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Megawatt Class Nuclear Space Power Systems (MCNSPS) Conceptual Design and Evaluation Report

Volume III—Technologies II: Power Conversion

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TABLE OF CONTENTS

4.5 POWER CONVERSION SYSTEMS

The major power conversion concepts to receive detailed consideration in this study are:

1) Rankine alkali metal vapor turbine-alternators;
2) In-core thermionic conversion;
3) Brayton gas turbine-alternators; and
4) Free piston Stirling engine-linear alternators.

In this volume the considerations important to the coupling of these four power conversion alternatives to an appropriate nuclear reactor heat source, together with the comparative performance characteristics of the combined systems meeting the MCNSPS requirements, are presented.

4.5.1 Rankine Alkali Metal Vapor Turbine-Alternator System

Introduction. It was recognized almost at the inception of the space nuclear electric power program that the heat rejection from a high temperature Rankine cycle system promised to give the lowest radiator size and weight of any of the systems under consideration. The cycle diagram and the schematic arrangement of the Rankine cycle are shown in Fig. 4.5.1.1. In this cycle, the working fluid is compressed as a liquid and heat is added to raise the liquid to its saturation temperature, as indicated in the cycle diagram by processes 1-2. Heat is added at constant pressure and temperature, causing a phase change from liquid to vapor, states 2-3. The vapor is expanded through a turbine and work is extracted, states 3-4. To complete the cycle, the waste heat is rejected at constant pressure and temperature, causing the vapor to condense back to the liquid state, during the process 4-1.

As a result of the boiling and condensing phase changes at constant temperature, the ideal saturated Rankine-cycle efficiency approaches that of the Carnot cycle. In addition, very little work is required to compress the working fluid in the liquid state. The principal advantages of the Rankine

RANKINE CYCLE THERMODYNAMIC DIAGRAM

AND SYSTEM CONFIGURATION

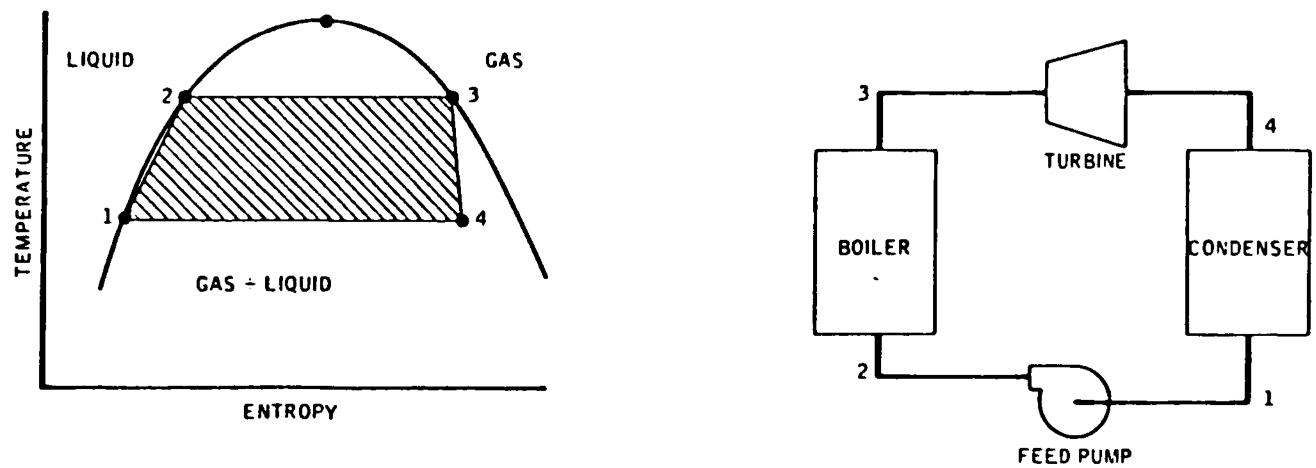

FIG. 4.5.1.1.

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Cycle are the high cycle efficiency and the isothermal heat rejection, both of which are important to minimizing the radiator area for a given heat source temperature. The principal disadvantages of liquid metal Rankine cycles are the relative complexity, heavy boilers and separators required to provide dry saturated or superheated vapor in mucri-gravity, vapor-liquid management in the separator, turbine, and condenser, liquid metal bearing requirements, numerous loops, pumps, heat exchangers, valves and ceramic alternator stator seals, inherent corrosion and erosion characteristics of the applicable working fluids.

Working Fluids

The reports and papers covering the extensive work carried out on high temperature Rankine cycles in the past 30 years have been reviewed. From this review it is evident that high temperature steam cycles entail such high pressures that the equipment would be much too heavy for space power applications. The upper temperature useable with higher boiling temperature organic fluids is limited to about 750 K by thermal decomposition; this temperature is too low to give an attractive system. Mercury was used in utility power plant systems for 40 years and was the choice for low power (3-30 kWe) SNAP space power systems. However, experience indicates that solution-type corrosion and mass transfer imposes an upper temperature limit of about 800 to 900 K for Fe-Cr-Ni alloys. Refractory metal alloys could be used for higher temperatures, but the vapor pressure of mercury increases to the point that equipment weights become excessive at boiling temperatures above about 1000 K. Alkali metals have been shown to be compatible with both the Fe-Cr-Ni and refractory metal alloys [23] to much higher temperatures and have lower vapor pressures than mercury, hence they are logical choices. Fig. 4.5.1.2 indicates the range of applicability of sodium, potassium, cesium and their potential radiator power densities. No other working fluids have since been found that would have both a suitab' vapor pressure and be compatible with a suitable structural alloy.

By 1959 completely independent studies at ORNL and NASA Lewis Lab indicated that potassium and cesium were the best candidates for space power systems ranging from 300 kWe to 1 MWe. The vapor pressure of rubidium is

TEMPERATURE RANGE OF APPLICATION OF SODIUM AND POTASSIUM WORKING FLUIDS IN SPACE RANKINE CYCLE

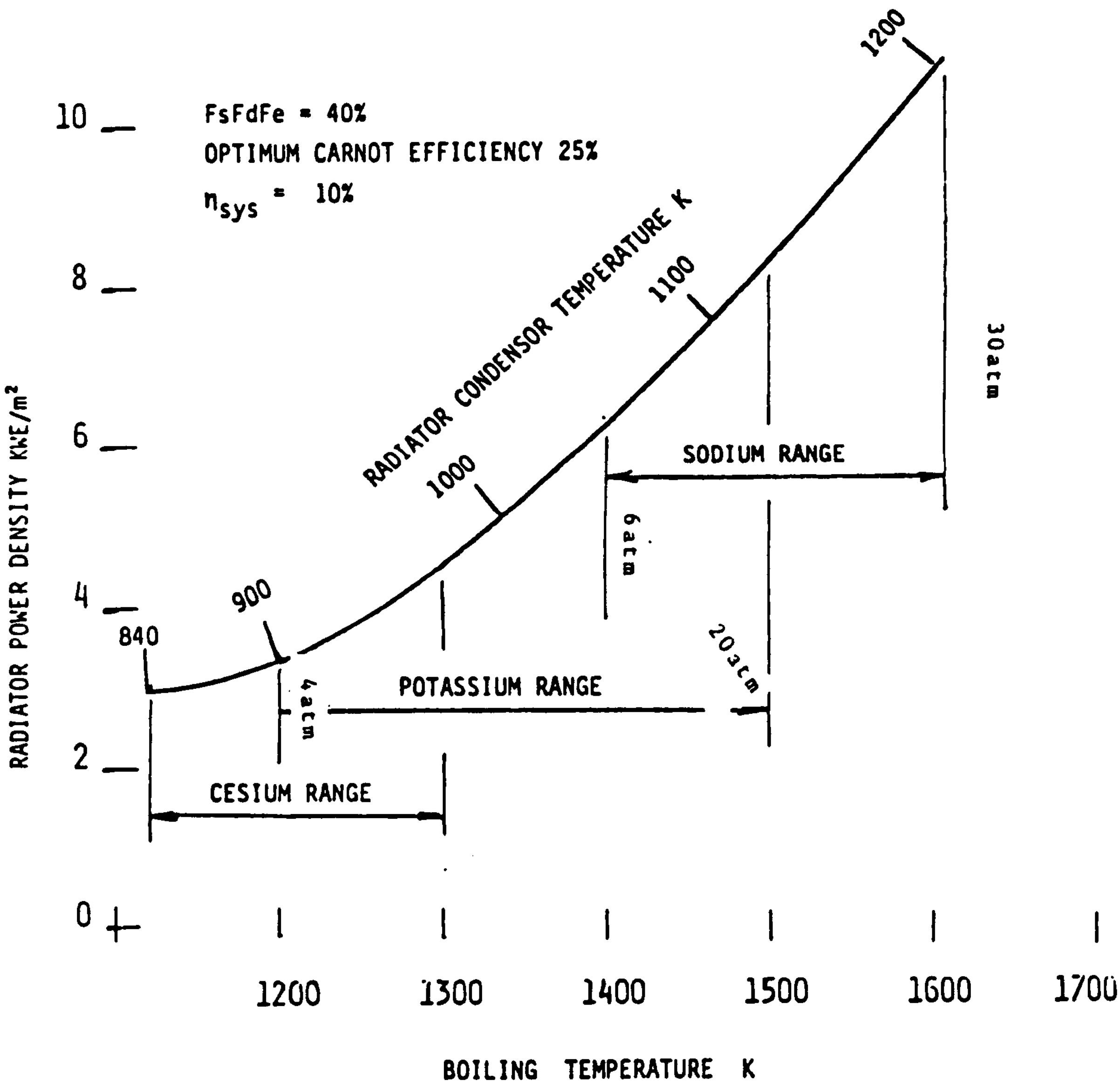

FIG. 4.5.1.2.

intermediate between the values for potassium and cesium, but it is difficult to obtain. Lithium was found to have such low vapor pressures at containable temperatures that the size of components and hence their weight became excessive. The diameter of the last stage of a 100 kW(e) lithium turbine, for example, would have to be almost 2 meters. A subsequent study at G.E. ANP in Evendale, Ohio reached similar conclusions. In comparing cesium and potassium it was found that the higher atomic weight of cesium gives a smaller and lighter turbine, but the turbine weight is a small fraction of the total power plant weight. On the other hand, potassium is much more readily available than cesium, less expensive, and far less subject to neutron activation. As a consequence, all three organizations chose potassium as the working fluid for the 300 kWe to 1 MWE power range.

In this program, for 10 MWe, sodium will be shown to provide lowest system mass, if suitable higher temperature (1600 K - 1800 K) reactor, fuel, pump and (low creep) turbine and boiler materials can be developed.

Over 100 reports are available describing liquid metal Rankine cycle concepts, materials work and components development that took place during the period 1955 to 1970. This cycle received over 200 million dollars (virtually all) of the space power development funding from 1955 to 1967. Most of the work was with the lower temperature SNAP 2 and SNAP 8 mercury systems. In the mid 1960's a healthy potassium Rankine component development program was under way. The potassium work was pointed toward 300 kWe to potentially 1 MWe and now serves as the technical basis for extrapolating to multimegawatt power outputs. To make the best use of this past work in this study, it was decided that past work [35-39] would be reviewed, correlated, and extrapolated to temperatures and power levels needed in this study. These correlations and extrapolations follow in this report.

In addition to components, design consideration must be given to coolant and working fluid loop arrangement. The major question in this regard is direct cycle versus indirect cycle.

Direct In-Core Boiling Cycle. The Rankine vapor cycle is similar to the Brayton gas cycle in that it can use either a direct cycle with the working fluid heated in the reactor core or an indirect cycle with a single phase liquid coolant loop and an external liquid to boiler or vapor heat exchanger. The direct and indirect cycles are illustrated schematically in Fig. 4.5.1.3.

Successful boiling reactors using water as a working fluid have been built for terrestrial low temperature, earth gravity power systems. However, a boiling reactor with liquid metal, as might be used in a 0-gravity space power system, requires a major technology development effort.

The direct in-core single pass boiling reactor is not reasonably capable of producing vapor superheat in a compact reactor because of the large void and heat transfer area requirement to achieve dry vapor for superheat. In order to prevent in-core boiling burnout in a fuel rod bundle core, a considerable liquid recirculation rate (approximately 10:1 mass ratio of coolant recirculation to vapor fraction) is frequently specified. In order to achieve low recirculation ratios, incore liquid-vapor separation, and boiling stability, the reactor should be "inside out", i.e., the fuel should be outside of the coolant channels, as in the heat pipe cooled "SPAR" reactor. In this manner, inlet coolant tube orificing can prevent "boiling disease" even in zero gravity, as was achieved in the Hanford and Savannha River tube type production reactors. In addition, a "calendria" type core will permit installation of spiral turbulators into each boiling tube as described in section 4.4 of Volume II. At the exit of the reactor, the vapor must be cyclone separated from the recirculating liquid and cyclone-chevron dried before entering a saturated vapor turbine. After partial expansion, the moisture must be extracted from the turbine to remove accumulated liquid. Another alternative is to admit prime hot vapor into later stages of the turbine in order to reheat the partially expanded vapor. Moisture extraction and reheat are required to prevent excessive long term turbine erosion in saturated metal vapor turbines. Turbine vapor extraction is also required for feed heating, better efficiency, and reactor temperature stability.

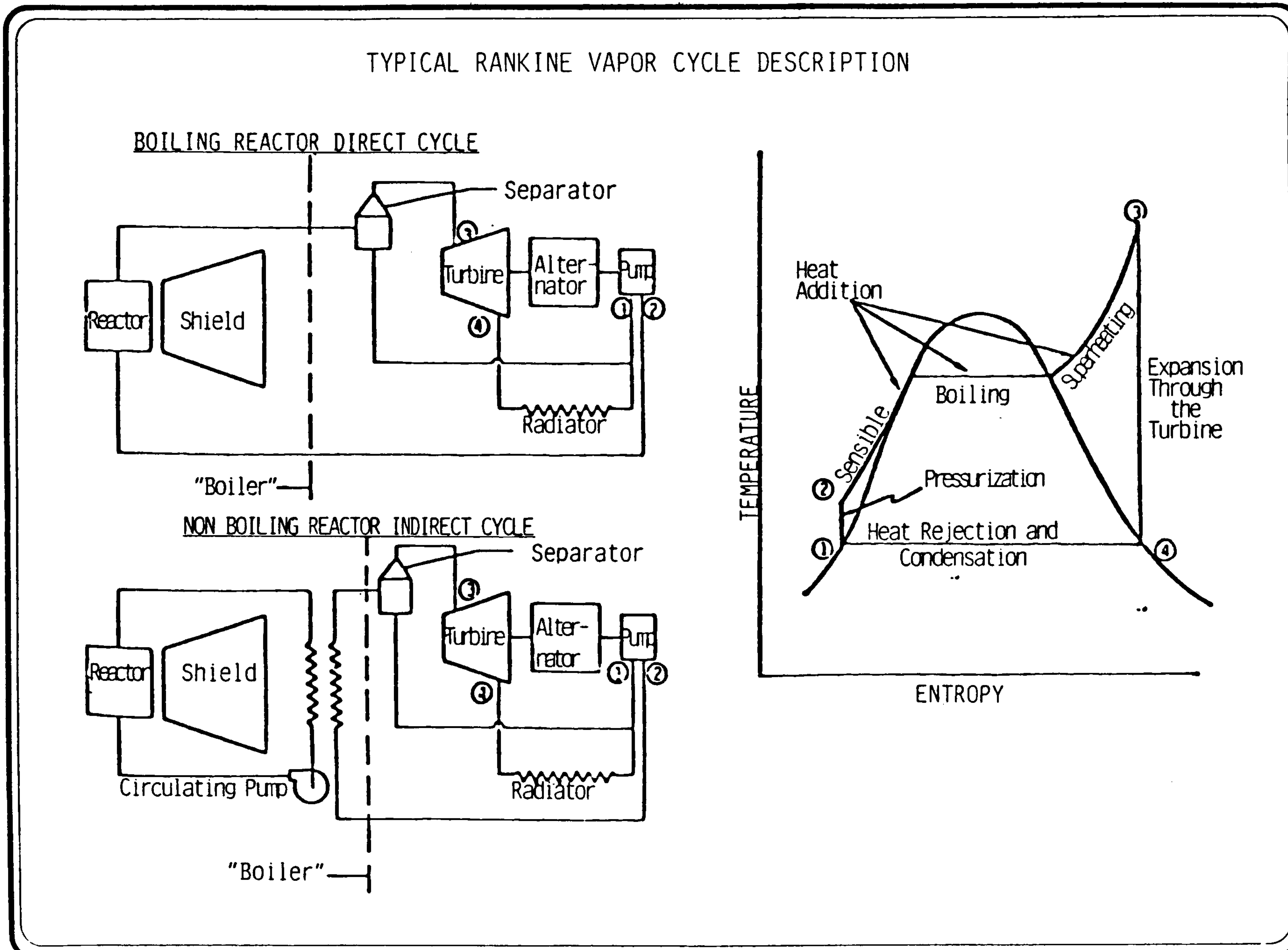
TYPICAL RANKINE VAPOR CYCLE DESCRIPTION
BOILING REACTOR DIRECT CYCLE
Separator
Reactor
Shield
Turbine
Alter-nator
Pump
Radiator
"Boiler"
NON BOILING REACTOR INDIRECT CYCLE
Separator
Reactor
Shield
Turbine
Alter-nator
Pump
Radiator
Circulating Pump
"Boiler"
TEMPERATURE
ENTROPY
Heat Addition
Boiling
Superheating
Sensible
Expansion Through the Turbine
Pressurization
Heat Rejection and Condensation

Fig. 4.5.1.3

The direct in-core boiling system is sensitive to fluctuating reactivity due to vapor (void) formation. Special attention is required to tailor the reactor design to have a nearly zero, or slightly negative, void coefficient of reactivity. The relatively high pressure (200-300 psi) on the reactor vessel will most likely require that the pressure vessel be located outside of the reflector control drums. This arrangement will greatly increase the complexity of reactor control.

A twin turbine, direct boiling, potassium reactor system schematic layout is shown in Fig. 4.5.1.4. The hot saturated vapor is cyclone separated and demisted in the reactor before being brought around the outside of the shield to a header and distributed to multiple turbo-alternator units. Recycled potassium liquid might be recirculated to the reactor inlet either by a very high temperature EM pump, a jet pump powered by high pressure condensate return, or by a vapor driven turbo-pump. Vapor extracted from the turbines is also reheated in a heat exchanger. Condensate is removed from the restartable-taper-cone condensers by EM pumps and delivered to canned-rotor or turbine driven boiler feed pumps. A heat rejection loop transfers waste heat from the taper cone condensers to the radiator. The main radiator is an extendable telescoping cylinder heat pipe type, which is shown packaged in the stowed configuration. In addition there is a bearing and alternator cooling heat rejection system which operates at a lower temperature.

The inherent primary advantage of the direct boiling system is lower reactor fuel temperatures for a given overall thermodynamic efficiency, if recirculation pumping power can be kept low. Because the use of lithium is precluded in the direct boiling system, a second advantage will be the lower corrosion sensitivity of refractory metals to potassium (as opposed to lithium). Finally, the direct cycle system avoids the weight and complexity of a separate loop, consisting of an intermediate heat exchanger and a high temperature lithium pump, components which are required for the indirect cycle approach.

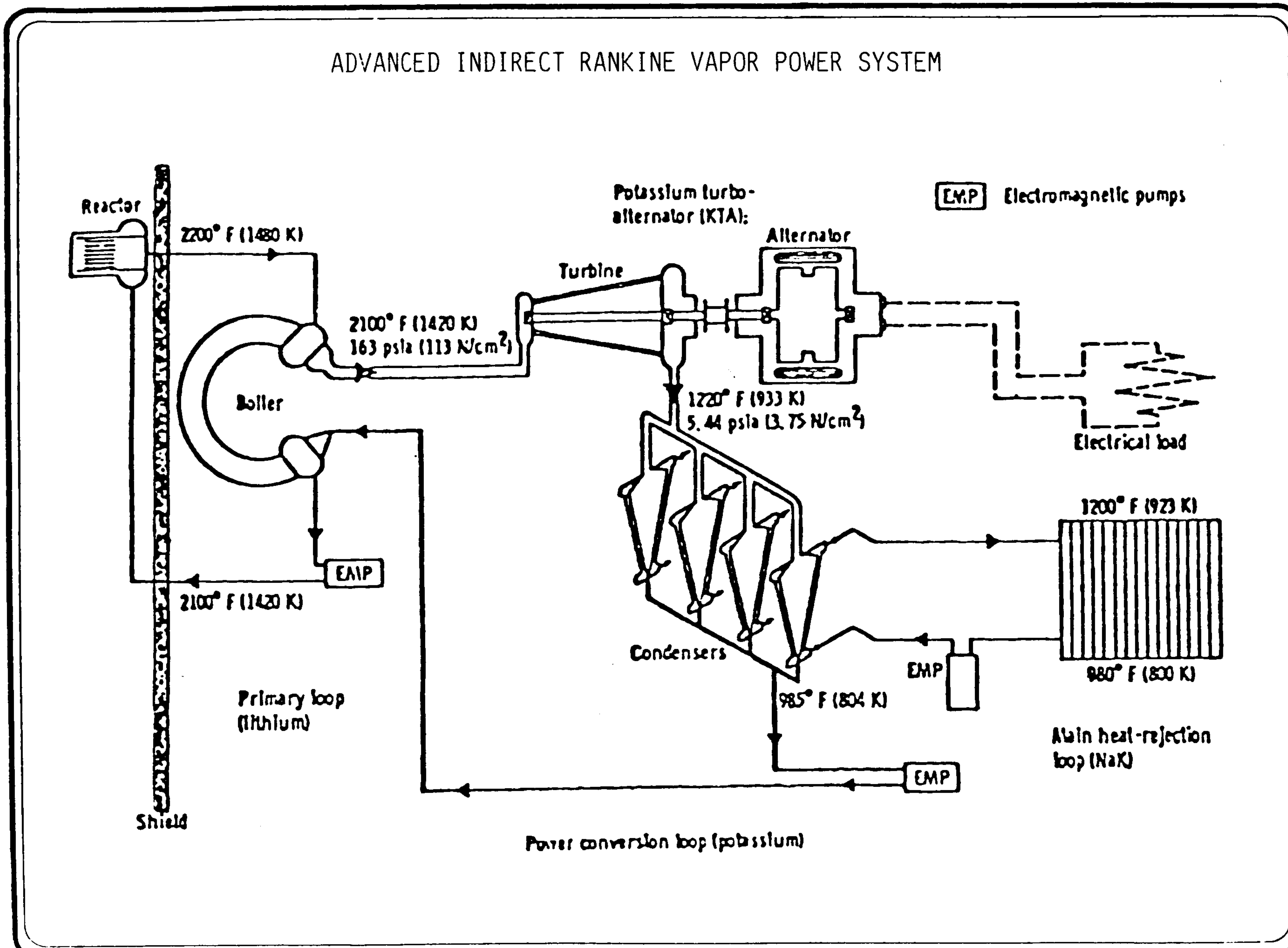
ADVANCED INDIRECT RANKINE VAPOR POWER SYSTEM
Reactor
2200° F (1480 K)
Shield
Boiler
2100° F (1420 K)
163 psia (113 N/cm²)
2100° F (1420 K)
EMP
Primary loop
(lithium)
Potassium turbo-
alternator (KTA):
Turbine
Alternator
1220° F (933 K)
5.44 psia (3.75 N/cm²)
EMP Electromagnetic pumps
Electrical load
Condensers
985° F (804 K)
EMP
1200° F (923 K)
980° F (800 K)
EMP
Main heat-rejection
loop (NaK)
EMP
Power conversion loop (potassium)

Fig. 4.5.1.4

<u>Indirect Systems</u>. The indirect Rankine cycle system, utilizing electromagnetically pumped liquid lithium coolant in the reactor or primary loop and an external once-through potassium boiler-superheater in the power conversion system loop, can be a much more straight forward engineering task. With only a pumped non-boiling primary loop, a compact, reflector controlled, fuel pin reactor is reasonable. The low lithium pressure will permit thin reactor vessel (double) walls with minimum loss of reflector control. Reactivity fluctuation is minimal, and boiling burnout and stability in the reactor can be avoided. The external once-through boiler-superheater, with liquid lithium as the heat source, is not subject to unstable flow distribution boiling burnout at the expense of increased boiler-superheater mass, slightly superheated vapor might be provided to minimize the need for turbine moisture extraction and reheat. Development will be more straight-forward but must deal with higher reactor temperatures and lithium metal containment for a given system efficiency and radiator size. An indirect cycle potassium Rankine power system flow schematic is shown in Fig. 4.5.1.5

For this type of system, multiple torroidal boiler superheaters produce the dry vapor with heat input from pumped lithium loops which cool the reactor. Redundancy in the ducting and boilers is maintained with multiple turbine generator units. Other components in the system would be similar to the design shown in Fig. 4.5.1.4, although the space taken up by the boiler units means that the entire system would be longer.

Another indirect cycle concept uses a modified "mini-heat-pipe" reactor (described previously in Section 4.4.4 of Volume II), rather than a straight-forward lithium cooled fuel pin reactor. This concept permits elimination of the double-walled primary reactor core vessel. Multiple lithium loops cool the <u>short</u> heat pipes in a cross-flow series-parallel design which eliminates the possibility of single point failures. The short heat pipes at high power density are more reasonable. The reactor can be made much larger for higher fuel inventory and power output. Individual heat pipe, radiation cooled, control rods can be considered without primary coolant vessel penetration.

CONCEPTUAL LAYOUT OF DIRECT BOILING REACTOR-
RANKINE VAPOR POWER SYSTEM

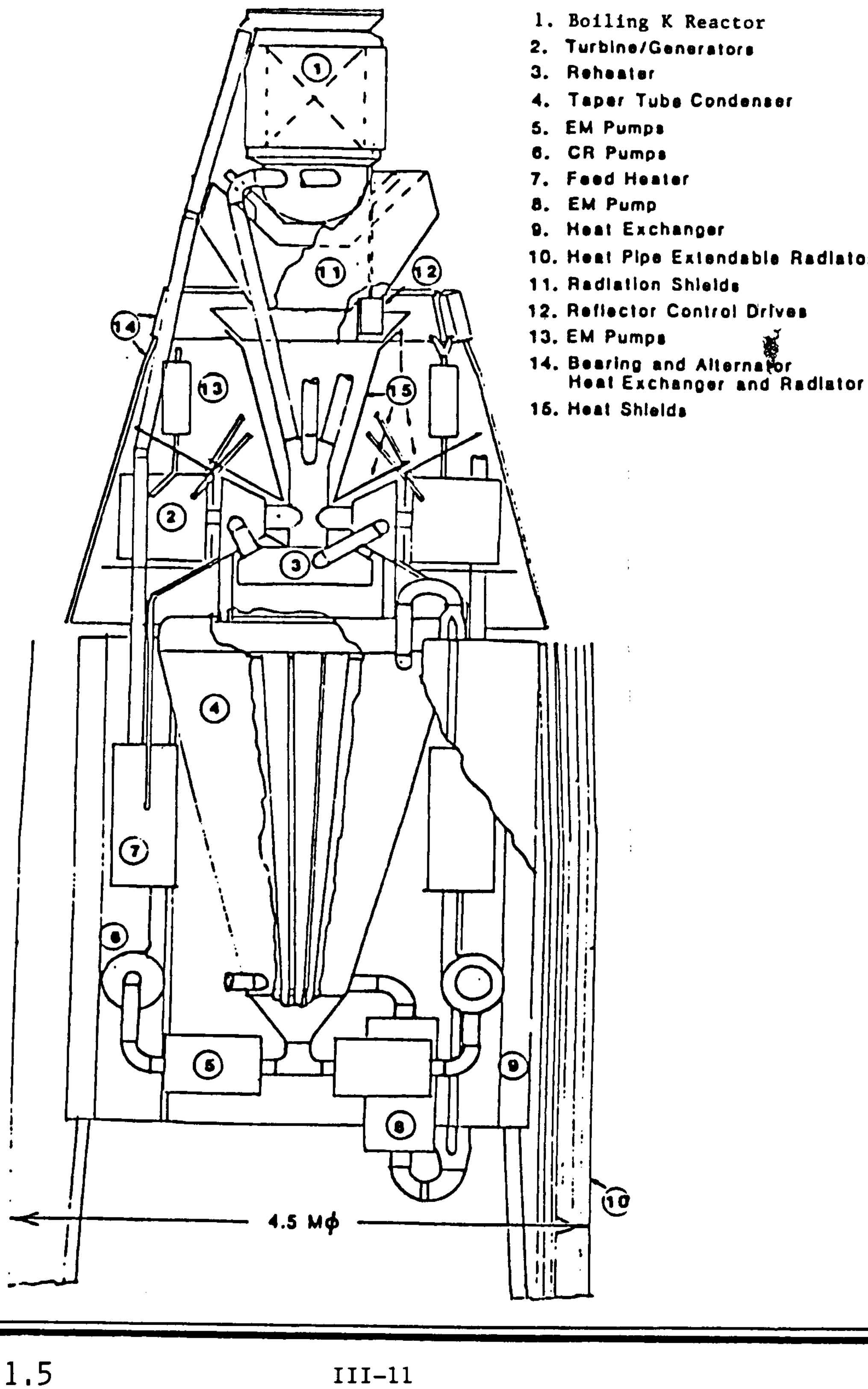

Fig. 4.5.1.5

Temperatures and Pressures. The peak cycle temperature for a liquid metal Rankine system is usually limited by the fuel cladding temperature, but this limit may also be a consequence of the allowable peak centerline temperature in the fuel itself. Although temperatures in excess of 1500 K might be achieved with liquid metal Rankine systems, it is expected that pressure containment will limit saturation temperatures to about 1400 K plus some small superheat for a potassium working fluid. At these temperatures, the vapor pressure of potassium is approximately 10-15 atm, which is reasonable for creep stress limitations in the high temperature components. The use of sodium as the working fluid would permit operation at 100 K higher temperature at the same pressure limits, if a low creep rate reactor core vessel (or boiler tube) material can be developed. The temperature of heat rejection can be varied for design optimization with the radiator size. However, as the heat rejection temperature is lowered, the condensing pressure is also lowered and the sizes of the last turbine stages, the condensers and the radiator are rapidly increased. Moisture erosion also becomes more significant.

Reactor and Shielding Considerations. For direct boiling Rankine systems, there are design difficulties in bringing the working fluid (large vapor pipes) around or through the shield, similar to the direct Brayton cycle problem. However, the mass of fluid which must be transported (and hence the sizes of the ducts) is smaller than the Brayton gas system and the radiation streaming may be less of a problem.

With an indirect Rankine cycle, a lithium metal primary loop would be used to bring the heat around the shield. This permits heat transport with a relatively small cross-section of fluid duct, so that the majority of the shield surface area can be left exposed for cooling. In addition, the liquid lithium in the ducts is a fair shielding (scattering) material itself, so that radiation streaming is reduced. With the pumped loop indirect cycle, there is no boiling in the reactor and the reactivity changes related to this are avoided.

Multiloop Systems. Rather than combining the condensing and waste heat radiating processes into a single vulnerable component, a shell-and-tube condenser can be used to reject heat to a third loop. This loop conveys the heat out to a radiator, resulting in a three-loop system similar to those used in utility steam power plants and in the SNAP-8 power unit. The programs at NASA Lewis Lab, G.E.-Evendale, and Pratt & Whitney also followed this course. A flow sheet for the three-loop system is shown in Fig. 4.5.1.6.

Electric Output. Electric output of the Rankine cycle system is similar to all other rotor-dynamic systems, in terms of producing a desired voltage. A particular characteristic of the Rankine system is the need for an auxiliary radiator to cool the canned stator generator and turbine bearings. This radiator can operate at relatively higher temperature than solid state components ($\approx$600 K).

Reliability and Modularity. The reliability of alkali metal Rankine cycle systems is primarily a question of high temperature fuel and component durability, corrosion resistance, and development of successful long life liquid metal bearings. Corrosion of high temperature components by the alkali metals in non-isothermal loops is greatly accelerated by the presence of small quantities of oxygen, nitrogen or carbon in the system [23,38]. It will be important to maintain the purity of the working fluid for a long operating life and to select materials that have very low temperature dependence on solubility in the liquid metals. The process of boiling the liquid metal continually re-distills impurities out in the portion where evaporation takes place. This can lead to transport of metal similar to the phenomena in high power heat pipes. However, the deposition of metal in the boiler portions of the system are less likely to clog small orifices, as occurs in the small passages of a heat pipe wick.

In order to achieve a reliable system, multiple boiler, turbo-alternator and condenser loops are anticipated. This is not difficult in the indirect case where the primary lithium flow can be divided and the alkali metal vapor turbines are small in both size and mass. Turbo-alternator units on the order of 1.5 to 3 MWe output were examined for extrapolation to 5 MWe and 10

THREE-LOOP SYSTEM WITH A LIQUID-COOLED REACTOR HEATING THE BOILER OF A RANKINE CYCLE LOOP COUPLED TO A SET OF INDIRECT RADIATOR LOOPS

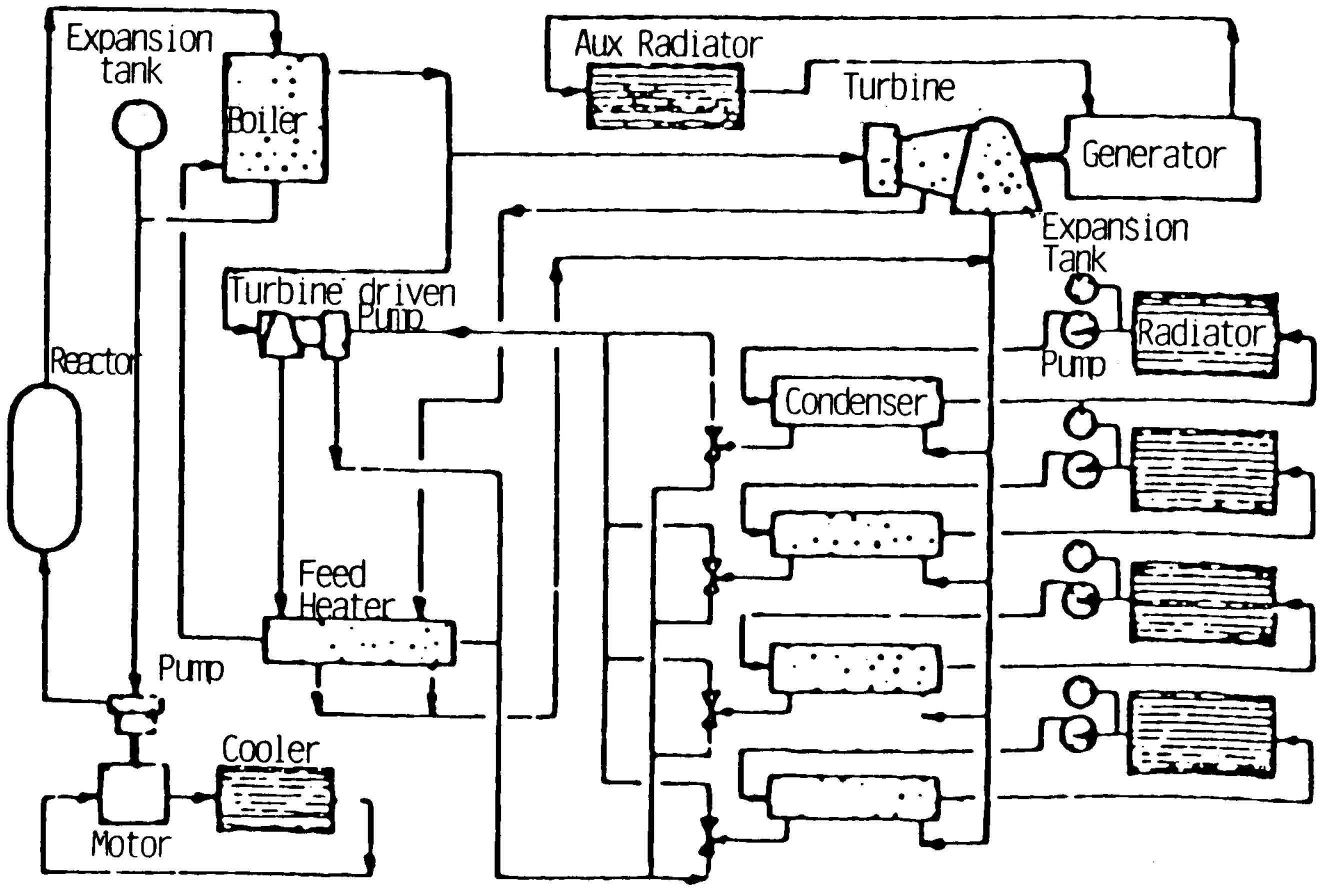

Fig. 4.5.1.6

MWe net output, respectively. All loops, pumps and turbo-alternators should be in near-perfect counter rotating pairs to minimize the vehicle disturbing torques.

The vital requirement for exceptionally high reliability, unattended operation in orbit raised a basic feasibility question at ORNL some years ago. Would it be possible to achieve the specified 90% probability for a year's operation without a forced outage? (An even more stringent reliability requirement of 95% for 5 years (44,000 hrs) is currently envisaged.) Extensive operating experience with complex systems indicate that such a high degree of reliability will be difficult to achieve in a dynamic system. Automobiles, for example, ordinarily require maintenance before they have run even 1000 hours, or about 50,000 miles at 50 mph. A study of the records of conventional steam plants and the Army Package Power Reactor systems and others, including hydroelectric units, was sobering: the best steam plants gave mean times to a forced outage of only around 500 hours, while the corresponding time for hydro units was about 1000 hours. This led to an extensive generic study [38] of the reliability of the basic types of equipment involved, i.e., valves, turbines, generators, motors, pumps, heat exchangers, instrumentation sensors, electronic controls, and electrical switchgear. The failure rates indicated that, even with the best quality control on each and every component, it was highly unlikely that the required reliability could be achieved unless the system is drastically simplified and the number of series connected components reduced to an absolute minimum. The only example found of a roughly comparable system demonstrating the desired degree of reliability was a household refrigerator. These appliances commonly give a mean time to failure of about 12 years. To achieve this, that system has been simplified to a minimum number of components: motor, pump, evaporator, condenser, expansion valve, thermostat, and a control switch with the requisite connecting piping and wiring. The refrigeration system is hermetically sealed, and the materials employed operate in a low temperature environment and are so compatible and so free of degradation, wear, or corrosion that no provision for additions to, or clean-up of, the working fluid is needed.

To get a degree of simplification in a space power plant, ORNL reduced the system to a single (direct) loop by using a boiling potassium reactor and a direct condensing radiator as shown in Fig. 4.5.1.7. [38] This eliminates not only some of the components of the typical three-loop system of Fig. 4.5.1.6, but also some of the instrumentation and control equipment as well. Analyses of outages in power plants have shown that the instrumentation and control equipment is the major cause of forced outages. The apparent simplified system of Fig. 4.5.1.7 could be achieved only if the formidable problems of a boiling potassium reactor could be solved.

As the programs on the 3-loop potassium Rankine cycle system (commonly referred to as the SNAP-50 system) proceeded at NASA, G.E., P & W, and AiResearch, the advantages of the direct-condensing radiator led to consideration of the 2-loop system of Fig. 4.5.1.8. This eliminated the temperature losses associated with the shell-and-tube condenser and the temperature drop in the NaK circuit for the radiator, thus increasing the mean radiator temperature about 100°C for a given turbine outlet temperature. This reduced the radiator size and weight and eliminated the NaK pump and its instrumentation and control equipment. These savings were offset in part by elimination of the redundancy provided by employing four parallel NaK radiator circuits. Additional armor was required on the radiator tubes to obtain the same low probability of an outage caused by a meteoroid puncture of the radiator.

Such a direct radiator at 1 to 10 MWe is not possible or reasonable. A 10 MWe Rankine space power system will utilize at least 1000m^2 of radiator area. The required micrometeorite protection on such an area is prohibitive. Reliable condensation and return of the condensate from a large distance in zero gravity would also be very difficult. In this study SPI will utilize zero gravity conical condensers incorporated into large long radiator heat pipes. (See conceptual design Section 6.1., Volume IV). In Figs. 4.5.1.9 thru 4.5.1.13 the results of previous potassium Rankine studies and development programs are correlated and extrapolated. Fig. 4.5.1.9 [40] provides the correlation and extrapolation of reactor size and mass for the various lithium cooled and boiling potassium reactors. On Fig. 4.5.1.10 the size and weight correlation for lithium to potassium boiler

SINGLE LOOP SYSTEM WITH A BOILING REACTOR AND A DIRECT CONDENSING RADIATOR

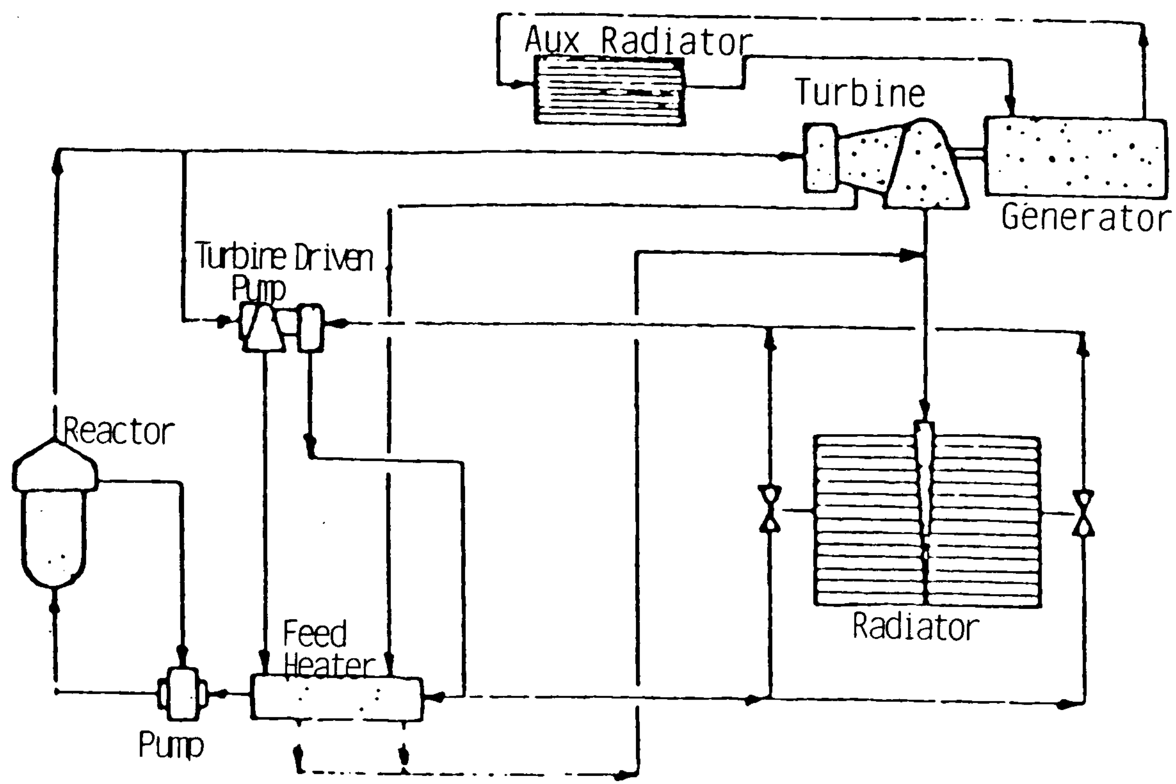

Fig. 4.5.1.7

TWO-LOOP SYSTEM WITH A LIQUID-COOLED REACTOR LOOP HEATING
THE BOILER OF A RANKINE CYCLE LOOP CONTAINING A DIRECT CONDENSING RADIATOR

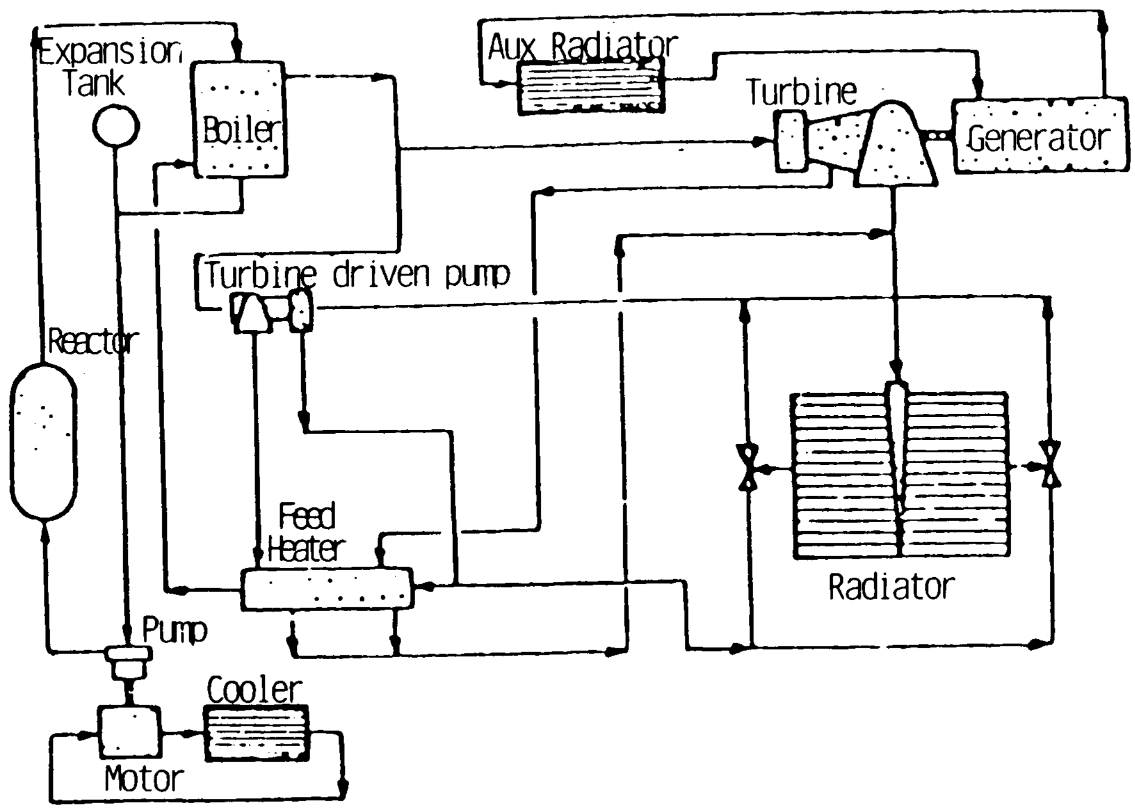

Fig. 4.5.1.8

REACTOR SIZE AND WEIGHT AS A FUNCTION OF THERMAL POWER OUTPUT

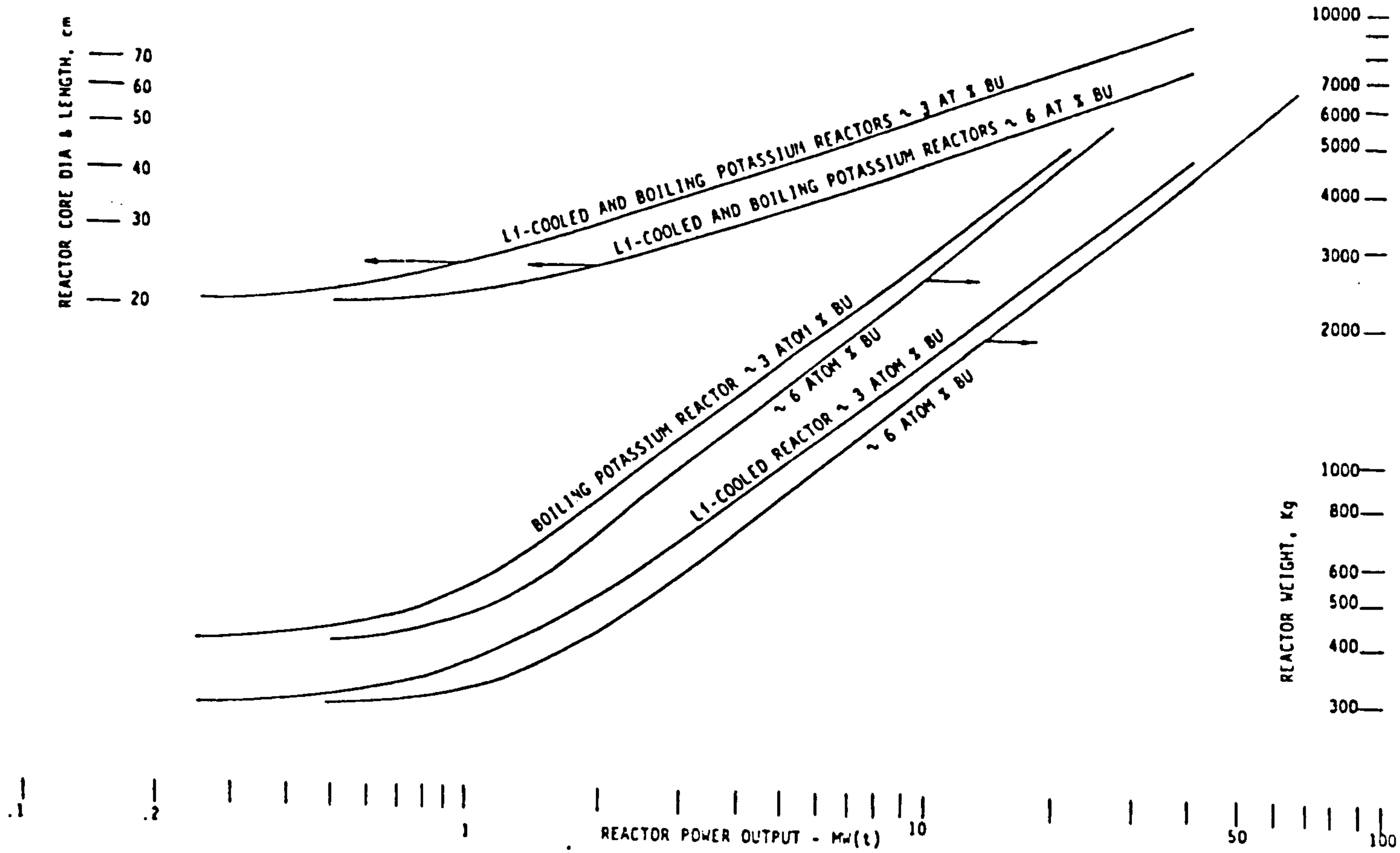

FIG. 4.5.1.9

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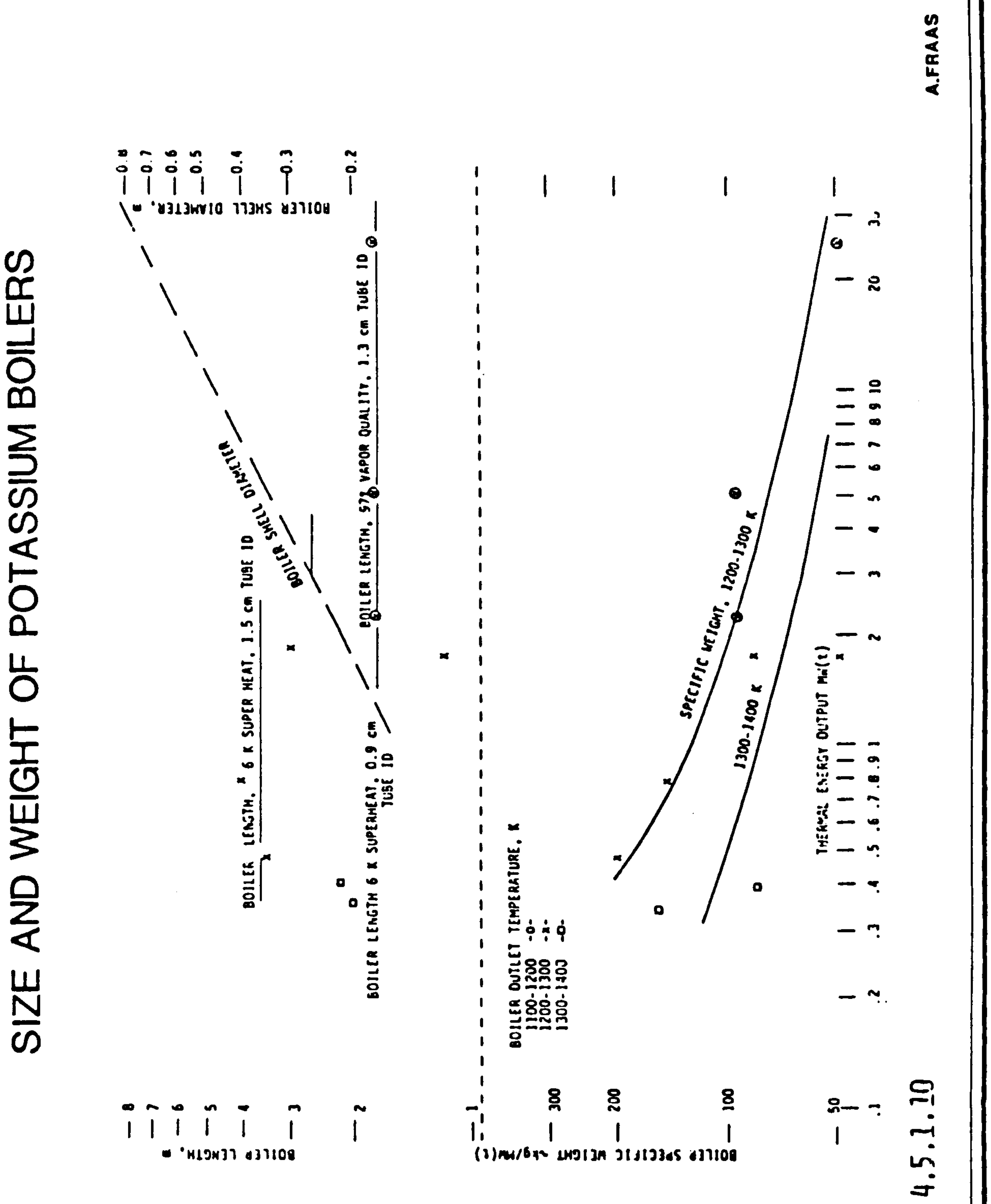

FIG. 4.5.1.10

Superheat and reheat boilers are assumed to be 3 times the size and mass of boilers.

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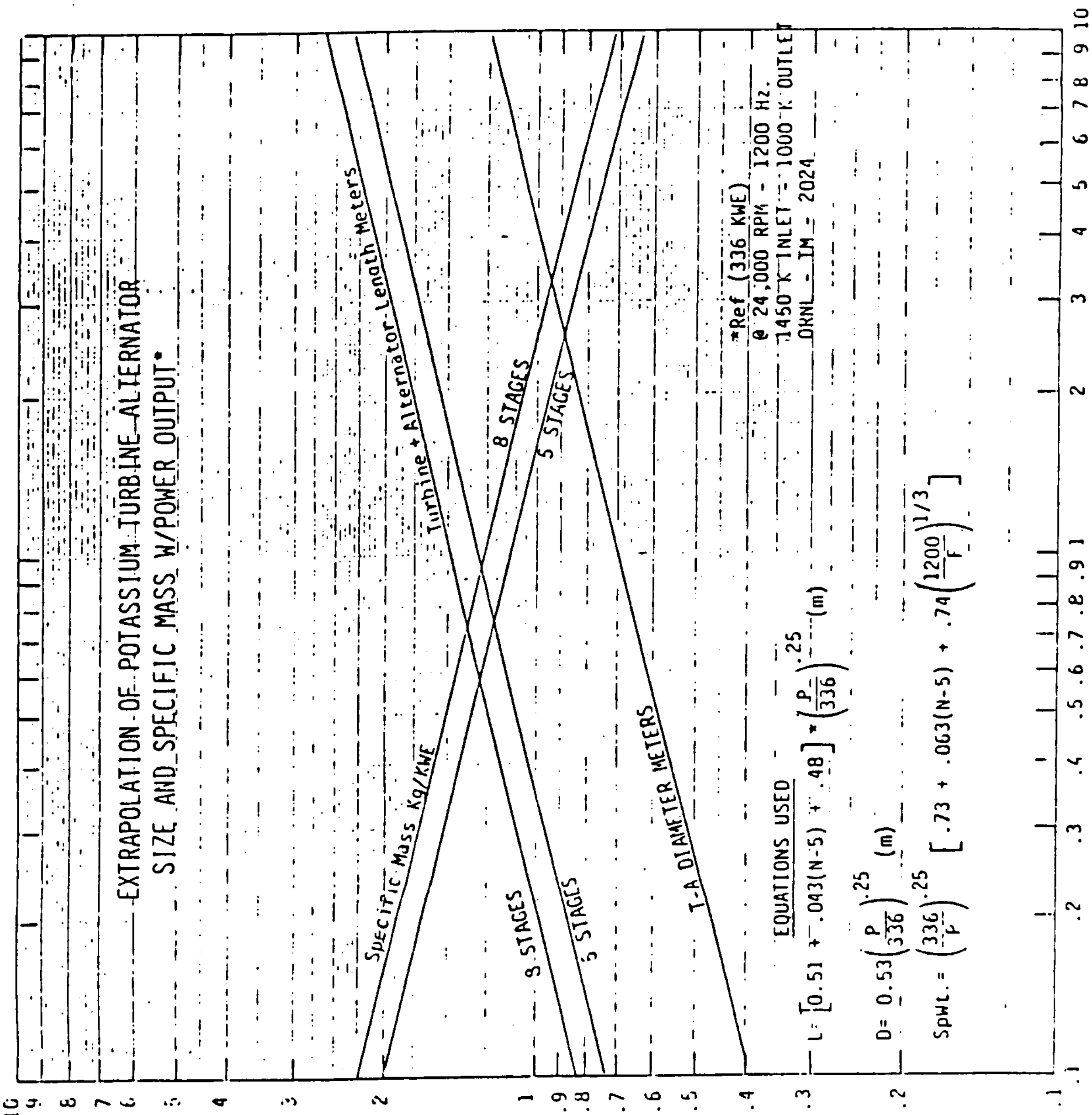

Fig. 4.5.1.1.11

EFFECTS OF REACTOR THERMAL OUTPUT ON THE WEIGHT OF FEED HEATERS FOR POTASSIUM RANKINE CYCLE SYSTEMS

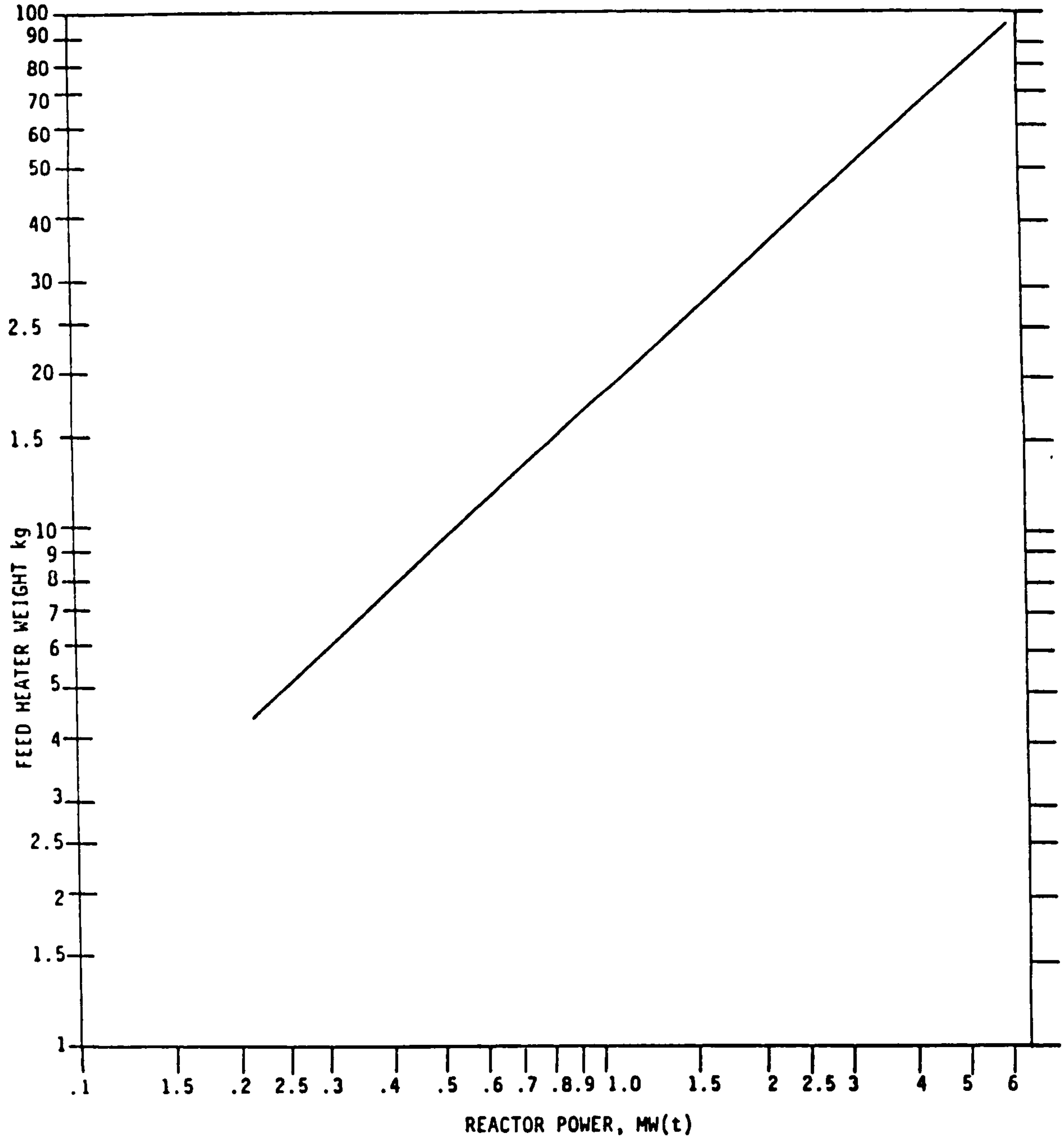

FIG. 4.5.1.12

WEIGHT AND EFFICIENCY OF BOILER FEED PUMPS

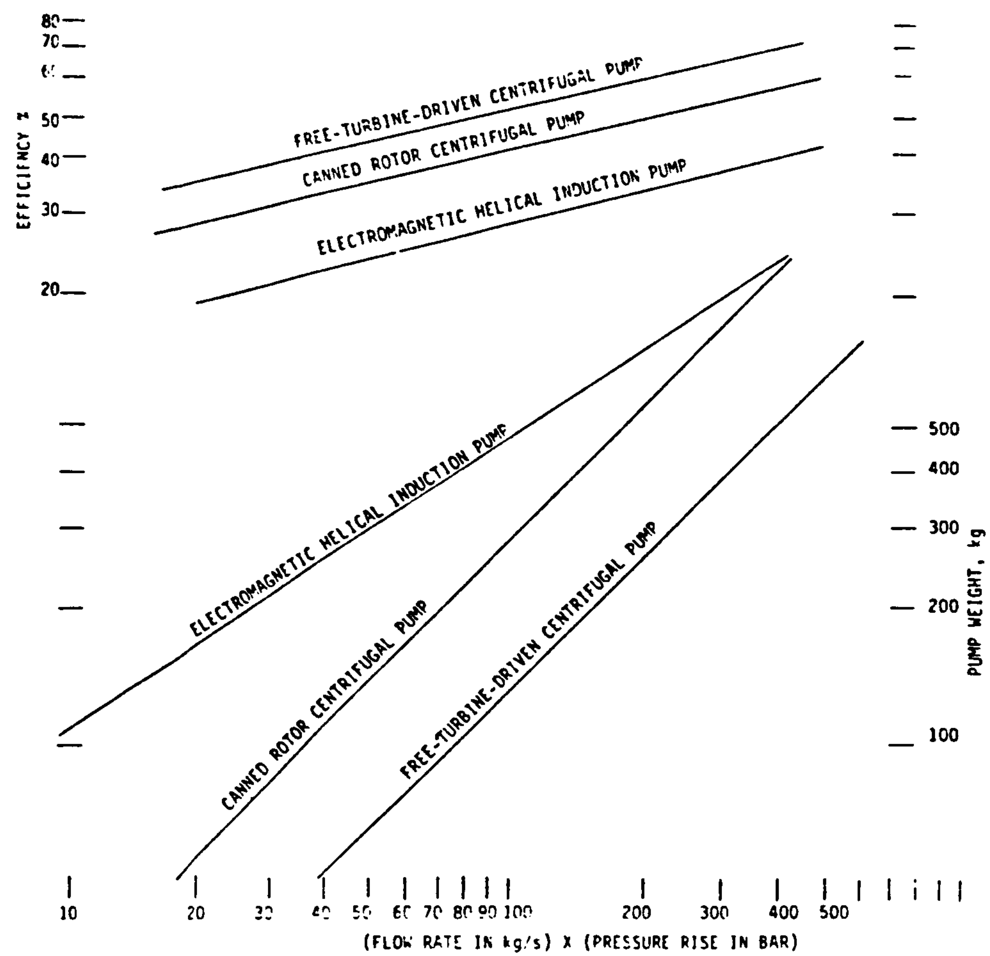

FIG. 4.5.1.13

A.FRAAS

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part design studies is presented. As indicated previously, a 10 MWe system will require a 40 to 50 MWt reactor, and large high-speed turbo-alternators will be coupled into counter rotating pairs.

If the power conversion system is divided into 4 parallel redundant systems, arranged into counter flowing and rotating pairs, then each unit will have to accommodate some 10 to 12 thermal megawatts.

From Fig. 4.5.1.10, four 10 megawatt boilers would have a mass of 2640 kg (4x10 MWt x 66 kg/MWt). The 2640 kg added to that of the 40 MWt Li cooled uranium nitride reactors (LUNR) of Fig. 4.5.1.9 places the LUNR mass nearly equal to the mass of the boiling potassium metal reactor (BMR). A final decision of whether to select a LUNR or BMR to provide heat to a Rankine vapor turbine cycle need not be made at this time. For purposes of comparing the Rankine cycle to the Stirling, Brayton or thermionic conversion cycles, the LUNR and BMR may be treated equally.

Fig. 4.5.1.11 correlates the Rankine turbine alternator mass and size from past studies [39].

Fig. 4.5.1.12 presents the feed heater correlation for mass and size [40].

Fig. 4.5.1.13 presents mass and efficiency correlations for the various pump types considered in past and current studies [36,40].

Rankine Cycle Performance Estimates. Analyses of Rankine cycle systems was carried out by doing a balance of the enthalpy at various points in the cycle as specified by Mollier charts for the particular working fluid. The data on these charts or tables are stored in a look-up routine in a system analysis computer program. Input parameters specifying the turbine and generator efficiencies, boiler feed and circulation pump work, pressure and thermal losses, etc. are entered to calculate the cycle performance. For liquid metals, the fluid state at the turbine exit typically represents a wet vapor condition. Both the enthalpy and the entropy values are interpolated between vapor and liquid states, based on the quality at that point. Techniques for handling a Rankine cycle in this way are similar for many types of working fluids, if the fluid properties are known as a function of temperature and pressure over the operating range.

The cycle high temperature limits are established by materials' creep strength, corrosion, and radiation induced swelling limits, and system life expectations.

The cycle lower temperature limits are established by radiator area and mass limitations imposed by the launch system capability and cost, and the deployed system mass, compactness and mobility requirements.

The characteristics of the telescoping heat pipe radiator described in Section 4.3 of Volume II were incorporated into the computer program. The cycle analysis establishes the cycle efficiency which is utilized to establish the system thermal power. The thermal power and cycle peak and minimum temperatures establish the fluid flow rates, pressure drops, pumping powers and the reactor, superheater, boiler, reheater, and feed heater sizes and masses. The reactor size and mass are determined by the procedure described in section 4.4 of Volume II.

The electric power requirement is used to establish the turbine-alternator mass and dimensions. For this preliminary analysis, the past program correlations and extrapolations on Figs. 4.5.1.9 thru 4.5.1.13 are utilized to establish component masses and sizes. Superheater and reheater sizes and specific masses are estimated and are optimistically assumed to be 3 times the size and mass of a boiler passing the equivalent heat.

Preliminary Calculated Results. The Rankine cycle efficiency decreases when the heat rejection temperature is raised, for a fixed value of the peak cycle temperature. However, the fraction of the ideal Carnot efficiency does not degrade. It even increases slightly. These effects are shown on Fig. 4.5.1.14, which illustrates the variation in the cycle efficiency as a function of the condenser temperature found in the previous Rankine cycle development programs. When all the aerodynamic, moisture, seal losses and generator inefficiencies have been included, the Rankine cycle systems produce nearly $\approx$60% of the ideal Carnot efficiency, as shown in these

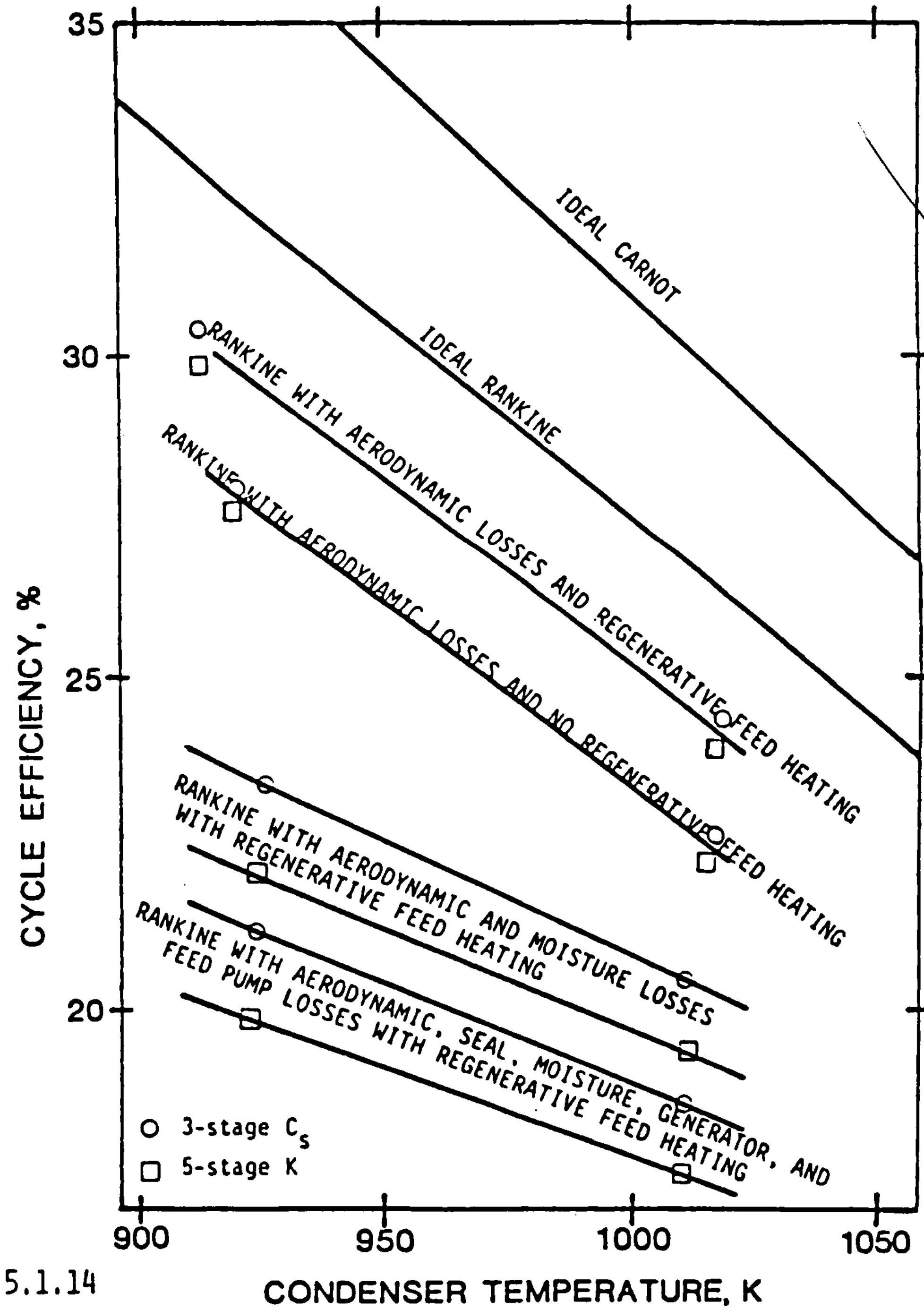


FIG. 4.5.1.14

Effects of Aerodynamic, Moisture, and Seal Losses on Cesium and Potassium Rankine Cycles with Regenerative Feed Heating, a Turbine Inlet Termperature of 1450 K, a Turbine Outlet Temperature of 994 K, and 14 K of Superheat.

figures. Typical overall efficiencies are in the vicinity of 19% for condenser temperatures in the range of 900-1000 K, and turbine inlet temperature of 1450 K. Reactor circulation pump, electrical power conditioning, regulation and transmission losses are not included.

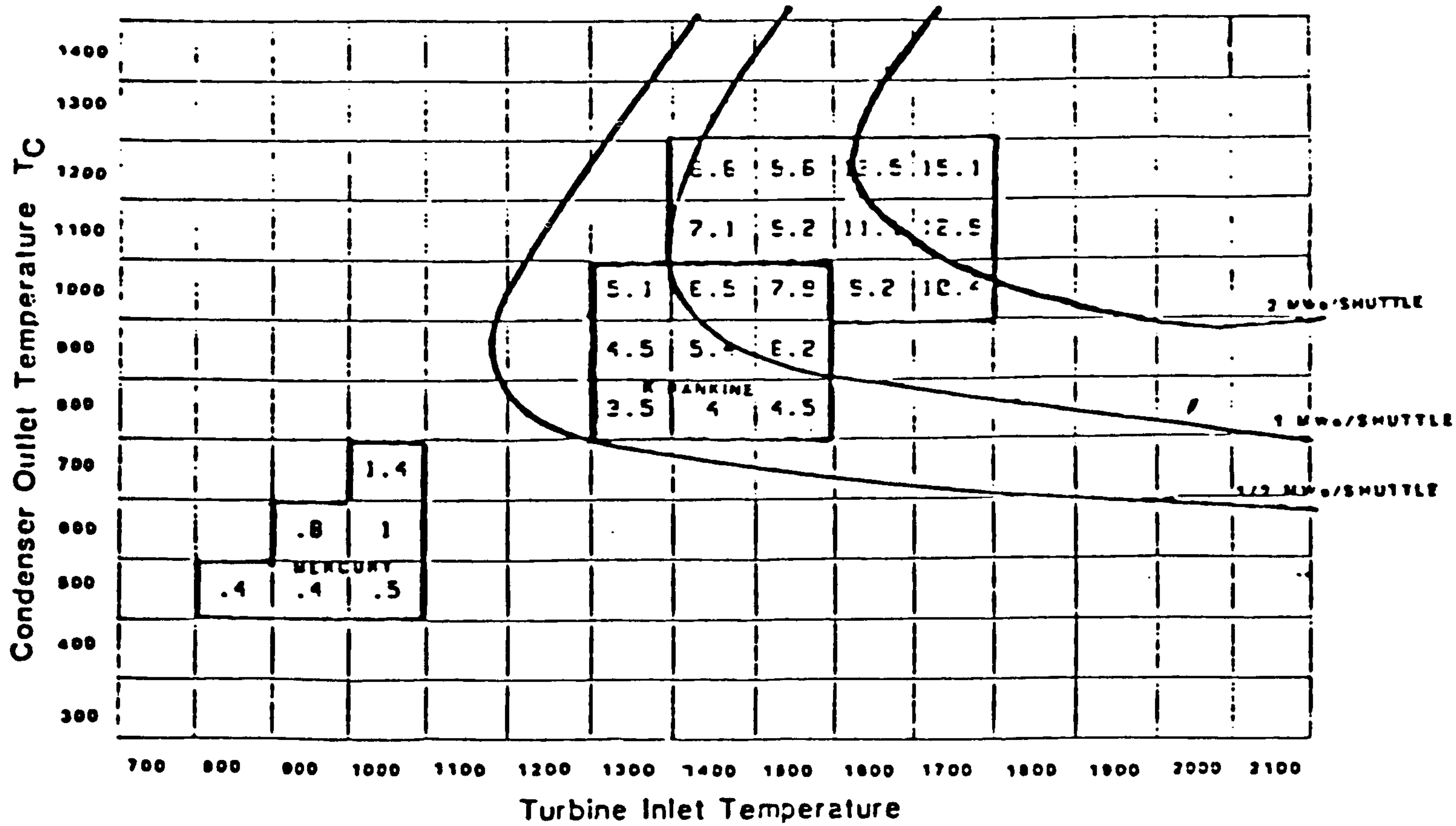
Probable Operating Regime. The alkali metal Rankine cycle system operates in a relatively small region of peak cycle and heat rejection temperatures. This is because of the rapidly changing vapor pressure of the fluid. In this limited range, it is possible to use a single value for the fraction of Carnot efficiency achieved in the cycle in order to calculate its performance. A preliminary calculation of P_e/A_R for the Rankine cycle system is shown in Fig. 4.5.1.15. The radiator is assumed to be at an average temperature 100 K below the condenser temperatures shown on the chart.

The data in Fig. 4.5.1.15 were generated using a constant 60% of carnot system efficiency. Based on the more detailed calculated results shown in Fig. 4.5.1.14, a value of approximately 75% has been estimated for the turbine expander efficiency only [41]. These values of component and device efficiencies seem reasonable for megawatt class machines that are used in the MCNSPS cycle selection analysis. Although the Rankine cycle is operated in a relatively small range of temperatures, its high heat rejection temperature yields a relatively high value of electric power for a given radiator.

Weight estimates of a 10 MWe Rankine power system are presented in Fig. 4.5.1.16. Turbine inlet temperature was limited to 1450K, believed to be the upper limit for ASTAR 811-C used as turbine blades. Using approximately 30,000 kg as the shuttle lift capacity, a 5 year life 10 MWe Rankine system could easily be lifted by 2 shuttle deliveries. The shuttle-stowable Rankine power conversion module is shown in Fig. 4.5.1.17. The main radiator subassembly (the telescoping radiator), can be lifted in the next subsequent shuttle.

EXAMPLE CALCULATION OF ELECTRICAL POWER PRODUCED PER UNIT RADIATOR AREA FOR THE RANKINE VAPOR CYCLE vs. TURBINE INLET AND CONDENSER TEMPERATURES

($T_{radiator} = T_{condenser}$ -100 K)



10 MWe RANKINE SYSTEM

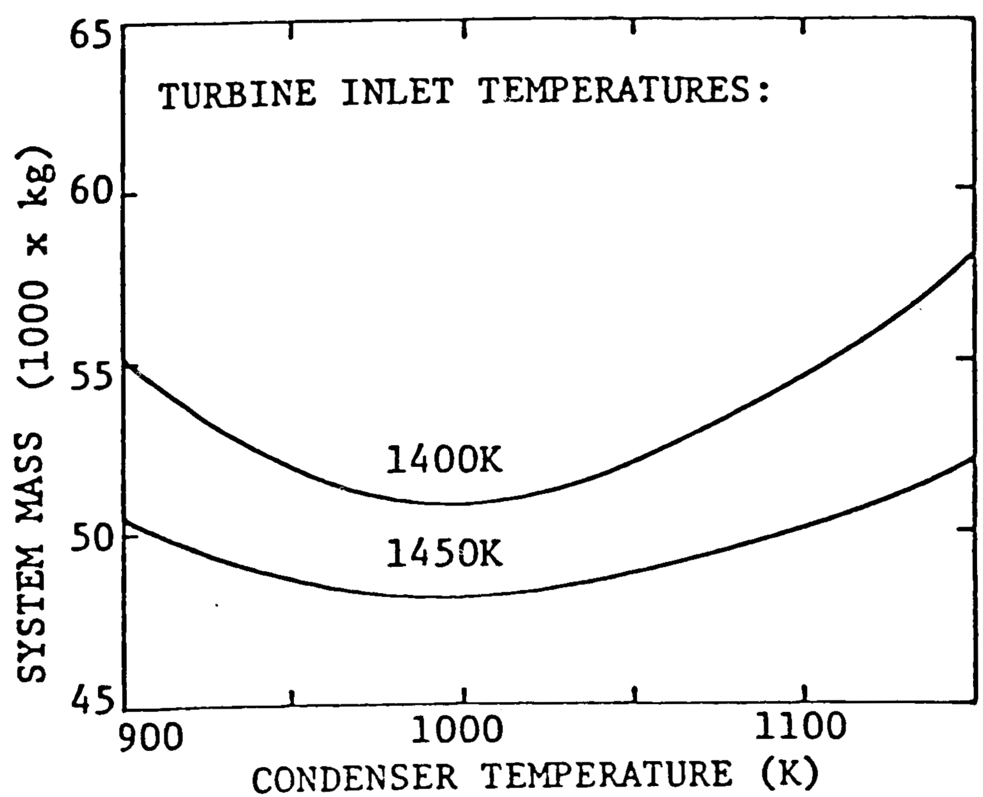


Fig. 4.5.1.16

10 MWe RANKINE CYCLE

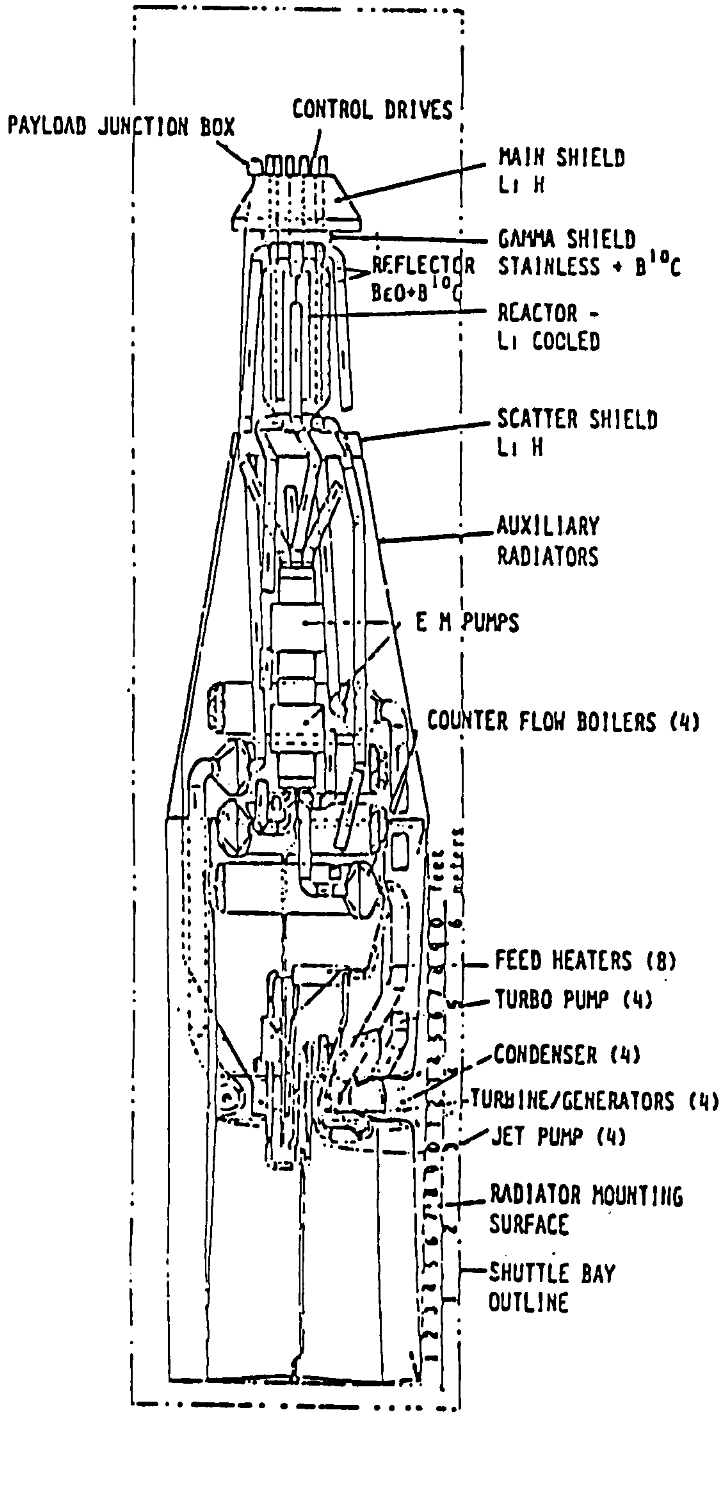


Fig. 4.5.1.17

4.5.2 In-core Thermionic Conversion System

General Characteristics

Preliminary comparison of thermo-electric versus thermionic direct power conversion systems indicated that thermionic systems had a substantial mass and radiator size advantage for space power systems above 100 kWe output. That mass and size advantage increases to factors of 2 to 4 as power output is increased to the multimegawatt range. Thermionic conversion is one type of static energy conversion, which has the potential to produce multimegawatt space nuclear power systems in mass and size competition with the other promising high power systems, which are dynamic. Direct thermionic conversion has the advantageous features of:

- Direct Conversion of Energy: Electric power is produced without the use of moving parts. This helps minimize wear-out, startup and restart mechanisms, and eliminates the problems of satellite inertial stability associated with dynamic systems.

- Modularity: The converters can be developed and produced as small modules, performance checked and assembled in series-parallel arrays of any desired size. This results in high reliability due to system redundancy and a lack of single point failure. It also reduces development time and cost, since development of the small module is greatly facilitated by lower unit costs and short iteration times.

- Good Conversion Efficiency with High Heat Rejection Temperatures: The thermionic converter has demonstrated attractive efficiencies and lifetimes at heat rejection temperatures exceeding 1100 K.

Reactor Types. A thermionic conversion system is potentially compatible with high-temperature liquid metal, heat pipe, and gas-cooled reactors. However, it is best capable of operating inside the reactor core with liquid metal coolant.

The results of this study have shown that in-core thermionic conversion is a preferred approach for MCNSPS applications. The in-core converter system concept is schematically illustrated in Fig. 4.5.2.1.

The in-core thermionic reactor (ITR) eliminates the need for a high temperature heat transfer loop between the reactor and energy conversion system. This has two advantages; it permits operation with a higher hot side temperature for the energy converter, and it allows most of the reactor (coolant, vessel, control drums, etc.) to operate at the energy converter cold side temperature. Heat transfer from the reactor around the shield is at the heat rejection temperature, minimizing problems of reactor and shield cooling. The ITR is the only known concept that can generate multi-megawatt power levels in shuttle launchable packages with near state-of-the-art reactor and component temperatures.

In comparing the feasibility and reliability of TI systems with other approaches, three considerations are paramount: fuel swelling has the potential for creating electrical short circuits between emitter and collectors, ceramic insulators within the reactor are subject to irradiation damage, and the presence of converter components within the core results in a larger reactor and correspondingly larger shield.

Thermionic Conversion Principles

Cycle. The thermionic energy converter is a non-mechanical gaseous-electronic device for converting heat directly into electric power by thermionic electron emission. In its simplest form the diode, shown schematically in Fig. 4.5.2.2, consists of two electrodes separated by a narrow gap (0.2-1mm), typically filled with cesium vapor at ≈1-10 Torr. Electrons are emitted from a hot electrode, the emitter, and are collected at a different potential by a colder electrode, the collector.

Heat is supplied to the emitter, to maintain a high enough temperature to emit electrons. The electrons cross the interelectrode gap and are collected by the collector. Heat is removed from the collector to maintain its temperature sufficiently low that it cannot emit electrons. The

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THE THERMIONIC IN-CORE CONVERTER REACTOR POWER SYSTEM (U)

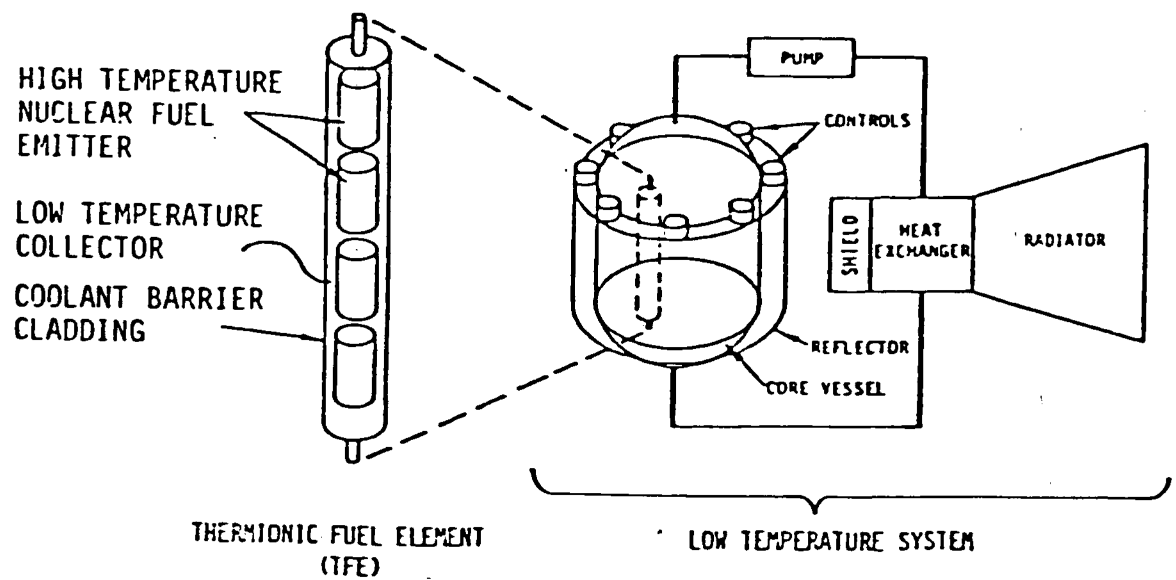

FIG. 4.5.2.1
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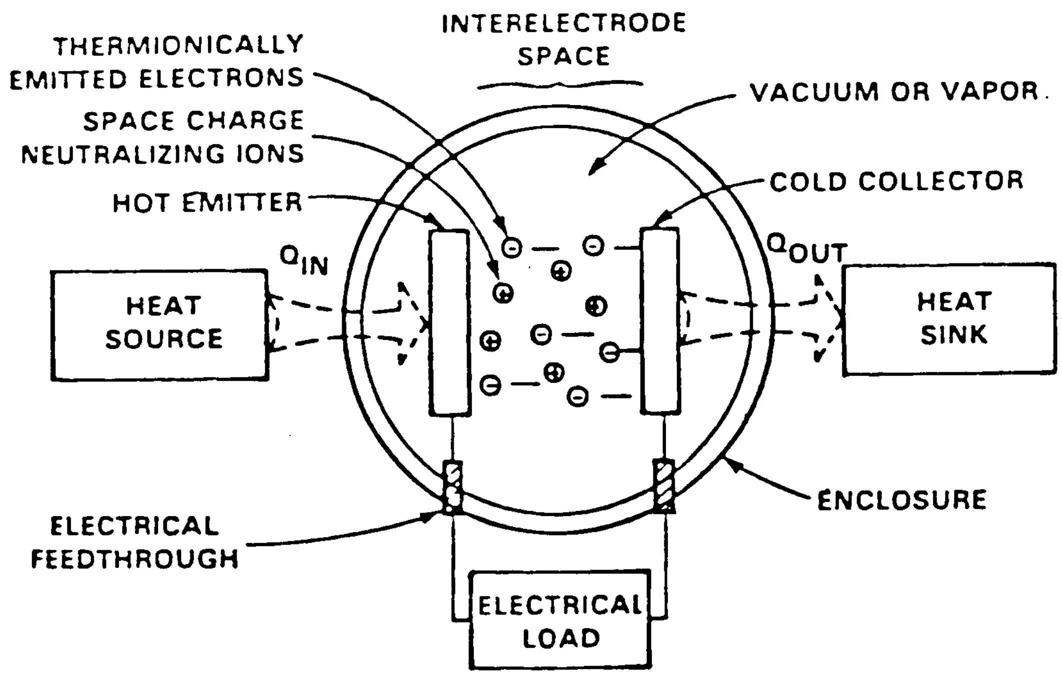
THERMIONIC CONVERTER SCHEMATIC



EFFECTIVE WORK = $Q_1 - Q_0$; EFFICIENCY $\alpha \dfrac{T_{HOT} - T_{COLD}}{T_{HOT}} \alpha\ \eta_{CARNOT}$

Typical Operating Regime

EMITTER TEMPERATURE :	1300 - 2000 K
COLLECTOR TEMPERATURE:	800 - 1100 K
ELECTRODE EFFICIENCY :	Up to 25%
POWER DENSITY :	1 - 10 W/cm^2

FIG. 4.5.2.2

collected electrons return to the emitter by flowing through the external load circuit.

Three operating modes are presently of practical importance and must be considered in a comprehensive review of potential systems.

1. Ignited Mode. Normal operation of a thermionic converter at practical current densities requires that ions be generated in the interelectrode space, typically accomplished by maintaining a low pressure cesium discharge in that space. In this mode, voltage drop across the discharge of about 0.5 volts is required, a substantial performance penalty.

2. Unignited Mode. At high emitter temperature ($\geq$2100 K) the ions can be generated thermally at the emitter instead of in a discharge, resulting in a much lower voltage loss (typically $\approx$ 0.1 volt). This mode of operation has seldom been tested in the U.S. because of the high emitter temperatures required, but it has been studied extensively in the USSR.

3. Quasi-Vacuum Mode. Operation with very close interelectrode spacings (2×10^{-3} - 10^{-2} mm) eliminates the need for ion generation, reducing the voltage drop across the gap to 0.1-0.2 volts. This operating mode is effective at emitter temperatures as low as 1200 K. Until recently no practical converters have been able to avoid electrical short circuits between the electrodes in this mode, but the new SAVTEC converter design at Rasor Associates has recently shown encouraging results.

Performance Analyses

A variety of analytical models of thermionic converter performance have been developed. The most precise and detailed of these model the transport of electrons and ions in the plasma, allowing accurate calculations of both electrical and thermal behavior. These are designated as fundamental models, eg. the ignited mode computer program, IMD-4, developed by Rasor Associates, Inc. IMD-4 contains enough detail of the physical processes involved to be successfully used for exploring approaches to advanced performance through "computer experiments", which have later been verified by laboratory testing. Another important use of the models is to serve as a basis of comparison in establishing the region of validity of more approximate calculational methods.

Idealized Model. The complexity and slow computing times of fundamental models make them poorly suited for system design studies. In addition separate computer programs must be used for each type of operating mode. Consequently, several approximate converter models have been used for calculating performance in MCNSPS studies. The first of these is a variant of an ideal diode model to which additional parameters have been added in order to describe non-ideal effects. This model will be described here to illustrate some of the relevant and important thermionic converter physics and to illustrate a simple method to calculate performance.

The internal electron potential of an ideal thermionic converter, called a motive diagram, is represented in Fig. 4.5.2.3. In the motive diagram ϕ_E and ϕ_C are the emitter and collector work functions, respectively. They represent the potential barrier which must be overcome by electrons leaving the electrode. The output voltage V of the converter is the difference between the emitter and collector Fermi levels.

Parametric analysis of a thermionic converter requires specifications of the following parameters:

1. Emitter Temperature, T_E (K)
2. Collector Temperature, T_C (K)
3. Current Density, J (A/cm^2)
4. Arc Drop, V_d (eV)
5. Collector Work Function, ϕ_c (eV)
6. Current Attenuation Factor, F_A

To calculate the diode output power with this model, the following steps are taken. First, the desired output current density J is selected along with the collector work function ϕ_C, the arc drop V_d, and the current attenuation

CHARACTERISTICS OF AN IDEAL THERMIONIC CONVERTER

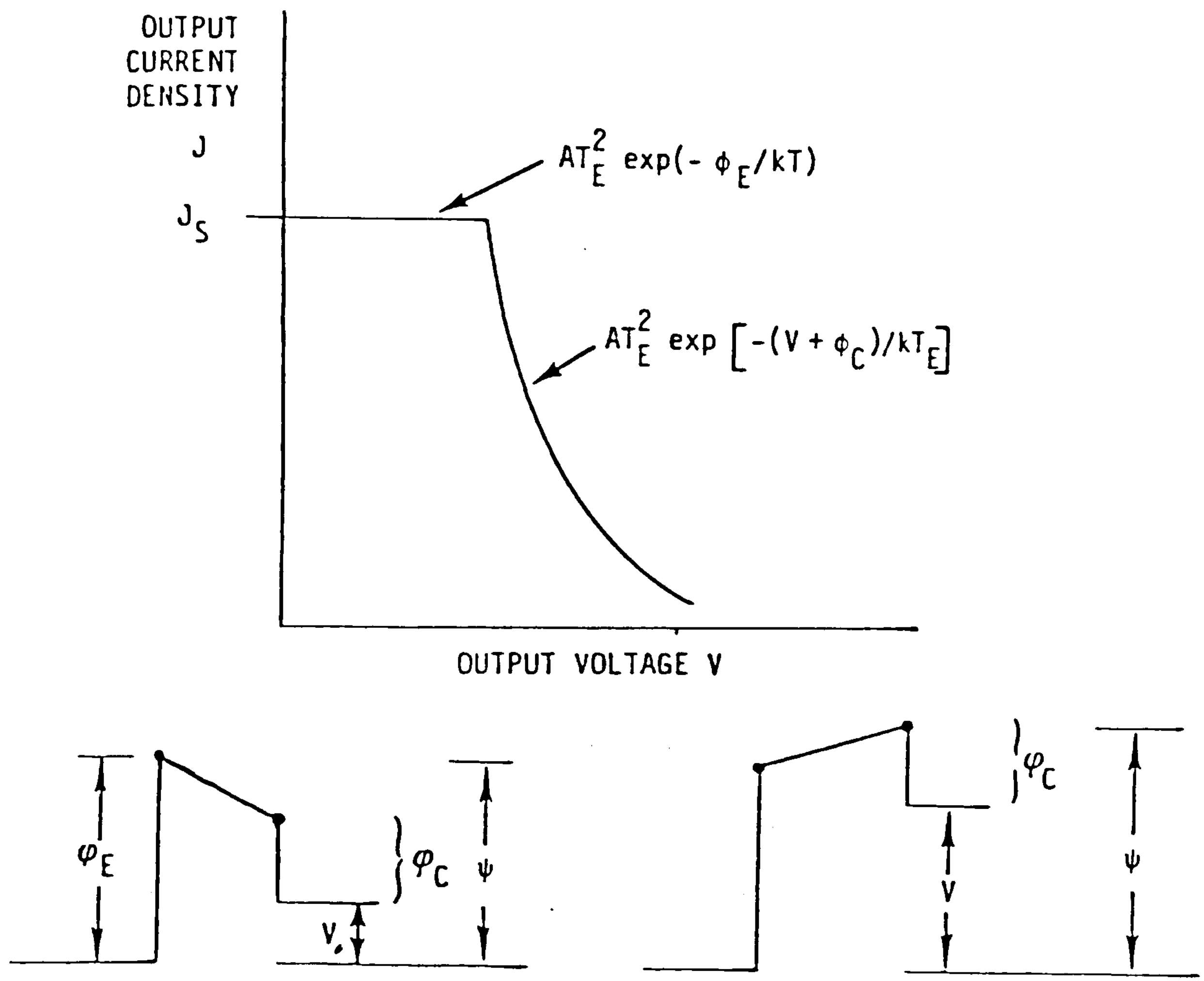


FIG. 4.5.2.3 CONSTANT J AS V INCREASES; DECREASING J AS V INCREASES

factor F_A. The temperatures of both emitter and collector must also be specified. The ignited mode operation is characterized by $V_d \approx 0.5$ eV and $F_A = 1.75$. Unignited mode has $V_d = 0$; but the value of J must be chosen to correspond to space charge suppression by emitter surface ionization only.

Next, the collector back emission, J_C is calculated from Eq. 1.

$$J_C = 120\ T_C^2 \exp(-\phi_C/kT_C) \qquad (A/cm^2) \qquad (1)$$

Next an effective barrier height, Ψ is calculated:

$$\Psi = kT_E \ln \left[\frac{120\ T_E^2}{(J+J_C F_A)}\right] \qquad (2)$$

The output voltage of the operating point is then calculated:

$$V_o = \Psi - \phi_C - V_d \qquad \text{(volts)} \qquad (3)$$

The thermal input power density P_{in}, which is comprised primarily of radiative heat transfer and electron cooling of the emitter, is given by:

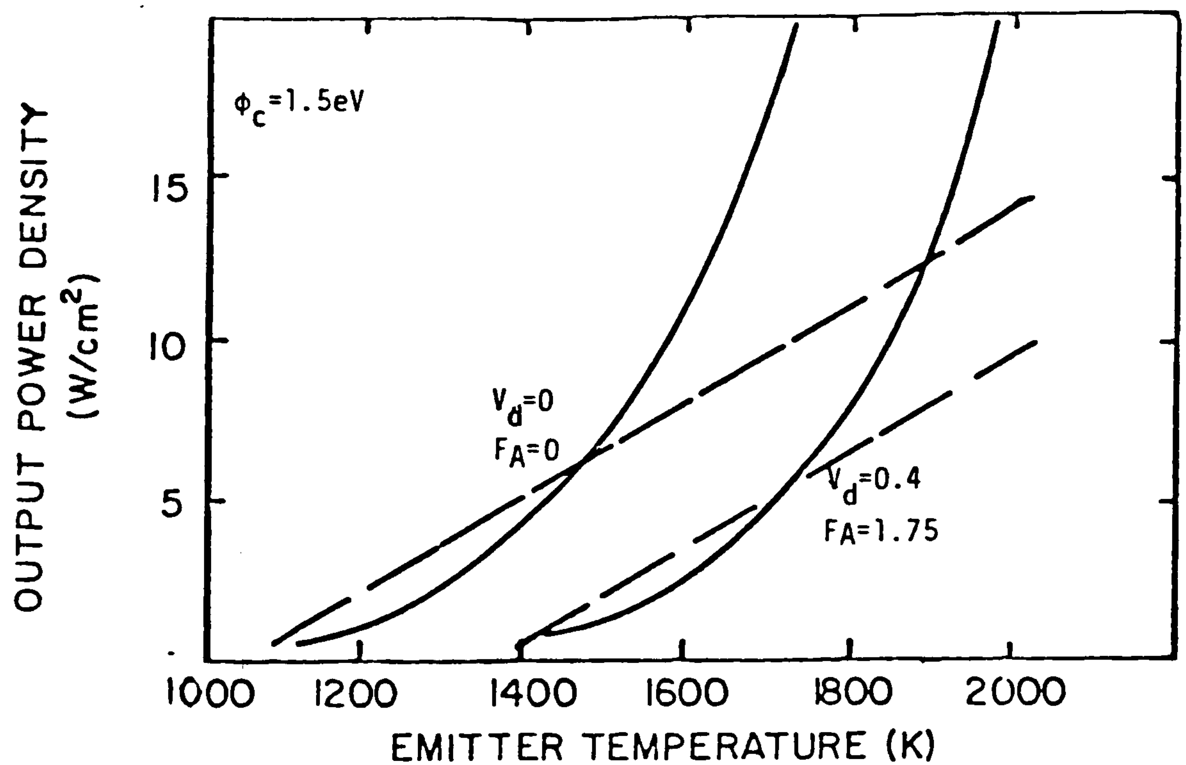
$$P_{in} = 1.6\times10^{-3}\ JT_E + 1.1\times10^{-12}\ (T_E^4 - T_C^4) \qquad (W/cm^2) \qquad (4)$$

The efficiency of the converter at its output leads includes terms for the electric output and thermal input along with other terms to account for resistive voltage drops and heat conduction in the lead. Approximately 10% of the input heat is conducted by an optimum lead and approximately 10% of the electric power is lost by Joule heating in the lead. Thus, the efficiency of a thermionic converter with optimized leads is given by:

$$\eta = \frac{0.9\ J\ V_o}{1.1\ P_{in}} \qquad (5)$$

Fig. 4.5.2.4 and Fig. 4.5.2.5 show the output power density and lead efficiency which can be expected for a thermionic converter with a collector temperature near 900 K. Performance curves are given for a current density

THERMIONIC CONVERTER POWER DENSITY



LEGEND:

——— MAXIMUM EFFICIENCY

— — — 10 AMP/cm^2

FIG. 4.5.2.4

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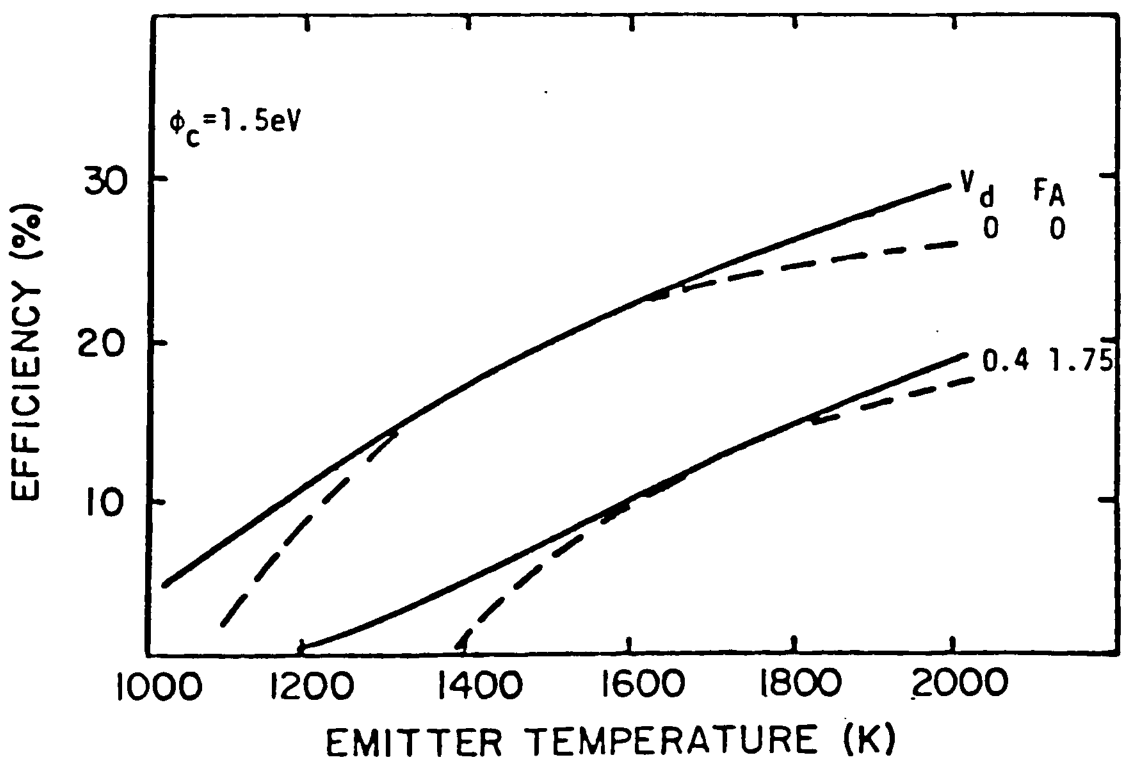
THERMIONIC CONVERTER EFFICIENCY (U)



LEGEND:

——— MAXIMUM EFFICIENCY

– – – – 10 AMP/cm^2

FIG. 4.5.2.5

11/14/84

of 10 A/cm². At high emitter temperatures, even higher current densities are optimum. The effect of arc drop on performance is also shown. The lower set of curves in each figure corresponds to an arc drop of 0.5 ev, approximately that which characterizes present fully optimized converters operating in the ignited mode. The higher performance set of curves correspond to zero arc drop, which is achievable in principle.

In fact, converters have been demonstrated which operate with near-zero arc drop. The close-spaced vacuum diode and the unignited cesium diode can operate at practical power densities, but they presently require the extremes of close interelectrode spacing or very high temperature technology, respectively.

Phenomenological Model. The ignited mode has the greatest practical importance in state-of-the-art thermionic conversion. This type of operation at 1850 to 1950 K emitter temperatures, which are compatible with long life Tungsten clad UO_2 fuel at high burnup, has been selected for the baseline MCNSPS thermionic system. Lighter, more compact megawatt class systems may be produced by using higher temperature unignited mode converters, but a major breakthrough emitter evaporation suppression and in nuclear fuel technology and venting would be required to achieve one to five year operating endurance. Analysis of the ignited mode can be treated with a more detailed model known as the phenomenological model. The phenomenological model has many characteristics of a correlation rather than a fundamental theory, although the relationships in its algorithm are based on physical principles. It has primarily been used in organizing parametric results of well-established experimental measurements, design studies, and systems analyses. Adjustable physical constants in the model can be set to correlate with experimental data. The model then provides good approximations of converter operating characteristics (i.e., current-voltage (J-V) curves) over at least the range of parameter variations in which the linkage of model assumptions to the experimental situation remains valid. The capabilities of the phenomenological model to predict ignited mode J-V curves are illustrated by Fig. 4.5.2.6a, in which experimental data for various cesium reservoir temperatures (Fig. 4.5.2.6b) are compared with values calculated by the phenomenological model.

COMPARISON OF RESULTS OF PHENOMENOLOGICAL MODEL TO EXPERIMENTAL DATA

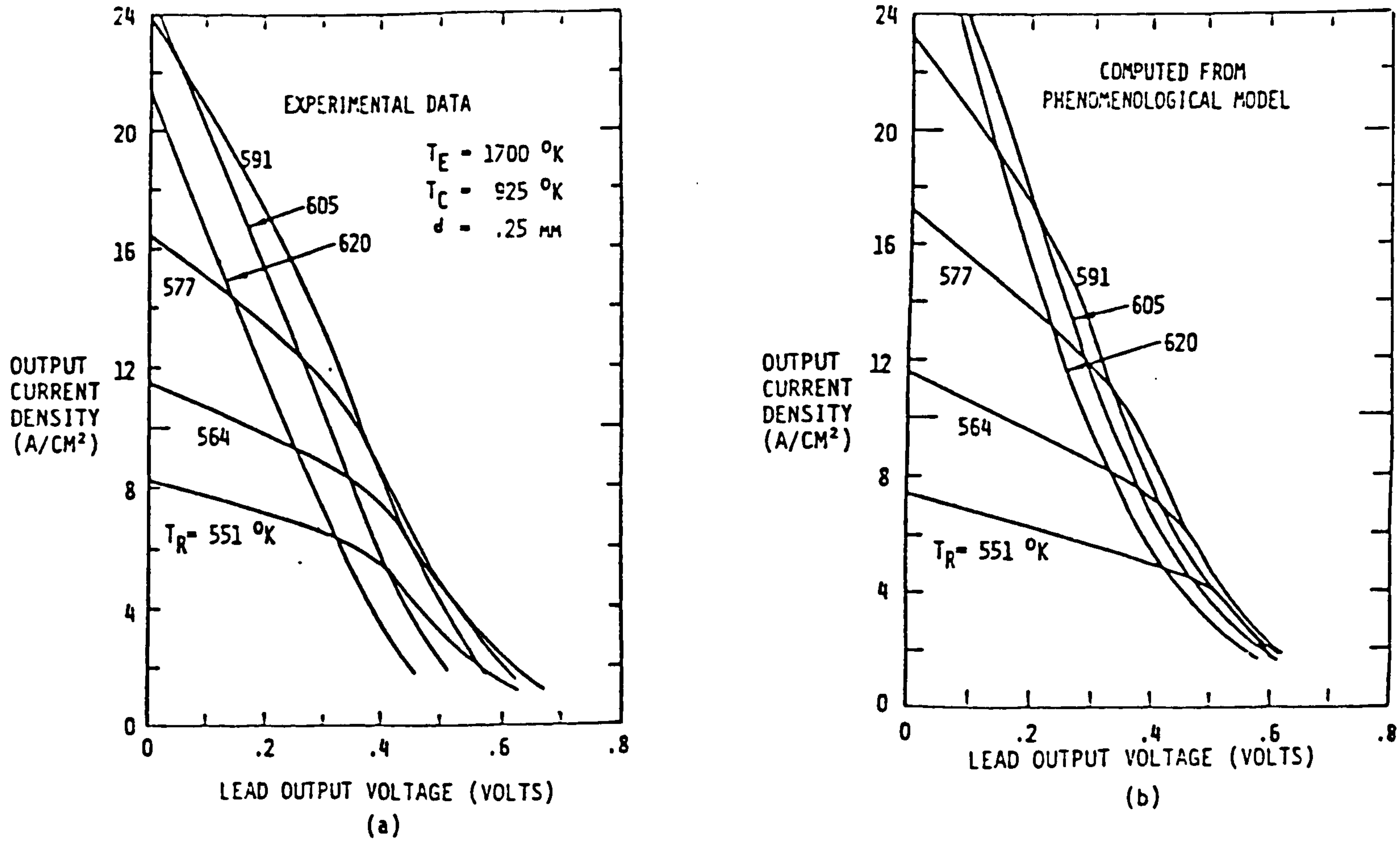


FIG 4.5.2.6

Subroutines which calculate ignited converter performance with the phenomenological model have been written in various computer languages for incorporation in systems analysis programs. This approach gives good results for ignited mode systems only.

Thermionic Hardware Experience. Practical nuclear fuel driven thermionic converters were very well demonstrated during the thermionic reactor space power program of the 1960s. The longest converter life test was designated LC-9, which was a cylindrical converter built and tested for NASA by General Atomic as part of the in-core nuclear space reactor program. LC-9 operated with electrical heating at an emitter temperature of 1970 K with stable performance for over five years. As shown in Fig. 4.5.2.7, LC-9 had an electrode efficiency of 17% and generated 8 W/cm^2 of output power (80 kWe/m^2). The converter was still performing stably when tests were terminated for programmatic reasons. This test well illustrates the long life capability of the thermionic conversion process. The tungsten/ cesium high temperature emitter combination forms an equilibrium system with no known degradation mechanisms except emitter evaporation. At typical converter operating conditions of 1850-1950 K in space power systems, such evaporation (30×10^{-8} m/yr at 2000 K) is unlikely to be a life limiting factor even for a century of operation.

The primary failure mechanisms for the converter are electrode distortion, which can cause a short circuit, and envelope leaks, which result in cesium loss. Both problems can be accommodated by proper design. Leaks occur typically at weld joints and more often at the ceramic-metal insulator-seal. Extensive development essentially eliminated such leaks by establishing weld and seal designs and fabrication practices for prototypical devices. Such practices must be modified and verified for each substantially different design, however. Fueled emitter distortion can also be controlled to a tolerable level by design and fabrication practice if verified by testing each new type of converter structure.

For in-core thermionic reactor designs the converter cell forms the fuel elements for the core. In this case the reactor fuel elements may consist of a series string of cylindrical thermionic converters, known as a TFE. The

LC-9 OUT-OF-PILE CONVERTER TEST HISTORY (U)

EMITTER TEMPERATURE: 1970 K

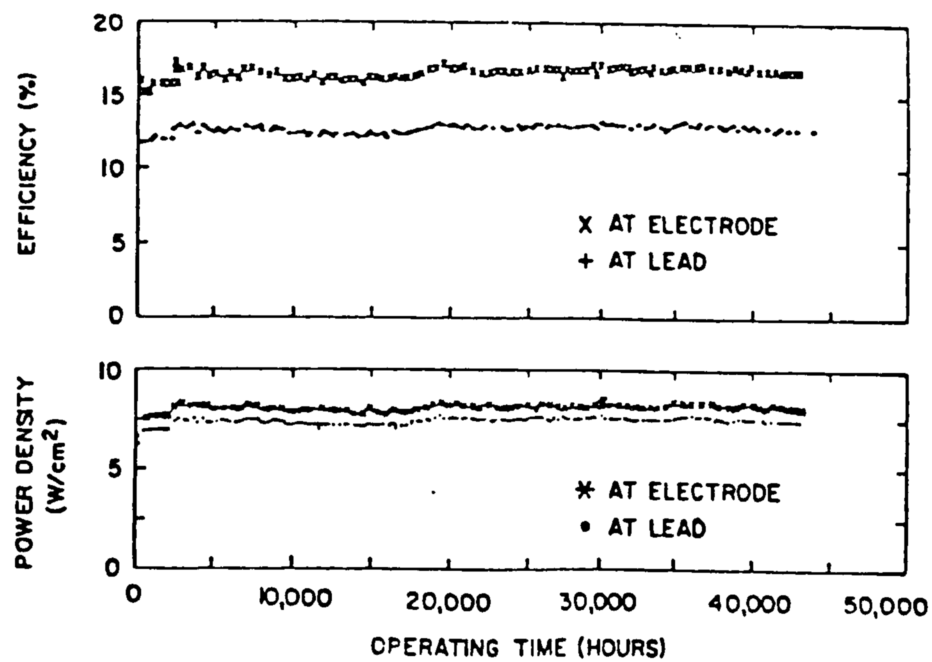


FIG. 4.5.2.7

11/14/84

cylindrical emitter of each converter contains the uranium fuel for the reactor. Such thermionic fuel elements were highly developed in the United States prior to termination of the space nuclear reactor power program in 1973. The primary life limitations encountered in in-core testing included: emitter cracking and short circuits due to swelling of the fuel, diffusion of carbide fuel components to the emitter surface with consequent performance loss, and leaks in the alumina insulator seals due to fast neutron damage. During the previous US program, lifetimes of prototypical TFE's were regularly exceeding 8000 hrs. In-core tests of TFE's with stable performance had reached 11,000 hours by the end of the program. Joint failures had largely been eliminated. Emitter swelling and radiation damage to ceramic components were life limiting at 12,000 and 20,000 hours, respectively, but a variety of approaches to solving these problems exist now and are being pursued in the present SP-100 predevelopment program. The longest life of an in-core thermionic converter was a thermionic fuel element operated at the Nuclear Research Center in Karlsruhe, Germany for 31,894 hours -- 23,240 hrs at full power.

Performance Measured: Power Density and Efficiency

The two critical performance parameters for a TFE are power density and efficiency. The bottom line for both is their value at the leads of the TFE, where interconnections to other TFE's and the rest of the system can be made. However, significant losses must be incurred in delivering power from the surfaces of the emitter and collector to the TFE leads, and similarly, thermal losses not associated with the emitter and collector surfaces exist in every TFE design. These losses are very design dependent, and as a result the performance of any particular TFE is a combination of two things: how well the thermionic conversion function is accomplished between the emitter and collector, and how well the TFE was designed to deliver this power to its leads. Typically, lead electrical values will be, at most, 90% of the corresponding electrode power due to voltage drops imposed by that portion of the leads which connect directly to the emitter. Similar thermal losses down the emitter lead set a ceiling on lead efficiency near 80% of electrode efficiency. Any further losses are the result of design trade-offs and are not imposed by physical limitations inherent to the converter.

Some of the design trade-offs that must be made are the voltage drop caused by conducting the current along the length of the emitter and the collector versus the amount of material thickness in the electrodes. This voltage drop can be made arbritarily small but the thicker electrodes displace fuel from the reactor core and add mass to the system. The cell aspect ratio (L/D) also affects the voltage drop. A typical compromise is to permit ~1-4% power loss in the electrodes. Other voltage losses occur in the TFE interconnects and the low voltage bus bars leading to the power conditioner. Again a few percent voltage loss is typically permitted in a system design.

Design dependent thermal losses analagous to the electrical lead losses also exist in every TFE design. Examples of these heat losses include conduction and thermal radiation from the bottom of the emitter to the top of the next cell, radiation from the top of the emitter through the emitter lead, and radiation from the outside of the large emitter lead itself. The advanced ThermoElectron G series design reduced these losses by using the bottom of the emitter as an active electrode, putting this heat to work instead of incurring the loss. Similarly, the G series uses a shorter emitter lead, reducing radiation losses from that surface. The West German ITR design carried this approach even farther by using an emitter lead that is both narrow and short. Using these type of design features make it possible to reduce the bypass heat to only a few percent. The cummulative effects of such parasitic losses are to degrade the overall system efficiency from the intrinsic electrode efficiency by about 25 or 30 % (e.g. 18% electrode efficiency yields 13% system efficiency).

The difference in performance that can be achieved in laboratory planar and prototypical cylindrical converters must also be considered when assessing the status of demonstrated TFE performance and the potential for improvements. Much of the research work in thermionic conversion is performed with converters that have small (~2 cm^2) flat opposed electrodes. These planar devices have uniform temperatures, uniform and easily adjusted interelectrode gaps, and very well defined electrode areas. They can be closely adjusted to give nearly fully optimized electrical performance, but their thermal performance cannot be reliably measured because of external radiation losses.

Cylindrical converters can be designed to have a minimal radiation loss problem, and thus they are used for efficiency measurements. However, input power may not be applied uniformly to the emitter and, as a consequence, there may be a variation in emitter temperature and current density over the surfaces. Finally, uniform spacing between emitter and collector is difficult to achieve. As a result the performance of a cylindrical converter cannot be fully optimized, and it will always have a somewhat poorer average electrode performance figure than the optimized value.

With these points in mind the data in Fig. 4.5.2.8 and Fig. 4.5.2.9 can be used to assess the state of development of the cylindrical thermionic converter in 1973 and today. The figures show the initial electrode performance for a wide variety of in-core and out-of core TFE's and cells built prior to 1973. All data have been normalized to optimized collector and cesium temperatures. Typically the output power density increases with current density, although at the lower temperatures (<1600 K) there is an efficiency penalty above ~6 A/cm^2. At higher temperatures (>1800 K), one must operate at or above 10 A/cm^2 to realize the highest efficiency. Consequently data are shown for both 7 A/cm^2 and 10 A/cm^2.

Also shown for comparison are performance data obtained in 1978 on two cylindrical converters built at Rasor Associates, Inc. (RAI) for JPL, and ThermoElectron data showing the average performance obtained in 1979 with five heat pipe converters. The GGA converters, Fig 4.5.2.9, had tungsten emitters, some with [110] and some with [100] orientation. Both niobium and molybdenum collectors are represented. The RAI converter had rhenium emitters and molybdenum collectors. In these converters a performance improvement was obtained using a structured (CVD-Re) emitter in one case and a structured (grooved) collector in the other. The TECO converters had a W(110) emitter and a sublimed molybdenum collector applied in a partial pressure of oxygen.

The electrode efficiencies which are presently available are illustrated in Fig. 4.5.2.10. The band shown encompasses the results from two structured electrodes built by RAI for JPL in 1978. Also shown is an envelope of the

CYLINDRICAL CONVERTER PERFORMANCE

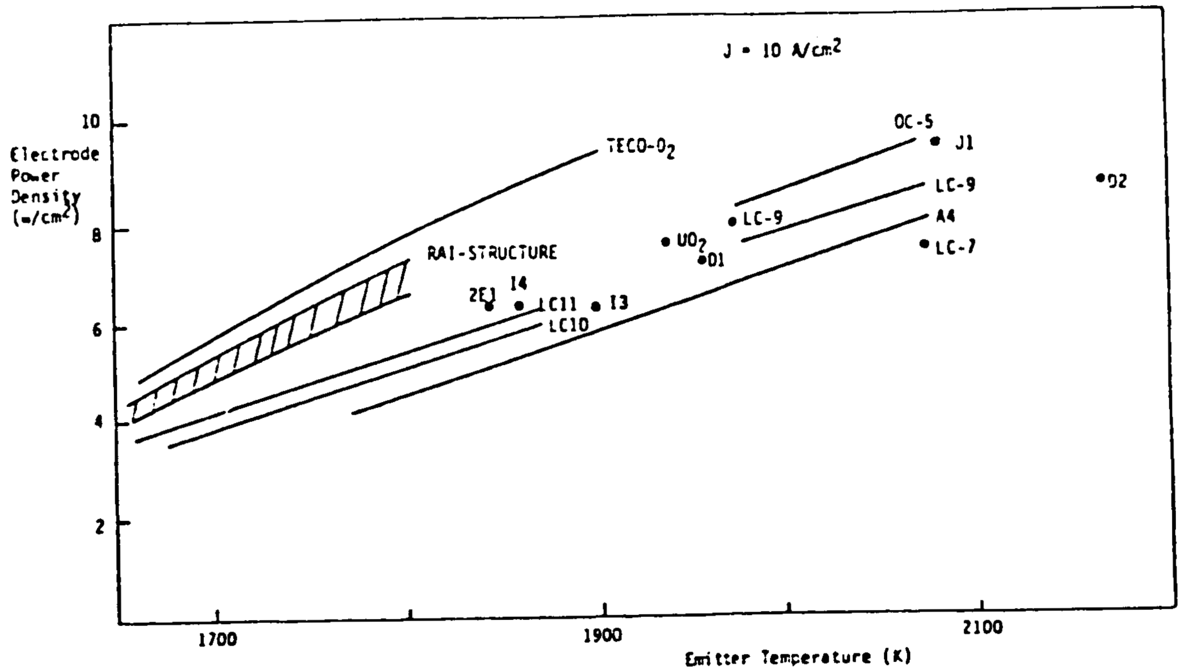


FIG. 4.5.2.8

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CYLINDRICAL CONVERTER PERFORMANCE

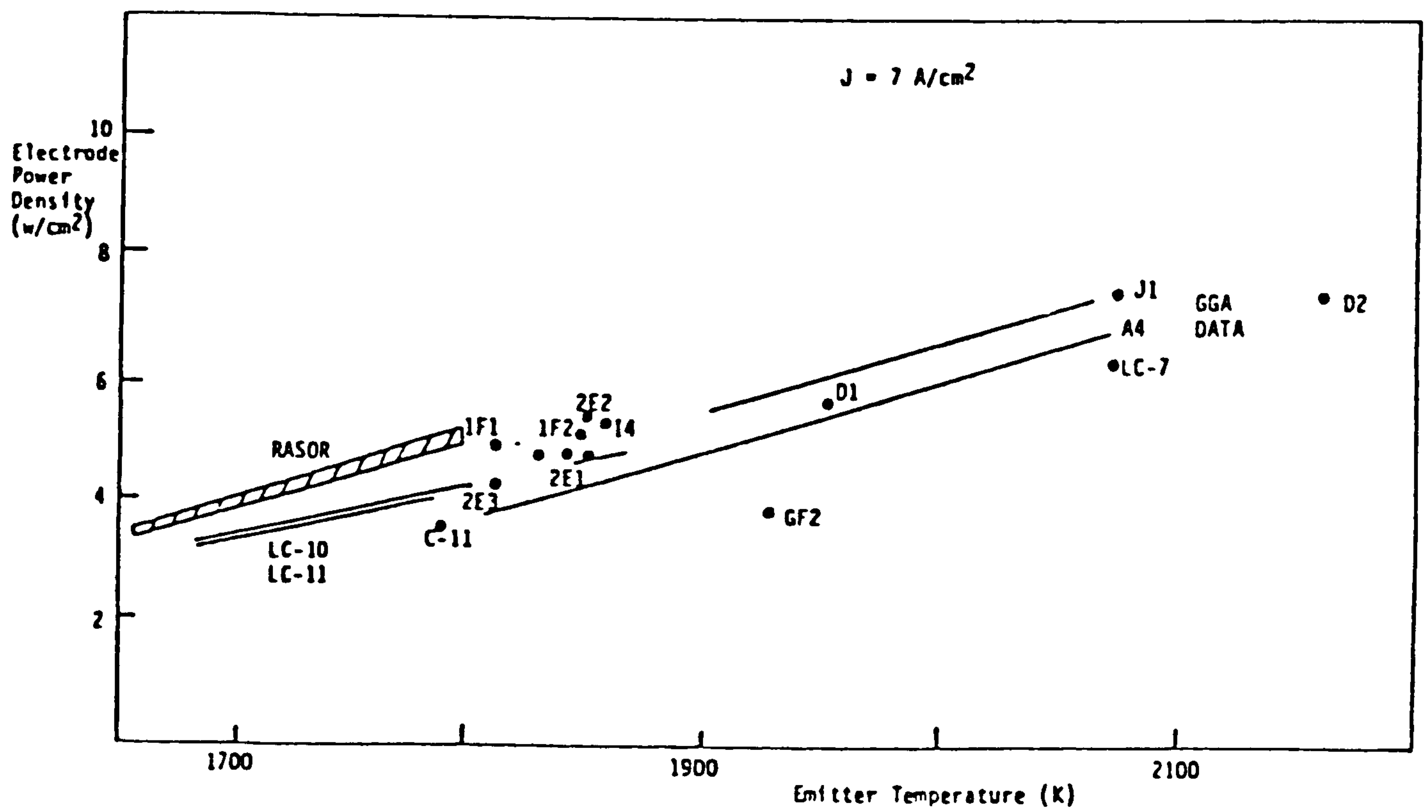
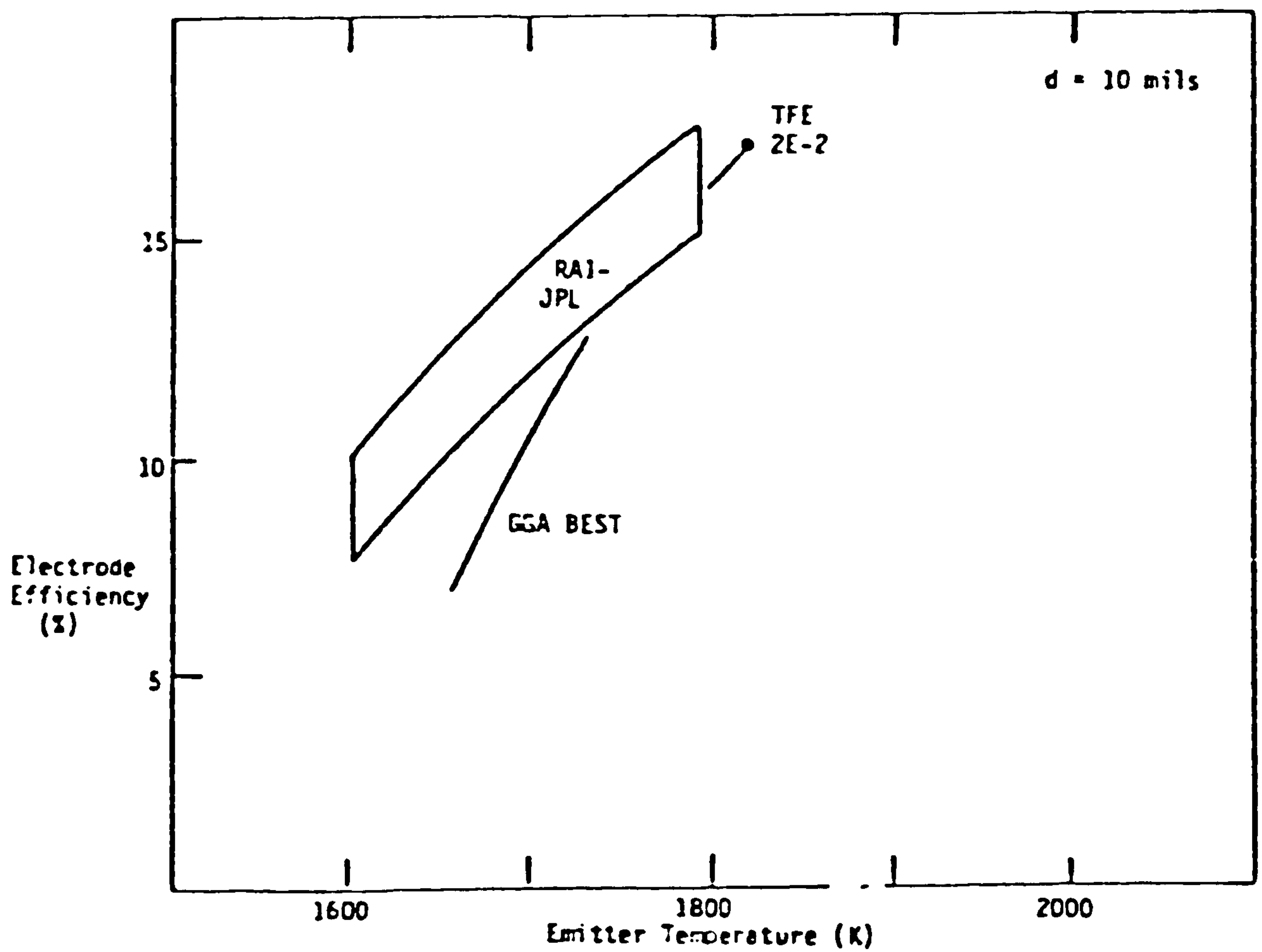


FIG. 4.5.2.9

CYLINDRICAL CONVERTER ELECTRODE EFFICIENCY



best data from GGA. The longest stable TFE test, TFE-2E2, was also one of the best performers.

Efficiency, in addition to its dependence on emitter temperature and current density, also depends strongly on collector temperature. The highest performance converters typically have low collector work functions and begin to lose efficiency at collector temperatures above 900 K (1160°F). Converters with higher collector work functions and relatively low peak efficiencies actually have superior performance at high heat rejection temperatures, 1050 K (1431°F) and above.

Two things should be noted from these data. First, good and reproducible performance can be obtained using cylindrical converters. For converters with similar electrodes, output power differences of less that 10% are to be expected. Second, the use of rhenium emitters and structured electrodes provides a performance advantage, particularly at very high power densities. This advantage corresponds to a 50 K increase in temperature at 7 A/cm^2 and 75-100 K at 10 A/cm^2.

Parametric Variation of MCNSPS Thermionic Systems

The reduction of converter efficiency with increasing collector temperatures requires an optimization tradeoff with the radiator mass. The mass of several 10 MWe system designs were calculated at various collector temperatures and are shown in Fig. 4.5.2.11. The optimum system mass is seen to occur at a collector temperature of about 1050 K. This optimum does not depend significantly on the emitter temperature.

The thermionic power conversion module is shown within the shuttle envelope in Fig 4.5.2.12. As with the Rankine system, the telescoping radiator can be lifted up in a second shuttle launch and mated to the thermionic power conversion module without welding.

10 MWe SYSTEM WEIGHTS
IN-CORE THERMIONICS

(POWER CONDITIONER WT.INCL.)

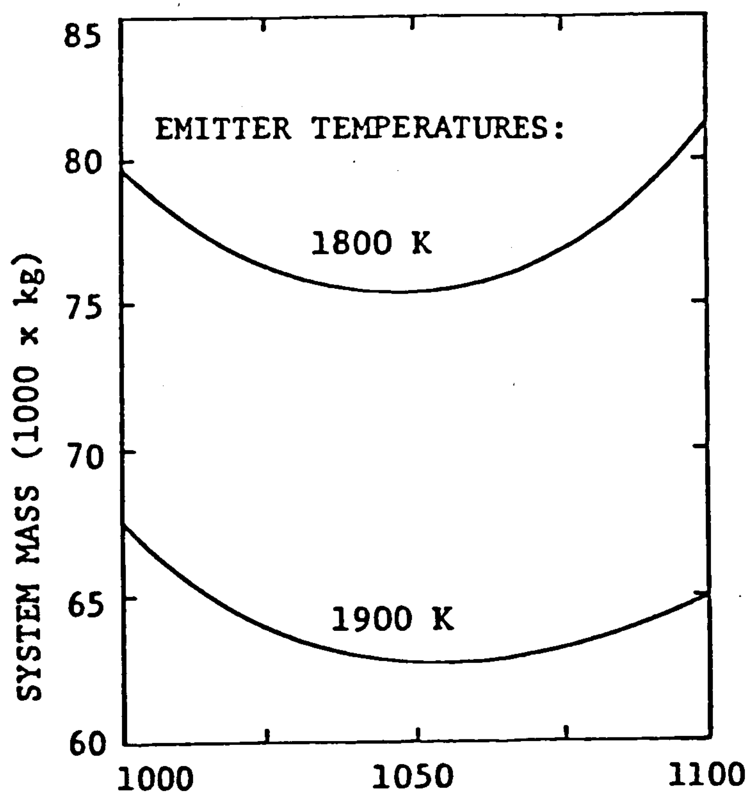


Fig. 4.5.2.11

10 MWe IN-CORE THERMIONIC SYSTEM

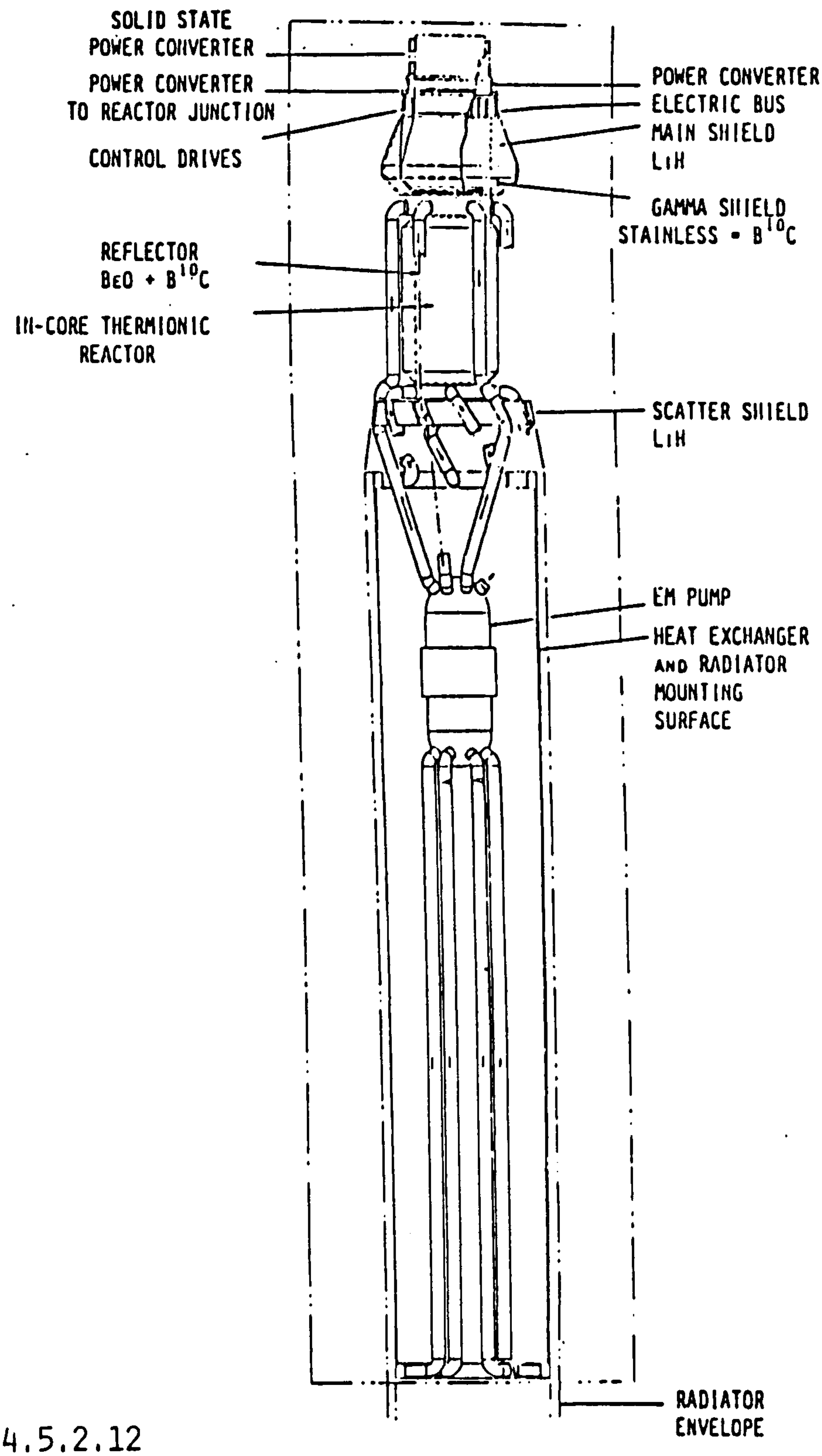


Fig. 4.5.2.12

4.5.3 Brayton Gas Turbine-Alternator System

Cycle Configuration. The Closed Brayton Cycle (CBC) gas turbine alternator space nuclear power system has the flexibility to be designed with a number of different loop configurations. Each configuration includes different combinations of gas and liquid metal cooled reactors and radiators.

Fig. 4.5.3.1 shows a one-loop Brayton system. In this configuration, the reactor is gas cooled. The reactor coolant is the turbo-compressor/alternator working fluid and is also the fluid which flows through the radiator. This configuration requires the least number of components and is the most simple design. The entire loop, and not just the turbo-compressor/alternator ducting, must be sized to minimize pressure drop. This results in large diameter ducting and components throughout the system. Since the volumetric heat capacity of a gas is significantly less than that of a liquid, relatively large piping/ducts are required. To protect those large areas of exposed ducting requires generous quantities of armor which leads to high weights. Particularly vulnerable to puncture is the radiator. In a one-loop configuration, a single puncture anywhere in the system would cause a complete loss of reactor coolant and turbo-compressor/alternator working fluid. Even if paying the armor penalty were acceptable, the existance of a half-acre of vulnerable, single point failure, surface area is not acceptable.

A three-loop arrangement is shown in Fig. 4.5.3.2. If there are redundant loops within the radiator loop, the puncture of a radiator only partially compromises the capacity of the radiator alone. The reactor coolant and turbo-compressor/alternator loops are unaffected. The three-loop concept permits use of optimum heat transfer and low pump power liquid metal fluids in the reactor and radiator loops. Three loops also minimizes loop piping/duct sizes, since pressure drop is dependent only on the components and piping within that particular loop. The disadvantages of the three-loop concept are that it is the most complex and results in a lowering of the effective radiator temperature because of the temperature drops across each of the heat exchangers; and the heat exchanger between the reactor and turbo-compressor/alternator loops must operate above the turbine inlet

ONE-LOOP BRAYTON CYCLE

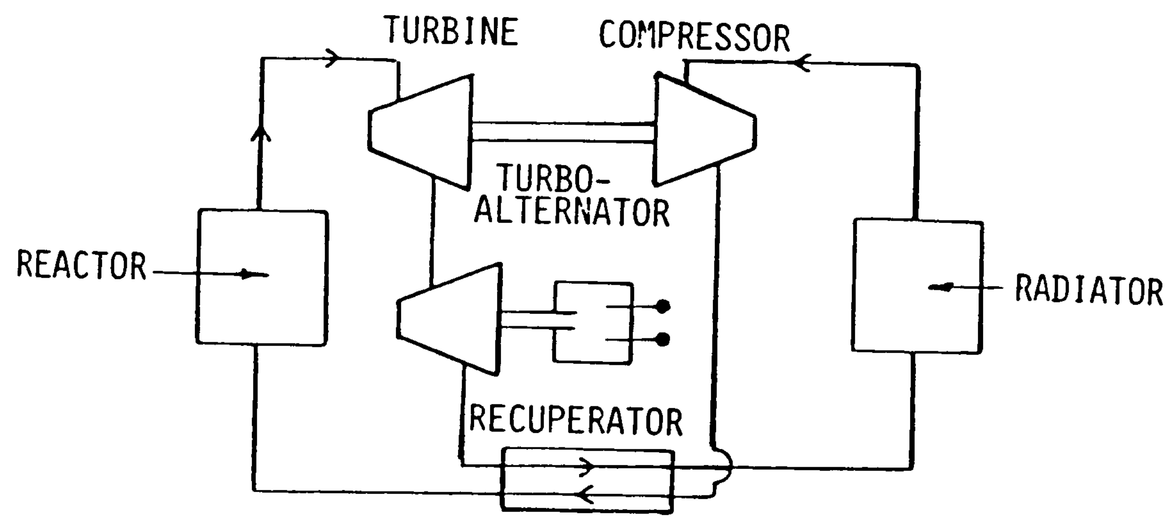


FIG. 4.5.3.1

THREE-LOOP BRAYTON CYCLE

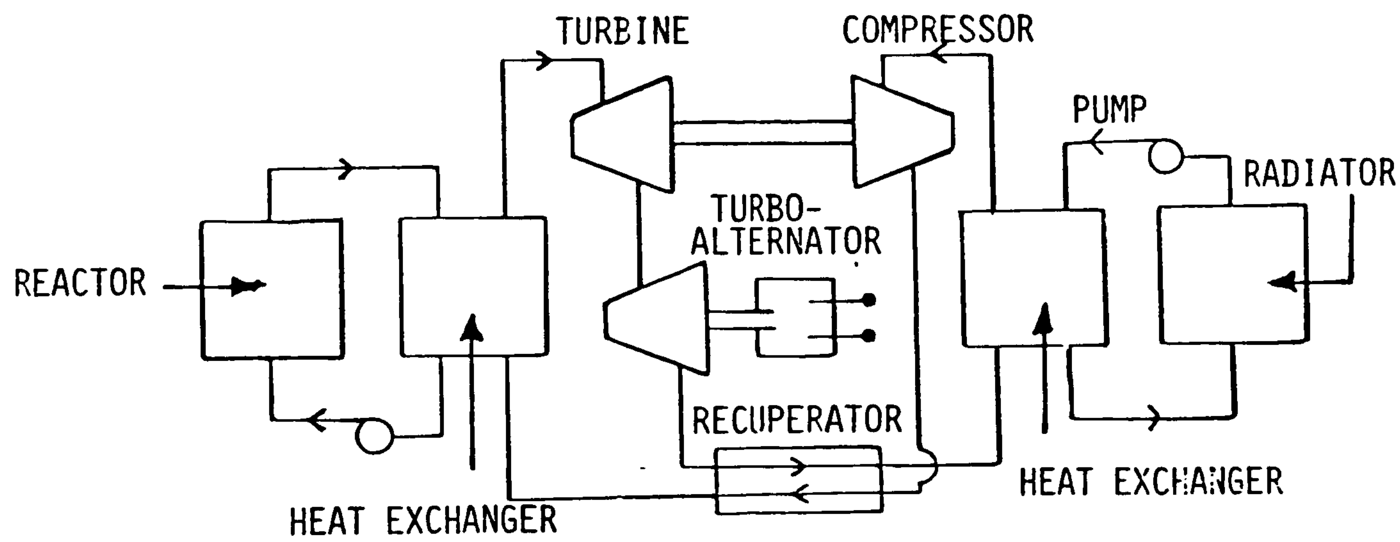


FIG. 4.5.3.2

temperature. This heat exchanger and the hot primary loop circulating pump, will probably constitute the least dependable components in the system.

The two-loop configuration is depicted in Fig. 4.5.3.3. The high-temperature heat exchanger and circulating pump between the reactor and the turbo-compressor/alternator loops has been removed. This necessitates a gas-cooled reactor, since the reactor coolant is also the turbo-compressor/alternator working fluid. As was pointed out in Volume II, there is a weight penalty for using a gas-cooled reactor for long endurance applications.

For a given reactor geometry and output level, pumping losses are large in a gas-cooled system relative to those in a liquid metal-cooled system, because the low volumetric heat capacity (cal/cc-C) of gases requires the pumping of large volumes of gas. The large pumping loss in turn requires a higher reactor gross output to compensate for the loss. Even at equal power ratings, the gas-cooled reactor core and vessel are larger (and heavier), and the ducting is larger than comparable liquid metal piping. Some of the weight gain realized in a gas-cooled reactor/two-loop concept is compensated by the weight saved by eliminating the high temperature heat exchanger and pumps. Elimination of the heat exchanger also reduces the reactor temperatures some 200 K, which substantially influences reactor and fuel life, reliability and mass. Or conversely, it permits raising the turbine inlet temperature some 200 K for the same reactor temperature limits. In addition, the reactor corrosion temperature limits of an inert gas-cooled reactor might be some 200 to 400 K higher than a liquid metal cooled reactor. Thus, a gas-cooled reactor might produce turbine inlet temperatures of some 1800 K whereas a lithium cooled reactor might produce gas turbine inlet temperatures of only 1400 to 1500 K.

Working Gas. The selection of a working gas in the direct Brayton cycle, which uses the turbine working fluid to cool the reactor as in the one-loop configuration is a compromise between heat transfer and turbo-machinery performances. He-Xe gas mixtures seem to offer significant advantages over single-gas selections. The He-Xe mixture has a very low Pr number, in the vicinity of 0.205. This is much lower than monatomic inert gases and is a

TWO-LOOP BRAYTON CYCLE

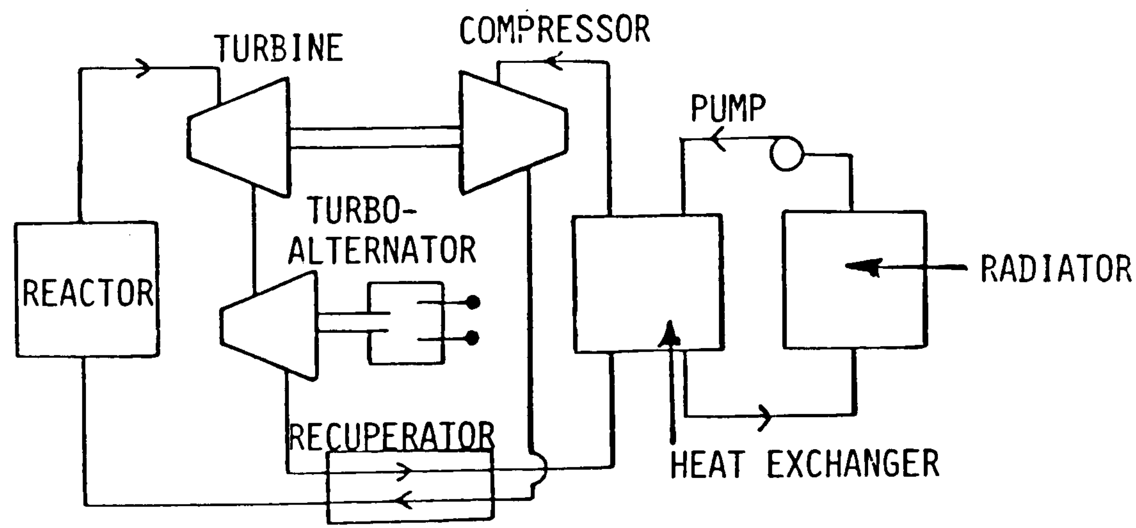


FIG. 4.5.3.3

measure of good heat transfer performance. Using a high molecular weight He-Xe gas mixture (effective molecular weight of 40) also makes the turbomachinery less complex (fewer stages).

Pressures. Although heat transfer and power density would both be improved by increasing the operating pressures, containment of the high pressure gases with components at high temperature usually requires a practical pressure limit. The maximum system pressure (compressor exit pressure) will be limited in this study to ≈68 atm (1000 psia). The compression ratio can be varied for optimization, but most closed Brayton cycle He-Xe space power systems optimize with a value of approximately 1.8 to 2.2. The CBC is particulary adaptable to operating at part load if gas can be summoned from the system to reduce pressure.

Electric Output. A particular advantage of the Brayton cycle, common to other dynamic systems, is the ability to generate A.C. electric power over a range of desired space system voltages, thus reducing the requirements for large switching power conditioning systems required for voltage transformation. Furthermore, the electric generating components can operate at higher temperatures than would be possible for a solid state power conditioning system.

Reliability and Modularity. Long-term reliability of rotating machines at high temperatures is always a question. Good reliability of Brayton systems with relatively low temperatures on the bearings and other critical areas has been demonstrated. However, at the required operating temperatures for high power MCNSPS, bearings must yet be demonstrated. It is anticipated that multiple counter rotating turbo-generator units would be required for mission confidence and torque balancing. For an overall power level in the 1 to 10 MWe range, turbo-alternator units of 1/4 to 2.5 MWe would be used. If the units are properly arranged, increased pressure and power output of surviving units can compensate for the failure of a single turbo-generator unit by providing auxilliary high pressure gas bottles.

Temperatures. The turbine inlet temperature is governed by the materials used for the first stage of the turbine rotor. Fig. 4.5.3.4 shows the temperature ranges, suggested by the Garrett Corporation, associated with three major types of turbine materials: superalloys; refractory metals; ceramics.

Mathematical Description. The principal components and temperature-entropy cycle for the single loop Brayton system are shown in Fig. 4.5.3.5. At low radiator temperatures * relatively high cycle efficiencies are possible for the Brayton system, upwards of 35% [1], and the cycle can obtain a fraction of ideal Carnot efficiency equal to 0.4 - 0.45. However, these high efficiencies are only obtained with a high Carnot efficiency at low heat rejection temperature with large radiators.

Analysis of a recuperated Brayton cycle requires the following parameters to be specified:

1. Peak cycle temperature (Turbine Inlet), T_H (K)
2. Minimum cycle temperature (Compressor Inlet), T_C (K)
3. Turbo-alternator efficiency, η_T
4. Compressor efficiency, η_C
5. Recuperator effectiveness, η_R
6. Specific heat ratio of the fluid, k
7. Compression ratio, P_2/P_1
8. Low temperature pressure loss, $\Delta P_1/P_1$
9. High temperature pressure loss, $\Delta P_2/P_2$

The compressor outlet temperature, T_2 is found by

$$T_2 = T_C \left[1 + \frac{1}{\eta_C} \left(P_2/P_1\right)^{\frac{k-1}{k}} - \frac{1}{\eta_C}\right] \cdot$$

The turbine outlet temperature T_4 is calculated from

$$T_4 = T_H - \eta_T T_H \left\{ 1 - \left[\frac{1 + \Delta P_1/P_1}{P_2/P_1(1 - \Delta P_2/P_2)} \right]^{\frac{k-1}{k}} \right\} \cdot$$

The regeneration temperature, T_x, is obtained from

$$T_x = T_4 \eta_R + T_2(1-\eta_R) \cdot$$

TECHNOLOGY ASSESSMENT
HIGH TEMP HOUSING MATERIALS

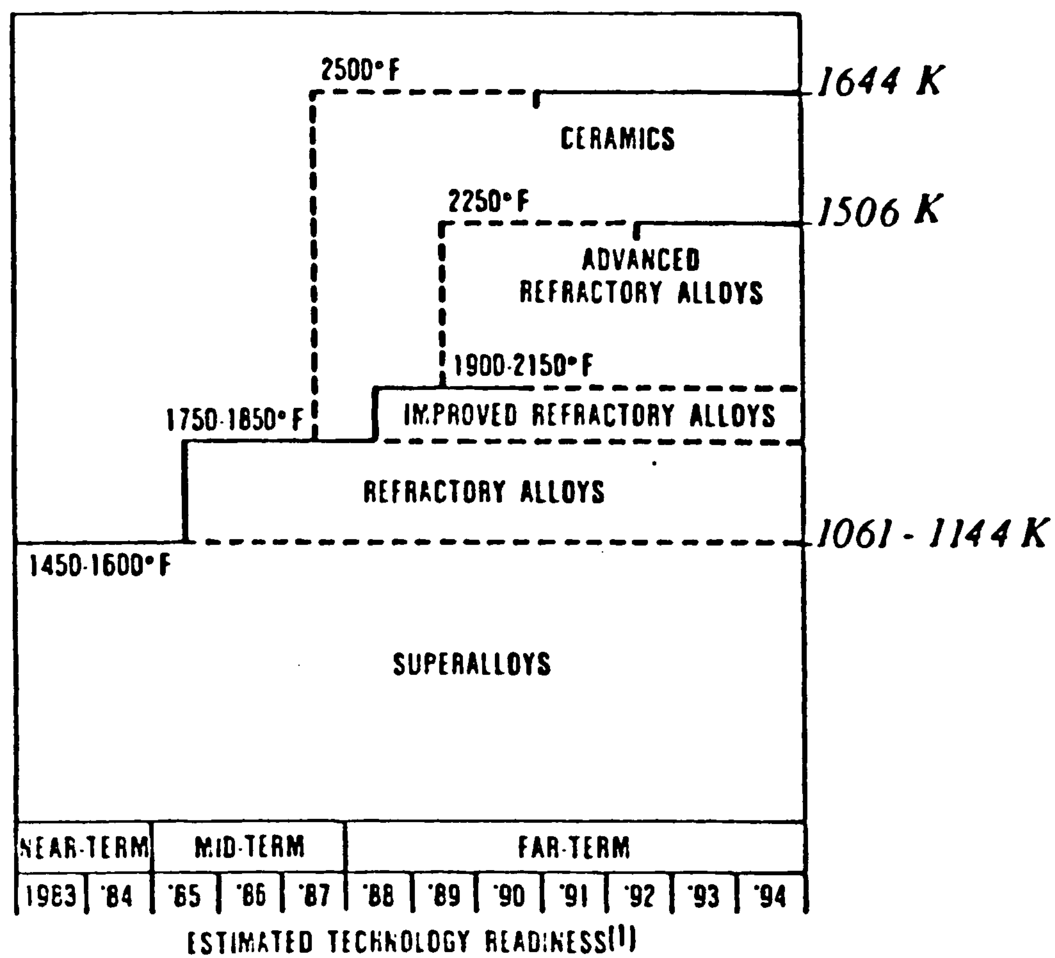


Fig. 4.5.3.4

THERMODYNAMIC CYCLE FOR RECUPERATED BRAYTON SYSTEM

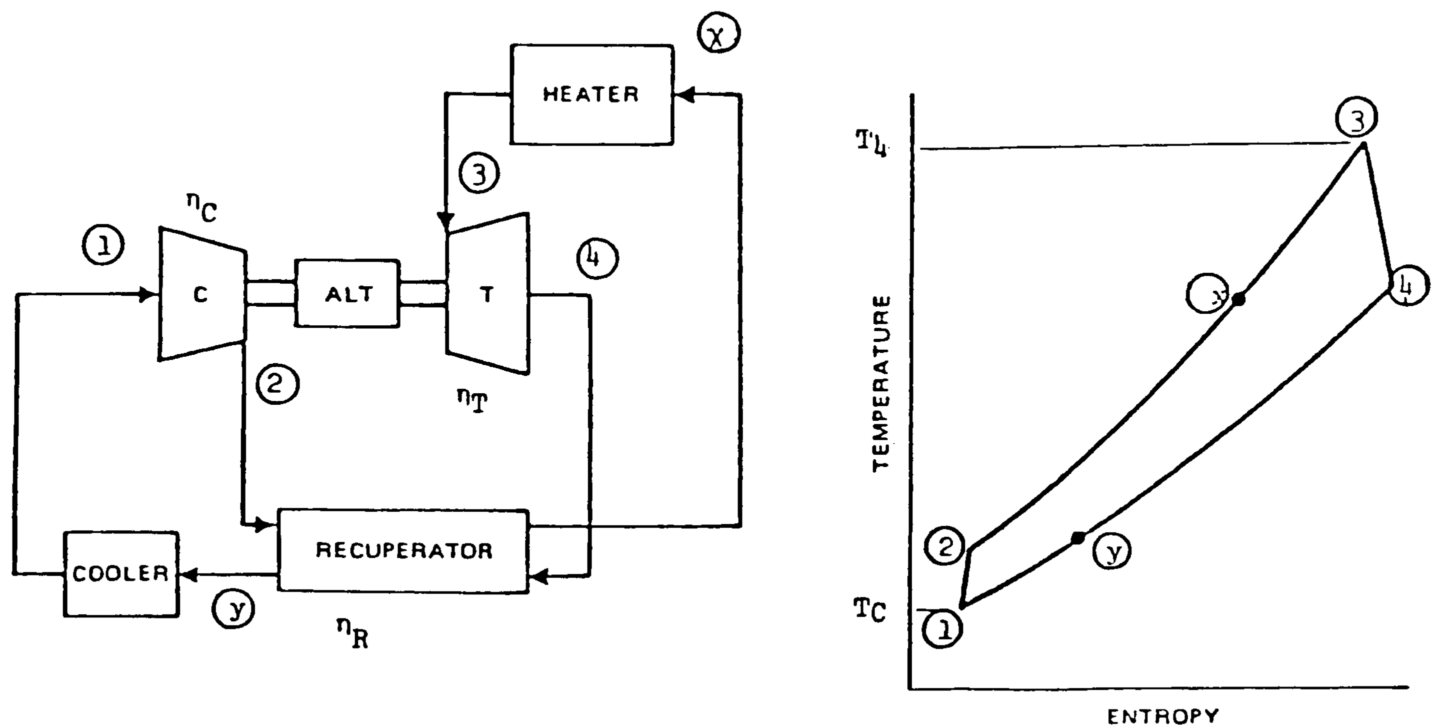


FIG. 4.5.3.5

11/3/84

These temperatures are used to calculate the cycle efficiency:

$$\eta_{cycle} = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_x)}$$

Calculated Results. The calculated efficiency of a Brayton cycle, using the methods described above, is shown in Fig 4.5.3.6. Both the device efficiency and the overall cycle efficiency for a Brayton system increase rapidly with the peak cycle temperature. However, if the lowest temperature in the cycle is raised from 300 K to 900 K, there is a drastic decrease in the system efficiency. This is a significant disadvantage for a high-power space system, because maximum power in a reasonable boost vehicle demands compact radiators using high temperature heat rejection. The Brayton cycle is relatively sensitive to the heat rejection temperature increase compared to other systems.

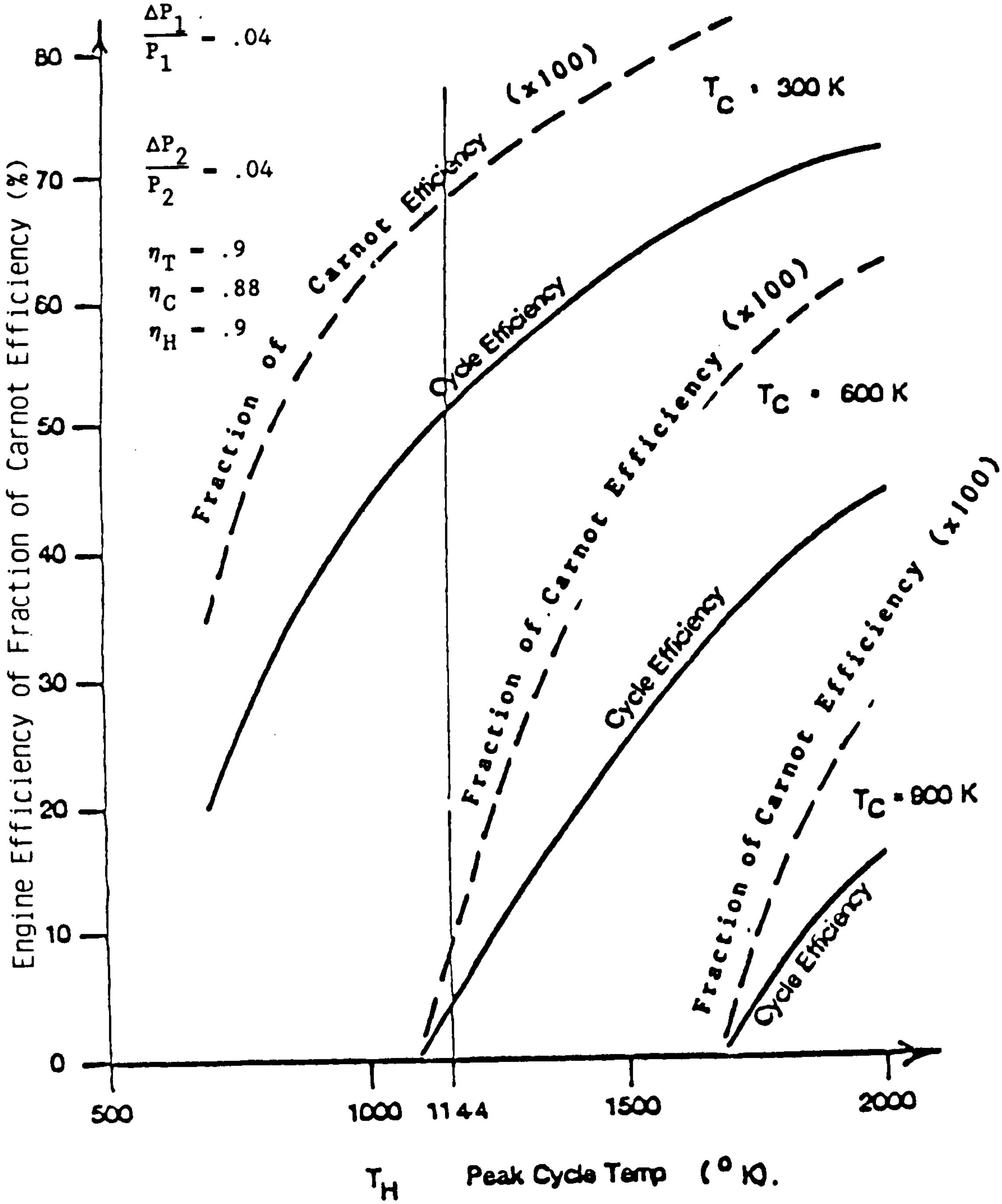
The curves in Fig. 4.5.3.6 are based on an example case, which was calculated using the estimated values of compressor, turbine, and regenerator efficiencies as shown on the Figure. The system efficiency is sensitive to these parameters as well as the pressure losses. The performance analysis of any actual system would require values of component efficiencies and pressure losses appropriate to the particular case.

Probable Operating Regime. A preliminary analysis of a thermal conversion cycle, solving for electric power generated (P_e) divided by the radiator area required (A_R) is found using

$$\frac{P_e}{A_R} = \frac{\eta \epsilon \sigma T_R^4}{1-\eta} \quad (kWe/m^2)$$

where: η = cycle efficiency; ϵ = emissivity; σ = Stephan-Boltzmann constant; T_R = heat rejection temperature

The cycle efficiency used in the above equation is obtained from the previous mathematical relationships. The results of this calculation are



Example calculation of performance of a regenerated Brayton cycle vs. peak cycle temperature as a function of compressor inlet temperature.

Fig. 4.5.3.6

shown in Table 4.5.3.1, where the values of P_e/A_R are printed in a matrix of temperatures representing the turbine inlet temperature and the compressor inlet temperature. The indicated zones on this figure are expected operating regions for the Brayton cycle in a space power system.

The data in Table 4.5.3.1 were generated by using efficiencies calculated with the component performance and pressure loss parameters shown in Fig. 4.5.3.6. Turbine and compressor efficiencies tend to be higher for larger machines, because the flow losses and leakage around the blades is a smaller fraction of the total power. As a consequence, turbo-alternator efficiencies of .9 and compressor efficiencies of .86 are achievable with larger units in the megawatt class. High pressure operation will help reduce the pressure loss in the heat exchange portions of the cycle, however, maximum compressor outlet pressure was limited to ≈1000 psia in this study. The regenerator effectiveness depends on a trade-off between massively large heat transfer area and performance. Nevertheless, it will be necessary to ensure that the pressure drops are maintained small, and values of $\Delta P/P = 8\%$ were used in the primary loop.

A particular feature shown in Table 4.5.3.1 is that the Brayton cycle has a distinct maximum heat rejection temperature for a given hot side temperature. The loss in the fraction of Carnot efficiency as the heat rejection temperature is raised is responsible for producing the maximum value in P_e/A_R. This behavior for the Brayton cycle is in contrast to other systems, which tend to develop a relatively constant fraction of the Carnot efficiency. This indicates that high power Brayton Systems must have a relatively low temperature radiator with a large area; and hence, a deployable, very lightweight heat-rejection system may be necessary.

As shown in Fig. 4.5.3.4, using present day superalloys, turbine inlet temperature is limited to ≈ 1144 K (1600°F). As seen in Fig. 4.5.3.6 a peak cycle temperature of 1144 K limits the radiator temperature to no more than ≈600 K with a cycle optimum being reached somewhere below this. A high radiator temperature rapidly degrades cycle efficiency while low radiator temperatures increase the size of the radiator necessary to reject the waste heat. Maximizing the value of kWe generated per m^2 of necessary radiator

T_C (at Compressor Inlet) \ T_H	700	800	900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100
1400									0	0	0	0	0	0	0
1300								0	0	0	0	0	0	0	0
1200	P_e/A_R (kWe/m²)						0	0	0	0	0	0	0	0	0
1100						0	0	0	0	0	0	0	0	0	2
1000					0	0	0	0	0	0	0	0	1.1	3.7	6.2
900				0	0	0	0	0	0	0	.5	2.4	4.3	6.1	8
800			0	0	0	0	0	0	.2	1.5	2.8	4.1	5.4	6.8	8.1
700		0	0	0	0	0	0	.9	1.8	2.6	3.5	4.4	5.3	6.2	7
600	0	0	0	0	0	.5	1	1.6	2.1	2.7	3.2	3.8	4.4	4.9	5.5
500	0	0	0	.2	.5	.9	1.2	1.5	1.8	2.2	2.5	2.8	3.1	3.4	3.8
400	0	.1	.3	.4	.6	.8	.9	1.1	1.2	1.4	1.6	1.7	1.9	2.1	2.2
300	.1	.2	.2	.3	4	.4	.5	.6	.7	.7	.8	.9	.9	1	1.1

T_H (at Turbine Inlet)

Example calculated electric power output per unit radiator area (P/A; KWe/M^2) for a regenerated Brayton cycle. (Radiator temperature assumed equal to compressor inlet.)

TAPLE 4.5.3.1

(kWe/m²) will locate the point of minimum radiator area. At a turbine inlet temperature = 1144 K, the Brayton system optimizes at $T_c \approx$ 500 K, operates at a cycle efficiency of 20%, and would require a radiator area of ≈ 14,250 m² (≈3.4 acres) in order to reject the 40 MW of waste heat. It is doubtful that any of the several heat rejection systems studied has the capability to deliver this required area in one or even two shuttles with the accompanying power conversion hardware. From this preliminary exercise, it is concluded that turbine inlet temperatures that exceed the material limits of superalloys will be required.

The Garrett Corporation was subcontracted to perform a computerized parametric analysis of a 10 MWe Brayton space nuclear power system. Garrett used an in-house, proprietary computer code which incorporates their extensive knowledge and experience in the Brayton cycle. Fig. 4.5.3.7 shows the ranges of major variables used in this analysis. The types of output information obtained from this analysis are shown in Fig. 4.5.3.8. Using this information from Garrett, the equations described in the Mathmatical Description portion of this section were used to determine intermediate cycle temperatures.

System weights were computed by combining the output from the Garrett computer code with the reactor, radiator and other miscellaneous data assembled by SPI. Fig. 4.5.3.9 compares the system weights of 1500 K and 1800 K turbine inlet temperature designs with varying recuperator effectiveness.

It is interesting to examine the 1800 K results. Fig. 4.5.3.10 shows the component weight breakouts of the three cases analyzed at 1800 K. Essentially the system which is lightest is the one with the lowest combination of radiator and recuperator weights. In comparing the 95% recuperator case with the zero recuperator case, the radiator weights are nearly identical. Although the zero recuperator has more heat to reject, it does so at a higher average radiator temperature. With radiator weights approximately equal, the zero recuperator has a significant weight advantage, since it doesn't have the weight associated with a recuperator.

GARRET PARAMETRIC COMPUTER ANALYSIS

OPERATING CONDITIONS INVESTIGATED

TURBINE INLET TEMPERATURE (K)	1244, 1500, 1800
COMPRESSOR INLET TEMPERATURE (K)	333 - 694
RECUPERATOR EFFECTIVENESS	0, .80, .95
ROTATING SPEED (RPM)	10,000 - 45,000
PRESSURE RATIO	1.8 - 4.3
COMPRESSOR SPECIFIC SPEED	.08, .09, .10
PRESSURE LOSS RATIO	.92, .94

FIG. 4.5.3.7

11/7/84

GARRET PARAMETRIC COMPUTER ANALYSIS

<u>OUTPUT INFORMATION</u>

COMPRESSOR EXIT PRESSURE

COMPRESSOR PRESSURE RATIO

WEIGHT OF ROTATING UNITS

WEIGHT OF RECUPERATOR

VOLUME OF ROTATING UNITS

VOLUME OF RECUPERATOR

CYCLE EFFICIENCY

FIG. 4.5.3.8

10 MWe BRAYTON SYSTEM MASSES

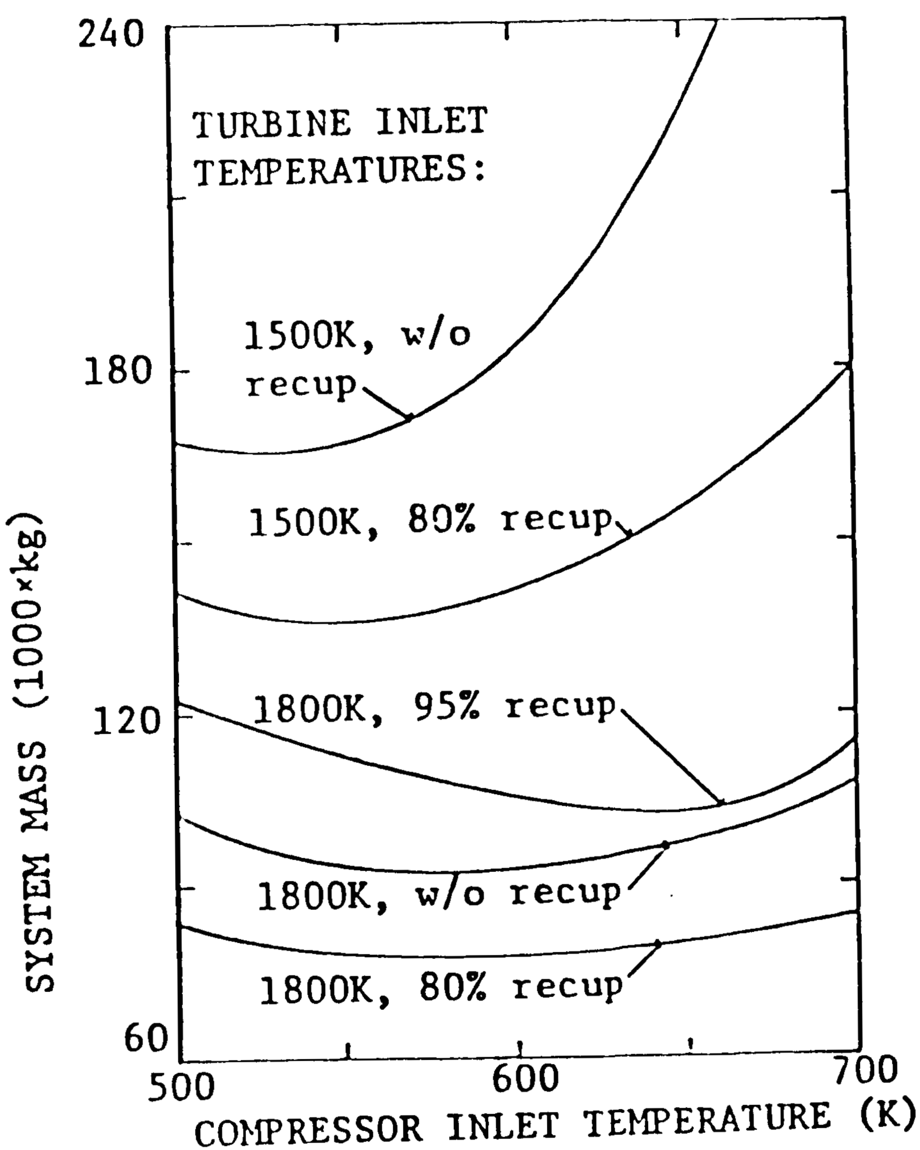


Fig. 4.5.3.9

BRAYTON CYCLE WEIGHT COMPARISON

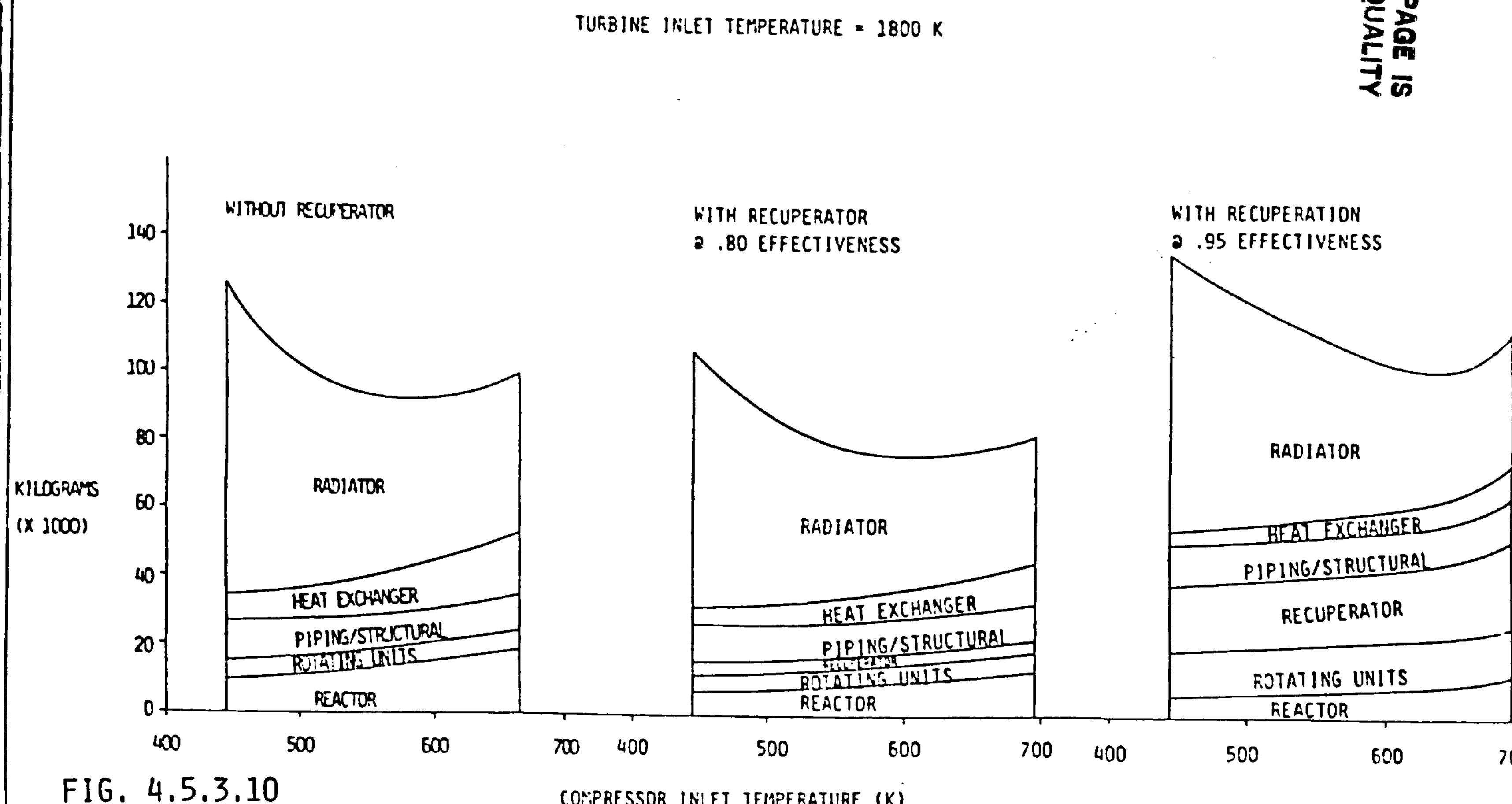


FIG. 4.5.3.10

11/8/84

The zero recuperator and 95% effective recuperator are at opposite ends of the spectrum. At the recommendation of Garrett, an 80% recuperator case was also analyzed. This case tends to bring out the advantages of both extremes and results in the lowest system weight. Because the recuperator is only 80% effective, it is much smaller than the 95% effective unit. It does, however, reduce the amount of heat which must be rejected and results in an average radiator temperature somewhere between the 0% and 95% effective recuperator cases.

Although not at the optimum compressor inlet temperature, Fig. 4.5.3.11 shows a comparison of cycle temperatures, cycle efficiency, reject heat and recuperator heat for the three degrees of recuperation; 0%; 80%; 95%.

Fig. 4.5.3.12 shows component weight breakouts for two cases at 1500 K turbine inlet temperature. Again the 80% recuperated case is lighter than the zero recuperator case.

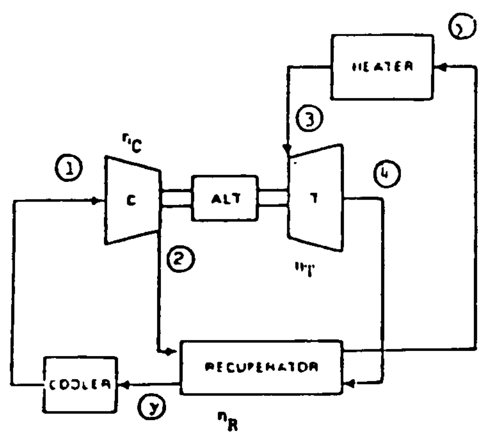
The number of shuttles necessary to launch the systems discussed is shown in Table 4.5.3.2 and 4.5.3.3 for 1800 K and 1500 K turbine inlet temperatures, respectively. As is shown, the optimum 1800 K system requires three shuttles while the 1500 K system requires 5 shuttles. At 1800 K turbine inlet temperature, 80% recuperation and a compressor inlet temperature of 556 K or 667 K, the entire power conversion system can be fit into one shuttle. This includes the reactor, turbo-compressor alternator, recuperator, piping, structure and miscellaneous. The heat exchanger and radiator require an additional two shuttles, not only from a weight standpoint but from a packageability standpoint. The power conversion system, packaged into the shuttle, is shown in Fig. 4.5.3.13.

Using the case of 667 K compressor inlet temperature, Table 4.5.3.2 shows a required radiator area of 2022 m^2. This area is calculated from the requirement to reject 38.0 MWt of heat over a temperature range of 1015 K-642 K (ΔT across heat exchanger = 25 K).

The wide range of the heat rejection temperature eliminates the possibility of using the telescoping type heat pipe radiator. Heat pipe working fluids

BRAYTON CYCLE

SYSTEM COMPARISONS AT 1800 K TURBINE INLET TEMPERATURE



	WITHOUT RECUPERATOR	WITH RECUPERATOR	
		ε = .80	ε = .95
T_3 (K)	1800	1800	1800
T_4	1151	1353	1465
T_y	1151	943	794
T_1	556	556	556
T_2	1049	840	759
T_x	1049	1251	1430
η (%)	20.8	29.5	35.5
Q_{recup} (MWt)	--	27.0	54.4
Q_{reject} (MWt)	40.8	25.6	19.5

FIG. 4.5.3.11

BRAYTON CYCLE WEIGHT COMPARISON

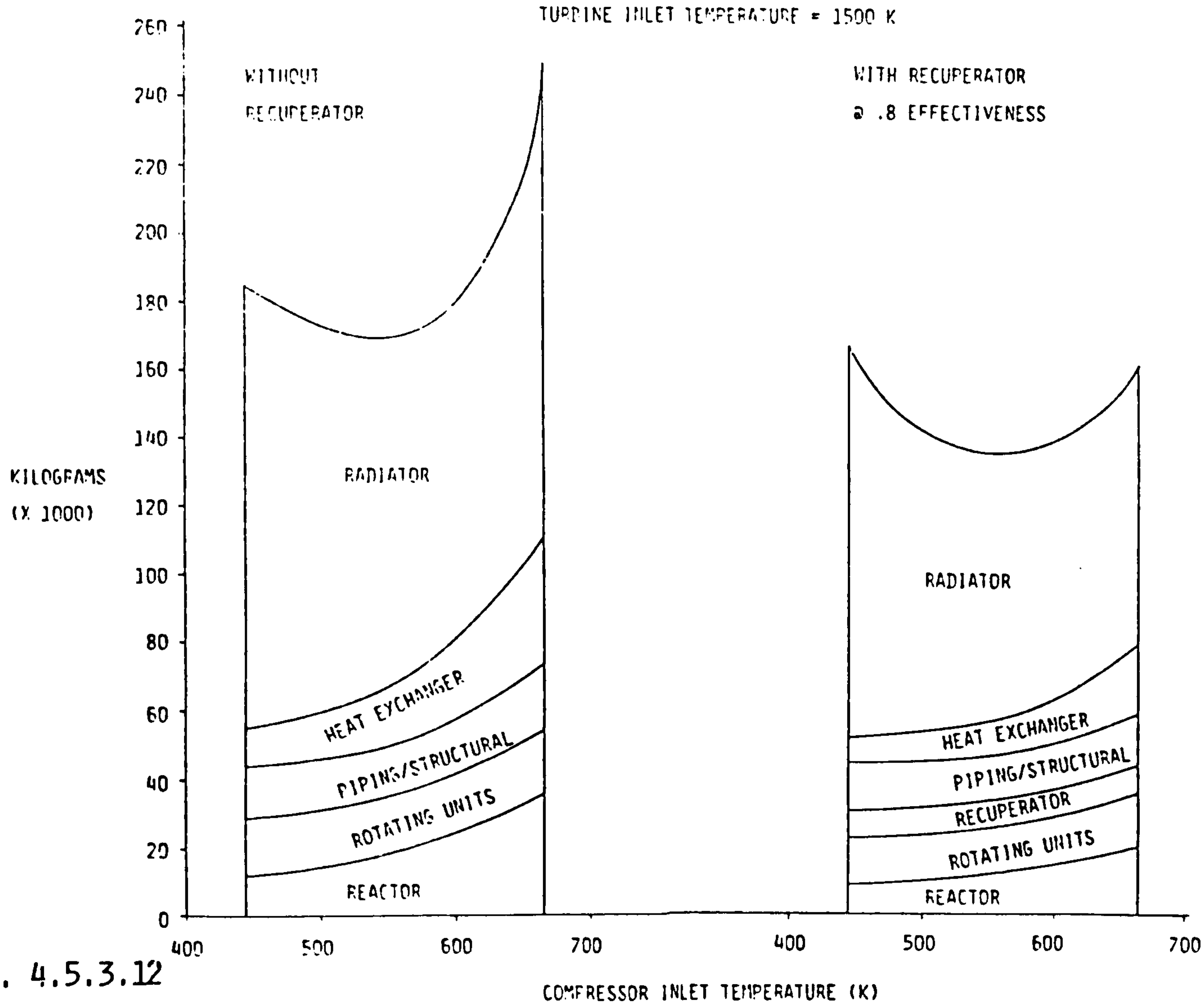


FIG. 4.5.3.12

11/7/84

BRAYTON CYCLE SHUTTLE DELIVERY

TURBINE INLET TEMPERATURE (K)	COMPRESSOR INLET TEMPERATURE (K)	RADIATOR AREA (m^2)	POWER MODULE MASS (kg)	RADIATOR+ HX MASS (kg)	SHUTTLE DELIVERY		
					POWER MOD.	RAD+HX	Σ
1800 @ 0% RECUP.	444	4943	26,376	99,722	1	4	5
	556	2826	28,338	65,150	1	3	4
	667	2403	35,253	65,361	2	3	5
1800 @ 80% RECUP.	444	4071	26,257	80,591	1	3	4
	556	2394	27,089	51,746	1	2	3
	667	2022	31,286	49,750	1+	2	3+
	694	1989	33,315	50,355	2	2	4
1800 @ 95% RECUP.	444	4429	50,611	85,903	2	3	5
	556	2917	52,807	59,596	2	2	4
	667	1979	58,800	45,055	2	2	4
	694	2192	64,757	51,585	3	2	5

TABLE 4.5.3.2

BRAYTON CYCLE SHUTTLE DELIVERY

TURBINE INLET TEMPERATURE (K)	COMPRESSOR INLET TEMPERATURE (K)	RADIATOR AREA (m^2)	POWER MODULE MASS (kg)	RADIATOR + HX MASS (kg)	SHUTTLE DELIVERY		
					POWER MOD.	RAD+HX	Σ
1500 @ 0% RECUP.	444	6988	44,229	140,293	2	5	7
	556	5334	50,052	118,552	2	4	6
	667	7064	73,440	172,964	3	6	9
1500 @ 60% RECUP.	444	6248	45,703	121,975	2	4	6
	556	4248	47,730	88,866	2	3	5
	667	4388	60,262	102,534	2+	4	6+

TABLE 4.5.3.3

10 MWe BRAYTON SYSTEM

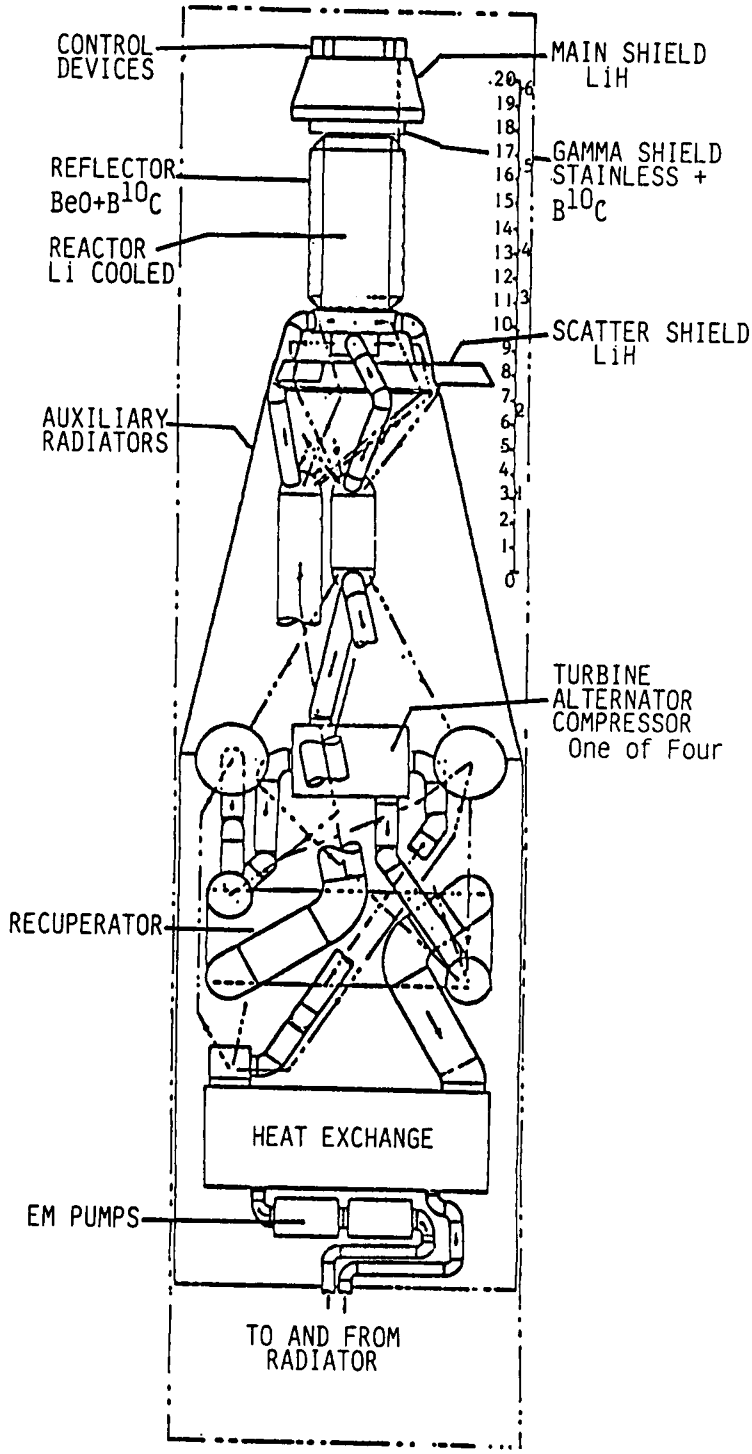


Fig. 4.5.3.13

used at the low end of the heat rejection temperature range do not have sufficient axial heat carrying capacity (kW/cm^2) to efficiently operate in the long heat pipe telescope-type of arrangement. The Brayton cycle matches quite well with the triform-type of heat pipe radiator, which utilizes several working fluids over a broad temperature range.

The system uses two coaxially-run pipes. The outer tube carries NaK out to the heat pipe evaporator sections and the inner pipe provides a return. As the NaK fluid temperature drops, due to sensible heat loss, the heat pipe working fluids and envelope are changed. Fig. 4.5.3.14 and Table 4.5.3.4 demonstrate these effects.

BRAYTON CYCLE DESIGN POINT

TRIFORM RADIATOR , 38 MWT,1015K–642K

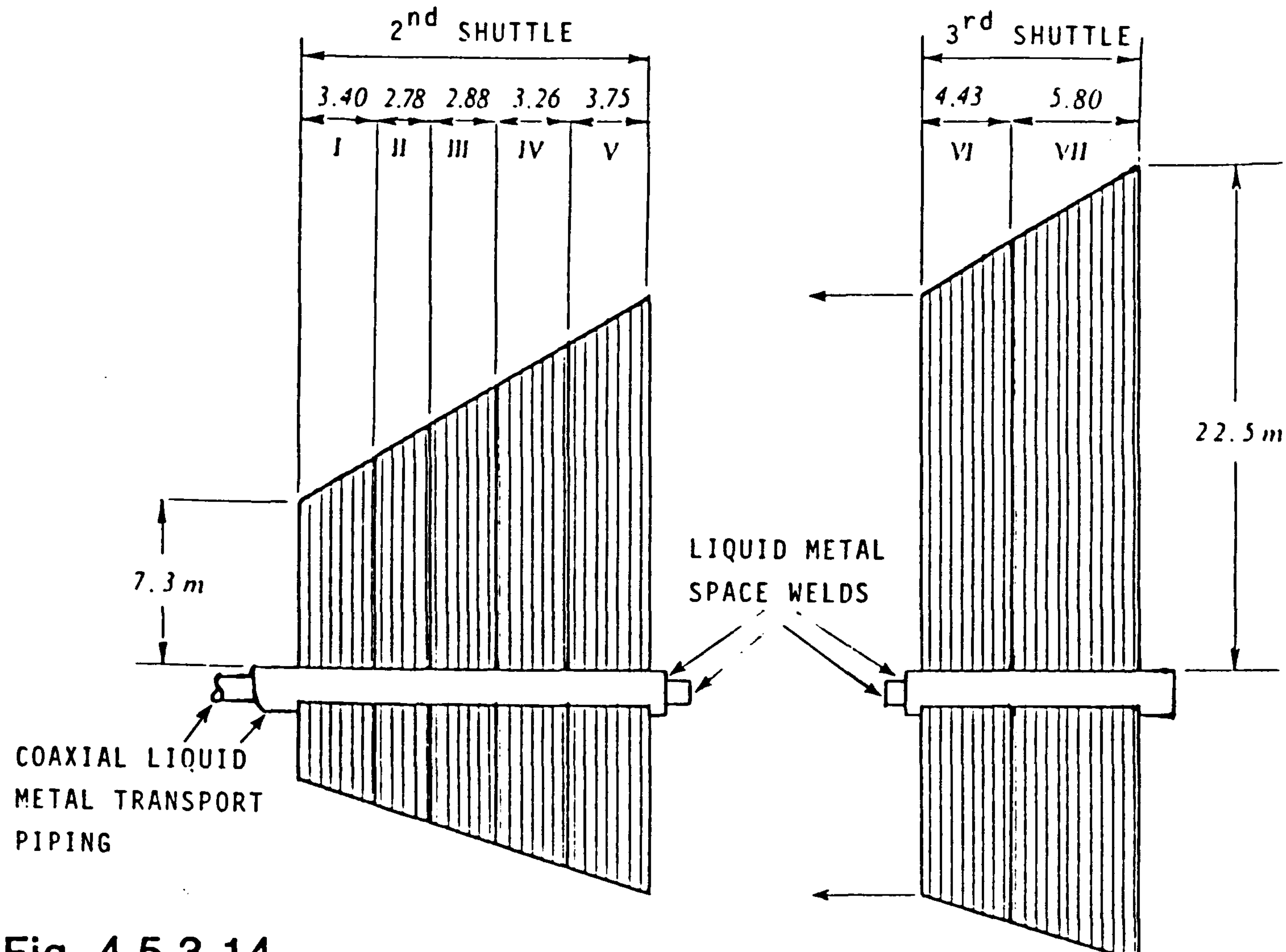


Fig. 4.5.3.14

SECTIONS ASTRONAUT-ASSEMBLED IN SPACE

TRIFORM RADIATOR BREAKDOWN

SECTION	$\bar{T}$	A	REJECT HEAT	WKG. FLUID/ PIPE MAT'L	SPECIFIC WT	WT
	(K)	(m^2)	(MWt)		(kg/m^2)	(kg)
I	983	147	6.63	K/Nb	23.8	3487
II	925	145	5.09	K/SS	21.1	3060
III	875	182	5.09	K/Ti	17.4	3163
IV	825	230	5.09	Cs/Ti	17.4	4008
V	775	295	5.09	Rb/Ti	17.4	5149
VI	725	389	5.09	Rb/Ti	17.4	6761
VII	671	634	5.92	Hg/Ti	17.4	11,033
		2022	38.0			36,661

+CONCENTRIC PIPE+NaK : 2,341

RADIATOR SYSTEM WEIGHT 38,975 kg

TABLE 4.5.3.4

11/7/84

4.5.4 Megawatt Class Stirling Gas Engines for Space

Introduction. Free Piston Stirling gas engines coupled to linear actuated alternators have an advantage for space application because they can be made to have a high efficiency, and designed to operate between any temperatures for which suitable construction materials can be found. However, very high power engines operating at the high temperatures required for megawatt class electric power systems have never been approached in Stirling engine technology.

In this section the range of suitable Stirling engine design is discussed and conceptual designs using first order design methods are presented to determine the approximate size of the machines and show the effect of temperature. A computer program, developed at Martini Energy Co. for design of free piston Stirling engines is used for these analyses. Based upon the first order designs, iterative methods, which have been specially modified to determine the approximate size and weight of the engine and the electric generator, are applied. Approximate optimization is done to determine the best design for each temperature level and frequency of operation, and critical endurance and reliability requirements are discussed. Finally, conclusions on this system, as it applies to the MCNSPS performance requirements, are given.

Stirling Engine Systems

General Requirements. As in the cases of the previous systems, the Stirling engine systems are in the 1 to 10 MWe range and are to have a 5 to 10 year life. This means that most common Stirling engine designs, which use mechanical seals, would not be suitable. Gas bearings, which also act as seals, would be suitable. Also, flexures which act as bearings and seals might be used at low temperatures. Because of the high neutron and gamma ray fluxes that are expected in the vicinity of the Stirling engine-generator, lubricants made from oils or plastics like Teflon are to be avoided. Also the electrical insulators must be ceramic for the purpose of radiation hardening as well as for high temperature operation. In order to fit within the power range requirement, the engine hot temperatures must be in the range from 1300 to 1600 K. These temperatures are dictated by the maximum practical temperature for the lithium cooled nuclear reactor with consideration for a reasonable temperature drop in the liquid metal pumped loop leading to the engine. It is also required that the heat sink temperature of the engine be in the 800 to 900 K range. This temperature is dictated by the radiator size and weight and a reasonable temperature drop in the heat pipes needed to transport the heat.

Long Life Engines. A Stirling engine generator system has been built and operated for over 8 years using a radioisotope heated electric power source [41,42]. Fig. 4.5.4.1 shows the concept [43]. Note that the power piston is a cold diaphragm which is attached to the armature of an oscillating electric generator. The displacer is supported by a spring (not shown). Because the entire machine is spring supported, the displacer picks up enough energy from the oscillation of the engine to keep the displacer oscillating properly. By paying proper attention to the fatigue limit, Mr. Cooke-Yarborough and his colleagues at Harwell have been able to attain at least 8 years life, with an efficiency much better than thermoelectrics. With their latest design they plan to produce 183 W of AC power from 1216 W of heat input at 819 K input temperature and 300 K sink temperature. This gives an efficiency of about 15%. This group has demonstrated the lifetime, but neither they nor anyone else has demonstrated the power output at the temperatures and power densities needed for megawatt scale Stirling engines [44]. A small, lower temperature, 3 kilowatt Stirling free piston engine built by Mechanical Technology Inc. (MTI) of Latham New York has been working at NASA-LERC for several thousand hours over the past year.

High Efficiency Engines. There are a number of possible US developers of high efficiency, free piston, Stirling engines with linear alternators that have a chance of also being light weight and long life. They are:

1. Sunpower Inc., Athens, Ohio
2. Mechanical Technology Inc., Latham, New York
3. Energy Research and Generation, Oakland, Calif
4. General Electric Company, Valley Forge, Penn.
5. Martini Engineering, Richland, Washington

LONG ENDURANCE ISOTOPE FUELED STIRLING GENERATOR

HARWELL, ENGLAND

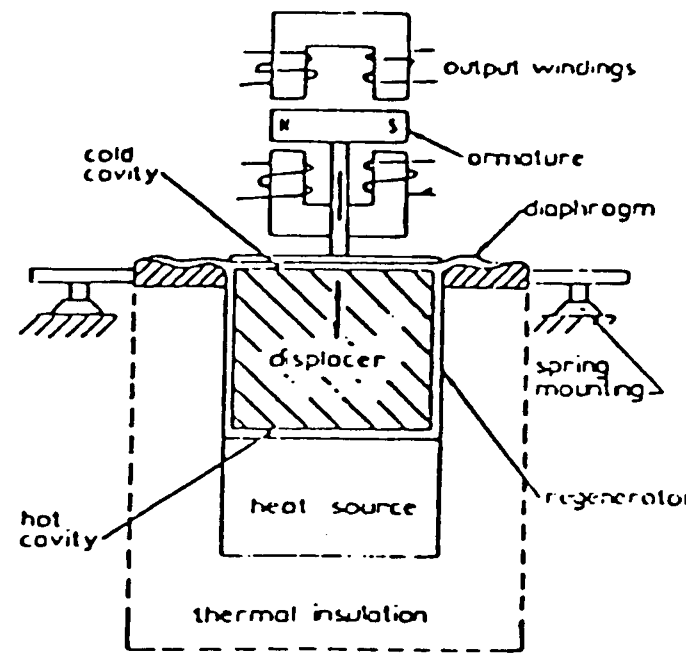


FIG. 4.5.4.1

In addition, the NASA-Lewis SP-100 office has assumed a management role in developing Stirling engines for space electric power. The in-house programs, as well as contracts with the above contractors and others are making active progress toward a goal of a 25 kWe size power source with a thermodynamic cycle efficiency of 70 percent of Carnot, temperature ratios on the order of 1.8 to 2.0, and a power conversion specific weight of 6 kg/kWe. The engine will use non-contacting gas bearings and a dynamically balanced system [45].

The basic SP-100 Stirling cycle program is predicated on a heat input temperature of 900 K, in order to utilize the LMFBR reactor technology. SP-100 would require five, 25 kWe engine-generators, each about a foot in diameter and 4 feet long. Initially it is planned to operate with a temperature ratio of 2, which translates to a heat rejection temperature of 450 K and a Carnot efficiency of 50%. An overall efficiency of 32% may be possible although 25% is expected. The latter is comprised of an efficiency of 67% of Carnot, a mechanical efficiency due to gas spring losses of 85%, and an alternator efficiency of 90%: (.50) (.67) (.85) (.90) x 100 =25.6%.

An SP-100 advanced program that would utilize a 1325 K Nb-1Zr reactor, Na or NAK cooled, is being considered. The engine temperature inputs would vary from 1200 to 1300 K. Gas operating pressures to 17 MPa (2500 psi) are being planned. Frequencies of 100 to 120 Hz are anticipated in order to achieve a maximum of 6-8 kg/kWe for the engine-alternator. The refractory metals Nb-1Zr, Cb-103 and FS-85 are being considered for cylinders. The engine temperature ratio is expected to be 1.8 to 2.0 for an average engine sink temperature of about 650 K and an average radiator temperature of about 600 K. A separate cooling loop may be required to cool the permanent magnet alternator to 500 to 550 K.

Sunpower. Although Sunpower Inc., of Athens, Ohio has not demonstrated very long life, they have built large, high efficiency engines that may be capable of attaining high efficiency and long life in a space environment. They have built and tested 1 kWe and 10 kWe machines. Their best 1 kWe machine, shown in Fig. 4.5.4.2, attained an efficiency of >32% heat to mechanical energy at >1250 watts output with helium at 7 MPa. The heat

SUNPOWER 1 KW FREE-PISTON STIRLING ENGINE GENERATOR

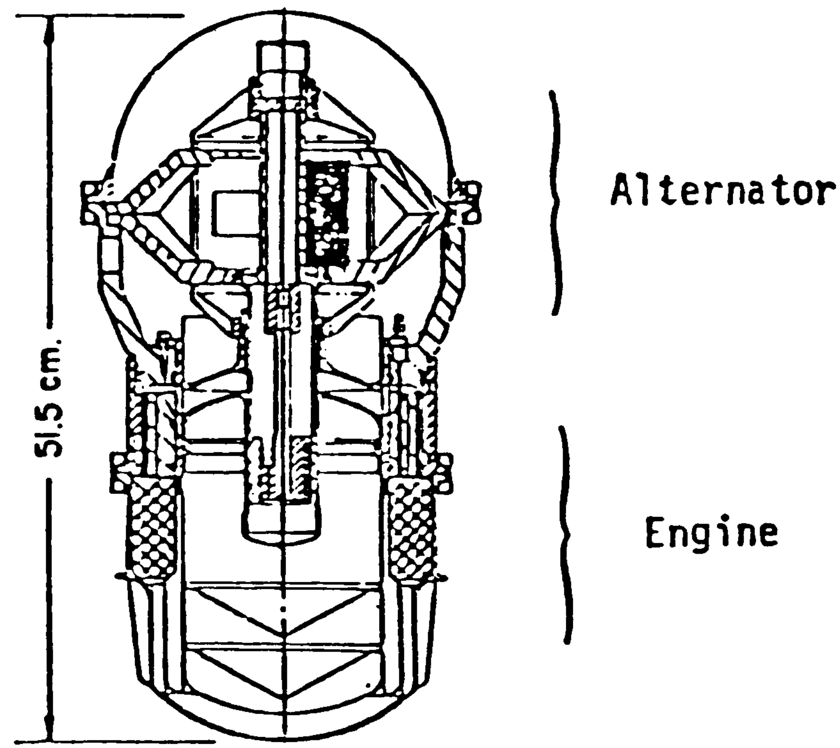


FIG. 4.5.4.2

source temperature was 923 K, and the heat sink temperature was 303 K. The testing was done independently at NASA-Lewis [46]. Sunpower has built a 20 kWe engine with a linear alternator load that has a projected efficiency of 44%. It operates at 50 Hz, employs 20 bars of helium pressure, and weighs 150 kg. It is designed to be solar heated [47]. Currently it is in Germany under test with a solar concentrator [48]. Sunpower is usually designing for low cost. However, for long life systems they plan to use rotating as well as oscillating parts to create a gas lubricated journal bearing and seal with no check valves needed. Sunpower is a contractor on the SP-100 program.

Sunpower has been working with free piston Stirling engine generators using their best technology to develop an efficient and light weight linear electric generator. The best that they have been able to do is 8 kg/kWe at 60 Hz with an efficiency of 90% at room temperature. The specific weight of the generator is approximately inversely proportional to frequency. Increasing the temperature above the normal 300 K range would greatly increase the system weight, particularly of the generator, if it is based upon a permanent magnet. They feel that 1 kg/kWe specific weight is simply not possible, even at room temperature [49].

Sunpower will be designing a 25 kWe engine generator for the SP-100 program managed by NASA-Lewis. The goal is a 930 K heat source temperature of 564 K and a radiator at nearly 550 K. A 6-8 kg/kWe specific weight and an overall efficiency of 25% is required [50].

<u>Mechanical Technology Inc. (MTI)</u>. This company has a program to develop a 3 kWe free piston Stirling engine-linear generator for the Army. They have 4 or 5 engines of this type under test on various programs. This Army power source is designed for 60 bar helium pressure and 60 Hz. It is expected to produce 3.1 kWe at 17.1% overall efficiency. The machine is heated by a diesel fuel burner. Fig. 4.5.4.3 shows the current concept.

MTI has a contract with NASA-Lewis on the SP-100 program to design and build a 25 kWe engine-generator. It will operate at 120 Hz and have an overall specific weight goal of 6-8 kg/kWe. The design heat source temperature is

MTI 3 KW FREE-PISTON STIRLING ENGINE GENERATOR

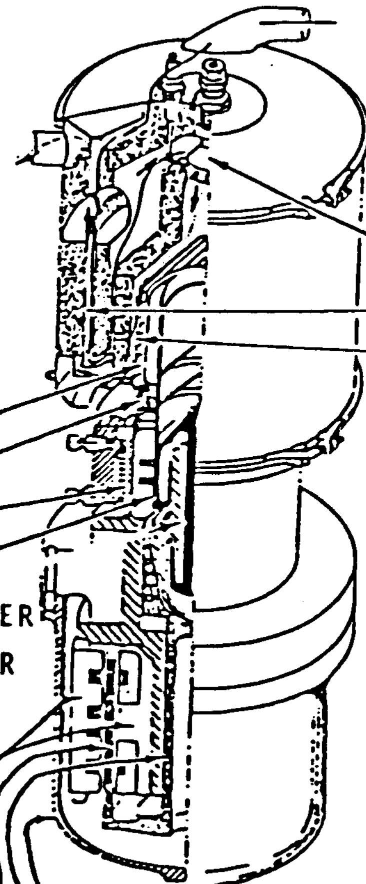


FIG. 4.5.4.3

1100 K from a liquid metal loop. The design heat sink temperature is 450 to 550 K in the form of a heat pipe. However, the first engine they will build will have a heat sink temperature of 120 °F (322 K) and a heat source temperature of 644 K. The practical limit for permanent magnet linear generators is considered to be 550 K. The barrier to going to higher temperature is the rapid loss in magnetism of all known permanent magnet materials [51].

Energy Research and Generation (ERG). Advanced high efficiency, low weight engine generator sets suitable for space electric power are being designed by ERG. To achieve full dynamic balance for their 25 kWe power source, ERG plans to use three in-line electrically driven displacers operating two opposed double acting power pistons that are integral with the permanent magnet plungers of the linear alternator [52]. Fig. 4.5.4.4 shows this concept. The central displacer has twice the mass of the two outside displacers. An electric motor drive moves the central displacer down while the two outside displacers are driven up. Therefore, there is balance in an up and down direction without any additional mechanism. During this part of the cycle there is a larger pressure between the power pistons than outside them. They are thus driven apart, balancing each other without any additional mechanism. On the return stroke complete balance is also maintained.

In connection with the electric generator, ERG has been attempting to produce a linear electric generator with a specific weight of 1 kg/kWe since 1973 [53]. ERG states that it is possible to do this by "eliminating back-iron and using a wrapped toroidal core and permanent magnets with field coil control." ERG now has a contract with NASA-Lewis to investigate the design, fabrication, testing, and demonstration of a light weight, high efficiency, compact linear alternator for free piston Stirling engine (FPSE) spacepower conversion systems. To quote the abstract [54]: "Based upon preliminary analysis, the linear alternator concept, when successfully developed and mated with a dynamically balance FPSE, offers a space-qualified electric power plant having potential specific weights of 1 kg/kWe and an overall bus-bar efficiency of 45% using 1500 K heat input by lithium heat pipes and

ERG STIRLING ENGINE GENERATOR CONCEPT

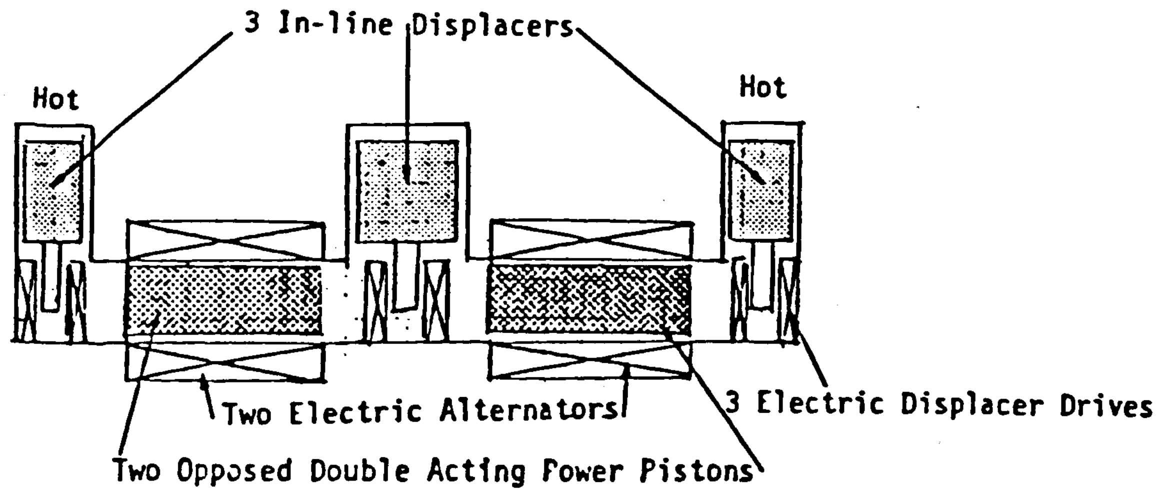


FIG. 4.5.4.4

750 K heat rejection by potassium heat pipe radiators. " The permanent magnet alternator must be separately cooled to 550 K.

The use of an electric drive on the displacers makes it possible to turn the engines on and off easily. This would be important in a space electric power source where spares could be installed and could be turned on or off, as needed by the controller. Both MTI and ERG have found that the use of an electrically driven displacer is the best way to control the engine, to get 90 degree phase angle for the most power, and to force a particular frequency if desired.

ERG claims that their engine will operate at 84% of the theoretical maximum Carnot efficiency. Others claim about 75%. However, the big difference is in the power density of the linear alternator. ERG claims 1 kg/kWe for a high temperature machine. Others claim 8 kg/kWe for a low temperature machine but are contracting with SP-100 for 6 kg/kWe high temperature machines. It appears that ERG has never been funded adequately to reduce their advanced ideas to deliverable hardware. However, component models and laboratory tests are encouraging. The ERG "thermizer" heat source and heat sink engine heat exchanger is novel, promising (at low operating pressure), and may permit fairly high frequency lower pressure operation.

ERG's proprietary novel linear alternator appears to be feasible, and is a technology advancement that must be achieved before free-piston Stirling engines can be seriously considered for multimegawatt application.

Martini Engineering. Martini has patented a new type of Stirling engine that uses a displacer with three layers. These layers act as heater, regenerator, and cooler. An advantage that would be important for space applications is the ability to operate at high frequency.

Fig. 4.5.4.5 shows the Martini concept as it would be applied to space electric power. A pair of these engine-generators would be positioned opposite the liquid metal pumped loop from the reactor.

MARTINI ENGINEERING 3 LAYER DISPLACER ENGINE GENERATOR CONCEPT

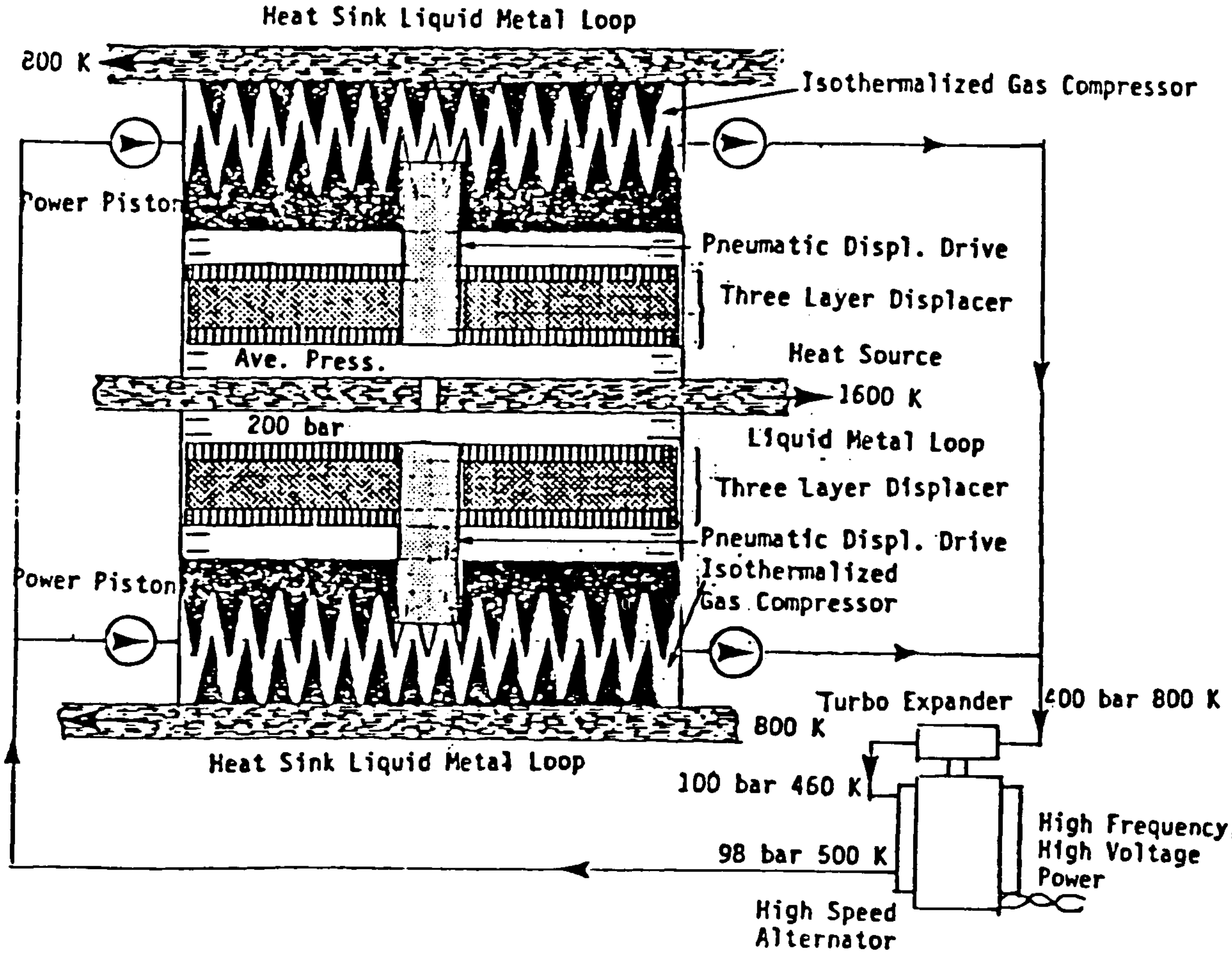


FIG. 4.5.4.5

At rest, the flexures, which are a series of leaf springs, support the three layer displacer at mid-stroke. At rest, leaf springs also support the power piston at mid-stroke. A floating labyrinth seal is used on both the displacer and power piston.

The power pistons compress gas. The gas compressor has interleaving fins in it so that the stationary side of the engine not only removes the heat of compression but also cools the power piston. The power piston gets the waste heat out of the engine by contacting the cooling layer of the displacer during part of each cycle. The cooling layer of the displacer is a perforated metal plate or a porous metal plate which collects the waste heat from the working gas of the engine as it passes through the plate.

A displacer drive piston passes through the power piston. This piston drives the displacer in the same way as is used in the usual free piston Stirling engine. However, in the usual free piston Stirling engine the displacer has to be carefully controlled so that it does not hit either end. In this concept the displacer must hit both ends to work properly. A snubbing action is built into the displacer so that it does not impact destructively.

During the time contact is made, the cooler layer is cooled by metal-to-metal conduction and by conduction across a very thin gas layer. During no contact, the cooler layer cools the gas that flows through it. This cooler has a much larger number of short but fine flow passages which transfer the heat with less flow loss than the usual cooler in a Stirling engine. The usual cooler is a shell and tube heat exchanger.

The heater layer must also spend some time each cycle close to, or touching, the hot plate. The heater layer is heated by metal-to-metal conduction and conduction through a very thin gas layer. The regenerator layer performs the usual function of storing heat as the gas moves back and forth and of insulating the hot space from the cold space. It has a much larger flow area than the usual regenerator. If needed, it can be convoluted for additional flow area and shorter flow path.

On the other side of the power piston is a gas compressor. The compressor is designed to have a large heat transfer area to volume ratio. The stationary fins in the compressor absorb the heat of compression as well as the waste heat from the engine. Two engines back-to-back will balance each other, even when they start out of balance. It was found by computer simulation that with the addition of equalizer tubes, connecting the working gas spaces and the gas pump spaces that are within one cycle, the two parts synchronize even when they are made to start off unsynchronized.

A pair of engine-pumps could operate a high speed hydraulic and a rotary electric generator. This rotary generator will be much smaller and lighter than a linear generator. A control in the turbine could be used for speed control. Another pair of engines operating another turbine and generator could be made to spin in the opposite direction to cancel out the gyroscopic effect. A bonus is that the expanding gas from the turbine would be cooler than the radiator. This gas would be used to keep the rotary generator cool for better operation at no increase in complexity. Also the turbine generator would be smaller and lighter than the linear alternator ever could be, because it moves continuously at a much higher speed.

It should be emphasized that there are many untried ideas in this concept. However, the basic concept of the engine acting as a pump to operate a common generator is a viable concept worthy of future evaluation. Because of time limitation, nothing more can be done with this untried concept in this report.

<u>Multimegawatt Conceptual Design</u>. In summary of the state-of-the-art, no Stirling engine presently exists that has the demonstrated life, power level, power density, temperature ratio, pressure, high temperature operation, or use of materials that would be required in a megawatt class engine system. In separate machines, the desired levels of lifetime and efficiency have been demonstrated as described above, but at low temperature, power level, power density, and at high temperature ratio. As a result, in order to evaluate the potential for Stirling engine use as a multimegawatt space power engine, an entirely new conceptual design was generated.

The Martini Engineering Company Stirling engine computer program developed for NASA-LERC and Argonne National Laboratories was utilized for this new design. The engine will be heated by liquid lithium from the reactor and cooled by NaK to a radiator. In order to achieve attractive radiator size and to be able to utilize effective potassium vapor heat pipe (800 K) radiant heat dissipation, engine cold end temperatures should be at least 900 K.

To achieve 70 to 75% of Carnot efficiency from a free piston Stirling engine alternator system, the ratio of hot to cold engine surface temperatures, (T_H/T_C), must be at least 1.8 to 2.0. Consequently, the reactor lithium must be 1600 to 1800 K. Due to an expected limit of about 1600 K for refractory metal systems containing lithium systems, and the high expected Stirling engine pressures of several hundred bars (1000's of psi), 1600 K was the highest engine temperature examined.

<u>Power Density</u>. To build a competitive multimegawatt class Stirling engine power source for space, the pressure and speed limits must be extended as far as possible, since power output is proportional to both pressure and speed. As pressure is increased, more heat must be transferred through the same area because there is more mass in the working fluid. As speed is increased, there is less time to accomplish the heat transfer. One must include the electric generator in the mass and power optimization, because the electric generator is by far the most massive part of the system. The mass of the electric generator goes down rapidly as speed increases.

<u>Engine Power Level</u>. The requirement is for 1 to 10 MWe and it would seem reasonable to accomplish this with from 4 to 40 cylinders, or about 250 kWe per cylinder. The only large Stirling engine study funded in recent years was done by three contractors and supervised by ANL [14]. This was for a 500 to 3000 hp coal fired, electric power source. These contractors, General Electric, Advanced Mechanical Technology Inc., and Foster-Miller Associates, picked between 65 and 105 kWe per cylinder. There is no increase in the size of the heater or cooler tubes, just their number. The flow area of the regenerator increases without increasing its length.

High Temperature Technology. High temperature operation for Stirling engines is quite new technology. United Stirling has been considering using ceramics to increase power density and efficiency and reduce costs since at least 1977 [55]. The Japanese have great hopes for a ceramic Stirling engine [56]. The Air Force is currently sponsoring work in this area [57]. Although all this work concerns high temperature, it is not applicable. The ceramics, like silicon carbide and silicon nitride, will take temperatures much higher than the 800 °C (1073 K) limit for air compatible metal alloys, but they are incompatible with alkali metals which will be used in the space power system. Also the high power density and the high efficiency result from having a low heat rejection temperature. The desired heat rejection temperature of 800 to 900 K for space power systems is only slightly less than the usual heat input temperature for a terrestial Stirling engine. Thus, the comparatively high efficiency and power density would not be realized.

Refractory metals must be used to be compatible with the alkali metal that adds and removes the heat, and to have enough strength to be useful. Fabrication technology development will be a major task of any development program. General Electric [58] is studying CVD-tungsten coated silicon carbide parts for an automotive Stirling engine design concept for NASA-LERC. Heat addition would be by means of lithium heat pipes. Testing of refractory metal engines must be done in a very good vacuum to simulate space conditions and to keep the refractory metals from oxidizing [23].

Reference Design Concept. An engine concept somewhat similar (an extrapolation) to the SP-100 FPSE concepts was chosen. Preliminary analysis indicated that utilization of known refractory metal creep properties led to very thick and heavy cylinder walls, creating an excessive thermal short circuit from the hot to the cold end of the engine. As just mentioned, ceramic engines could not be used because of their long term incompatibility with liquid lithium. As an alternative possibility, graphite or silicon carbide fiber is assumed wound around a tungsten, tantalum or molybdenum thin-walled cylinder, and CVD-tungsten, tantalum or molybdenum is assumed deposited between layers of fiber wrap. Some 2400 small, u-shaped tungsten or molybdenum tubes would have to be installed through the fiber wound layers of CVD-W to provide for lithium heat transfer to the helium working fluid. Similar u-tubes are required for heat rejection to a deployable NaK loop radiator, about 1/2 acre in size. The economizer was incorporated into the displacer piston. See Fig. 4.5.4.6 for a cylinder layout.

Table 4.5.4.1 presents a typical computer input-output from the design study. From this table note that:

Efficiency. The overall efficiency is the gross electric power divided by the total heat to the engine. It does not include insulation losses outside the engine or system pumping, transmission or control losses.

Temperatures. The assumed input temperature is 1600 K on the ID of the heater tube. Based upon the heat flux through the heater tubes, the temperature differential across the heater tubes is calculated to be only 1.84 K. To this must be added the film drop in the liquid metal. On the helium side, a gas temperature of 1569 K is assumed to be constant throughout the cycle. This temperature is determined by iteration until the heat that can be transferred is equal to the heat that must be transferred. The same considerations apply on the cold side.

Weights. Weights are determined directly from the volume of the metal and its density. The hot parts are calculated on the basis of Astar 811c at a density of 16.8 g/cc. The alternator is assumed to have a lumped density of 7.0 g/cc. The specific weight of the alternator is assumed to be inversely proportional to frequency. The assumed 8 kg/kWe at 60 Hz and 300 K is known to be about the state-of-the-art based upon a survey of those who have designed and built them. However, if the engine can be made to run at 240 Hz, the specific weight of the alternator would be only 2 kg/kWe.

Lengths. The overall length of the Stirling engine system, consisting of one engine, alternator and bounce space (dominated by the alternator length) should be somewhat less than 2 meters. Two of these systems must be installed in line and not exceed the 4 meter width of the shuttle compartment. To a first approximation, the length and the diameter of the engine system are related to the working gas volume of the engine, which in

100 kWe STIRLING ENGINE MODULE

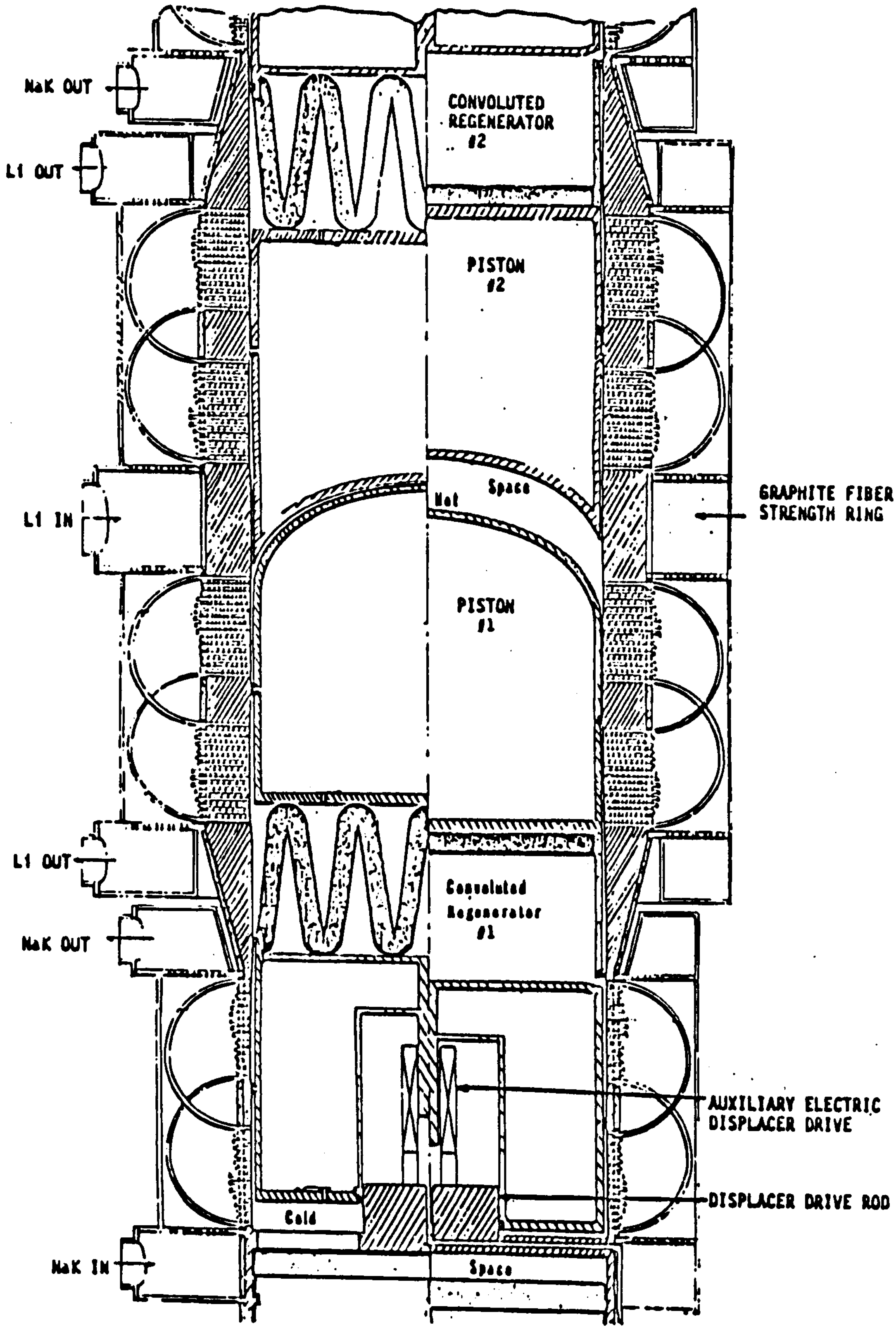


FIG. 4.5.4.6

MARTINI STIRLING ENGINE CODE

INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.	10
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24
Engine operating conditions:	
Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1600.00
Metal temperature of gas cooler, K	900.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.60
Number of radiation shields in displacer,	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo.P.Pist.	93492.31	Thermodynamic	349508.00
Thermo.Displ.	50469.00	Adiabatic Corr.	15287.58
Adiabatic Corr.	-25819.71	Reheat loss	53506.61
Heater flow loss	15610.66	Shuttle loss	44.07
Regen.flow loss	21071.83	Appendix loss	23008.89
Cooler flow loss	13764.10	Temp.swing loss	182.10
Dsp.Dr.Pwr.Reqmt.	-16.01	Cyl. Wall Cond.	170.49
Net Engine Power	67672.60	Displcr Wall Cond.	4687.45
Alternator Loss	6767.26	Regen. Wall Cond.	4862.90
NET ELECT.POWER	60921.35	Cyl. Gas Cond.	1.97
OVERALL EFFICIENCY, %	14.12	Regen. Mtx. Cond.	6282.33
Temperatures, K		Rad.Inside Displ.	.00
OD Gas Heater	1601.84	Flow Fric. Credit	-26154.58
Effect.Hot Gas	1568.73	Total Heat to Eng.	431388.60
Effect.Cold Gas	922.57	Displ.Dr. Heat	-16.01
OD Gas Cooler	699.73	Engine Cooling	363700.00
Weights, kg		Alternator cooling	6767.26
Hot Cylinder	39.22	Lengths, cm	
Heater tubes	15.36	Engine	32.65
Reg. Wall	1.55	Alternator	55.39
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	94.79
Displacer	35.39	Diameters, cm	
Displ.Drive Rod	.77	Eng.Cyl.OD	25.02
Displ.D.R.Support	2.67	OD Ann.Regen.	30.30
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	18.09	Hot cylinder	2.51
Alternator	121.81	Cold cylinder	.40
TOTAL WEIGHT	242.47	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.05

TABLE 4.5.4.1

turn is proportional to the kilowatts generated. Therefore, the engine system is shortened by making it fatter, while maintaining the same working gas volume, and hence the power and efficiency will be, to first order, unchanged. To scale the power to the desired range, see Appendix A.

Diameters. The largest diameter of the engine system is controlled by the outside diameter of the heat exchanger arrays. The engine cylinder diameter of the alternator should be the same as the engine, so that a simple pressure vessel can enclose both, although this is not essential. The alternator could be much shorter and fatter as long as the volume is the same.

Results of Calculation. The Martini Stirling Engine computer design program was installed, operated to verify validity of the calculations, and used to provide a first order design of a system appropriate to meet MCNSPS requirements. Appendix A contains the complete computer printout of all the final results. By iteration, an acceptable engine design of about the right size and power was devised, if fairly low radiator temperatures are used. Because of the large number of heater and cooler tubes and because of the convoluted regenerator, a high frequency engine of 240 Hz provides good efficiency at an attractive specific weight. Table 4.5.4.2 summarizes the major results. All engine designs are for 400 bar (~6000 psi) pressure, a 20 cm diameter cylinder, and a 1.5 cm displacer and power piston stroke. In all cases the displacer drive piston diameter was adjusted so that the electric power requirement to drive the displacer was negligible.

Critical Endurance and Reliability Issues. In selecting and performing the conceptual designs described in this section, only those concepts with the prospect for inherently good endurance and reliability characteristics were

SUMMARY OF RESULTS
MEGAWATT CLASS STIRLING ENGINES FOR SPACE

Heat Source Temp.K	Heat Sink Temp.K	Overall Length cm	Specific Weight kg/kW(e)	Net Power kW(e)	Overall Effic. %	Run # Apdx.B
1600	900	95	4.0	61	14	10
1600	800	122	3.3	91	19	9
1600	700	153	3.0	125	24	8
1600	600	188	2.8	163	28	6
1600	500	227	2.6	207	31	4
1500	900	78	4.4	42	10	13
1500	800	106	3.4	73	16	15
1500	700	137	3.0	108	22	16
1500	600	172	2.7	146	26	17
1500	500	212	2.6	190	29	18
1400	900	60	5.6	23	6	26
1400	800	89	3.6	55	13	25
1400	700	120	3.0	89	19	24
1400	600	155	2.7	127	24	23
1400	500	195	2.5	171	27	22
1300	900	41	41.2	2	0	29
1300	800	70	4.1	33	8	30
1300	700	101	3.1	68	15	31
1300	600	137	2.7	107	21	32
1300	500	177	2.6	151	25	33

TABLE 4.5.4.2

chosen. These designs include several different types of free-piston Stirling engines coupled to linear generators. By choosing this type over kinematic engines and smaller rotary generators, one obtains the following advantages:

1. No seal problem. Hermetically sealed. Internal seals are not critical.
2. Low bearing loads. Gas bearings are adequate and radiation sensitive materials are avoided.
3. Small parts count. With fewer parts than with a kinematic engine, the reliability should be higher.

There remain significant endurance and reliability issues. Designing and building a high performance engine out of ceramic fiber reinforced refractory metals will present some new and unusual problems:

1. The endurance limit of the refractory alloys must be reliably known and allowed for.
2. Thermal stress in the metals must be evaluated and properly taken into account because of the high heat flux.
3. High vacuum, high temperature joining techniques must be used throughout. Helium is the only practical working gas and it permeates elastomers which might survive the radiation environment.
4. Gas bearing technology must be checked. Most gas bearings rotate. These oscillate and may rotate as well for some concepts.

Stirling Space Power Systems. The Stirling engine will be heated by lithium metal pumped through the cylinder head heat exchanger. The lithium is assumed to be heated to a maximum temperature of 1570 K in a uranium nitride fueled fast reactor. In order to achieve reasonable primary loop pumping power, the reactor coolant temperature rises 100°C across the core. The coolant flows to approximately 56 double (or 112 single end to end) cylinder engines. The approximately 220 liters/sec (3500 gpm) flow might be distributed to the engines in parallel or in series. Reasonable size, velocities and design require that the flow be largely in parallel. Thus, each engine pair receives at least 4 liters/sec (all parallel). Such an arrangement would require a 100°C temperature gradient around the cylinder head. This excessive gradient should be reduced to prevent cylinder warpage and misalignment. Thus 2 to 4 engine pairs may be plumbed in series to form

14 to 28 parallel circuits. Engine thermal distortion would be reduced with only 25 to 50°C temperature gradients per cylinder.

The Stirling cycle work by W. Martini was previously typified in Table 4.5.4.1 and summarized in Table 4.5.4.2 and the 20 cm base Stirling engine module concept was shown in Fig. 4.5.4.6. The high-frequency (240 Hz), very-high-pressure, 400 bar (≈6000 psi), low-pressure-ratio design achieves fair performance (≈15% efficiency) at temperature ratios greater than 1.75. Good efficiencies of 20% are projected for temperature ratios of 2. The high frequency and high power density results in about 3 to 4 kg/kWe engine-alternator specific weights at 2.0 to 1.75 temperature ratios, respectively. Lower specific weights and higher frequencies at higher temperature ratios are of little interest for megawatt space power. Real systems have many losses that are not seen in ideal studies. Table 4.5.4.3 provides a summary of preliminary total system estimated weight per net electrical kilowatt output for the Martini Stirling engine data taken from Table 4.5.4.2. Note here that $T_{heat\text{-}source}$ of Table 4.5.4.2 is equal to $(T_{ho} + T_{hi})/2$ of Table 4.5.4.3. Likewise $T_{heat\text{-}sink}$ of Table 4.5.4.2 is equal to $(T_{co} + T_{ci})/2$ of Table 4.5.4.3. Also note that the $T_{radiator}$ average is equal to T_c-100. This provides for reactor and heat rejection loop temperature rises of 100 K from inlet to outlet.

The gross electric power output allowance was 10% greater than the net output in order to allow for pumping power, power conditioning, transmission losses and 5-year system degradation. The estimated kg/kW_{net} are listed for the radiator system, the engine-alternator system and the reactor-shield-pumping system. In order to produce 10 MWe net after 5 years, at least 11 to 12 MWe gross must be generated at BOL in order to provide power for the primary pump loop, the heat rejection pump loop, auxiliary-alternator cooling pumps, power conditioning, bus bar losses and some small number of engine failures or degradation. Consequently, the Martini engine-alternators alone, required for 10 megawatts EOL net output, could not be carried in one shuttle. This engine has enormous materials problems due to high temperature creep, cyclic stress and materials compatibility.

SPECIFIC WEIGHT OF STIRLING ENGINE SYSTEMS

	Th_o	Th_1	Tc_o	Tc_1	η_T	T_R	R_s (kWt/m^2)	P_N/m^2	kg/kWNet Rad	kg/kWNet E-A	kg/kWNet R-S-P	TOTAL
	1600	1500	900	800	.15	750	16	2.57	3.89	4.07	3.33	13.0
	1600		850	750	.18	.700	12	2.4	4.18	3.74	2.32	12.3
X	1600		800	700	.205	650	9	2.11	4.74	3.47	2.44	12.25
	1600		750	650	.228	600	6.7	1.8	5.56	3.33	2.19	12.74
	1600		700	600	.25	550	4.7	1.42	7.02	3.15	2.0	14.0
	1500	1400	900	800	.115	750	16	1.89	5.29	4.4	4.35	16.15
	1500		850	750	.15	700	12	1.92	5.19	3.96	3.33	14.35
	1500		800	700	.175	650	9	1.74	5.78	3.52	2.86	14.0
	1500		750	650	.205	600	6.7	1.57	6.37	3.33	2.44	13.96
	1500		700	600	.228	550	4.7	1.26	7.92	3.13	2.19	15.23
	1400	1300	900	800	.07	750	16	1.09	9.13	5.39	7.14	24.9
	1400		850	750	.11	700	12	1.35	7.42	4.51	4.55	18.95
	1400		800	700	.14	650	9	1.33	7.51	3.69	3.57	16.99
	1400		750	650	.173	600	6.7	1.27	7.85	3.41	2.89	16.27
	1400		700	600	.20	550	4.7	1.07	9.36	3.14	2.5	14.25

$$Q = \text{Reactor kWt} = \frac{1.1\ (\text{kWeN})}{\eta_T} = \frac{1.1}{\eta_T} \text{ per kWeNet}$$

$$\text{Radiator Area} = A_R = \frac{Q - 1.1P_N}{\text{kWt}/m^2} = \frac{1.1(1/\eta - 1)}{R_s} = \frac{m^2}{P_{Net}}$$

Rad = Radiator
E-A = Engine Alternator
R-S-P = Reactor-Shield-Pumps

TABLE 4.5.4.3

Fig. 4.5.4.7 shows a layout minus radiator for the 10 MWe net EOL output for 56 cylinder pairs and 112 displacer-alternators.

Speculation. The Benson (Energy Resource Group) engine concepts utilize more internally configured heat input and output exchangers. With lower pressures and higher pressure ratios, ERG expects good engine efficiencies at lower, more attractive temperature ratios of about 1.4 to 1.6. The lower pressures might reduce the materials creep problem to a manageable level. ERG powers the alternator bounce space with a smaller auxiliary displacer piston. Thus, ERG proposes a low-mass, linear alternator concept that might achieve engine-alternator weights of 1-1/2 to 3 kg/kWe. Such specific weights might be competitive with lower pressure potassium Rankine engine specific weights. However, the liquid-metal connected systems (i.e., reactor-shield-power conversion) at 10 MWe net will still not be lifted in one shuttle. Space assembly and liquid-metal-filled pipe welding, or a much larger booster, will be required.

Conclusion. High-temperature, high-power Stirling engines in these performance and specific weight ranges are highly speculative. No known refractory metals have low enough creep rates at temperatures from 1500 to 1600 K. Ceramic fiber or other ceramic reinforcing of refractory metals will be required, which represents an entirely new technology in itself.

10 MWe STIRLING SYSTEM

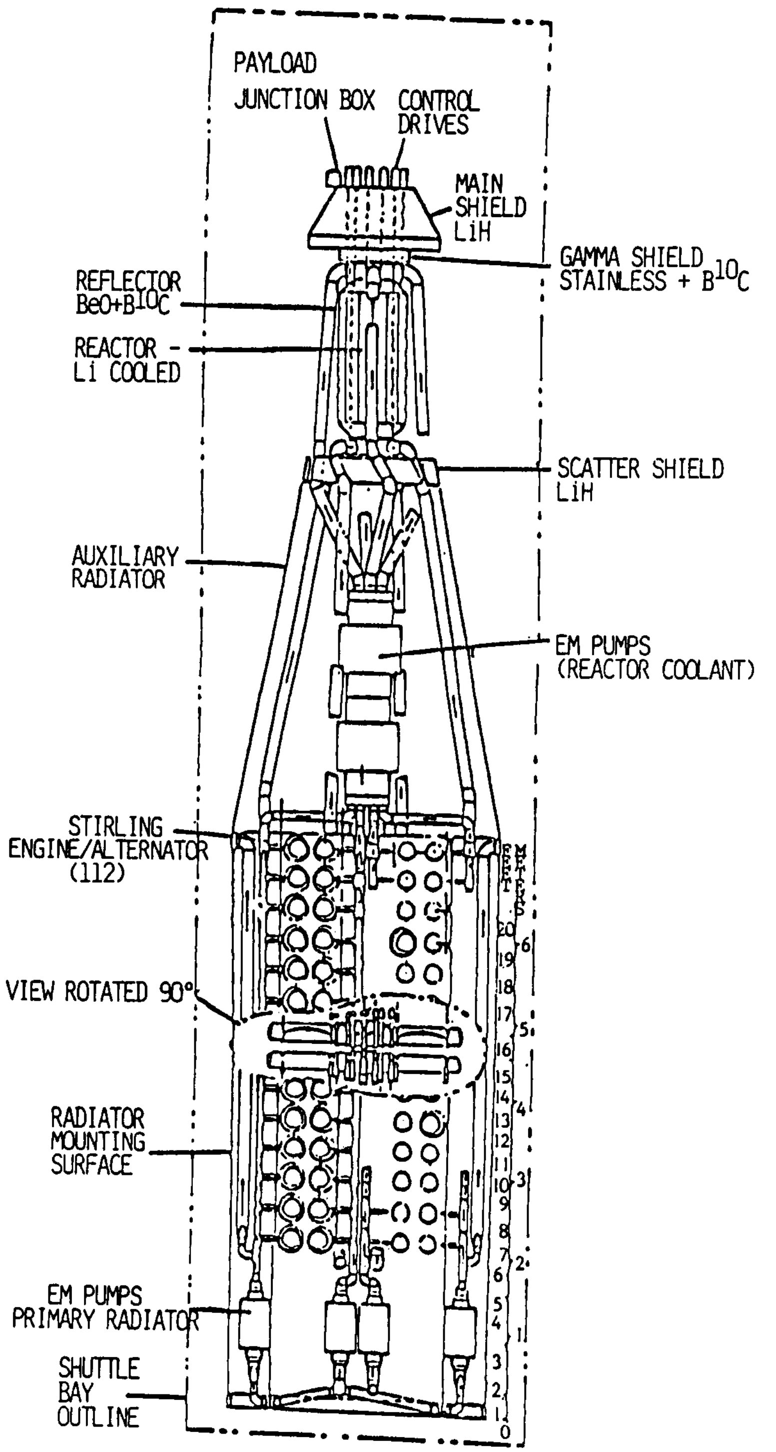


Fig. 4.5.4.7

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APPENDIX A (U)

Stirling Engine Design (U)

(U) First Order Design. First order designs are done using approximate formulas which tell the designer about how big and how efficient the Stirling engine would be. It says nothing about how the engine should be designed. There are a number of approximate formulas collectively termed the Beale equations [1]. The one that is most appropriate was proposed by J. Senft:

$$W = 0.035 \times F \times PM \times VC \times (TE - TC)/(TE + TC)$$

Where:
- W = Engine power output, watts
- F = Operating frequency, Hz
- PM = Mean cycle pressure, bar
- VC = Displacement of the power piston, cc
- TE = Heat source temperature, K
- TC = Heat sink temperature, K

(U) Assume a frequency of 60 Hz and a mean cycle pressure of 200 bar (2900 psia). Design for 250,000 watts (e). Assume a 90% efficiency electric generator. Thus the desired engine power would be 277,778 watts (m). Based upon these assumptions the displacement (VC) of the power piston would be as shown below:

Heat Sink Temperature	Heat Source Temperature 1300 K	1400 K	1500 K	1600 K
800 K	2778 cc	2425 cc	2173 cc	1984 cc
900 K	3638 cc	3042 cc	2646 cc	2363 cc

UNCLASSIFIED

UNCLASSIFIED

(U) The indicated efficiency of a practical Stirling engine is about 75% of the Carnot efficienty [2]. This efficiency depends very little on absolute temperature and will be used for approximate design. Assume again 90% efficiency for the electric generator. Thus the overall efficiency from heat to electric power is 67.5 % of the Carnot efficiency. The expected actual overall efficiency is given below:

Heat Sink Temperature	Heat Source Temperature 1300 K	1400 K	1500 K	1600 K
800 K	26%	29%	32%	34%
900 K	21%	24%	27%	30%

UNCLASSIFIED

(U) From the information we have obtained on the SP-100 program we can make some first order estimates about the size of the engine generator. The engine-generator is about half engine and half generator on a volume basis. On a weight basis it is 60% generator and 40% engine [3]. The SP-100 engine runs at 120 Hz so the specific weight of the generator is about 4 kg/kW(e). Assume that the density of the generator, being almost all solid metal, is 5 kg/liter. This would mean that a 250 kW(e) engine-generator would weight 1667 kg or 6.7 kg/kW(e). If the engine-generator were 5 times as long as it is in diameter, it would be 55 cm in diameter and 277 cm long. To get a more accurate idea of the engine design and the weight and size of the engine and generator, a second order analysis was undertaken.

(U) Second Order Design. We will first describe the tools that are available to do second order designs. Then we will develop a design like that being offered by both Sunpower and MTI (See Figs. 4.5.4.2 and 4.5.4.3) which is more or less standard. However we will add some refinements that are proprietary and could be patentable. We will find one design that seems to give good results and then see how it applies over the full range of requested temperatures.

(U) Linear Engine, Linear Generator Design. Figure A.1 shows the concept which is similar to both the Sunpower and the MTI design. The design will first be described. Then the independent inputs and the output display will be described using a sample output. The rest of the outputs are given in detail in Appendix A and abstracted in a table. Finally, design improvements will be discussed.

(U) Concept Description. The concept shown in Fig. A.1 is basically a free-displacer, free-piston engine. The displacer is pneumatically driven with optional assist and control by an electric drive. The power piston is also the armature of the electric generator. Heat is supplied to the tubular hair pin type heaters from the reactor by a pumped lithium loop. Each heater tube is the same shape and the same length. Heat is removed from the tubular coolers by another colder lithium loop. Because of the Curie point limitation to the magnetic materials in the generator, a separate coolant loop cools the linear generator to 500 K. An insulation layer on the power piston protects the electric generator from the "cold" end of the engine. Both the displacer and the power are sealed by gas bearings. In this concept hollow volumes in the displacer accumulate the maximum working gas pressure and apply it to a static type seal and gas bearing. In Fig. A.1 these are shown in concept and are not designed for load carrying capacity. The electric drive or generator is also made to rotate both the power piston and the displacer. This rotation creates a journal gas bearing which also acts as a seal.

(U) Two engine generators will be operated as a pair for balancing. They will be on the same center line with their hot ends joined. Fig. A.1 just shows one of these engines. This engine pair shares a common engine cylinder wall. If both are as shown in Fig. A.1, then there needs to be a cylinder in the middle with a spacer and no heat exchanger tubes. However, if one displacer has a convex cylinder head, as shown, and the other has a concave cylinder head, the two displacers would mesh. Important space and weight would be saved.

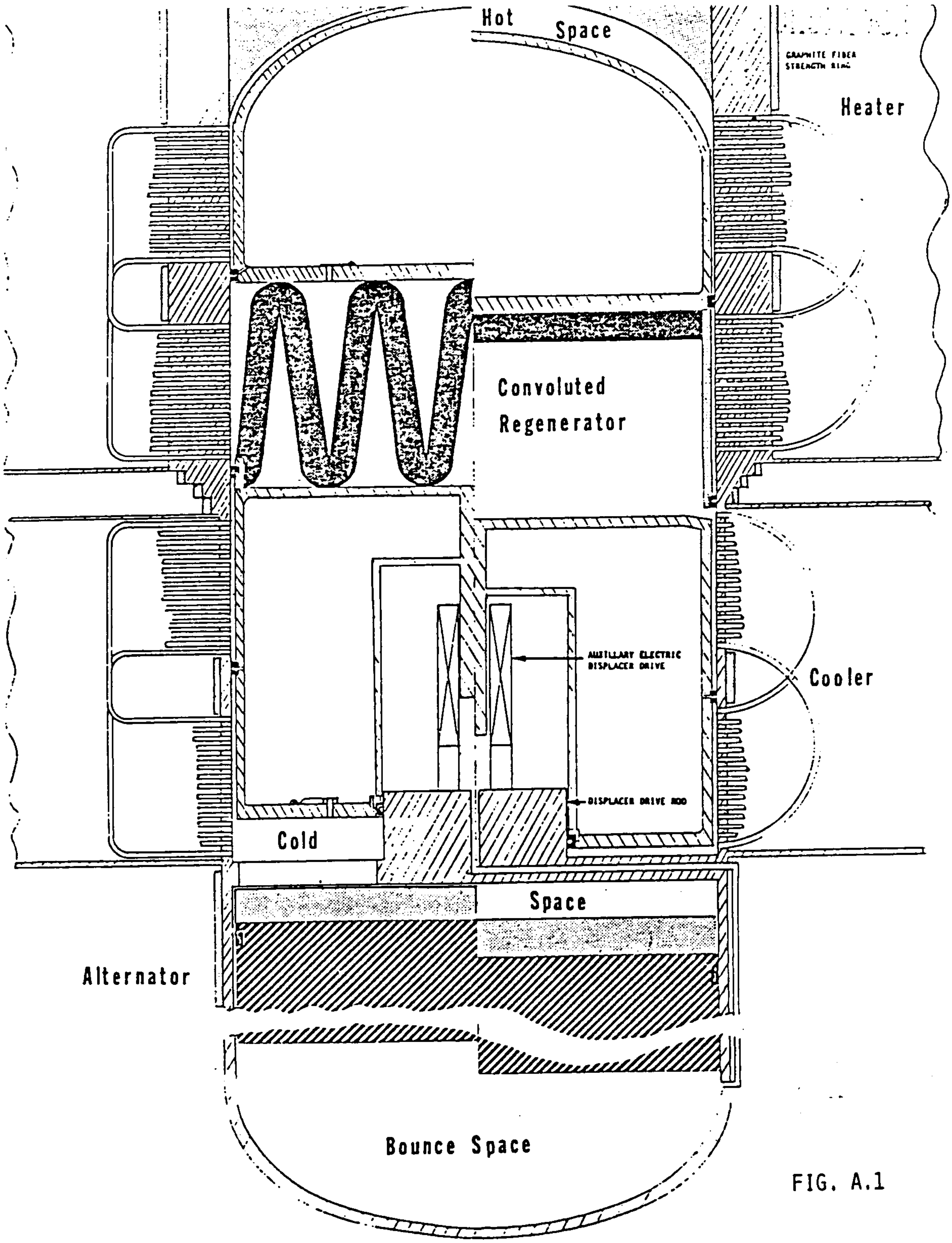
Hot Space
GRAPHITE FIBER STRENGTH RING
Heater
Convoluted Regenerator
AUXILIARY ELECTRIC DISPLACER DRIVE
Cooler
DISPLACER DRIVE ROD
Cold
Space
Alternator
Bounce Space

FIG. A.1

(U) Note that in Fig. A.1 the cross section shows the displacer and the power piston all the way up on the left side and all the way down on the right side. Also two different designs are shown for the hair pin heat exchangers. The squared off design at the left could not be built without some changes because the vertical part of each hair pin would interfere with neighboring tubes. The design on the right is the same as is used in the consortium Stirling engine being tested in England. The tubes, each in the form of a semicircle, can be placed so that they do not interfere with each other. These semicircular tubes would have less flow loss. The velocity heads would be 1.5 instead of 2.5 as is now used.

(U) In Fig. A.1 the engine cylinder wall is inconveniently thick. The wall thickness was calculated using the creep strength for Astar-811c. It is for 1% creep in 7 years. It was assumed that it would have a working strength of 500 MPa below 950 K. Above 950 K the following formula for the working strength was used.

$$WSTR = \exp\left[30.3741 - 3.5236 \ln (T)\right]$$

where WSTR = working strength, MPa

T = temperature, K

This wall thickness would be twice as thick as shown if it were not for the use of graphite fiber hoops which take the hoop stress. The metal only has to withstand the axial stress. The 2.51 cm thickness is for 1600 K. At 1300 K the wall thickness is down to 1.21 cm. Information obtained from the Great Lakes Carbon Co. indicated a tensile strength of 2758 MPa for graphite fiber. Apparently in the range being considered there is no dependence on temperature and there is negligible creep. This information must be checked. Possibly a better design would be to use tie rods outside the hot zone to hold the engine pair together and then add more and stronger strength bands using some type of cladding for the refractory fiber if necessary.

(U) Having the two hot ends of the engines in a common cylinder saves the weight of the end head. It also assures that both engines will operate in a counterbalanced mode. Computer simulation calculations at Martini Engineering showed that engines so connected would get in step within one cycle, even though they were intentionally started out of step. Of course the matching parts must have the same mass and the alternators musc be connected in parallel for the counter balance to work.

(U) The use of the hair pin heater tubes starting and ending in the cylinder wall is unique in Stirling engine design so far as is known. The reason for it is to minimize the engine diameter and make possible fiber wrap strengthening without having to deal with heads. To do this the regenerator had to be placed inside the displacer in a volume that would otherwise be empty. An additional improvement is the use of a convoluted regenerator which also is unique as far as is known. The regenerator works better to allow higher speed operation when the flow area is large, the flow path is small and the regenerator quite dense. A convoluted regenerator would be difficult to build so that one would have no appreciable leaks around the regenerator matrix. For a space power engine it would be well worth it.

(U) Because of the high reject temperature required in this engine design, it will be difficult and may be impossible to make a good electric displacer drive. In the engine designs described in this report we adjusted the diameter of the displacer drive piston so that the power applied to the displacer almost exactly equaled the flow loss through the heater, regenerator and cooler. Therefore, the electric displacer drive may not be needed. However there needs to be a more rigorous analysis of the pneumatic displacer drive. The timing of the forces applied during the cycle, the flow losses, and the inertia of the displacer determine the phase angle and the displacer stroke. They cannot be specified in advance. A more detailed analysis must be done to determine how much electric drive power is really needed to attain the desired 90 degree phase angle. Usually the phase angle is lower.

(U) Description of Needed Inputs. In calculating the performance of the RE-1000 free piston Stirling engine for NASA-Lewis, 94 numbers had to be input. Fifteen of these were for control of the calculation. The rest were dimensions and operating conditions. In this calculation certain dimensions will be set ahead of time and decisions made about how other dimensions relate to get as small a list as possible of truly independent inputs. The list given below is divided into these three parts:

(U) A. Set Inputs

1. Identity of metal of construction, best possible.
2. Working gas = helium because hydrogen would leak out.
3. Fraction of open area in spider supporting displacer drive piston = 0.5.
4. Phase angle = 90 degrees, best value.
5. Velocity head loss due to entrance, exit and bend in both heater and cooler = 2.5, right for design.
6. Density of alternator = 7 g/cc, estimate based upon iron and copper almost solid.
7. Emissivity of radiation shields = 0.2

(U) B. Independent Inputs

1. Charge pressure of working gas.
2. Inside wall temperature of gas heater tubes.
3. Inside wall temperature of gas cooler tubes.
4. Alternator temperature.
5. Engine and alternator frequency.
6. Maximum stroke of displacer or power piston.
7. Inside diameter of engine cylinder.
8. Gap between displacer and cylinder wall.
9. Displacer rod diameter.
10. Number of radiation shields in displacer (included but now no longer relevant).
11. Number of rows of heater and cooler tubes.
12. Radial length of heater tubes.
13. ID of heater tubes.

14. Square pitch between heater or cooler tubes.
15. Length of seal.
16. Diameter of wire in regenerator matrix
17. Porosity of regenerator.
18. Ratio of flow area to face area in regenerator.
19. Radial length of cooler tubes.
20. ID of cooler tubes.
21. Specific weight of alternator.
22. Efficiency of alternator.

(U) C. Derived Inputs

1. Working gas properties -- based upon helium
2. Metal properties.
3. Thickness of engine cylinder -- depends upon diameter, charge pressure and metal properties.
4. Thickness of heater tubes -- same as #3.
5. Thickness of cooler tubes -- same as #3
6. Thickness of outer regenerator wall -- average of hot and cold engine cylinder.
7. Thickness of inner regenerator wall -- currently set at 0.5 mm.
8. Thickness of displacer wall -- to take twice value of pressure swing in tension.
9. Thickness of cold pressure vessel heads -- currently same as wall thickness.
10. Total number of heater or cooler tubes -- depends upon engine diameter and number of rows of tubes.
11. OD of regenerator -- depends upon engine diameter and number of rows of tubes.
12. Working stroke of displacer or power piston -- 90 % of maximum stroke.
13. Length of displacer -- depends upon the length of the heater, regenerator, cooler, pressure, number of rows of tubes, and maximum stroke.
14. Volume of power piston bounce space -- 5 times the power piston stroke volume.

15. Volume of displacer bounce space -- 4 times the maximum displacement of displacer drive rod.

16. Thickness of spider -- 0.1 of engine diameter.

(U) Note that the 79 dimensions and operating conditions needed to describe the RE-1000 engine has been reduced to 22 truly independent dimensions and operating conditions for both the engine and the alternator. This has been done by ignoring secondary effects such as leakage, making judicious choices for things that cannot really be changed, and by relating some inputs to others.

(U) Method of Calculation. The isothermal analysis is used because it is fast and has been calibrated against published engine data [4] to within ± 10%. It has been extended to include calculation of weights and sizes. It has been made specific for the internal convoluted regenerator design.

(U) Descriptions of Outputs. Samples of the computer output are attached at the end of this appendix. Note that the independent inputs are given first and then the outputs are given. These outputs will now be discussed so that the reader will know what they represent. Note that the output is divided into power, heat requirement, efficiency, temperatures, weights, lengths, diameters, thicknesses, and numbers. Each of these subdivisions will now be discussed.

(U) Power. This isothermal second order analysis is built on the assumption that there is a basic thermodynamic power output and heat input that can be calculated by assuming that the gas spaces in the engine all have a known temperature for the cycle which does not change during the cycle. In this case the motion of the power piston and the displacer are both known in advance. Therefore, the pressure of the working gas at each point in the cycle can be calculated. This pressure applied to the area of the power piston as it moves to and fro creates the thermodynamic power piston power. The same pressure applied to the displacer drive piston as the displacer moves to and fro creates the thermodynamic power applied to the displacer which is used to overcome flow losses.

(U) The adiabatic correction makes this program the equivalent to a more complicated and time consuming program. The basic isothermal program assumes that the hot space and cold space of the engine are at fixed temperatures do not change during the cycle. In reality, these temperatures do change during the cycle. In reality, these temperatures do change during the cycle. These changes have an important effect on the true power output especially at low temperature ratios. We found that the difference between the true power and the isothermal power depends chiefly upon the temperature ratio and the dead volume ratio. This relationship has been precalculated and stored in a table. These two ratios are determined and then the adiabatic correction is determined quickly by interpolation in the table.

(U) The flow losses are determined by first approximating the flow through the different parts of the engine as a constant flow for part of the cycle, no flow for part of the cycle, the same constant flow back, and then no flow to complete the cycle. These constant flows and the fraction of the cycle time that they occur is determined for the heater and the cooler. For the regenerator, an average of these two is used. Standard flow loss and heat transfer coefficients are used.

(U) The displacer drive power requirement is the electrical watts that are needed to drive the displacer in addition to that supplied by the thermodynamic displacer power. We tried to make this small by adjusting the displacer drive rod diameter because it may be difficult to make a high temperature displacer driver.

(U) The net engine power is the thermodynamic power piston power corrected by the adiabatic correction.

(U) The alternator loss is the power loss due to copper and iron losses in the alternator.

(U) The net electric power is the power generated by the alternator less the electric power needed to operate the displacer. The efficiency of the displacer electric drive is assumed to be the same as the electric alternator. If for a particular case the displacer drive is calculated to produce power, this power is added to the power from the alternator.

(U) Heat Requirements. The thermodynamic heat requirement is calculated from the integral of the engine pressure and the hot gas volume.

(U) There is also an adiabatic correction for the heat input as well as the power output. It is calculated the same way.

(U) Because the regenerator is not perfect, additional heat must be added to the working gas to reheat it every time it comes back into the hot space. This is always a major loss.

(U) The shuttle loss comes about because two surfaces with an axial temperature gradient, like the displacer and the engine cylinder, move to and fro. Conduction back and forth across the gas gap causes additional heat loss.

(U) The appendix loss is caused by gas being pressured into the crack between the displacer and the cylinder wall, cooling foo and then coming back at lower pressure but colder into the hot space. The appendix loss can be decreased and the shuttle loss increased for a net gain up to a point by decreasing the gap between the displacer and the cylinder wall. However, this gap in this case, acts as a flow passage and should not be too small.

(U) The temperature swing loss accounts for the fact that the regenerator has limited heat capacity and therefore has some temperature swing as the gas moves back and forth. The regenerator therefore is not as good as it would be assuming unlimited heat capacity.

(U) All the conduction terms are calculated by simple straight conduction across the part of the engine that takes the temperature difference. This conduction is the same whether the engine is operating or not.

(U) The flow friction credit takes into account that the flow loss in the heater and half of the regenerator is converted back into heat and therefore reduces the heat that would otherwise be required from the heat source.

(U) The total heat supplied to the engine is the denominator for calculating efficiency. It is also assumed to be supplied to the engine through the heater tube wall. In some engine designs it might be argued that some heat loss terms would not go through the gas heater.

(U) The engine cooling must pass through the gas cooler at the heat sink temperature.

(U) The alternator cooling is the amount of heat that must be removed from the alternator at a lower temperature than the main radiator to a special low temperature radiator.

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INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	4
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1600.00
Metal temperature of gas cooler, K	500.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.20
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	237034.60	Thermodynamic	443621.90
Thermo. Displ.	57493.31	Adiabatic Corr.	23152.50
Adiabatic Corr.	-7825.66	Reheat loss	132593.80
Heater flow loss	24680.27	Shuttle loss	60.12
Regen. flow loss	19192.60	Appendix loss	90026.80
Cooler flow loss	13394.25	Temp. swing loss	743.64
Dsp. Dr. Pwr. Reqmt.	-251.33	Cyl. Wall Cond.	258.27
Net Engine Power	229208.90	Displcr Wall Cond.	3641.52
Alternator Loss	22920.89	Regen. Wall Cond.	7366.69
NET ELECT. POWER	206539.40	Cyl. Gas Cond.	2.64
OVERALL EFFICIENCY, %	30.56	Regen. Mtx. Cond.	8423.44
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1602.89	Flow Fric. Credit	-34276.57
Effect. Hot Gas	1556.25	Total Heat to Eng.	675814.60
Effect. Cold Gas	523.03	Displ. Dr. Heat	-251.33
OD Gas Cooler	499.64	Engine Cooling	446354.50
Weights, kg		Alternator cooling	22920.89
Hot Cylinder	39.22	Lengths, cm	
Heater tubes	15.36	Engine	32.65
Reg. Wall	1.55	Alternator	187.61
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	227.01
Displacer	19.14	Diameters, cm	
Displ. Drive Rod	.69	Eng. Cyl. OD	25.02
Displ. D. R. Support	2.67	OD Ann. Regen.	30.30
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	46.34	Hot cylinder	2.51
Alternator	412.58	Cold cylinder	.40
TOTAL WEIGHT	545.16	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.05

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	6
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1600.00
Metal temperature of gas cooler, K	600.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.30
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	193455.50	Thermodynamic	413326.70
Thermo. Displ.	55065.42	Adiabatic Corr.	20846.36
Adiabatic Corr.	-12286.09	Reheat loss	103961.90
Heater flow loss	21477.54	Shuttle loss	57.32
Regen. flow loss	19634.02	Appendix loss	62644.02
Cooler flow loss	13646.57	Temp. swing loss	500.39
Dsp. Dr. Pwr. Reqmt.	-341.43	Cyl. Wall Cond.	237.37
Net Engine Power	181169.40	Displcr Wall Cond.	4159.59
Alternator Loss	18116.94	Regen. Wall Cond.	6770.65
NET ELECT. POWER	163393.90	Cyl. Gas Cond.	2.52
OVERALL EFFICIENCY, %	27.73	Regen. Mtx. Cond.	8029.20
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1602.52	Flow Fric. Credit	-31294.55
Effect. Hot Gas	1560.53	Total Heat to Eng.	589241.50
Effect. Cold Gas	622.23	Displ. Dr. Heat	-341.43
OD Gas Cooler	599.68	Engine Cooling	407730.70
Weights, kg		Alternator cooling	18116.94
Hot Cylinder	39.22	Lengths, cm	
Heater tubes	15.36	Engine	32.65
Reg. Wall	1.55	Alternator	148.29
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	187.69
Displacer	22.55	Diameters, cm	
Displ. Drive Rod	.71	Eng. Cyl. OD	25.02
Displ. D. R. Support	2.67	OD Ann. Regen.	30.30
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	37.94	Hot cylinder	2.51
Alternator	326.10	Cold cylinder	.40
TOTAL WEIGHT	453.72	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.05

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INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.	8
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24
Engine operating conditions:	
Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1600.00
Metal temperature of gas cooler, K	700.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.40
Number of radiation shields in displacer,	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FP6E-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	155939.20	Thermodynamic	388491.20
Thermo. Displ.	53184.52	Adiabatic Corr.	18765.11
Adiabatic Corr.	-17440.53	Reheat loss	82678.39
Heater flow loss	19066.58	Shuttle loss	53.77
Regen. flow loss	20105.46	Appendix loss	44223.99
Cooler flow loss	13768.92	Temp. swing loss	349.26
Dsp. Dr. Pwr. Reqmt.	-270.63	Cyl. Wall Cond.	215.69
Net Engine Power	138498.60	Displcr Wall Cond.	4420.42
Alternator Loss	13849.86	Regen. Wall Cond.	6152.04
NET ELECT. POWER	124919.40	Cyl. Gas Cond.	2.36
OVERALL EFFICIENCY, %	23.85	Regen. Mtx. Cond.	7531.33
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1602.24	Flow Fric. Credit	-29119.31
Effect. Hot Gas	1563.87	Total Heat to Eng.	523764.30
Effect. Cold Gas	722.02	Displ. Dr. Heat	-270.63
OD Gas Cooler	699.70	Engine Cooling	384995.00
Weights, kg		Alternator cooling	13849.86
Hot Cylinder	39.22	Lengths, cm	
Heater tubes	15.36	Engine	32.65
Reg. Wall	1.55	Alternator	113.36
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	152.76
Displacer	26.38	Diameters, cm	
Displ. Drive Rod	.73	Eng. Cyl. OD	25.02
Displ. D. R. Support	2.67	OD Ann. Regen.	30.30
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	30.48	Hot cylinder	2.51
Alternator	249.30	Cold cylinder	.40
TOTAL WEIGHT	373.29	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.05

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	9
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1600.00
Metal temperature of gas cooler, K	800.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.50
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	122949.80	Thermodynamic	367538.50
Thermo. Displ.	51685.23	Adiabatic Corr.	16936.30
Adiabatic Corr.	-21899.48	Reheat loss	66392.90
Heater flow loss	17166.66	Shuttle loss	49.60
Regen. flow loss	20587.43	Appendix loss	31942.90
Cooler flow loss	13798.56	Temp. swing loss	250.18
Dsp. Dr. Pwr. Reqmt.	-147.31	Cyl. Wall Cond.	193.36
Net Engine Power	101050.30	Displcr Wall Cond.	4603.93
Alternator Loss	10105.03	Regen. Wall Cond.	5515.14
NET ELECT. POWER	91092.57	Cyl. Gas Cond.	2.18
OVERALL EFFICIENCY, %	19.26	Regen. Mtx. Cond.	6945.35
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1602.02	Flow Fric. Credit	-27460.37
Effect. Hot Gas	1566.52	Total Heat to Eng.	472909.90
Effect. Cold Gas	822.19	Displ. Dr. Heat	-147.31
OD Gas Cooler	799.72	Engine Cooling	371712.30
Weights, kg		Alternator cooling	10105.03
Hot Cylinder	39.22	Lengths, cm	
Heater tubes	15.36	Engine	32.65
Res. Wall	1.55	Alternator	82.71
Res. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	122.11
Displacer	30.65	Diameters, cm	
Displ. Drive Rod	.75	Eng. Cyl. OD	25.02
Displ. D. R. Support	2.67	OD Ann. Regen.	30.30
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	23.93	Hot cylinder	2.51
Alternator	161.89	Cold cylinder	.40
TOTAL WEIGHT	303.62	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.05

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INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.	10
Graphic option 0=no; 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24
Engine operating conditions:	
Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1600.00
Metal temperature of gas cooler, K	900.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.60
Number of radiation shields in displacer,	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		**HEAT REQUIREMENT, WATTS**	
Thermo.P.Pist.	93492.31	Thermodynamic	349508.00
Thermo.Displ.	50469.00	Adiabatic Corr.	15287.58
Adiabatic Corr.	-25819.71	Reheat loss	53506.61
Heater flow loss	15618.66	Shuttle loss	44.87
Regen.flow loss	21071.83	Appendix loss	23008.89
Cooler flow loss	13764.10	Temp.swing loss	182.10
Dsp.Dr.Pwr.Reqmt.	-16.01	Cyl. Wall Cond.	170.49
Net Engine Power	67672.60	Displcr Wall Cond.	4687.45
Alternator Loss	6767.26	Regen. Wall Cond.	4862.90
NET ELECT.POWER	60921.35	Cyl. Gas Cond.	1.97
OVERALL EFFICIENCY, %	14.12	Regen. Mtx. Cond.	6282.33
Temperatures, K		Rad.Inside Displ.	.00
OD Gas Heater	1601.64	Flow Fric. Credit	-26154.58
Effect.Hot Gas	1568.73	Total Heat to Eng.	431388.60
Effect.Cold Gas	922.57	Displ.Dr. Heat	-16.01
OD Gas Cooler	899.73	Engine Cooling	363700.00
Weights, kg		Alternator cooling	6767.26
Hot Cylinder	39.22	**Lengths, cm**	
Heater tubes	15.36	Engine	32.65
Reg. Wall	1.55	Alternator	55.39
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	94.79
Displacer	35.39	**Diameters, cm**	
Displ.Drive Rod	.77	Eng.Cyl.OD	25.02
Displ.D.R.Support	2.67	OD Ann.Regen.	30.30
Cooler Tubes	1.24	**Wall Thicknesses, cm**	
Cold Cylinder	18.09	Hot cylinder	2.51
Alternator	121.81	Cold cylinder	.40
TOTAL WEIGHT	242.47	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.05

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	13
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1500.00
Metal temperature of gas cooler, K	900.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.70
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	76365.90	Thermodynamic	339692.10
Thermo. Displ.	50350.87	Adiabatic Corr.	14323.94
Adiabatic Corr.	-29025.59	Reheat loss	49374.72
Heater flow loss	15706.00	Shuttle loss	37.61
Regen. flow loss	20267.86	Appendix loss	20016.69
Cooler flow loss	14513.15	Temp. swing loss	176.75
Dsp. Dr. Pwr. Reqmt.	151.27	Cyl. Wall Cond.	145.39
Net Engine Power	47340.31	Displcr Wall Cond.	3461.77
Alternator Loss	4734.03	Regen. Wall Cond.	3419.34
NET ELECT. POWER	42455.01	Cyl. Gas Cond.	1.65
OVERALL EFFICIENCY, %	10.35	Regen. Mtx. Cond.	5266.86
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1501.41	Flow Fric. Credit	-25839.93
Effect. Hot Gas	1470.93	Total Heat to Eng.	410076.90
Effect. Cold Gas	922.22	Displ. Dr. Heat	151.27
OD Gas Cooler	899.73	Engine Cooling	362887.90
Weights, kg		Alternator cooling	4734.03
Hot Cylinder :	31.24	Lengths, cm	
Heater tubes	12.23	Engine	32.65
Reg. Wall	1.28	Alternator	38.75
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	78.15
Displacer	30.65	Diameters, cm	
Displ. Drive Rod	.79	Eng. Cyl. OD	24.00
Displ. D. R. Support	2.67	OD Ann. Regen.	30.28
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	14.54	Hot cylinder	2.00
Alternator	85.21	Cold cylinder	.40
TOTAL WEIGHT	186.22	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.04

INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.	15
Graphic option 0=no. 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24
Engine operating conditions:	
Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1500.00
Metal temperature of gas cooler, K	800.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.60
Number of radiation shields in displacer,	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS	
Thermo. P. Pist.	105861.90
Thermo. Displ.	51620.38
Adiabatic Corr.	-24398.17
Heater flow loss	17283.33
Regen. flow loss	19778.89
Cooler flow loss	14581.32
Dsp. Dr. Pwr. Reqmt.	25.73
Net Engine Power	81463.72
Alternator Loss	8146.37
NET ELECT. POWER	73291.62
OVERALL EFFICIENCY, %	16.26
Temperatures, K	
OD Gas Heater	1501.55
Effect. Hot Gas	1468.80
Effect. Cold Gas	821.75
OD Gas Cooler	799.72
Weights, kg	
Hot Cylinder	31.24
Heater tubes	12.23
Reg. Wall	1.28
Reg. Matrix	6.16
Cyl. Wall	.21
Displacer	26.38
Displ. Drive Rod	.77
Displ. D. R. Support	2.67
Cooler Tubes	1.24
Cold Cylinder	20.50
Alternator	146.63
TOTAL WEIGHT	249.32
Number of heater tubes	2356

HEAT REQUIREMENT, WATTS	
Thermodynamic	357481.40
Adiabatic Corr.	16015.05
Reheat loss	62109.37
Shuttle loss	42.43
Appendix loss	28546.00
Temp. swing loss	246.31
Cyl. Wall Cond.	168.31
Displcr Wall Cond.	3449.42
Regen. Wall Cond.	3958.40
Cyl. Gas Cond.	1.86
Regen. Mtx. Cond.	5941.67
Rad. Inside Displ.	.00
Flow Fric. Credit	-27172.77
Total Heat to Eng.	450787.40
Displ. Dr. Heat	25.73
Engine Cooling	369349.40
Alternator cooling	8146.37
Lengths, cm	
Engine	32.65
Alternator	66.68
Bounce Space	6.75
TOTAL LENGTH	106.08
Diameters, cm	
Eng. Cyl. OD	24.00
OD Ann. Regen.	30.28
Wall Thicknesses, cm	
Hot cylinder	2.00
Cold cylinder	.40
Alter. cylinder	.40
Heater	.04

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	16
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1500.00
Metal temperature of gas cooler, K	700.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.50
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	138907.20	Thermodynamic	378170.20
Thermo. Displ.	53180.17	Adiabatic Corr.	17876.08
Adiabatic Corr.	-19187.56	Reheat loss	78286.38
Heater flow loss	19220.76	Shuttle loss	46.71
Regen. flow loss	19291.35	Appendix loss	40414.66
Cooler flow loss	14586.36	Temp. swing loss	348.26
Dsp. Dr. Pwr. Reqmt.	-90.78	Cyl. Wall Cond.	190.71
Net Engine Power	119719.60	Displcr Wall Cond.	3341.80
Alternator Loss	11971.96	Regen. Wall Cond.	4485.16
NET ELECT. POWER	107838.40	Cyl. Gas Cond.	2.05
OVERALL EFFICIENCY, %	21.53	Regen. Mtx. Cond.	6542.33
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1501.72	Flow Fric. Credit	-28866.44
Effect. Hot Gas	1466.24	Total Heat to Eng.	500837.90
Effect. Cold Gas	721.48	Displ. Dr. Heat	-90.78
OD Gas Cooler	699.71	Engine Cooling	381027.50
Weights, kg		Alternator cooling	11971.96
Hot Cylinder	31.24	Lengths, cm	
Heater tubes	12.23	Engine	32.65
Reg. Wall	1.28	Alternator	97.99
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	137.39
Displacer	22.55	Diameters, cm	
Displ. Drive Rod	.75	Eng. Cyl. OD	24.00
Displ. D. R. Support	2.67	OD Ann. Regen.	30.28
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	27.19	Hot cylinder	2.00
Alternator	215.50	Cold cylinder	.40
TOTAL WEIGHT	321.03	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.04

INDEPENDENT INPUT VALUES

Program control parameters:
Case number defined by operator.____________________ 17
Graphic option 0=no, 1=yes.______________________ 1
Conv. criteria (Frac. change in integrals.)______ .005000
Number of time steps per cycle.________________ 24
Engine operating conditions:
Average working gas pressure, bar______________ 400.00
Metal temperature of gas heater, K_____________ 1500.00
Metal temperature of gas cooler, K_____________ 600.00
Pressure vessel temp. of alt. and b. space, K ___ 500.00
Engine speed, Hz_______________________________ 240.00
Cylinder dimensions and materials
Maximum displacer and power piston stroke, cm ___ 1.50
Diameter of power piston and engine cyl., cm ____ 20.00
Gap between displacer and cylinder wall, cm _____ .10
Displacer rod diameter, cm _____________________ 7.40
Number of radiation shields in displacer. ______ 10
Heater, regenerator, cooler:
Number of heater and cooler tube rows __________ 15
Radial length of heated hair pin tubes, cm ______ 5.00
ID of heater tubes, cm _________________________ .20
Square pitch for heater or cooler tube array, cm_ .40
Seal length, cm ________________________________ .50
Diameter of wire in matrix, MICRONS ____________ 20.00
Porosity of matrix, PER CENT __________________ 70.00
Ratio of flow area to face area in regenerator __ 6.00
Radial length of cooled hair pin tubes, cm_______ 5.00
ID of cooler tubes, cm _________________________ .20
Linear generator parameters:
Specific weight of alternator at 60 Hz, kg/kW(e)_ 8.00
Efficiency of the alternator, per cent __________ 90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo.P.Pist.	176470.90	Thermodynamic	402715.00
Thermo.Displ.	55131.77	Adiabatic Corr.	19956.14
Adiabatic Corr.	-14686.45	Reheat loss	99396.80
Heater flow loss	21679.87	Shuttle loss	50.38
Regen.flow loss	18812.40	Appendix loss	57756.91
Cooler flow loss	14498.55	Temp.swing loss	504.19
Dsp.Dr.Pwr.Reqmt.	-156.61	Cyl. Wall Cond.	212.48
Net Engine Power	161784.50	Displcr Wall Cond.	3160.44
Alternator Loss	16178.45	Regen. Wall Cond.	4997.27
NET ELECT.POWER	145762.60	Cyl. Gas Cond.	2.21
OVERALL EFFICIENCY, %	25.81	Regen. Mtx. Cond.	7058.43
Temperatures, K		Rad.Inside Displ.	.00
OD Gas Heater	1501.94	Flow Fric. Credit	-31086.07
Effect.Hot Gas	1463.05	Total Heat to Eng.	564724.20
Effect.Cold Gas	621.63	Displ.Dr. Heat	-156.61
OD Gas Cooler	599.68	Engine Cooling	402783.10
Weights, kg		Alternator cooling	16178.45
Hot Cylinder	31.24	Lengths, cm	
Heater tubes	12.23	Engine	32.65
Reg. Wall	1.28	Alternator	132.42
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	171.82
Displacer	19.14	Diameters, cm	
Displ.Drive Rod	.73	Eng.Cyl.OD	24.00
Displ.D.R.Support	2.67	OD Ann.Regen.	30.28
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	34.55	Hot cylinder	2.00
Alternator	291.21	Cold cylinder	.40
TOTAL WEIGHT	400.67	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.04

INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.	18
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24
Engine operating conditions:	
Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1500.00
Metal temperature of gas cooler, K	500.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.30
Number of radiation shields in displacer,	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	220117.00	Thermodynamic	432669.80
Thermo. Displ.	57642.42	Adiabatic Corr.	22314.97
Adiabatic Corr.	-9605.65	Reheat loss	127998.30
Heater flow loss	24948.44	Shuttle loss	53.35
Regen. flow loss	18358.53	Appendix loss	84632.66
Cooler flow loss	14275.84	Temp. swing loss	757.06
Dsp. Dr. Pwr. Reqmt.	-66.24	Cyl. Wall Cond.	233.49
Net Engine Power	210511.40	Displcr Wall Cond.	2924.37
Alternator Loss	21051.14	Regen. Wall Cond.	5491.38
NET ELECT. POWER	189526.50	Cyl. Gas Cond.	2.34
OVERALL EFFICIENCY, %	29.14	Regen. Mtx. Cond.	7475.38
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1502.23	Flow Fric. Credit	-34127.70
Effect. Hot Gas	1458.89	Total Heat to Eng.	650425.40
Effect. Cold Gas	522.33	Displ. Dr. Heat	-66.24
OD Gas Cooler	499.65	Engine Cooling	439847.80
Weights, kg		Alternator cooling	21051.14
Hot Cylinder	31.24	Lengths, cm	
Heater tubes	12.23	Engine	32.65
Reg. Wall	1.28	Alternator	172.31
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	211.71
Displacer	16.12	Diameters, cm	
Displ. Drive Rod	.71	Eng. Cyl. OD	24.00
Displ. D. R. Support	2.67	OD Ann. Regen.	30.28
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	43.07	Hot cylinder	2.00
Alternator	378.92	Cold cylinder	.40
TOTAL WEIGHT	493.85	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.04

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	22
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1400.00
Metal temperature of gas cooler, K	500.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.42
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo.P.Pist.	201795.80	Thermodynamic	421071.50
Thermo.Displ.	57956.74	Adiabatic Corr.	21421.09
Adiabatic Corr.	-11360.90	Reheat loss	122773.50
Heater flow loss	25211.74	Shuttle loss	46.80
Regen.flow loss	17512.21	Appendix loss	78626.38
Cooler flow loss	15251.54	Temp.swing loss	768.22
Dsp.Dr.Pwr.Reqmt.	20.83	Cyl. Wall Cond.	208.91
Net Engine Power	190434.80	Displcr Wall Cond.	2183.99
Alternator Loss	19043.48	Regen. Wall Cond.	4029.61
NET ELECT.POWER	171370.50	Cyl. Gas Cond.	2.06
OVERALL EFFICIENCY, %	27.48	Regen. Mtx. Cond.	6558.26
Temperatures, K		Rad.Inside Displ.	.00
OD Gas Heater	1401.70	Flow Fric. Credit	-33967.84
Effect.Hot Gas	1361.56	Total Heat to Eng.	623722.40
Effect.Cold Gas	521.64	Displ.Dr. Heat	20.83
OD Gas Cooler	499.65	Engine Cooling	433308.40
Weights, kg		Alternator cooling	19043.48
Hot Cylinder	24.50	Lengths, cm	
Heater tubes	9.59	Engine	32.65
Reg. Wall	1.05	Alternator	155.87
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	195.27
Displacer	13.46	Diameters, cm	
Displ.Drive Rod	.74	Eng.Cyl.OD	23.14
Displ.D.R.Support	2.67	OD Ann.Regen.	30.26
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	39.56	Hot cylinder	1.57
Alternator	342.78	Cold cylinder	.40
TOTAL WEIGHT	441.95	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.03

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	23
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1400.00
Metal temperature of gas cooler, K	600.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.52
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	158093.30	Thermodynamic	391489.30
Thermo. Displ.	55347.15	Adiabatic Corr.	19001.81
Adiabatic Corr.	-17000.38	Reheat loss	94267.33
Heater flow loss	21875.08	Shuttle loss	43.68
Regen. flow loss	17976.95	Appendix loss	52635.52
Cooler flow loss	15434.53	Temp. swing loss	505.42
Dsp. Dr. Pwr. Reqmt.	-67.33	Cyl. Wall Cond.	187.79
Net Engine Power	141093.00	Displcr Wall Cond.	2351.99
Alternator Loss	14109.30	Regen. Wall Cond.	3622.29
NET ELECT. POWER	127051.00	Cyl. Gas Cond.	1.92
OVERALL EFFICIENCY, %	23.56	Regen. Mtx. Cond.	6119.89
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1401.47	Flow Fric. Credit	-30863.55
Effect. Hot Gas	1365.57	Total Heat to Eng.	539363.30
Effect. Cold Gas	621.04	Displ. Dr. Heat	-67.33
OD Gas Cooler	599.69	Engine Cooling	398203.00
Weights, kg		Alternator cooling	14109.30
Hot Cylinder	24.50	Lengths, cm	
Heater tubes	9.59	Engine	32.65
Reg. Wall	1.05	Alternator	115.49
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	154.89
Displacer	16.12	Diameters, cm	
Displ. Drive Rod	.76	Eng. Cyl. OD	23.14
Displ. D. R. Support	2.67	OD Ann. Regen.	30.26
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	30.93	Hot cylinder	1.57
Alternator	253.97	Cold cylinder	.40
TOTAL WEIGHT	347.19	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.03

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	24
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1400.00
Metal temperature of gas cooler, K	700.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.62
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	120466.30	Thermodynamic	367263.00
Thermo. Displ.	53312.27	Adiabatic Corr.	16896.81
Adiabatic Corr.	-22084.44	Reheat loss	73356.95
Heater flow loss	19366.29	Shuttle loss	39.88
Regen. flow loss	18463.14	Appendix loss	36285.45
Cooler flow loss	15480.08	Temp. swing loss	344.66
Dsp. Dr. Pwr. Reqmt.	-3.08	Cyl. Wall Cond.	165.93
Net Engine Power	98381.89	Displcr Wall Cond.	2468.07
Alternator Loss	9838.19	Regen. Wall Cond.	3200.68
NET ELECT. POWER	88546.77	Cyl. Gas Cond.	1.75
OVERALL EFFICIENCY, %	18.56	Regen. Mtx. Cond.	5586.83
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1401.30	Flow Fric. Credit	-28597.86
Effect. Hot Gas	1368.62	Total Heat to Eng.	477012.20
Effect. Cold Gas	721.03	Displ. Dr. Heat	-3.08
OD Gas Cooler	699.71	Engine Cooling	378627.20
Weights, kg		Alternator cooling	9838.19
Hot Cylinder	24.50	Lengths, cm	
Heater tubes	9.59	Engine	32.65
Reg. Wall	1.05	Alternator	80.53
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	119.93
Displacer	19.14	Diameters, cm	
Displ. Drive Rod	.78	Eng. Cyl. OD	23.14
Displ. D. R. Support	2.67	OD Ann. Regen.	30.26
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	23.46	Hot cylinder	1.57
Alternator	177.09	Cold cylinder	.40
TOTAL WEIGHT	265.89	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.03

INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.________________	25
Graphic option 0=no, 1=yes.______________________	1
Conv. criteria (Frac. change in integrals.)______	.005000
Number of time steps per cycle.__________________	24
Engine operating conditions:	
Average working gas pressure, bar_______________	400.00
Metal temperature of gas heater, K______________	1400.00
Metal temperature of gas cooler, K______________	800.00
Pressure vessel temp. of alt. and b. space, K ___	500.00
Engine speed, Hz_________________________________	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm ___	1.50
Diameter of power piston and engine cyl., cm ____	20.00
Gap between displacer and cylinder wall, cm _____	.10
Displacer rod diameter, cm ______________________	7.72
Number of radiation shields in displacer, ______	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows ___________	15
Radial length of heated hair pin tubes, cm ______	5.00
ID of heater tubes, cm __________________________	.20
Square pitch for heater or cooler tube array, cm_	.40
Seal length, cm _________________________________	.50
Diameter of wire in matrix, MICRONS _____________	20.00
Porosity of matrix, PER CENT ____________________	70.00
Ratio of flow area to face area in regenerator __	6.00
Radial length of cooled hair pin tubes, cm_______	5.00
ID of cooler tubes, cm __________________________	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)_	8.00
Efficiency of the alternator, per cent __________	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS	
Thermo. P. Pist.	67407.69
Thermo. Displ.	51682.29
Adiabatic Corr.	-26240.60
Heater flow loss	17388.82
Regen. flow loss	18956.05
Cooler flow loss	15434.49
Dsp. Dr. Pwr. Reqmt.	107.86
Net Engine Power	61167.09
Alternator Loss	6116.71
NET ELECT. POWER	54942.52
OVERALL EFFICIENCY, %	12.84
Temperatures, K	
OD Gas Heater	1401.16
Effect. Hot Gas	1371.08
Effect. Cold Gas	821.30
OD Gas Cooler	799.72
Weights, kg	
Hot Cylinder	24.50
Heater tubes	9.59
Reg. Wall	1.05
Reg. Matrix	6.16
Cyl. Wall	.21
Displacer	22.55
Displ. Drive Rod	.80
Displ. D. R. Support	2.67
Cooler Tubes	1.24
Cold Cylinder	16.95
Alternator	110.10
TOTAL WEIGHT	195.82
Number of heater tubes	2356

HEAT REQUIREMENT, WATTS	
Thermodynamic	346869.90
Adiabatic Corr.	15032.13
Reheat loss	57352.21
Shuttle loss	35.50
Appendix loss	24926.26
Temp. swing loss	240.06
Cyl. Wall Cond.	143.47
Displcr Wall Cond.	2514.09
Regen. Wall Cond.	2767.43
Cyl. Gas Cond.	1.56
Regen. Mtx. Cond.	4972.72
Rad. Inside Displ.	.00
Flow Fric. Credit	-26866.85
Total Heat to Eng.	427988.50
Displ. Dr. Heat	107.86
Engine Cooling	366929.30
Alternator cooling	6116.71
Lengths, cm	
Engine	32.65
Alternator	50.07
Bounce Space	6.75
TOTAL LENGTH	89.47
Diameters, cm	
Eng. Cyl. OD	23.14
OD Ann. Regen.	30.26
Wall Thicknesses, cm	
Hot cylinder	1.57
Cold cylinder	.40
Alter. cylinder	.40
Heater	.03

INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.	26
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24
Engine operating conditions:	
Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1400.00
Metal temperature of gas cooler, K	900.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.82
Number of radiation shields in displacer,	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	57877.68	Thermodynamic	329344.40
Thermo. Displ.	50350.45	Adiabatic Corr.	13273.21
Adiabatic Corr.	-31842.90	Reheat loss	44843.77
Heater flow loss	15781.42	Shuttle loss	30.62
Regen. flow loss	19449.43	Appendix loss	16874.79
Cooler flow loss	15326.83	Temp. swing loss	169.34
Dsp. Dr. Pwr. Reqmt.	230.25	Cyl. Wall Cond.	120.50
Net Engine Power	26034.77	Displcr Wall Cond.	2469.65
Alternator Loss	2603.48	Regen. Wall Cond.	2324.38
NET ELECT. POWER	23201.05	Cyl. Gas Cond.	1.35
OVERALL EFFICIENCY, %	5.98	Regen. Mtx. Cond.	4287.33
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1401.06	Flow Fric. Credit	-25506.13
Effect. Hot Gas	1373.12	Total Heat to Eng.	388233.30
Effect. Cold Gas	921.89	Displ. Dr. Heat	230.25
OD Gas Cooler	899.73	Engine Cooling	362428.70
Weights, kg		Alternator cooling	2603.48
Hot Cylinder	24.50	Lengths, cm	
Heater tubes	9.59	Engine	32.65
Res. Wall	1.05	Alternator	21.31
Res. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	60.71
Displacer	26.38	Diameters, cm	
Displ. Drive Rod	.82	Eng. Cyl. OD	23.14
Displ. D. R. Support	2.67	OD Ann. Regen.	30.26
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	10.81	Hot cylinder	1.57
Alternator	46.86	Cold cylinder	.40
TOTAL WEIGHT	130.29	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.03

INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.	29
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24
Engine operating conditions:	
Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1300.00
Metal temperature of gas cooler, K	900.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.97
Number of radiation shields in displacer,	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		**HEAT REQUIREMENT, WATTS**	
Thermo. P. Pist.	37725.83	Thermodynamic	316397.60
Thermo. Displ.	50562.21	Adiabatic Corr.	12116.68
Adiabatic Corr.	-35696.34	Reheat loss	39965.28
Heater flow loss	15838.46	Shuttle loss	23.89
Regen. flow loss	18611.31	Appendix loss	13670.31
Cooler flow loss	16205.94	Temp. swing loss	159.69
Dsp. Dr. Pwr. Reqmt.	103.89	Cyl. Wall Cond.	95.85
Net Engine Power	2029.49	Displcr Wall Cond.	1679.59
Alternator Loss	202.95	Regen. Wall Cond.	1510.29
NET ELECT. POWER	1722.66	Cyl. Gas Cond.	1.05
OVERALL EFFICIENCY, %	.47	Regen. Mtx. Cond.	3345.47
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1300.78	Flow Fric. Credit	-25144.11
Effect. Hot Gas	1275.30	Total Heat to Eng.	365821.50
Effect. Cold Gas	921.68	Displ. Dr. Heat	103.89
OD Gas Cooler	699.73	Engine Cooling	363895.90
Weights, kg		Alternator cooling	202.95
Hot Cylinder	16.87	**Lengths, cm**	
Heater tubes	7.39	Engine	32.65
Res. Wall	.86	Alternator	1.66
Res. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	41.06
Displacer	22.55	**Diameters, cm**	
Displ. Drive Rod	.85	Eng. Cyl. OD	22.42
Displ. D. R. Support	2.67	OD Ann. Regen.	30.25
Cooler Tubes	1.24	**Wall Thicknesses, cm**	
Cold Cylinder	6.61	Hot cylinder	1.21
Alternator	3.65	Cold cylinder	.40
TOTAL WEIGHT	71.06	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.02

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	30
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1300.00
Metal temperature of gas cooler, K	800.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.87
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	67262.13	Thermodynamic	335629.40
Thermo. Displ.	51969.55	Adiabatic Corr.	13901.92
Adiabatic Corr.	-30422.16	Reheat loss	52053.91
Heater flow loss	17476.70	Shuttle loss	28.84
Regen. flow loss	18113.64	Appendix loss	21081.52
Cooler flow loss	16359.34	Temp. swing loss	230.68
Dsp. Dr. Pwr. Reqmt.	-22.08	Cyl. Wall Cond.	116.86
Net Engine Power	36839.96	Displcr Wall Cond.	1767.94
Alternator Loss	3684.00	Regen. Wall Cond.	1872.90
NET ELECT. POWER	33178.05	Cyl. Gas Cond.	1.27
OVERALL EFFICIENCY, %	8.21	Regen. Mtx. Cond.	4040.12
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1300.86	Flow Fric. Credit	-26533.52
Effect. Hot Gas	1273.37	Total Heat to Eng.	404193.80
Effect. Cold Gas	821.01	Displ. Dr. Heat	-22.08
OD Gas Cooler	799.72	Engine Cooling	367331.80
Weights, kg		Alternator cooling	3684.00
Hot Cylinder	18.87	Lengths, cm	
Heater tubes	7.39	Engine	32.65
Reg. Wall	.66	Alternator	30.15
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	69.55
Displacer	19.14	Diameters, cm	
Displ. Drive Rod	.83	Eng. Cyl. OD	22.42
Displ. D. R. Support	2.67	OD Ann. Regen.	30.25
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	12.70	Hot cylinder	1.21
Alternator	66.31	Cold cylinder	.40
TOTAL WEIGHT	136.38	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.02

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	31
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1300.00
Metal temperature of gas cooler, K	700.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.77
Number of radiation shields in displacer,	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	100356.20	Thermodynamic	355703.90
Thermo. Displ.	53687.30	Adiabatic Corr.	15837.16
Adiabatic Corr.	-24846.38	Reheat loss	67810.45
Heater flow loss	19493.57	Shuttle loss	33.31
Regen. flow loss	17615.79	Appendix loss	31827.37
Cooler flow loss	16454.42	Temp. swing loss	337.55
Dsp. Dr. Pwr. Reqmt.	-137.24	Cyl. Wall Cond.	141.38
Net Engine Power	75509.79	Displcr Wall Cond.	1770.73
Alternator Loss	7550.98	Regen. Wall Cond.	2227.73
NET ELECT. POWER	68096.05	Cyl. Gas Cond.	1.46
OVERALL EFFICIENCY, %	15.06	Regen. Mtx. Cond.	4666.34
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1300.96	Flow Fric. Credit	-28301.46
Effect. Hot Gas	1271.02	Total Heat to Eng.	452056.00
Effect. Cold Gas	720.58	Displ. Dr. Heat	-137.24
OD Gas Cooler	699.71	Engine Cooling	376408.90
Weights, kg		Alternator cooling	7550.98
Hot Cylinder	18.87	Lengths, cm	
Heater tubes	7.39	Engine	32.65
Reg. Wall	.86	Alternator	61.81
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	101.21
Displacer	16.12	Diameters, cm	
Displ. Drive Rod	.81	Eng. Cyl. OD	22.42
Displ. D. R. Support	2.67	OD Ann. Regen.	30.25
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	19.46	Hot cylinder	1.21
Alternator	135.92	Cold cylinder	.40
TOTAL WEIGHT	209.70	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.02

INDEPENDENT INPUT VALUES

Program control parameters:	
Case number defined by operator.	32
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24
Engine operating conditions:	
Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1300.00
Metal temperature of gas cooler, K	600.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00
Cylinder dimensions and materials	
Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.67
Number of radiation shields in displacer,	10
Heater, regenerator, cooler:	
Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20
Linear generator parameters:	
Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	138019.70	Thermodynamic	379573.30
Thermo. Displ.	55824.75	Adiabatic Corr.	17977.93
Adiabatic Corr.	-19073.40	Reheat loss	88535.74
Heater flow loss	22055.06	Shuttle loss	37.22
Regen. flow loss	17123.07	Appendix loss	47542.80
Cooler flow loss	16460.16	Temp. swing loss	503.33
Dsp. Dr. Pwr. Reqmt.	-207.17	Cyl. Wall Cond.	163.31
Net Engine Power	118946.30	Displcr Wall Cond.	1707.35
Alternator Loss	11894.63	Regen. Wall Cond.	2573.35
NET ELECT. POWER	107258.80	Cyl. Gas Cond.	1.64
OVERALL EFFICIENCY, %	20.90	Regen. Mtx. Cond.	5215.00
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1301.09	Flow Fric. Credit	-30616.60
Effect. Hot Gas	1268.09	Total Heat to Eng.	513214.40
Effect. Cold Gas	620.49	Displ. Dr. Heat	-207.17
OD Gas Cooler	599.69	Engine Cooling	394060.90
Weights, kg		Alternator cooling	11894.63
Hot Cylinder	18.87	Lengths, cm	
Heater tubes	7.39	Engine	32.65
Reg. Wall	.86	Alternator	97.36
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	136.76
Displacer	13.46	Diameters, cm	
Displ. Drive Rod	.79	Eng. Cyl. OD	22.42
Displ. D. R. Support	2.67	OD Ann. Regen.	30.25
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	27.06	Hot cylinder	1.21
Alternator	214.10	Cold cylinder	.40
TOTAL WEIGHT	292.79	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.02

INDEPENDENT INPUT VALUES

Program control parameters:

Case number defined by operator.	33
Graphic option 0=no, 1=yes.	1
Conv. criteria (Frac. change in integrals.)	.005000
Number of time steps per cycle.	24

Engine operating conditions:

Average working gas pressure, bar	400.00
Metal temperature of gas heater, K	1300.00
Metal temperature of gas cooler, K	500.00
Pressure vessel temp. of alt. and b. space, K	500.00
Engine speed, Hz	240.00

Cylinder dimensions and materials

Maximum displacer and power piston stroke, cm	1.50
Diameter of power piston and engine cyl., cm	20.00
Gap between displacer and cylinder wall, cm	.10
Displacer rod diameter, cm	7.55
Number of radiation shields in displacer.	10

Heater, regenerator, cooler:

Number of heater and cooler tube rows	15
Radial length of heated hair pin tubes, cm	5.00
ID of heater tubes, cm	.20
Square pitch for heater or cooler tube array, cm	.40
Seal length, cm	.50
Diameter of wire in matrix, MICRONS	20.00
Porosity of matrix, PER CENT	70.00
Ratio of flow area to face area in regenerator	6.00
Radial length of cooled hair pin tubes, cm	5.00
ID of cooler tubes, cm	.20

Linear generator parameters:

Specific weight of alternator at 60 Hz, kg/kW(e)	8.00
Efficiency of the alternator, per cent	90.00

Martini Eng. Isothermal Analysis of FPSE-Alternator Power System

POWER, WATTS		HEAT REQUIREMENT, WATTS	
Thermo. P. Pist.	182062.40	Thermodynamic	408744.40
Thermo. Displ.	58248.68	Adiabatic Corr.	20432.91
Adiabatic Corr.	-13567.25	Reheat loss	116827.70
Heater flow loss	25479.65	Shuttle loss	40.48
Regen. flow loss	16659.20	Appendix loss	71980.41
Cooler flow loss	16345.22	Temp. swing loss	776.30
Dsp. Dr. Pwr. Reqmt.	261.54	Cyl. Wall Cond.	184.53
Net Engine Power	168495.10	Displcr Wall Cond.	1929.19
Alternator Loss	16849.51	Regen. Wall Cond.	2907.71
NET ELECT. POWER	151384.10	Cyl. Gas Cond.	1.78
OVERALL EFFICIENCY, %	25.41	Regen. Mtx. Cond.	5673.36
Temperatures, K		Rad. Inside Displ.	.00
OD Gas Heater	1301.26	Flow Fric. Credit	-33809.25
Effect. Hot Gas	1264.26	Total Heat to Eng.	595689.50
Effect. Cold Gas	520.97	Displ. Dr. Heat	261.54
OD Gas Cooler	499.66	Engine Cooling	427455.90
Weights, kg		Alternator cooling	16849.51
Hot Cylinder	18.87	Lengths, cm	
Heater tubes	7.39	Engine	32.65
Reg. Wall	.86	Alternator	137.92
Reg. Matrix	6.16	Bounce Space	6.75
Cyl. Wall	.21	TOTAL LENGTH	177.32
Displacer	13.46	Diameters, cm	
Displ. Drive Rod	.76	Eng. Cyl. OD	22.42
Displ. D. R. Support	2.67	OD Ann. Regen.	30.25
Cooler Tubes	1.24	Wall Thicknesses, cm	
Cold Cylinder	35.72	Hot cylinder	1.21
Alternator	303.29	Cold cylinder	.40
TOTAL WEIGHT	390.62	Alter. cylinder	.40
Number of heater tubes	2356	Heater	.02

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16. Abstract

This study was conducted in 1984 under the direction of the NASA Lewis Research Center, for the Triagency (DARPA, NASA, DOE) SP-100 program office. The objective was to determine which reactor, conversion and radiator technologies would best fulfill future Megawatt Class Nuclear Space Power System Requirements. Specifically, the requirement was 10 megawatts for 5 years of full power operation and 10 years system life on orbit. A variety of liquid metal and gas cooled reactors, static and dynamic conversion systems, and passive and dynamic radiators were considered. Four concepts were selected for more detailed study. Namely:

1) A gas cooled reactor with closed cycle Brayton turbine-alternator conversion with heat pipe and pumped tube-fin heat rejection.
2) A Lithium cooled reactor with a free piston Stirling engine-linear alternator and a pumped tube-fin radiator.
3) A Lithium cooled reactor with a Potassium Rankine turbine-alternator and heat pipe radiator.
4) A Lithium cooled incore thermionic static conversion reactor with a heat pipe radiator.

The systems recommended for further development to meet a 10 megawatt long life requirement are the Lithium cooled reactor with the K-Rankine conversion and heat pipe radiator, and the Lithium cooled incore thermionic reactor with heat pipe radiator.

17. Key Words (Suggested by Author(s))
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