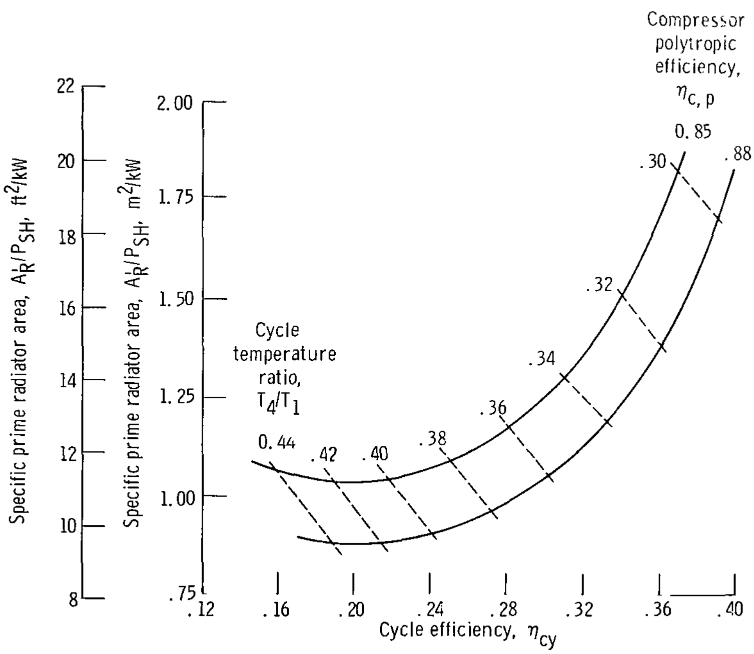

Figure 16. - Effect of compressor efficiency on thermodynamic-cycle performance. Turbine-inlet temperature, 1144 K (2060° R); turbine polytropic efficiency, 0.89; system loss pressure ratio, 0.94; recuperator effectiveness, 0.925; waste-heat-exchanger effectiveness, 0.95; capacity-rate ratio of waste-heat exchanger, 0.90; effective radiator sink temperature, 278 K (500° R); surface hemispherical emittance, 0.90.

Turbine-Inlet Temperature

The effect of turbine-inlet temperature on the envelope of specific prime radiator area as a function of cycle efficiency is shown in figure 17. Each curve assumes a system designed for that particular turbine-inlet temperature. The three temperatures shown are representative of levels which might be achieved with different reactors. The turbine-inlet temperature of 894 K (1610° R) assumes the use of the zirconium hydride (SNAP-8) reactor which is presently under development. The turbine-inlet temperature of 1422 K (2560° R) may be achievable with a second generation of the advanced lithium-cooled reactor. Comparison of the curves indicates the large increase in required radiator area at the low-turbine inlet temperature. As turbine-inlet temperature increases, the minimum-area point occurs at lower cycle temperature ratios and higher cycle efficiencies.

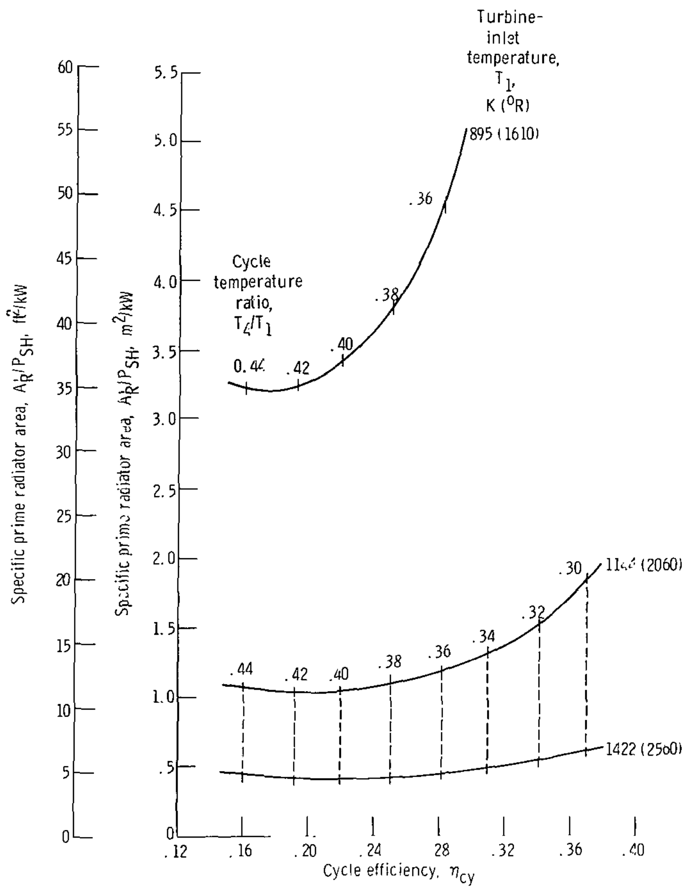

Figure 17. - Effect of turbine-inlet temperature on thermodynamic-cycle performance. Compressor polytropic efficiency, 0.85; turbine polytropic efficiency, 0.89; system loss pressure ratio, 0.94; recuperator effectiveness, 0.925; waste-heat-exchanger effectiveness, 0.95; capacity-rate ratio of waste-heat exchanger, 0.90; effective radiator sink temperature, 278 K (500° R); surface hemispherical emittance, 0.90.

APPENDIX C

POWER-CONVERSION-MODULE LOSS ESTIMATES

The assumptions and procedures used in estimating the various parasitic losses in the Brayton conversion system are presented in this section. Loss estimates were based on the design values of the 2- to 15-kilowatt electric Brayton power system under test at 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. Half of this fluid goes to the compressor bearing and is directed back into the cycle at the compressor rotor tip. This portion of the flow introduces only a small loss and was disregarded. The other half of the bleed flow supplies the turbine bearing cavity and is directed 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, which is 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 the flow on the cold side and 1-percent lower flow through the heat-source heat exchanger result in a loss of 3 percent of the gross shaft power.

Bearing, Windage, and Seal Losses

Alternator windage loss estimates were calculated by the method outlined in reference 2.

24 000-rpm turbomachinery. - Gas bearings operating at compressor discharge pressure are assumed. Based on the 2- to 15-kilowatt electric Brayton power system, the bearing-friction and seal losses are estimated to be 2 percent of the maximum gross shaft power of the module. Alternator windage losses of 4 kilowatts at maximum power were calculated for a reference Lundell alternator with a 7.5-inch-diameter rotor.

36 000-rpm turbomachinery. - At a rotational speed of 36 000 rpm and with the alternator cavity at compressor discharge pressure, the windage loss is 10 kilowatts. To reduce the alternator heating effect, the use of an ejector pump and low-clearance seals to reduce cavity pressure was assumed. Preliminary calculations indicate that an ejec-

tor pump using 1.2 percent of the cycle working fluid at compressor discharge pressure could reduce alternator cavity pressure to a point where windage losses are 6 kilowatts. The use of 1.2 percent of the working fluid for pumping represents an additional loss of 1.8 percent of the gross shaft power. Bearing and seal losses were assumed to be 2 percent of the maximum gross shaft power of the module.

Thermal Losses

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

Control Power

Control-power requirements for the system were estimated to be 500 watts plus 3 percent of the alternator gross output.

Pumping Power

The use of electromagnetic pumps in the reactor coolant and intermediate loops was assumed. The overall pump efficiencies were 10 and 7 percent, respectively. In the waste-heat-rejection loop, use of a motor-driven, rotating pump having an overall efficiency of 0.35 was assumed. The following additional assumptions were used for computing the pump power:

Reactor coolant loop:

- Coolant . . . . Lithium
- Reactor outlet temperature, K (^{o}R) 1222 (2200)
- Coolant temperature rise through reactor, K (^{o}R). 55.5 (100)
- Loop pressure drop, N/m^2 abs (psia) . . . . 3.45×10^4 (5)

Intermediate loop:

- Coolant . . . . Sodium-potassium eutectic (NaK)
- Maximum coolant temperature, K (^{o}R). 1183 (2130)
- Coolant temperature rise in lithium-to-NaK heat exchanger, K (^{o}R) 244 (440)
- Loop pressure drop, N/m^2 abs (psia) . . . . 3.45×10^4 (5)

Radiator loop:

- Coolant . . . . Organic
- Loop pressure drop, N/m^2 abs (psia) . . . . 2.07×10^5 (30)

Pump power was assumed to vary with module power level.

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