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An Assessment of Dynamic Energy Conversion Systems for Radioisotope Heat Sources

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AN ASSESSMENT OF DYNAMIC ENERGY CONVERSION SYSTEMS FOR RADIOISOTOPE HEAT SOURCES

by

Gary R. Thayer
Carolyn A. Mangeng

ABSTRACT

The use of dynamic conversion systems to convert the heat generated in a 7500 W(t) ^{90}Sr radioisotopic heat source to electricity is examined. The systems studied were Stirling; Brayton Cycle; three organic Rankines (ORCs) (Barber-Nichols/ORMAT, Sundstrand, and TRW); and an organic Rankine plus thermoelectrics. The systems were ranked for a North Warning System mission using a Los Alamos Multi-attribute Decision Theory code. Three different heat source designs were used: case I with a beginning of life (BOL) source temperature of 640 C, case II with a BOL source temperature of 745°C, and case III with a BOL source temperature of 945°C. The Stirling engine system was the top-ranked system of cases I and II, closely followed by the ORC systems in case I and ORC plus thermoelectrics in case II. The Brayton cycle system was top-ranked for case III, with the Stirling engine system a close second. The use of ^{238}Pu in heat source sizes of 7500 W(t) was examined and found to be questionable because of cost and material availability and because of additional requirements for analysis of safeguards and critical mass.

I. INTRODUCTION

A. Background

The Office of Defense Waste Utilization of the Department of Energy (DOE) has asked Los Alamos National Laboratory to assess the use of dynamic energy conversion systems to convert the heat generated by a radioisotope heat source

to electricity. This radioisotope dynamic power system (RDPS) would be used to supply electricity to remote terrestrial sites. The dynamic systems being considered are a Brayton cycle, an organic Rankine cycle (ORC), and a Stirling engine; also being considered is an ORC cascaded with thermoelectrics.

Solid-state thermoelectric converters have previously been used with radioisotope heat sources, and about 100 of these radioisotope thermoelectric generators (RTGs) ranging in power from 0.075 to 150 W(e) have been built for terrestrial applications.¹ Thermoelectrics have been used with radioisotope heat sources because they offered a light-weight, compact, and very reliable conversion system. These characteristics were especially important for space applications, where most of the impetus for thermoelectric development originated. However, when terrestrial applications with power levels above 1 kW(e) are considered, the expense and low conversion efficiencies (~8%) of thermoelectrics make dynamic systems an attractive alternative. Also, the development of dynamic systems has progressed far enough that a reliable RDPS with a conversion efficiency greater than 20% is possible.

The importance of conversion efficiencies can be seen if one looks at isotope supply and radioisotope heat-source cost. The primary isotope considered for terrestrial applications is ⁹⁰Sr. This isotope is obtained from processing defense nuclear wastes. At present there is available about 325 kW(t) of ⁹⁰Sr with a sufficient concentration to be usable in heat sources.* The estimates are that an additional 230 kW(t) of ⁹⁰Sr will be available in the future. Teledyne is constructing a 500 W(e) RTG that will use ~7500 W(t)** of ⁹⁰Sr; therefore the total amount of ⁹⁰Sr resources that will be available for RTGs is about 37 kW(e). One proposed application for these large RTGs would require about 2 kW(e) per site, so enough ⁹⁰Sr is available for only 18 sites, less than half of the estimated 40 sites proposed. Thus, although there is at present a surplus of ⁹⁰Sr, if any use for large RTGs is identified, the supply of available ⁹⁰Sr could be quickly exhausted. Using 20%-efficient RDPSs could stretch the ⁹⁰Sr supply by a factor of 2-1/2 and might also allow an additional 810 kW(t) of low concentration ⁹⁰Sr material to be usable.

*Information provided by Earl Wheelright, Pacific Northwest Laboratories, Richland, Washington, June 1984.

**Information provided by John Vogt, Teledyne Energy Systems, Timonium, Maryland, July 1984.

The cost for the heat source for the Teledyne RTG is estimated to be in the neighborhood of \$550,000.* This heat source cost represents the major cost in a radioisotope power system. The heat source cost is the same whether the conversion system is a thermoelectric or a dynamic conversion system. Thus a 20%-efficient RDPS will have a considerably lower cost per kilowatt-hour (electric) output than will an 8%-efficient RTG. Because an RDPS would stretch the available resources of ^{90}Sr and lower the cost per kilowatt-hour for the power source, the development of an RDPS should make radioisotopic power systems more attractive for remote power applications.

It is important that the suitability of, and future demand for, radioisotope power sources be determined because decisions will soon need to be made on whether or not to recover ^{90}Sr from future wastes at Hanford and Savannah River. In general, the cost to recover ^{90}Sr from the waste streams and store it is equal to the cost incurred in disposing of the ^{90}Sr (see Wheelright footnote, p. 2). The primary incentive for recovering the ^{90}Sr would be its future use for radioisotope power sources.

Another isotope commonly used in the past in RTGs is ^{238}Pu , which is primarily used in the space program. This isotope is produced by irradiating ^{237}Np in a reactor. However, approximately forty ^{238}Pu RTGs have been manufactured for terrestrial use. Most of these have had a power output of 0.075 W(e).¹ Although some ^{238}Pu RTGs with a power level of 150 W(e) have been produced for terrestrial use,¹ no ^{238}Pu power sources in a size comparable to the Teledyne heat source [7500-W(t)] have been built. A part of this report will examine the use of ^{238}Pu in 7500-W(t) sizes for terrestrial applications to determine if they can be considered as a possible substitute for ^{90}Sr .

B. RDPSs

The concept of using dynamic conversion systems with a radioisotope heat source has been considered for a number of years, but very little hardware has been constructed. The dynamic conversion systems themselves have been under development for numerous applications, and in some cases "off the shelf" equipment exists that could be used in an RDPS. The major question for the dynamic systems is whether they will be able to operate with the required

*Information provided by John Vogt, Teledyne Energy Systems, Timonium, Maryland, September 1984.

reliability, as few systems of the envisioned size and type have had uninterrupted operation over long periods of time. Also, requirements exist for the design of heat transport systems, heat rejection systems, and safety systems. None of the requirements appear to be impossible to meet, but developmental work would have to be done.

1. Previous RDPS Work. The previous major effort with RDPSs was in 1976-1978 when NASA-DOE commissioned studies to examine dynamic systems with an output of ~1 kW(e) for space applications using a ^{238}Pu heat source. These studies looked at four systems--a Brayton cycle, a Stirling engine, and two organic Rankine converters (ORCs) using different working fluids (Dowtherm A and toluene). In some cases hardware was constructed and some initial tests were done using electric heaters. However, the program was terminated before lifetime testing or testing of proposed improved systems could be done. A summary of the four proposed systems is given in a paper by John Boretz.²

We were able to find only one example where a dynamic conversion system was actually tested with a radioisotope heat source. In 1974, the Harwell Laboratory in England constructed a 60-W(e) Stirling engine with a ^{90}Sr heat source. This engine has run continuously since that time with no reported problems.³ However, this particular engine design results in a fairly low efficiency device (8-1/2%) and is not being considered as a candidate in the latest studies. Pacific Northwest Laboratories (PNL) has initiated a project to construct a 1-kW(e) RDPS using a Stirling engine. The proposed engine would be the favored commercial design type for high efficiency in the 1-kW(e) output range. If the project is successful, it would demonstrate the feasibility of a Stirling engine/radioisotope combination (See Wheelright footnote, p. 2). Another application being investigated for a Stirling engine/radioisotope combination is an implantable power source for an artificial heart.⁴ This device only requires a 2-W output. Studies examining the scaling up of the design to the 15-kW(e) range have been done.⁵

2. Status of Dynamic Systems. Considerable development work has been devoted to the three candidate conversion systems, generally for use with other heat sources. Systems using a burner and combustion of conventional fuels for heat have been designed for all three systems. Also the solar energy program has been examining all three systems for use with parabolic or mirror concentrators. The design of systems for use with combustors or solar power is

generally more complicated than would be the case for a radioisotope heat source.

In a combustion system, the weak link in the system is usually the combustor, so systems built with a combustion heat source have not had a record of extended operating experience. Usually in the test sequence, a number of failures occur that are attributable to problems in the combustion system. Also combustion systems are usually designed so that they may have a variable output and may also be started and stopped. The solar system designs also are for variable output, placing more severe requirements on the conversion system than would be the case for an RDPS because in an RDPS the conversion system would probably be operated at constant full power during its lifetime and any extra power would be shorted to a resistor. Thus systems designed successfully using combustion or solar power for a heat source should be easily adaptable to a radioisotope heat source.

Brayton cycle machines have been designed for solar systems, and some hardware has been built.⁶ Generally, the power outputs are larger and the gas inlet temperatures higher than is the case for the RDPS application. A Brayton cycle machine was built for the 1978 NASA-DOE radioisotope space power study, and it was operated for 1000 hours. Its operating temperature was slightly higher than is available with the present capsule design. Some redesign of the Brayton cycle machine would be needed to build an optimized system for the present ⁹⁰Sr capsule design.

ORC systems with lower temperature working fluids have reached the commercially available stage. Propane-fueled ORC units were chosen to provide power on the Alaskan pipeline, and over 100 units have been installed and operated with no failures attributable to the ORC system. Fossil-fueled ORC systems are also powering a number of telecommunication systems from sub-Saharan Africa to the Trans-Siberian Railway.⁷ These operating systems are designed for high-reliability and operate at lower temperatures than envisioned for an RDPS. Operation at higher temperature requires a different working fluid (toluene) from that used in the commercially available units and a slightly different design. Thus the past experience with ORC systems may not be directly applicable to the system envisioned for an RDPS.

Stirling engines have a long history of development. At the present, the favored design for a reliable high-efficiency system is the free-piston design. (The design being used in the British radioisotope experiment is a

diaphragm-type machine). A free-piston Stirling engine with a 1 to 1.5-kW(e) output is presently commercially available for a research engine.⁸ This is the engine being considered for the PNL system. The engine is also undergoing extended-operation testing at NASA-Lewis.⁹ Other types of Stirling engine designs are available, but they are generally for higher power levels and, except for the low-efficiency diaphragm design, do not have the reliability potential of the free-piston design.

3. Other Design Requirements. This report is primarily concerned with the dynamic conversion systems, although there are a number of other design concerns for a complete RDPS system. The three main areas of concern are (1) heat transport from the heat source to the engine, (2) heat rejection, and (3) safety.

In an RTGs, heat is transported from the radioisotope heat source to the thermoelectric units by conduction. In an RDPS, three methods are commonly envisioned for heat transport: (1) direct conduction, (2) pumped fluids or gas, and (3) heat pipes. Direct conduction would use a heat accumulator block of a metal with good heat conduction properties. The heat would then be conducted directly to the engine from this metal block. This method would work best for the Stirling and ORC systems. The pumped-fluids method would pump a fluid or gas in a heat exchanger around the radioisotope capsules and then to the engine. This system could be used on the Brayton or ORC system. If heat pipes are used for heat transport, they would be inserted into a heat accumulator block containing the radioisotope capsules and would transport the heat to the engine. All three systems could use heat pipes.

Heat rejection is generally done with an air condenser. A fan could be used to decrease the condenser size at the expense of reliability and overall efficiency. Depending on requirements, either the working fluid or another fluid or gas can be used in the condenser.

Drop and impact safety is a concern with all radioisotope systems both RTGs and RDPSs, and solutions found for RTG systems should be applicable to RDPS's. However, RDPSs have another safety concern not shared by an RTG. If a thermoelectric element should fail, an RTG simply has to increase the normal mode of heat rejection by the maximum of 8%. This is a simple problem to solve. However, in an RDPS about 95% of the heat generated by the radioisotopes must be transported from the source through the working fluid of the engine to the heat rejection system. Thus safety systems must be incorporated in the design that

will cool the heat source if the engine fails or if the working fluid is lost. Also, loss of some of the heat transport system must be considered in some designs. This would include loss of the fluid in the heat rejection system, if applicable. The most stringent safety requirements will be those applying to transport, and special systems may have to be designed for this. Several ideas have been presented for safety systems, and these will have to be tested.

4. Other Conversion Systems. Two other conversion systems, a sodium-heat engine and a Nitinol (Joule-effect or memory-metal) engine, were examined briefly but not included in the analysis. Both system were too far from having a proven design to be considered in this study. The sodium-heat engine has the possibility of providing a no-moving-part conversion system that has good efficiencies. However, problems such as corrosion, exist.¹⁰ A working Nitinol engine has been built by McDonald Douglas,¹¹ but as of now, the concept only works with a low ΔT between the hot and cold reservoirs. Thus, the engine would not be able to take advantage of even the moderately high temperatures available with the radioisotope heat source.

II. PROJECT DESCRIPTION

A. Overview

The purpose of this project was to investigate the possibility of using a dynamic system to convert to electricity the heat generated in a ^{90}Sr radioisotope heat source and to determine which of the possible dynamic systems would be best suited for this application. Mission requirements needed definition in order to rank the systems, and the mission was assumed to be to power a site in either remote Alaska or in northern Canada. Because of the uncertainty surrounding the heat-source design, three heat-source configurations were considered. Cases I and II use a state-of-the-art radioisotope capsule design, and case III uses an improved capsule design. These designs are discussed in Sec. II.B.

Seven different conversion systems were considered along with the various cases and are listed below.

- Case I Brayton cycle
- Sundstrand ORC design

- Composite Barber-Nichols/ORMAT ORC design
- TRW ORC design
- Stirling engine
- Case II Brayton cycle
- Sundstrand ORC design with TAGS/PbTe thermoelectrics topping cycle
- Stirling engine
- Case III Brayton cycle
- Sundstrand ORC design with SiGe thermoelectrics topping cycle
- Stirling engine

The ranking of the systems for the three cases was obtained from a Multi-Attribute Decision Theory code (MADT) developed at Los Alamos.

B. Heat Source Design Characteristics

The reference heat source for the RDPS is a ^{90}Sr heat source using nine 800-W(t) WESF capsules. This was the preliminary design for the 500-W(e) RTG by Teledyne,¹² and it would provide 7200 W(t) at beginning of life (BOL) and 6308 W(t) after 5.5 years or end of life (EOL). A five-year lifetime was chosen because temperature data for the heat source was available from Teledyne.¹² A longer lifetime would significantly decrease the levelized life-cycle costs, but would also require a higher initial power for the source and a designing of the conversion systems to operate with a larger source temperature and power variation.

The temperature limit was chosen to be 800°C at the liner of the ^{90}Sr capsule. This temperature was generally agreed to be the present temperature limitation for the ^{90}Sr fuel (see Wheelright footnote, p. 2). However there was uncertainty whether to meet impact safety criteria by enclosing the ^{90}Sr capsule in another container (strength member), or whether the needed impact protection could be supplied by a redesigned liner or by the heat accumulator block. The Teledyne preliminary design used an additional container,¹² whereas the PNL design assumed that the ^{90}Sr capsule could be put directly into the heat accumulation block (see Wheelright footnote, p. 2).

Because of this uncertainty in the design requirements for the heat source, this study considered three cases in the ranking. Case I is the Teledyne design with a strength member enclosing an 800 W(t) Westinghouse Electric strontium flouride (WESF) capsule. The available temperatures are those calculated by

Teledyne.¹² Case II uses the same WESF capsule with an 800°C cladding temperature limit; however, the capsule is put directly into a heat accumulator block. The available temperatures for this case were calculated by Coyne Pringer of the Los Alamos National Laboratory.¹³ Case III assumed a 1000°C limit on the capsule liner, which is the limit expected for a redesigned WESF capsule with melt-cast strontium-fluoride (see Wheelright footnote, p. 2).

C. Power System Characteristics

The power system mission was assumed to be that of a power source for a microwave relay tower in the NWS in northern Alaska and Canada. The system would be required to operate in an Arctic environment but could be in an enclosed building. The heat sink would be the ambient air. The system would be transported to the site using a helicopter.

The electrical output for all systems was chosen to be a constant power at Vdc. Any variable power requirements from the load would be handled by electronic controls that would short the excess power to a resistor. All of the conversion systems produce alternating current that is then rectified to produce 28 Vdc. Thus the 28-Vdc requirement produces a penalty for all the systems. Each of the systems has a different preferred output where it would have a higher efficiency than at 28 Vdc. In order to produce a meaningful comparison, the traditional thermoelectric voltage of 28 Vdc was chosen. However, one must be aware that in each system more power could be obtained by designing the load to operate on an output other than 28 Vdc.

D. Ranking the Systems

The ranking of the various systems was done with the Multi-Attribute Decision Theory Code MADT. The advantage of using this code is that it produces a traceable, detailed record of the ranking process. Thus the effect on the ranking produced by changing assumptions or system characteristics can easily be obtained. Some sensitivity studies were done to show some of these effects.

The advantage of using MADT is that it produces a traceable record of the elements that went into the ranking of the systems. The detailed descriptions of the process used in this report is presented in Sec. IX.

III. HEAT SOURCE AND DYNAMIC SYSTEMS DESCRIPTION

A. Heat Source

Three heat source cases are used in this study. They are all based on the original Teledyne design for a 7200-W(t) ⁹⁰Sr isotopic heat source.¹² This

design used nine 800-W(t) WESF capsules; each capsule is a cylinder of hot-pressed strontium-fluoride clad in 0.137-in.-thick Hastalloy C-276. The capsule has a diameter of approximately 2-1/4 in. and is about 19 in. long. The capsules are to be inserted into a heat accumulator block (Teledyne design--tungsten) that will provide biological shielding and a heat-flow path (Fig. 1).

The actual design for the heat source being fabricated will use three Sentinel capsules containing approximately 2500-W(t) hot-pressed strontium-fluoride inserted into a tungsten heat-accumulator block (Fig. 2). The total loading will be 7500-7650 W(t) (see Vogt footnote, p. 2). The cost for the radioisotope heat source in this study will be based on the configuration using the Sentinel capsule, whereas the technical details will use the nine 800-W(t) WESF capsule design. These bases are used because costs are only available for the heat source with Sentinel capsules. The cost differential between the two designs should not be large, and because all the dynamic conversion systems are using the same heat source, should have minimal effect on the comparisons of the dynamic systems.

Case I is the Teledyne design using WESF capsules with the ^{90}Sr capsule enclosed in a strength member made of Hastalloy X (Fig. 3). This was done to meet impact protection standards. With this design, temperatures in the heat accumulator block were restricted to 560-650°C.¹² PNL is planning to build a heat source using WESF capsules with an output of 3600 W(t). This heat source design is the basis for case II. PNL predicted they would be able to obtain temperatures of 750-800°C in the heat-accumulator block by not enclosing the WESF capsule in a strength member. The calculated temperature drops show that a heat-accumulator block temperature of 745-765°C can be achieved at BOL.¹³

Case III is WESF capsules redesigned with cast-melt strontium-fluoride for the fuel form. With these changes, cladding surface temperatures of approximately 1000°C are achievable (see Wheelright footnote, p. 2). If no strength member is used for the redesigned capsules, heat-accumulator block temperatures of 945-965°C can be obtained at BOL.

The assumption was made that the heat accumulator block temperature would decline by 30°C over the five-year lifetime of the heat source. This is the temperature reduction predicted in the Teledyne calculations.¹² Another assumption is that heat losses through the insulation would be 5%. This is the approximate heat loss calculated by Teledyne for their RTG design.¹²

Fig. 1. Initial Teledyne design for 7200-W(t) ^{90}Sr heat source (dimensions in inches).

Fig. 2. Final design for Teledyne 7500 W(t) ^{90}Sr heat source (dimensions in inches).

Fig.3. WESF capsule and Teledyne strength-member design (dimensions in inches).
(WESF = Westinghouse electric strontium fluoride.)

B. Brayton Cycle

The Brayton cycle conversion system is based on the Mini-Brayton Rotating Unit (Mini-BRU) constructed by Garret AiResearch Manufacturing Company.¹⁴ This system was constructed for NASA for the space-power-system program and underwent about 1000 h of testing before the program was cancelled. A larger unit using the same design principles has undergone about 10 000 h of testing. The Mini-BRU consists of a radial-flow turbine, a radial-flow compressor, and a radial-gap alternator mounted on a common shaft (Fig. 4). The working fluid would be a helium-xenon mixture with a molecular weight of 83.8. In a machine optimized for a 1400°F turbine inlet temperature, the common shaft would rotate at 52,000 to 60,000 rpm.* The common shaft is supported on gas-lubricated foil bearings. These are capable of long-life operation, especially in steady-state operation. A regenerator is used to reduce the waste heat rejected to the atmosphere.

The system parameters given in Table I are typical parameters for a 7200 W(t) source, a turbine inlet temperature of 870°C, and a heat-rejection temperature of 25°C. For the Brayton cycle systems, heat pipes were used to transport the heat from the ⁹⁰Sr heat source to the engine. Details of the heat-pipe design are given in Sec. V. The specific value for efficiencies for the three cases is given in Sec. X.

Interfacing the Mini-BRU to the heat source is one of the major design uncertainties in this system. The Brayton cycle needs high temperatures in order to operate efficiently, so the heat exchanger from the heat source would have to be designed to obtain the maximum gas exit temperature (or turbine inlet temperature) possible. At the same time the flow losses would need to be kept to a minimum as flow losses also strongly affect the system performance (see footnote below). Complicating the heat transport is the fact that the gas mixture has fairly poor heat transfer characteristics. At 1400°C the efficiency vs temperature curve has a slope of 0.045% per degrees centigrade,¹⁴ so any reduced gas temperature will strongly affect the system performance. The design of this heat exchanger will be one of the major developmental needs for a Brayton cycle system.

In addition, the heat-rejection temperature also has a strong effect on system performance. Table II gives the power output and efficiencies of a Mini-

*Information provided by Ted Mock, Garrett Turbine Engine Co., Phoenix, Arizona, August 1984.

Fig.4. Cut-away of Mini-BRU Brayton cycle.

TABLE I
TYPICAL MINI-BRU OPERATING PARAMETERS

Component	Flow (lb/s)	T _{in} (°R)	T _{out} (°R)	P _{in} (psia)	T _{out} (psia)	Eff	P/P (%)	Comments
Compressor	0.363	542	665	71.7	107.0	0.761		Bearing heat added r _c * = 1.491
Alternator HX	0.363	665	679	107.0	106.9		0.1	Bleed flow exits
Recuperator HP	0.355	679	1768	106.9	106.6	0.975	0.22	
Heater	0.355	1768	2060	106.6	106.4		0.2	
Turbine	0.355	2060	1819	106.3	72.8	0.832		r _t ** = 1.461
Recuperator LP	0.363	1796	723	72.6	72.3	0.955	0.46	
Radiator	0.363	723	537	72.3	71.7		0.70	

Note: Net Output 2.079 kW Net System Efficiency 0.289
Control Output 0.059 kW Working Fluid Xe/He (83.8)
Gross Output 2.138 kW Shaft Speed 52,000 rpm
Bearing Loss 0.154 kW Lost Pressure Ratio 0.98
Windage Loss 0.085 kW Alternator Efficiency 0.92
Recuperator Total Pressure Drop = 0.69%

*r_c = compressor pressure ratio

**r_t = turbine pressure ratio

TABLE I'
MINI-BRU PERFORMANCE FOR VARYING HEAT REJECTION TEMPERATURE

Cold Temperature (°C)	Output Power [kW(e)]	Input Power ^a [kW(t)]	Efficiency ^b (%)
27	1.38	5.50	25
16	1.57	5.79	27.2
4	1.78	6.10	29.2
-7	2.00	6.44	31%

Notes: Turbine inlet temperature 870°C.
See mock footnote, p. 14, for source for varying heat rejection temperature.

^a If system pressure gas volume, and flow rate is kept constant the amount of heat entering the working fluid is dependent on the cold temperature.

^b AC output.

BRU for various heat-rejection temperatures assuming a 870°C turbine inlet temperature.

The Brayton cycle system is also sensitive to compressor losses, so the working fluid flow-path through the heat-rejection system would need to be as short as possible. Garret AiResearch would prefer a gas-liquid heat exchanger on the heat-rejection side of the system, and a small, compact, gas-liquid heat exchanger has been designed (see footnote p. 14).

In an actual system, the Mini-BRU would be optimized to use all the power produced by the ^{90}Sr heat source, with some small changes in the efficiencies. Because the system optimization includes designing for a specific heat rejection temperature, the Brayton cycle system would operate most efficiently in an environment that would provide a constant temperature heat sink.

In general, considerable development work needs to be done to interface a Brayton-cycle system with a ^{90}Sr heat source and with a heat-rejection system. Almost certainly a redesigned heat source would be needed. Also a mission such as a sea-bed or body power source where a constant temperature heat sink is available would be a more promising application.

The Brayton cycle system has the capability for a long life and reliable operation. As noted earlier, all the rotating parts are on one shaft and are supported by foil bearings that experience very little wear in steady-state operation.¹⁴ Unless there are parasitic loads, these bearings would be the only moving parts in the system. The machinery is rotating at very high speeds, so some care may have to be taken during transportation. However, the same principle is used for aircraft engines, so the system has the capability to be fairly rugged. The system would have to be protected against loss of load either by electronics that would dump the load to a resistor or by a bypass valve that would bypass the working fluid around the turbine (see footnote p. 14).

Maintenance on the system would be minimal. The Mini-BRU was designed for unattended operation in space for 7 years, so 5 years of unattended operation is reasonable.¹⁴ However, because of the sensitivity to heat rejection temperatures, cleaning the heat rejection fins may be required at sometime in the 5-year interval to prevent system output degradation. If maintenance personnel were on site, maintenance would probably consist of checking pressure at various places in the system, checking turbine inlet temperatures, and cleaning the heat rejection fins. Conceivably the helium could preferentially leak from the

system, but this would be detectable by lower turbine inlet temperatures and reduced pressures. If this occurred the system would probably have to be depressurized and then refilled. This would require considerable maintenance time. System repressurization, if there is not preferential helium leakage, would require a minimum amount of time.

The efficiency of the system drops quite rapidly when less than normal operating temperatures are encountered. Also, the system would have to be re-optimized for smaller output powers and temperatures. However, reoptimization only involves resizing the nozzle and optimizing the gas pressure and would not represent a major cost item (see Mock footnote p. 14).

There is considerable gas turbine experience in areas such as aircraft engines and auxiliary power units. However, the RDPS application would be of a smaller power level and at lower temperatures than normal. The lower operating temperature enhances the probability that the system reliability would be met. Obtaining the system efficiency will probably depend more on the design of the heat source and heat-rejection interfaces, as both of these interfaces are critical to the system performance. The need for stringent design requirements would also lower the probability of meeting the cost projections, even though a Mini-BRU has been built and cost projections for the actual conversion system can be expected to be fairly accurate.

Because of the need for low heat-rejection temperatures and because of the gas-working-fluid characteristics, the possibility exists that the Brayton system would require a larger heat rejection system than other conversion systems. Most of the Brayton RDPS weight is in the heat source. Most of the size is taken up by the isotope source and working-fluid heat exchanger. However, the size could be reduced by redesigning the heat exchanger.

In summary, the Brayton cycle is a potentially reliable, efficient system for an RDPS. It requires both high-source and low-heat rejection temperatures to achieve the projected efficiencies. These requirements plus the poor heat transfer characteristics of the working fluid and the sensitivity to compressor losses put a premium on heat-exchanger design. Redesign of the heat source would also be required to achieve the high turbine-inlet temperatures.

C. Organic Rankine

Three ORCs in a total of five systems have been considered in this study. The base system was obtained through information supplied by Sundstrand

Corporation.* The Sundstrand system was also used in the two design options that cascaded thermoelectrics and an ORC. One alternate ORC system was compiled from information supplied by TRW,** and another from information supplied by Barber-Nichols Engineering and ORMAT Systems Inc.†

The Sundstrand system is an upgrade of Sundstrand's operational REMCOM ORC system; however, it uses toluene instead of Dowtherm A. The TRW system has only been tested with a "bread-board" design using Dowtherm A. Results of this test were extrapolated for the proposed high efficiency (25%) ORC. The Barber-Nichols ORMAT system represents the capabilities of commercially available ORC systems.

Appendix A is a report by Barber-Nichols Inc. of Arvada, Colorado, on the possible design for an ORC system. The ground rules for the report were that the system use the proposed 7200 W(t) teledyne source and off-the-shelf equipment. Thus the design represents a system that not only has a minimum of developmental costs but also has the least technical risk in meeting the design value. In this design, toluene is pumped around the outside of the shield in the area at present occupied by the thermoelectrics. A forced-draft condenser is used as the heat rejection system and a 200°F condenser temperature is assumed. This design would require very little developmental work. If a redesign of the heat rejection system could be done to lower the condenser temperature, higher efficiencies would be obtained. Also, if the condenser-fan motor could be eliminated, the parasitic losses would be reduced the net efficiency of the unit increased.

The three ORC systems are shown in Figs. 5-7, and the system parameters for the three systems are given in Table III. Toluene is used as the working fluid, and a maximum temperature of 400°C (750°F) is assumed. Heat transport is assumed to be by pumped fluids, although Sundstrand prefers heat pipes for heat transport, thus protecting toluene from a radiation environment and eliminating radiological decomposition.

*Information provided by Douglas Lacy, Sundstrand Energy Systems, Rockford, Illinois, October 1984.

**Information provided by John Boretz, TRW, Redondo Beach, California, July 1984.

†Information provided by Nicholas Christopher, ORMAT Systems Inc., Hopkin, Mass., September 1984.

Fig. 5. Sundstrand concept for an isotope-fueled organic Rankine cycle engine system.

Fig. 6. TRW organic Rankine cycle engine.

Fig. 7. Barber-Nichols organic Rankine cycle engine.

TABLE III
CHARACTERISTICS OF ORGANIC RANKINE CYCLE SYSTEMS

Component	Sundstrand	TRW	Barber Nichols/ORMT ^a
Efficiency (%)			
Turbine	67	75	58
Pump	29	65	25
Regenerator	95	95	80
Alternator/Rectifier	90	87	88
Losses (W)			
Bearing and Seal	130		250
Parasitic Power	50		-
Pressure (psi) ^a			
Pump Input	1.341	6.5	8.81
Pump Output	600		435
Regenerator Output	550		420
Turbine Inlet	500		395
Turbine Outlet	1.610	10	9.01
Temperatures (°C)			
Condenser Output	43	84	93
Regenerator Input	61		100
Regenerator Output	228		215
Turbine Inlet	393	399	378
Turbine Outlet	282		329
Condenser Input	72		146
Turbine Speed (rpm)	54,000	85,000	65,000

^aPsi values from Barber-Nichols study (see Appendix A).

Different concepts have been used for the three cases in the ranking of the RDPS concepts. In case I, the low temperature case, the ORC systems ranked are the three systems (Sundstrand, TRW, and Barber-Nichols/ORMAT) described in Table III. For cases II and III, the Sundstrand ORC system has been cascaded with two different types of thermoelectrics, one with a higher temperature limitation than the other. The characteristics of the cascaded systems were obtained from viewgraphs furnished by Sundstrand and Teledyne Corporation. Because an ORC can only be run with peak temperature at 400°C, thermoelectric topping cycle makes use of the higher temperatures and gives a larger overall efficiency.

ORC systems are in widespread use, although typically they are used for low-temperature applications of up to 300°C. Also the systems have been built with an emphasis on reliability at the expense of efficiency, so the ORC systems on the market now have too low a temperature limit for this application and are low-efficiency (10%) devices. The systems proposed for this application use

working fluid at about 400°C, thus allowing an increased efficiency. However, there is not much experience in using toluene at high temperatures, so the reliability is uncertain. There is also a wide range (12-25%) in the efficiencies that the various proposed ORC systems are predicted to achieve. Unfortunately, the organizations that have built the most ORC systems predict the lower efficiencies.

The operating ORC systems have experienced a high degree of reliability. Over 100 ORC units are used on the Alaskan Pipeline, and no failures attributed to the ORC have been experienced.⁷ However, as mentioned above, most commercial ORC systems operating have a lower peak temperature and use a different working fluid. The main uncertainty in the ORC is not the machinery but the working fluid. A temperature of 400°C is the maximum temperature at which toluene is stable. Therefore, if there are hot spots in the system, there might be some toluene decomposition and noncondensable gas could be introduced, increasing the condenser pressure and reducing the efficiency. Heavy sludge-type organics could also be produced at hot spots. The sludge could be removed in filters, but an excessive buildup would increase flow resistance and thus increase the parasitic pumping power. Careful designing is needed to eliminate "hot spots" in the boiler.

Another possible complication is that the toluene may be operating in a radiation environment at 400°C. The combination of high temperature and radiation may cause more decomposition than predicted. Tests simulating long-term operation at 400°C in a radiation environment may need to be performed to show that reliability can be obtained. Heat transport by heat pipe, as suggested by Sundstrand, may be necessary to keep the toluene out of a radiation environment.

The ORC system must be protected against loss of load. Typically, loss of load will cause a rapid increase in the turbine speed. At about 25% overspeed, the turbine will begin to rub against the housing, stopping the turbine.* Although this does not completely destroy the ORC system, it would have to be refurbished at a central location before it could operate again. The speedup of the turbine would happen so fast that there is no mechanical way of bypassing flow or relief valves that can prevent the rubbing.* Therefore, if loss of load

*Information provided by Robert Nichols, Barber-Nichols Engineering Inc., Arvada, Colorado, September 1984.

is a possibility, a safety circuit will need to be designed to electronically shift the load to a resistor to prevent damage to the ORC.

The ORC systems are designed to be maintenance-free for 5 years. However, if toluene decomposition is a problem, scheduled maintenance may be needed to bleed off noncondensables and replace filters. The system may also have to be refilled with toluene. Field maintenance of this type may present a problem. The ORC system must be kept free of oxygen because of toluene decomposes in the presence of oxygen, and it must be kept very clean because of the close tolerances in bearings and turbines. Adhering to the necessary standards in the field may be difficult. Toluene is flammable at temperatures existing in the ORC and heat source, so fire precautions will need to be taken. The same care is needed in manufacturing, but in a system run of 100 h, any problems created by contamination will show up. Therefore part of the acceptance criteria will include an initial test on each unit.

The efficiency at the ORC system is sensitive to the condenser temperature. In the Barber-Nichols design, lowering the temperature from 200°F to 0°F would increase the rectifier output from 0.91 kW to about 1.2 kW (see Nichols footnote, p. 24). Therefore, by putting some effort into the condenser design, it is possible that a significant improvement can be obtained in the output efficiency. Also, in order to maintain the required output power, cleaning the heat-rejection fins may be necessary during the 5-year lifetime. In applications where the condensing temperature will vary, such as in an air condenser, the output of the ORC system will vary with the condensing temperature.

The efficiency of an ORC system has the smallest slope vs temperature of the three systems. Because the ORC system is generally operating at a peak temperature below the achievable maximum temperature for the heat source, a redesign at the heat-source transport system may allow the ORC to operate at its temperature limit, even with low quality isotopic fuel.

In general an ORC system is the most mature of the three systems. Many hundreds of ORC systems are in operation today. Neither Sundstrand nor TRW have constructed prototypes. The ORC systems proposed for this application are significantly different from the operational systems, so the validity of past experience is questionable. Decomposition of the toluene is the main

reliability concern, and achievable efficiencies are the main operational concern.

The sizes given for the ORC systems vary as do the efficiencies. The Sundstrand system is the largest, but it was designed for terrestrial use with natural convection cooling. The TRW has the smallest system, but it was designed for space use and uses forced convection cooling. The Barber-Nichols design is intermediate in size, being designed for terrestrial use but having forced convection cooling.

In summary the ORC system is probably the easiest system to interface with the present Teledyne ⁹⁰Sr design. The thermoelectrics can be replaced with tubes for the ORC boiler. Because of the extensive operating experience, it would seem that the ORC will be a very reliable system; however, what efficiencies can be achieved is open to question. Because the ORC is operating at a temperature below the maximum achievable temperature of the heat source, it is a very versatile system and could be adapted for use with low-grade isotopic fuel. High-temperature heat sources could be more fully utilized by cascading thermoelectrics and an ORC system.

D. Stirling Engine

The Stirling engine system in this study is based on a 1-kWe, free-piston Stirling engine (RE-1000) built by Sun-Power, Inc. This is the same engine that is proposed for the RDPS being built at Pacific Northwest Laboratories (see Wheelright footnote, p. 2). The engine is an off-the-shelf item, although it is considered as research equipment rather than as a commercial item. Helium is the working fluid, and electricity is generated with a linear alternator. The Sun-Power test engine has a rather inefficient alternator (80%) and achieves a maximum efficiency at a head temperature of about 600°C.* Alternate designs could increase the alternator efficiency to 90% and could achieve a higher efficiency at higher temperatures. The design used in this study for case III (945-965°C temperatures) was the Stirling system proposed for the space-isotope power project that operated with a higher efficiency and at higher temperatures than the present Sun-Power design.¹⁵

The system parameters for the Sun-Power Stirling engine and the space-isotope power project Stirling (SIPS)¹⁵ engine are given in Table IV. A drawing

*Information provided by David Berchowitz, Sunpower Inc., Athens, Ohio, August 1984.

TABLE IV
FREE-PISTON STIRLING ENGINE PARAMETERS

	<u>Sun-Power RE-1000</u>	<u>SIPS</u>
Speed (Hz)	30	32.5
Pressure (psi)	1015	1381
Piston Stroke (cm)	2.54	2.2
Working Fluid	helium	helium
Diameter (cm)	34	
Length (cm)	116	

by Mechanical Technology Inc. for a free piston Stirling engine is shown in Fig. 8. The engine was assumed to be thermally connected to the heat source by conduction through a copper block as in the PNL design. Head temperatures were those calculated by PNL for their RDPS (see Wheelright footnote, p. 2). For the higher temperatures, the SIPS information was used. The efficiencies for temperature up to 600°C were obtained from Sun-Power test information (see footnote p. 26). A 90%-efficient alternator design was assumed for case II and case III, and the existing 80%-efficient alternator for case I. Heat rejection was assumed to air either through fins attached to the Stirling engine or through a heat-pipe heat-exchanger system. No auxiliary pumps or compressors were assumed to be necessary.

The possibility also exists for producing a hydraulic output from a Stirling engine. This output could then be used to drive conventional rotating machinery such as pumps or generators. This system was developed for the artificial heart program, and recently studies have been done by the University of Washington on scaling the system up to the 1- to 2-kW(e) size and larger.⁵

The major uncertainty associated with the free-piston Stirling engine is the reliability. The free-piston design includes sliding surfaces and moving pistons that must be kept lubricated and sliding seals that must be kept reasonably tight. In addition, auxiliaries such as a fan for heat rejection or a pump for moving fluids to the heat-source or heat-rejection system must be separate. Note also that these auxiliaries would also add other pieces of moving machinery. The present design which uses a copper block to transport heat to the engine and uses fin rejection of heat to the air does not use any auxiliaries; however, in an optimized design auxiliaries may be necessary.

Fig. 8. Schematic of free-piston Stirling engine.
(Illustration by courtesy of Mechanical Technology
Inc., Latham, New York.)

There has been an RDPS built in England (Harwell) that used a Stirling engine, and it has been running steadily since 1974--so long-term operation with a Stirling engine is possible.³ However this is a low-efficiency design and is not the design being proposed now for an RDPS application. The primary experience with the Sun-Power engine has been at NASA-Lewis. There a similar engine has been operated over 1000 h and an extended 10 000-h test is underway. During the 1000-h test, the engine was torn down and rebuilt many times, so extended, unattended operation has not been demonstrated for this engine.¹⁶

There has been a significant amount of effort put into the development of Stirling engines. Most of this effort has been concentrated on developing Stirling engines for automotive use. In order to compete with present engines, the automotive application puts stringent requirements on the engine in terms of cost, efficiency, starting changing load, durability, etc. Thus much of the effort has gone into a significantly different system with different problems than are present in the RDPS application. Therefore, it is not clear how much of the past efforts on Stirling engines have benefited the engine design being used here.

There would be no need to make special provisions for loss of load with a Stirling engine. Because the engine is a harmonic device, it will not speed up sufficiently to cause engine damage. It will simply transfer the incoming heat directly to the heat rejection portion of the system. Therefore, all that will be needed will be to oversize the heat-rejection system to handle the normal heat rejected plus the portion of the incoming heat normally converted to electricity.

The routine maintenance for the system is likely to consist of checking the pressure and lubrication reservoir, cleaning the heat-rejection fins, and perhaps monitoring the piston position. The maintenance may also involve adding more helium to repressurize the system. Because there are sliding surfaces and seals, the possibility exists that an engine will need to be overhauled after a few years of steady operation. The overhaul would involve many of the processes that a diesel engine requires, such as new seals (rings), reboring the cylinders, etc. If an overhaul were needed in the 5-year lifetime of the system, the engine would be removed from the heat source and sent to a central facility for refurbishing and another engine would be substituted.

With a conduction heat transport system, the engine removal would involve removing insulation, unbolting the engine, putting in a replacement engine

reinstalling moorings and insulation, and starting the new engine--using 3 to 4 hours in all. This should not be necessary more than once, if at all, during the five-year lifetime of the system.

The Stirling engine has an advantage over the other systems in that the efficiency does not appear to be as sensitive to hot and cold temperatures, making it more versatile in terms of isotope requirements and heat rejection requirements.

The efficiency factors used for Stirling engines in this study are conservative compared to projected efficiencies of 30 to 40% for a Stirling engine system. Therefore the probability that the projected efficiencies would be met is very high. In fact the efficiency used in case I is a measured efficiency. The only uncertainty in this case is whether the given temperatures can be reached in a large system. The PNL RDPS on which the temperature prediction is based is projected to have about half the size of the heat source for this study. Scaling to the 7200-W(t) size may result in a larger temperature drop because the Stirling engine head area is nearly the same size, but twice as much energy must be transported to the head. A redesign using heat pipes, two engines, etc., may remove this problem.

The case II efficiencies were the same efficiencies as in case I except a 90%-efficient linear alternator was assumed rather than the 80%-efficient alternator used in the measurements in case I. This is not a very large extrapolation and should be obtainable with a little developmental effort.

The case III efficiencies were obtained from the space isotope power program. This efficiency estimate is also conservative compared to the projections. However, this is not a new technology and the working engines at this size have not achieved this efficiency, so it is unclear how much of a development program is needed to produce machines having this efficiency.

Because the Stirling engines being proposed are off-the-shelf items and the proposed heat transport methods are fairly simple, the cost estimate should be good. The only uncertainty lies in the heat-rejection system design and possibly in a new alternator design.

As was mentioned above, the big unknown with Stirling engines is the reliability. Extended, unattended operation with free-piston Stirling engines has not been reported. Existing programs to make an RDPS with a Stirling engine at PNL and to do some extended operation tests at NASA-Lewis may answer some of these concerns in the next few years.

Again as with the other systems, the primary weight component is the isotope heat source, and the size depends highly on the heat-rejection system. If the Stirling engine is bolted directly to the heat accumulator block, a mechanically strong, compact system should result, especially if fins attached to the cold end of the engine can be used for heat rejection.

The Stirling engine holds the promise of producing a compact, efficient, and versatile RDPS system, but the main uncertainty is whether the engine can be made reliable enough for this application and still retain the high efficiencies. Existing testing programs may provide the answer to this question. Certainly considerable effort has been put into the development of Stirling engines. Design codes exist, and design principles are known, so it would seem possible that a Stirling engine could be developed for this application.

IV. HEAT TRANSPORT METHODS

In an RDPS, about 95% of the heat produced by the radioisotope heat source must be transported through the working fluid of the conversion system. Therefore a heat transport method to transport the heat from the heat source to the conversion system must be considered in the design. In this study we have investigated three heat transport methods: pumped fluids, heat pipes, and conduction through metal.

A. Pumped Fluids

The pumped-fluid method of heat transport is most applicable to the ORC system. The ORC design already has a pump to move the working fluid from the condenser to the boiler, so an additional pump does not need to be put in the system. Also a liquid is being pumped, so pumping losses, although significant, are not as great as they would be if a gas were pumped. A design using pumped fluids was done for an ORC system by Barber-Nichols Engineering. The detailed design is given in Appendix A.

One method for possibly reducing pumping losses is to use one of the other heat-transport methods; however, the ORC system has a requirement for a minimum boiler surface area because of heat-flux considerations. Therefore the same length of tubing may be needed if the fluid is pumped through the heat source or through a heat exchanger coupled with another heat-transport method.

The pumped fluid does not seem to be practical for either the Stirling engine or the Brayton cycle. For the Stirling engine an additional pump must be

supplied, creating an auxiliary power loss and introducing another failure point. The Brayton cycle has a compressor in the power loop that could be used to pump gas through the source. However pumping the gas a considerable distance would result in parasitic pumping loss. The Brayton cycle design is very sensitive to losses in the compressor loop, and if the gas was pumped through the source, high pumping losses would result. And if liquid were pumped to a heat exchange for the Brayton cycle, an additional pump would be required, introducing additional failure modes and parasitic power losses.

B. Heat Pipes

Instead of pumping the fluid through the heat source, the heat itself could be used to drive fluid motion through heat pipes. The basic configuration for a heat pipe transport system would place the isotope capsules in a machined heat-accumulator block that would also contain the heat pipes (Fig. 9). Gaps between the isotope capsules, the heat-accumulator block, and the heat pipes could be

Fig. 9. Schematic of heat source design used for calculation of heat-pipe temperatures.

filled with helium. Extra heat pipes could be included in the design for safety, or the heat pipes could be designed so that in normal operation the heat would be removed in the middle of the pipe, and if the conversion system failed, the heat would be transported to the top of the heat pipe for rejection to the atmosphere.

A preliminary design of a heat pipe system was done by Coyne Prenger of Los Alamos.¹³ For his analysis the isotope capsule was assumed to have a surface temperature of 800°C with a gap of 1.4 mm between the capsule and the heat accumulator block. The heat accumulator block was copper, and the imbedded heat pipes had a gap of 0.5 mm between the surface and the block. A temperature of 700°C was obtained at the top of the heat pipe. The temperature drops for the design are shown in Fig. 10; a negligible heat loss through the insulation is assumed.

Fig. 10. Temperature drops in radioisotope heat source using heat-pipe heat transport.

The design chosen for the heat pipes was one using eight sodium heat pipes, each with an outside diameter of 2.5 cm. The evaporator length for the heat pipes is 48 cm, and the condenser length is 100 cm. The inside of the heat pipe is knurled to a depth of 0.25 mm to provide a liquid return path for the condensate and ensure uniform wetting of the heat-pipe wall. The predicted performance limits of this design are shown in Fig. 11. The heat pipes would be operating significantly below their flood limit, so the heat pipe axial temperature gradients would be negligible.¹³

The heat pipe transport method could be used with all three systems, as the top of the heat pipes could be put into a heat exchanger that would transfer heat into the working fluids of the conversion systems. Because the temperature decrease is small from the capsule surface to the top of the heat pipe, the systems that require a high temperature for efficient operation can be used.

C. Heat Conduction

The simplest method of transporting the heat is to conduct the heat through a metal such as copper. Although the temperature drops are greater than for a heat-pipe system, the output temperatures are high enough to run an ORC or Stirling engine system.

The PNL design for an RDPS with a Stirling engine uses this metal-conduction method. The heat from the radioisotope capsules is conducted into a copper heat-accumulator block and through that block to the collar at the Stirling engine. Temperatures of 600⁰ are calculated for the collar (see Wheel-right footnote, p. 2).

The advantages of this method are that (1) the design is mechanically strong because the conversion system is connected to the heat source by a large block of metal and (2) problems such as corrosion and loss of heat-transport fluid do not occur because there is no moving fluid. Thus the PNL design would probably be a more rugged heat pipe or pumped fluid system.

The problems with the method is that the ΔT is too large for the Brayton cycle design unless a redesigned heat capsule with a higher surface temperature is used. Also the PNL design is for a source with a heat output of 3600 W(t). Scaling the design to larger powers may create problems in transporting the heat and result in larger ΔT 's for the system.

Fig. 11. Heat pipe performance limit.

D. Summary

In summary, the most versatile heat-transport system appears to be heat pipes. This method produces the highest temperatures outside the heat source with a minimum of parasitic losses, and it can be used with all three conversion systems. Conducting the heat to the conversion system through a metal block will produce a simple rugged system and may be applicable to Stirling engines and ORC systems. However, it may have unacceptable temperature drops for large sources.

Heat transport using pumped fluids is only practical when an ORC is used as the conversion system. The pumped fluid system can use the existing hardware and design for the Teledyne heat source and therefore would entail the least development cost of the three heat-transport systems, but parasitic losses using the pumped fluids are relatively high, reducing the efficiency of the unit.

V. HEAT REJECTION SYSTEMS

One of the major engineering problems in an RDPS is designing the heat-rejection system. In an RTG, the heat is transported to the surface of the heat-source container, which provides a large area for heat rejection to the atmosphere, especially if fins are used. In an RDPS, the heat is concentrated in the working fluid, which typically has a small volume and surface area. This concentrated heat must be transported to a device with a large enough surface area so that the heat can be rejected to the atmosphere. Other heat sinks such as water or ground can be considered for heat rejection, but for this study only the difficult case, heat rejection to air, is considered. Because heat rejection to the atmosphere requires a large surface area, the heat-rejection system will generally be the largest component in an RDPS. The problems involved are much the same as with the heat transport system; that is, how to transport the heat, this time from the conversion system to the heat rejection system, without encountering prohibitive parasitic power losses. In addition, the conversion systems, especially the Brayton cycle, are very sensitive to the cold-side temperature; so the heat rejection system must be designed to cool the working fluid to as near ambient air temperature as possible. Probably only the heat rejection system will require maintenance (i.e., cleaning of the radiators or condenser coils).

One design of a heat-rejection system for an ORC is presented in Appendix A. The system uses a fan to pull air through the condenser coils. This design results in a compact unit at the penalty of introducing parasitic power losses for the fan motor and providing an additional rotating machine that could fail. Natural convection could be used instead of forced convection as in the Sundstrand design (Fig. 12), but this would require a larger condenser and more working fluid, resulting in an increase in pumping losses. A detailed trade-off study would be needed to determine which condenser would have the smallest power loss. Although both of the above systems were designed for an ORC, the same advantages and problems would apply in the other dynamic-conversion systems.

A second heat-rejection design is to put a heat exchanger in the condenser or cool portion of the dynamic-conversion system. The heat would be transferred from the dynamic-conversion-system working fluid to another medium for the heat-rejection system. The dynamic-conversion system designers would probably prefer that the heat-rejection system use a liquid as the heat transport medium. The improved heat-transfer properties of a liquid would reduce the amount of piping in the condenser or cool portion of the conversion system and would result in cooler temperatures and reduced pumping losses. Garrett AiResearch has designed a compact gas-to-liquid heat exchanger for the Brayton cycle system that would decrease the pumping losses in the system (see footnote p. 14). Heat pipes could be utilized to transport the heat from the liquid to fins for rejections into the atmosphere.

A third design would be heat conduction through a metal block to fins. This system would probably be most useful in a Stirling engine system because it would not require any pumps or fluids. However, the ΔT for this method may be too large for the Brayton cycle. Also, because the ORC system needs the surface area for a condenser, it may not be practical for that system.

In conclusion, the heat-rejection system represents a major design problem in an RDPS. The heat must be rejected to the atmosphere with as little pumping or parasitic load loss as is possible to still keep the cold portion of the conversion system at as low a temperature as possible. Temperature is especially critical for the Brayton cycle because substantial improvements in efficiency can be achieved if low temperatures are obtained at the cold leg of the system.

Fig. 12. Sundstrand design for a terrestrial radioisotope dynamic power system (RDPS).
(ORC = organic Rankine cycle.)

Of the three systems above, the AiResearch system with a gas-to-liquid heat exchanger plus heat rejection through heat pipes and fins seems to be the most versatile. Pumping the dynamic conversion system working fluid through an atmospheric heat dump is only practical for the ORC system. The Stirling engine would require additional pumps and the Brayton cycle would experience unacceptable pumping losses. Conduction through metal to fins appears practical only for the Stirling engine. The other two systems will need more surface area for cooling the working fluid.

The heat-rejection system will undoubtedly have to be designed with a particular mission and conversion system specified. The design will have to incorporate trade-offs between pumping and parasitic power losses, value of the cold leg or condensing temperature, reliability concerns, maintenance requirements, ambient temperatures, humidity, etc.

VI. SAFETY SYSTEMS

The safety analyses for an RTG are concerned primarily with the integrity of the isotope capsule under impact and in high-temperature fires. An RDPS will presumably have to meet these same impact and high-temperature criteria. In addition, because 95% of the heat produced by the heat source will pass through the working fluid of the converter, additional safety systems will be required to remove the isotope heat if the normal heat transport path is inoperable, either through converter failure, loss of the working fluid, or loss of the heat rejection system. The RDPS is also more sensitive to loss of electrical load than is an RTG.

A. Impact Safety

The method by which impact safety is met for the isotope capsules will have a large effect on the design of the RDPS. If the heat accumulator block or a redesigned liner can provide the required impact safety, higher temperatures can be achieved for the Stirling engine and the Brayton cycle. If an additional strength member is required, as in the Teledyne design, the available temperatures will be much lower and will significantly lower the efficiency of the Stirling and Brayton systems.

B. Converter Failure

In order to protect against converter failure, loss of working fluid, or loss of the heat-rejection system, redundant heat-transport mechanisms must be

incorporated. Two general methods have been suggested to accomplish this--heat pipes and insulation removal.

The heat pipe method would use heat pipes in the heat source that are not connected with the conversion systems or the normal heat-rejection system. These would be designed so they would not operate at the normal heat-source temperature. However, if the heat-source temperature rose, the pipes would begin to transport the heat out of the heat source. One variation of this would place the converter in the middle of the heat pipe and the fins on the top of the pipe. During normal operation, heat would be removed from the heat pipe by the converter. In the event of a converter failure, the heat pipe would transport the heat to the fins at the top of the pipe where the heat would be dumped to the atmosphere. Using this system would reduce the requirement for redundant heat pipes. One problem with using heat pipes is that they would probably be required to stick out of the heat source where they would be vulnerable to breakage either during transport or from the event that caused the converter failure.

One way of eliminating the problem of breaking heat pipes is to design a safety system that uses removal of the insulation to provide an alternate heat removal path. In order to maximize the heat flowing through the converter, an RDPS will be highly insulated. By removing this insulation, the heat can be conducted to the surface of the heat source and dissipated. A number of methods have been suggested for this.

One design used springs that were held in compression by a fusible link. When the link melted because of a rise in heat source temperature, the insulation was pushed away from the source, allowing natural convection around the source to dissipate the heat. Another design suggested by PNL is to use vacuum-foil insulation around the heat source. This insulation could be removed by flooding the insulation with helium. A fusible valve or link could be used to control this system. The insulation could be replaced by pumping out the helium (see Wheelright footnote, p. 2). One difficulty with the vacuum-foil insulation is that its structural properties are not very good and could be damaged when the system is being transported. This condition would not be a safety problem, but it would reduce the efficiency of the unit because of the additional heat loss. Removal of the insulation during transportation could be veneficial--the dynamic system would become inoperable and problems of transporting an operating conver-

sion system would be reduced. Once the RDPS was installed, the insulation could be put back and the conversion system started.

C. Loss of Load

The simplest solution to the loss-of-load problem would be to design electronics that would dump the electrical load to a resistor if the external load was lost. An ORC system or a Brayton cycle would require such a system, although a Brayton cycle could possibly use a turbine by-pass valve that would open upon loss of load. Discussions with Stirling engine designers have indicated that a Stirling engine would not need any protection from loss of load.

D. Toxicology and Fire Safety

Most of the safety issues in radioisotope heat sources concern the containment of the radioisotopes themselves. However, with dynamic systems there can be additional safety issues pertaining to the working fluid. Stirling systems and Brayton cycle systems use inert gases as the working fluid, and the only additional safety problems for these systems involve ones associated with moderately compressed gases. The ORC systems recommended for this application use toluene as the working fluid. This is a flammable organic liquid that is also toxic. Table V gives some of the toxicity and flammability limits for the toluene.* These values are no worse than many other commonly handled organics. Also the amount of toluene in the system will probably be 1 to 2 gallons, which minimizes the potential hazard. ORC systems with much larger inventories of toluene have been safely operated. The main safety concerns would probably be from spillage if the liquid were added on site or from a hot vapor jet if a leak developed in the system during operation. These threats posed by the toluene would be a concern during transportation but should present no greater problem than many other commonly handled organics.

E. Summary

RDPSs will have some unique problems from RTGs in the area of safety and fault tolerance. Therefore, designers will have to consider these problems by either including one heat pipe or a combination of redundant heat pipes, insulation removal, or turbine bypass circuits. Conceptually, these do not appear hard to design, but analysis, testing, and certification of these safety systems will certainly add to the development time and cost for an RDPS.

*Information provided by Milton Krupka, Los Alamos National Laboratory, Los Alamos, New Mexico, July 1984.

TABLE V
TOLUENE TOXICITY AND FLAMMABILITY DATA

<u>Toxicity</u>	
LD ₅₀ (orally in rats)	7.53 ml/kg
Concentration limits	
8-hour time-weighted average	200 ppm
Acceptable ceiling conc.	300 ppm
Acceptable 10-min peak conc.	800 ppm
<u>Flammability</u>	
Ignition temperature	536°C
Limits of flammability	1.27% by volume (lower)
	6.75% by volume (upper)
Heat of combustion	10,142 cal/g
Minimum ignition energy (est.)	0.25 mJ
Vapor density	3.14 ^a
<u>Vapor pressures</u>	
6.4°C	10 mm
90°C	480 mm
110.6°C	760 mm

^aDensity of vapor relative to air at one.

VII. OTHER CONVERSION SYSTEMS

In addition to the Stirling engine system, ORC system, and the Brayton cycle system, we found two other systems that may have promise for conversion systems for isotopic heat sources; however, they are not at a stage of development where they can be evaluated. The two systems are a sodium heat engine and a Nitinol engine.

A sodium heat engine (SHE) uses liquid sodium to produce electricity in a thermally regenerative cell. A diagram of the system is shown in Fig. 13.¹⁰ The electricity is generated directly with no moving parts. A more complete description of the cells can be found in Ref. 10. Estimates are that efficiencies from 30 to 42% can be achieved in advanced cells. At a hot temperature of 800°C, efficiencies at 25 to 30% are projected for near-term designs.

Fig. 13. Sodium heat engine schematic.¹⁰

The SHE has several advantages:

1. It has no moving parts.
2. It could be inserted directly into an isotopic heat source much like a sodium heat pipe.
3. The system is able to operate at almost maximum efficiency with a cold temperature of 300°C , reducing the problems encountered in heat rejection system designs.
4. High temperatures of 800 to 1000°C are adequate for high-efficiency operation.

However, the system is at an early stage of development, and material and design problems must be solved before the predicted operating efficiencies can be approached. The SHE does offer promises and should be considered in a long-term approach for conversion systems for isotope heat sources.

The Nitinol engine uses the joule heat effector thermal memory in metals to generate motion.¹¹ When the metal nitinol is heated past a transition temperature, the metal will return to a previous shape. Thus the metal can be deformed at low temperature and then heated to return it to an original configuration. Mechanical energy can be extracted during this process. Nitinol engines are predicted to operate at about 30% of Carnot.*

The advantages of Nitinol engines are the following:

1. The working metal provides good heat-transfer characteristics.
2. The metal is corrosion-resistant.
3. Long-term operation with no metal fatigue has been demonstrated.
4. Low ΔT 's (50°C) can be used to drive the engine.

The primary disadvantage for isotope heat sources is that the hot temperature is limited to less than 200 to 300°C ; therefore, the higher temperatures available in an isotope heat source cannot be used effectively. However, isotope heat sources do have the potential for use on low density or low ^{90}Sr fuel where high temperatures are difficult to obtain. These sources could also be used to provide additional power in a bottoming cycle for use in auxiliary fans, motors, etc.

*Information provided by J. L. McNichols, McDonnell Astronautics Co., Huntington Beach, California, July 1984.

VIII. PLUTONIUM-238 USE IN LARGE TERRESTRIAL HEAT SOURCES

Plutonium has been used extensively in RTG heat sources for space applications. Heat-source sizes up to about 4 kW(t) have been constructed for space application.¹ In the 1978 space dynamic power system studies, power levels up to 7 kW(t) were considered.² However, construction of 7.5-kW(t) ^{238}Pu heat sources for terrestrial applications must take into consideration cost, safeguards, and safety concerns that are not applicable to space power sources. Also, if construction of more than a few sources are planned, the availability of ^{238}Pu from the production facilities is a concern.

A. Physical and Chemical Properties

Plutonium has a melting point of 640°C . The oxide melts at 2365°C plus or minus 30°C . The specific power of pure ^{238}Pu is 0.56 W/g compared to 0.933 W/g for ^{90}Sr . The specific power of the plutonium oxide fuel is 0.41 W/g, and its power density is 4.64 W/cc. For ^{90}Sr the specific power is 0.27 W/g, and for strontium that is 40% ^{90}Sr the specific power is 1.1 W/cc. The isotopic content of heat source grade ^{238}Pu is given in Table VI.

TABLE VI
ISOTOPIC CONTENT OF HEAT SOURCE GRADE ^{238}Pu

<u>Isotopes</u>	<u>Weight (%)</u>
^{236}Pu	<0.001
^{239}Pu	14.19
^{240}Pu	2.04
^{241}Pu	0.42
^{242}Pu	0.15

Note: The ^{238}Pu available from Savannah River is now 83.2% pure. Other plutonium isotopes are present in the weight percents noted in the table.

Plutonium-238 is an alpha emitter and requires minimal shielding. Shielding costs are considerably less than for a ^{90}Sr source; therefore a ^{238}Pu source would be smaller and lighter than a ^{90}Sr source. In addition, concerns with radiolysis of organic fluids would not be applicable to a ^{238}Pu heat source.

Plutonium is compatible with refractory metals such as rhenium and with noble metals such as iridium that are used for cladding, but the temperature limit for chemical compatibility is about 200°C lower for refractory metals than for noble metals.

B. Heat Source Characteristics

For purposes of comparing a ^{238}Pu heat source to the Teledyne ^{90}Sr design, the General Purpose Heat Source (GPHS) ^{238}Pu pellet was selected. The GPHS pellet is a ceramic plutonium-oxide cylinder that is 2.75-cm long and 2.75 cm in diameter and has a thermal power of 62.5 W. The plutonium oxide fuel is produced at Savannah River, where it is pressed into pellets and loaded into iridium clads. The iridium blanks are produced by Oak Ridge National Laboratory and the clads are manufactured at the Mound Facility. The iridium-clad pellets are encapsulated at the Mound Facility, where the GPHS-RTGs are manufactured.

One hundred and sixteen GPHS pellets would be required to obtain the 7200 W specified in this study. Approximately 15.5 kg of ^{238}Pu (at an isotope purity of 83%) would be required per unit to achieve 7200 W(t). The units would be arranged in a heat accumulator block in a fashion similar to the Teledyne 500-W(e) design. Because a strength member would be required for this design, the pellet would be contained within a graphite impact shell (GIS).

The temperature at the GIS surface would be approximately 1200°C . The maximum allowable surface temperature is 1950°C , so a 750°C safety margin exists. In case III a capsule surface temperature of 1000°C was assumed for the ^{90}Sr capsule. If the 1200°C temperature was available, the primary beneficiary would be the Brayton cycle system. The Brayton cycle would be able to run at its design turbine inlet temperature of 875°C and should be able to achieve an efficiency of approximately 28 to 30%. In fact, a maximum turbine inlet temperature of approximately 1025°C could be obtained. A ceramic Brayton cycle system is under development that could utilize these temperatures and possibly achieve even higher efficiencies.*

*Information provided by Ronald Niell, Garrett Turbine Engine Company, Phoenix, Arizona, August 1984.

C. Plutonium Availability and Heat-Source Production Capability

At present, there is approximately 12 kg of ^{238}Pu in stock, of which 8 to 9 kg are committed to programs at Los Alamos.¹⁸ Approximately 10 kg of 80 to 90% ^{238}Pu exists in the form of scrap. Savannah River is not scheduled to produce any ^{238}Pu for the next two fiscal years. Any new requirements for ^{238}Pu will have to be reviewed by the Office of Materials Production of the US Department of Energy. They establish the isotope production rates to apply at Savannah River in order to meet overall programmatic needs and priorities.

Production of 50 to 100 7200-W(t) units would require 750 to 1550 kg of ^{238}Pu . The current limiting GPHS production rate at DOE facilities is about 200 units/yr, limited by the fuel fabrication and cladding operations at existing Savannah River facilities. This production capability translates into a maximum of two 7200-W(t) units/yr.

Oak Ridge is currently producing around 50 iridium blanks/month, or enough for 25 fuel-clads; the Mound Facility manufactures clads at about the same rate. Savannah River has a production capability of 200 pellets/yr including clad welding; and Los Alamos has a production capability of about 20 pellets/yr. Currently, the Mound Facility produces an average of 2 GPHS modules a week, each module containing four 62.5-W units or about 400 62.5-W units/yr.

D. Cost

Historically, ^{238}Pu has been provided to defense programs as a government supplied equipment (GSE) at no cost to the user. Its cost on the open market from Oak Ridge Isotope Sales is \$1800/g. The materials and manufacturing cost of the iridium-clad is \$7900/set, assuming 1 set in 8 is destroyed during testing or other procedures.* The cost of producing a single GPHS pellet, including welding and nondestructive testing, was estimated by Los Alamos to be \$20,000 for each 62.5-W unit** and by Savannah River to be about \$100,000/unit.¹⁸ The cost of encapsulating the GPHS unit in a GIS at the Mound Facility would be approximately \$6600/unit, not including gauging or outgassing but including the cost of the fine-weave pierced-fabric (FWPF) graphite, machining, and thermal processing. The total cost of a 7200-W(t) ^{238}Pu heat source would therefore be \$4.2 million when using the Los Alamos estimates (\$34,500 for each 62.5-W unit

*Information provided by Ernest Johnson, Mound Laboratories, Kansas City, Missouri, September 1984.

**Information provided by Gary Rinehart, Los Alamos National Laboratory, Los Alamos, New Mexico.

plus \$200,000 for heat source fabrication), or \$14.4 million when using the Savannah River estimates (\$124,500 per unit).

A cost advantage of the ^{238}Pu is that it has a longer half life (87.8 yr vs 28.6 yr for ^{90}Sr); it would take 17 yr for the power level in a ^{238}Pu source to decrease as much as the power decreases in 5 yr for a ^{90}Sr source. Assuming a system lifetime of 17 yr for ^{238}Pu and a conversion efficiency of 30%, the lifetime cost for a ^{238}Pu power system would be approximately \$15/kWh compared to approximately \$9/kWh for a comparable ^{90}Sr system with 25% efficient conversion and a 5-yr lifetime.

Costs provided above were given for existing production rates and would be significantly different if new facilities would be required for GPHS production.

E. Safeguards Safety

Plutonium-238 is classified as a special nuclear material (SNM) by the DOE, and all aspects of its use are closely monitored and carefully prescribed for the protection of national security. For example, there are special requirements for physical protection and for control and accountability of plutonium, and these requirements are more stringent as the quantity of plutonium increases. The 15.5-kg quantity described above would fall in Category I, the most stringent category of all. SNM requirements, however, do allow exceptions to the rules where alternative measures can be demonstrated to meet standards.

The minimum bare-sphere critical mass of 80% ^{238}Pu oxide (20% ^{239}Pu) is 25.5 kg.¹⁸ The minimum reflected critical mass would be approximately half that number or 12.75 kg, and the water-moderated mass would be somewhat higher than 12.75 kg.* The difference in isotope composition of plutonium oxide available today would have a modest effect on the critical mass, but a difference in the density of the plutonium configuration would have a very large effect. The critical mass figures presented here were computed for plutonium oxide fuel density, about 11.5 g/cm³, and would be from 5 to 8 times greater if the fuel density were reduced to 2 to 3 g/cm³, as it would be when configured in an RTG, for example. Although a 7200-W(t) ^{238}Pu heat source would contain more than a critical mass of the ^{238}Pu mixture, a noncritical configuration would not be difficult to maintain in the heat source. However, analyses would have to be performed to show that the source would remain subcritical in accident situations.

*Information provided by David R. Smith, Los Alamos National Laboratory, Los Alamos, New Mexico, September 1984.

The radiological safety concerns for ^{238}Pu would be much the same as those for ^{90}Sr . That is, analyses would have to be done to show that the capsule would not rupture under prescribed impact and fire conditions.

F. Summary

In summary, various points can be made.

1. A ^{238}Pu heat source would have higher temperatures and allow higher efficiency conversion.
2. A ^{238}Pu heat source would have a longer half-life, reducing power leveling problems and giving a longer power-system life.
3. Even with the above two advantage, ^{238}Pu power costs would be approximately 60% greater than for a ^{90}Sr heat source.
4. A 7200-W(t) heat source would contain more than a minimum critical mass of ^{238}Pu mixture. Therefore, additional analyses would have to be done to insure that the source would not go critical under accident scenarios. Also, safeguards questions would have to be addressed.

IX. RANKING

The ranking of the various systems were done with the Multi-Attribute Decision Theory (MADT) code (see footnote p. 7). This code combines attribute measures for various attributes and the attribute weight factors to produce a ranking of the different systems. First, the code requires a number of attributes that can be used to judge the various systems.

These primary attributes can then be further subdivided, and in the second step, weight factors are assigned to each primary attribute and to the subdivisions of these attributes. These weight factors indicate the importance of that particular attribute or subdivision for the mission.

The third step is to select the methods to determine attribute measures for each of the attributes. these attribute measures can be specific numbers, such as system efficiency, cost, size, etc., or the attribute measures can be obtained from a subjective judgment as to how well the system meets the attribute. In the latter case, the systems are ranked on a scale of 1 to 10 as to how they satisfy the attribute. Once the methods for determining the various attribute measures have been chosen, the specific numbers for each attribute measure for each system are obtained.

The last step is to choose utility functions for each of the attribute subdivisions. The utility function converts the attribute measure for that subdivision to a utility value (UV) that goes from 0 to 1 and is a common unit for all attributes. This utility value can be used to combine attribute subdivisions to obtain a ranking. For example, for lifetime costs the attribute measure would be a number that is in dollars per kilowatt-hour. In order to combine this with another attribute measure such as size (measured in cubic feet), both of the attribute measures for cost and size are converted to a common unit of the utility value by a separate utility function. The utility function consists of a curve of utility value vs an attribute measure (i.e., dollars per kilowatt-hour) from which a utility value can be obtained for each attribute measure.

The utility values found from the utility functions are then multiplied times the weight factor for the attribute subdivision to obtain a score. All the scores from the attribute subdivisions are then added to give a figure of merit (FOM) as given in Eq. 1.

$$FOM = \sum_{\substack{\text{all} \\ \text{attributes}}} UV \times WF \quad . \quad (1)$$

A. Selection of Attributes

For this analysis the criteria were selected assuming a hypothetical mission to power a microwave relay tower in the North Warning System. The criteria selected to judge the systems were the following:

1. Costs: The system costs should be as low as possible.
2. Reliability: The power system should start and operate at its rated power during its operating lifetime with a minimum of failures.
3. Maintenance: The system should not require maintenance more than once a year. Maintenance time should be minimized.
4. Efficiency: The efficiency of the system should be as high as possible in order to preserve radioisotope resources and to allow lifetime stretch of the system.

5. Technical Risk: The system should rely as much as possible on available and proven equipment and designs.

6. Size/Weight: The system should be helicopter-transportable either as a unit or as modules.

These criteria were then divided into secondary attributes. Fig. 14 gives the attribute tree used in this study, and Fig. 15 shows the weights assigned to primary attributes.

In the chart the criterion cost was broken down into three areas--development, lifetime, and initial costs. The total costs of a system is usually considered to be development costs plus all the lifetime costs for the individual units built. Because the ultimate number of RDPS units that will be built is not known, development costs for a system were included separately, and lifetime costs were then calculated on a per-unit basis. Initial costs are also part of the lifetime costs. However, initial costs were included separately because generally if two systems have identical lifetime costs the one with the lower initial cost is more attractive.

Reliability was divided into three areas--output degradation, operating failure, and startup failure. It is obviously important that a system does not fail when it is in a remote location as repair is difficult and time consuming. It is also important that the system output will not degrade more rapidly than is planned; it may not mean an immediate repair trip, but it may require premature system replacement, or it may reduce the ability of the system to provide all the necessary power. A failure at startup will not jeopardize the ability of the system to ultimately provide power, but it will result in additional costs and deployment time for the project.

Maintenance requirements were divided into two parts--man-hours and frequency. In general the length of time needed to do the preventative maintenance on a remote system is not important as long as it doesn't require staying at a site for more than one day or doesn't reduce the number of sites that can be visited in a day. However, the number of times the facility must be visited is an important consideration. Because the sites are remote, there is travel time and inconvenience in visiting the sites to be taken into account.

The efficiency attribute is concerned with the affect of the conversion efficiency on resource availability. The first of the two divisions, efficiency at operating temperature, is concerned with the number of power units that can be built from the limited isotope resources. The larger the number of units

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Fig. 14. Attribute tree used for ranking.
(FOM = figure of merit.)

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WF^a = WEIGHT FACTOR

OWF^b = OVERALL WEIGHT FACTOR

Fig. 15. Weights assigned to primary attribute.
(FOM = figure of merit; WF = weight factor;
OWF = overall weight factor.)

that can be produced, the more attractive an RDPS will become for applications that have a requirement for a large number of units. The second division, efficiency at lower than operating temperatures, is concerned with using more dilute ^{90}Sr . There is a large amount of strontium available that has an isotopic content of less than 25%. This strontium could be used in a heat source but could not produce the temperatures of that of the mixture with more than 25% ^{90}Sr . Thus, if the conversion system has a reasonable efficiency at lower temperatures, strontium mixtures with less than 25% ^{90}Sr could be used with the same heat source design and still produce a usable output.

Technical risk is concerned with the amount of testing and experience there is in using the various systems. The estimates for the technical risk are divided into three areas--reliability, efficiency, and cost--and are more likely to be accurate if numerous systems have been built and tested.

Because the system must be transported to the site by helicopter, the total system size and weight must be less than the maximum that a helicopter can accommodate, or the system must be built in modules. Again, because a remote site has been chosen for the mission, the fewer trips to the site that are needed for construction the better.

B. Ranking and Assigning Weight Factors to the Attributes

The ranking of the criteria was done by first assembling the criteria in order of importance as follows:

- Cost
- Reliability
- Technical Risk
- Maintenance
- Efficiency
- Size/Weight

These criteria were then compared by pairs to determine their relative importance (Fig. 15).

Cost was assumed to be the most important criterion, with lifetime costs being the most important component (Fig. 16). Depending on the importance of the mission or the amount of backup available, costs may be more or less important than the second-ranked attribute, reliability. For this mission the assumption was that there will be redundant communication links, so reliability is not of overriding importance.

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Fig. 16. Weight factor assigned to cost.
(WF = weight factor; OWF = overall weight factor.)

For reliability, the most important component was the probability of operational failure. The probability of output degradation was assumed to be less important because warnings of reduced output would be available and repair personnel could be dispatched. The weight factors given the reliability criterion portions are shown in Fig. 17.

Technical risk was chosen as the third most important criterion after costs and reliability. Most of the systems proposed have not demonstrated either the necessary reliability or the projected efficiencies. If the final systems are not able to meet their prediction for efficiency and reliability, their ranking in the first two criteria areas would be reduced. The weight factors of the three components of technical risk are given in Fig. 18.

Maintenance requirements were rated fourth in this application because all systems will require a minimum of maintenance over the five-year lifetime. There is a possibility that an annual visit to the site may be necessary to inspect the towers, electronics, buildings, etc. If this annual trip is necessary for other systems at the site, an annual maintenance requirement for the power system is not burdensome unless the time requirements are large. The weight factor for the two maintenance components are given in Fig. 19.

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Fig. 17. Weight factors for reliability attribute.

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Fig. 18. Weight factors for technical risk criteria.

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Fig. 19. Weight factors for maintenance attribute.

The efficiency criterion only applies to the effects on isotope resource utilization. This is important for applications requiring total amounts of ^{90}Sr close to the amount of available resources. Thus, the higher efficiencies may allow an RDPS to meet the total mission requirement, whereas in a system with lower efficiency, the available isotopes will not be adequate for the total mission. Because the probability is small that the mission requirements and the available resources will match closely enough that efficiency will greatly affect the choice of systems, the efficiency criterion was rated next to last. The weight factors of the efficiency components are given in Fig. 20.

The size/weight criterion was rated last because even though helicopter transportation is assumed, the system could be broken down into components and delivered in two trips. This would be an inconvenience but not an important factor in the system application. If heavy-lift helicopters were available, the importance of this criterion would be reduced even more. Fig. 21 gives the weight factors assigned to the individual size and weight components.

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Fig. 20. Weight factors for efficiency attribute.

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Fig. 21. Weight factors for size/weight attribute.

C. Method to Determine the Attribute Measures

The method used to obtain the attribute's measures for the systems depend on the nature of the information available. In some cases, specific numbers such as cost, efficiency, weight, etc., are available to use as attribute measures. In other cases, subjective judgment is all that is available to determine how the system will meet the criteria. In the latter case, cardinal ranking on a scale of 1 to 10 is used to judge the systems' abilities. The score for the system comprises these judgments and the specific numbers and are combined with the utility function to obtain utility values for the ranking process. The method for obtaining the attribute measures for the various attributes are given in Table VII.

TABLE VII
METHODS FOR OBTAINING ATTRIBUTE MEASURES

Attribute	Measurement Method
1. Cost	
1.1 Developmental	Cardinal ranking
1.2 Lifetime	Dollars-per-kilowatt hours assuming constant 1500 W(e) output
1.3 Initial	Dollars-per-kilowatt cost at factory gate assuming 1500 W(e) output at EOL
2. Reliability	
2.1 Probability of operational degradation	Cardinal ranking
2.2 Probability of operational failure	Cardinal ranking
2.3 Probability of start-up failure	Cardinal ranking
3.0 Maintenance	
3.1 Frequency	Visits per year
3.2 Man-hours	Man-hours required per visit
4.0 Efficiency	
4.1 Operating temperature	Projected operating efficiency
4.2 Low temperature	Slope of efficiency vs temperature at operating temperature
5.0 Technical risk	
5.1 Reliability	Log of total operational hours for a system
5.2 Efficiency	Difference between projected efficiency and achieved efficiencies
5.3 Cost	Cardinal ranking
6.0 Size/Weight	
6.1 Weight	Weight in pounds
6.2 Size	Size in cubic feet

The development costs were scored by using cardinal ranking, because little information was available on the actual developmental costs needed for the systems. The ranking took into account the state of the technology for the systems and the amount of experience with the various systems. The lifetime costs were costs per kilowatt hour assuming that the power system has a constant output of 1500 W(e) for 5 yr. The initial cost assumed an output of 1500 W(e) at EOL. Both of these costs are factory-gate costs and do not include transportation or site installation costs. All systems were sized to produce 1500 W(e) at EOL by varying the heat-source size.

The costs for each system were calculated assuming an output of 1500 W(e) at EOL. Standardizing the EOL output requires that the heat source size be changed for each system because of the differing efficiencies of the systems. The EOL capacity was chosen because this would be the power requirement of the facility and this represents the useful output of the RDPS unit over its lifetime. An identical output was chosen for all systems because this output produced the most realistic cost comparison between the different system's efficiencies. Using the standard output will result in a less accurate cost for the heat source, but as most systems would require a redesign of the heat source anyway, the decreased accuracy is not a significant problem. Another more serious problem with this approach is that the costs for a heat source tend to increase in steps. For example the cost of adding 10 watts per capsule is small, but if an additional capsule is needed, the cost will increase considerably. For this study the additional cost includes a capsule cost component even if a fraction of a capsule is needed.

Cardinal ranking was used in scoring all the reliability criteria for the primary reason that specific reliability numbers were not available for the systems; some systems had not been built and others had had no long-life testing performed on them. The rankings were obtained by comparing the reliability experience of systems that were related to the proposed systems. Furthermore, the complexity, number, and type of failure possibilities for each system was considered in the scoring of the various systems.

The measure for maintenance was based on the frequency of maintenance calls per year and the number of hours required. Because very little information was available on maintenance requirements, the systems were assigned frequency-of-maintenance according to their sensitivity to low temperature necessitating the cleaning of heat rejection fins. Those systems that need a lower temperature

cold-side would require more frequent cleaning to maintain their performance. The man-hours required for maintenance were based on the complexity of the monitoring equipment that would be necessary to monitor the operation of the system.

The measure for the system operating-temperature efficiency was the projected efficiency at the end-of-life temperature, neglecting heat losses through heat-source insulation. The efficiency at low source temperature was characterized by the slope of the efficiency vs temperature curve at the EOL operating temperature.

Information on the operating experience of the different systems was used to characterize the technical risk criteria. The log of the reported number of operating hours on the specific system proposed was used for rating the technical risk for reliability. The total operating hours was used rather than continuous hours because most of the systems are set up for research purposes, and long lifetime runs have not been attempted. No information was generally available as to whether the failures experienced were attributable to external equipment, such as burners, monitoring equipment, etc., or to the system of interest. For technical risk efficiency, the number used for characterization was obtained by taking the difference between the projected efficiency and the actual efficiency achieved by an applicable system. The technical risk for cost was a cardinal ranking, again primarily because of the lack of hard information on prototype costs.

The measure for the size and weight were the projected size and weight of the various systems. The size and weight used included the changes in heat source and heat-rejection size to accommodate a 1.5-kW(e) output with the various efficiencies of the systems.

D. Finding Attribute Measures

The actual values used for each attribute measure is given in Appendix B. What will be described in this section is the procedure by which the attribute measures were obtained.

1.0 Cost. The developmental utility function cost was a cardinal ranking. The criteria for the ranking are listed below.

<u>Attribute Measure</u>	<u>Numerical Equivalent</u>
Large developmental costs needed	0
Extensive developmental costs needed	2
Above-average developmental costs needed	4
Below-average developmental costs needed	6
Minimal developmental costs needed	8
No developmental costs needed	10

The initial cost and lifetime costs were calculated using the Teledyne 7500-W(t) ⁹⁰Sr heat source as the cost basis for the heat sources and varying the heat-source size so all systems produced 1500 W(e) at EOL or 5 yr. The cost data assumptions are summarized below.

1. Heat-source size will be varied so that all systems will produce 1500 W(e) at EOL.
2. Heat loss through insulation is 5%.
3. Heat source costs are based on the Teledyne 7500-W(t) source (see footnote p. 2).
4. A 7500-W(t) heat-source cost is \$300,000 (without isotopes).
5. Material costs for the heat source are 20% of total cost.
6. Encapsulation costs for the 7500-W(t) ⁹⁰Sr source are \$50,000 for each of the 2,500-kW(t) capsules.*
7. Material and encapsulation costs are directly proportional to heat-source size.
8. Heat-rejection system costs are \$500/kW(t) where the kilowatt (thermal) is the size of the heat source.
9. Isotope (⁹⁰Sr) costs are \$14/W.
10. Heat-pipe costs are \$500/kW(t) of the heat source.**

*Information provided by Bill McMullen, Office of Defense Waste Utilization, DOE Albuquerque, New Mexico, September 1984.

**S. V. Jackson, W. L. Kirchner, M. C. Krupka, and G. R. Thayer, "North Warning System Alternate Power Supply Assessment," Los Alamos National Laboratory report (to be published).

11. Thermoelectric cost are \$17/W assuming 10% conversion efficiency from the total heat-source power.*

Items that should be mentioned for costs follow.

1. The cost of the ^{90}Sr is given as \$14/W--the present cost from the isotope sales organization at Oak Ridge. Savannah River has estimated that the future costs for recovering ^{90}Sr will be about \$180/W.¹⁹ Thus, depending on the pricing policy and whether or not stockpiled ^{90}Sr is used, the cost penalty for increasing the heat source size may be significantly larger.
2. All costs were done on a basis of 50 units produced. Single-purpose and prototype units will obviously have significantly higher costs.
3. Encapsulation costs were estimates based on those for sentinel 2500 W(t) capsules. If new processes or WESF capsules are used, there could be a change in the costs.
4. The costs are factory-gate costs. They do not include transportation to the site, site construction costs, shelter building costs, etc.

2.0 Reliability. The utility functions for reliability used cardinal ranking. The cardinal ranking criteria are listed below.

<u>Attribute Measure</u>	<u>Numerical Equivalent</u>
Failure of the power system once is almost certain during lifetime.	0
Numerous vulnerable failure possibilities exist in the power system; therefore, failure is likely during lifetime.	2
Only a few vulnerable failure possibilities exist; therefore failure is possible during lifetime.	4
Only a few failure possibilities exist, and no experience exists to show that these possibilities are not vulnerable to failure; therefore, failure may be possible during lifetime.	6

*Information provided by K. F. Paulovich, "Thermoelectric Generators for Short Range Radar Installations," G. A. Technologies, La Jolla, California, October 28, 1983.

Only a few failure possibilities exist, and experience has shown that the possibilities are not vulnerable to failure; therefore failure is unlikely during lifetime. 8

Failure during the lifetime of the power system is extremely unlikely. 10

3.0 Maintenance. There was very little information provided on maintenance for the various systems. This is probably because of the lack of experience in running these systems for long periods of time. Therefore, the values given for maintenance were chosen so as not to significantly affect the outcome of the decision process. The frequency of maintenance was given as 0.5 visits/yr for the ORC and the Brayton and 0.4 visits/yr for the Stirling. These values were chosen by assuming that the primary maintenance item would be to clean the heat-rejection surfaces. The systems (ORC and Brayton) that were more sensitive to the low temperature were assumed to require slightly more frequent cleaning of the heat-rejection surfaces to maintain their output. Normally, at BOL when the power output is large, cleaning would not be needed; but near EOL, clean surfaces may be needed to maintain rated output. Therefore, maintenance requirements may change as the system ages.

The number of hours required for maintenance was chosen by assuming that cleaning the heat-rejection surfaces would take about 2 hours. The Brayton cycle and the ORC systems would only require the checking of a pressure gauge or two to check operation. The Stirling engine and ORC plus thermoelectrics may involve more elaborate checks, adding another hour for maintenance.

One important consideration not included is the nonroutine maintenance required if the system output is degrading too fast or the system parameters are outside of specifications. To be able to provide any meaningful number for these situations would require a failure analysis and corresponding corrective action for the various systems. These data were not supplied for the different systems.

4.0 Efficiencies. The operating temperature efficiencies were those provided by the various system proponents degraded for a 30°C temperature loss from BOL to EOL. These efficiencies do not include heat loss through the heat source insulation. In some cases, the only information available was for an alternating current output. In these cases, a rectifier efficiency of 95% was assumed in order to change the power to 28 Vdc. The efficiencies did not include the power use in external electrical controls or in auxiliary motors for forced-convection heat-rejection.

The slope of the curve of efficiency vs temperature were obtained from curves provided by the system proponents or by curves appearing in a paper by John Boretz on the systems developed for space use.²

5.0 Technical Risk. The values used in the technical risk for reliability and efficiency were gotten from information obtained from proponents and from published papers. The number of operational hours generally were obtained from descriptions of a system that closely matched the proposed system for an RDPS. Thus for Stirling engines, the value used for operational hours was not a summation of all free-piston Stirling engine experience but was for the number of operating hours for a single, free-piston Stirling-engine test at NASA-Lewis.¹⁶ This test was done because to aggregate all operating hours for a system would be impossible and because proponents of a system generally report the most significant achievements of the system in terms of operating life, reliability, etc.

The number used for operating hours did not take into account whether or not the system operated continuously over that time period. Most of these system failures reported in the tests were caused by failures of auxiliary systems or monitoring equipment. We did not try or have the information to produce a continuous operation time with the effects of these outside failures removed.

The technical risk of obtaining cost was done with a cardinal ranking. The criteria for the ranking were the following:

<u>Attribute Measure</u>	<u>Numerical Equivalent</u>
No similar systems to the proposed system have been produced.	0
Similar systems have been build, but a prototype for this system would have to be developed.	2
A prototype system has been built that could be adapted to this application.	4
An off-the-shelf system would have to be scaled and redesigned.	6
Slight modification of an off-the-shelf system could be used.	8
The systems could use an off-the-shelf system with no modification.	10

6.0 Size/Weight. Most of the weight of the systems is in the heat-source. The assumption made to obtain the system weight was that the heat-source weight was directly proportional to the heat output needed to produce 1.5 kW(e) at EOL with the various system efficiencies. The basis for the weight of the system was the shipping weight given by Teledyne for their source (see Vogt footnote, p. 2).

The heat-rejection systems probably represent the largest component of the ROPS. Because heat-rejection systems have not been very well defined for the various concepts, the shipping configuration size given by Teledyne was used as a basis for finding the size of the various RDPs. It was assumed that the volume occupied by each system would be directly proportional to the heat-source size for the system.

E. Selecting Utility Functions

The utility functions convert the different measures described above to a common unit--utility value. The utility functions are expressed as a graph of utility vs the attribute measure of the particular criterion. If cardinal ranking is used to evaluate the systems, the code automatically generates a straight line with a 45° slope and an X axis of 1 to 10. In all cases the utility values go from 0 to 1. A discussion of the utility functions follows.

1. Attribute 1--Cost (Fig. 22). Development costs were scored using cardinal ranking. The utility functions for the other costs were chosen to provide the largest possible spread in the utility values for the costs. Thus a utility value of one is given to costs that are lower than any of the projected costs. At the number for the lowest cost system, the utility value begins to decrease linearly, reaching zero at the highest cost system. This gives a maximum spread for the cost utility values.

2. Attribute 2--Reliability (Fig. 23). Reliability was scored using cardinal ranking.

3. Attribute 3--Maintenance (Fig. 24). The utility function for maintenance frequency was linear with a utility value given to the requirement of zero maintenance visit per year. A utility value of zero was given for a requirement of one maintenance visit per year.

The utility function for man-hours required for maintenance is a step function. If the maintenance requirement is less than 4 h, the maintenance trip can be combined with electronic maintenance (4 h assumed) and still be done in a one-day trip. Therefore, the segment of 1 to 4 h was given a utility value of one. If maintenance requires 4 to 8 h, it was assumed that a special trip would

Fig. 22. Utility functions for cost attributes.

**2.1 OUTPUT DEGRADN.
(CARDINAL RANKING)**

**2.2 OPERL FAILURE
(CARDINAL RANKING)**

**2.3 STARTUP FAILURE
(CARDINAL RANKING)**

Fig. 23. Utility functions for reliability attributes.

Fig. 24. Utility functions for maintenance attributes.

have to be made for the power system maintenance or an on-site overnight stay would be necessary to accomplish both electronics and power-system maintenance. This region was given a utility value of 0.6. If maintenance requires 8 to 16 h, an overnight stay or extra trip is required for the power-system maintenance alone, and the segment was given a utility value of 0.2. More than 16 h was given a utility value of zero.

4. Attribute 4--Efficiency (Fig. 25). This utility function for operating temperature efficiency started at the lowest efficiency (10%) with a zero utility value and increased linearly to a utility value of one at 30%. This is the full range of efficiencies and provides the maximum separation of utility values. In the utility function for the low source temperature efficiency a slope of zero for the system efficiency vs temperature curve was given a utility value of one. The utility value then decreases linearly with an increase in slope of the system efficiency temperature curve until a slope of $0.05\%/^{\circ}\text{C}$ is given a utility value of zero.

5. Attribute 5--Technical Risk (Fig. 26). Because the measure of the technical risk of obtaining reliability is the logarithm of the number of operational hours, the utility function was given a curve to reduce the utility value of the lower values. The value 5, or 100,000 h operating time, was given a utility value of one. The utility values then decrease to a utility value of zero at zero operational hours. The combination of the logarithm and nonlinear utility function was chosen so that the wide range of operating-hour experience could be given proportionally more credit than a semilogarithmic relationship. The utility function for technical risk of obtaining efficiency was chosen to be a linear relationship covering the range of values of the difference between the projected efficiency and the achieved efficiency. If the two efficiencies were the same, a utility value of one was assigned; and if they had an absolute difference of ten percent, a utility value of zero was assigned. Cardinal ranking was used to score the technical risk of obtaining cost.

6. Attribute 6--Size and Weight (Fig. 27). The shape of the size and weight utility functions is a step function with a slight slope on the step. The utility value decreases linearly to 0.8 at the maximum load for one helicopter. When two helicopters are required to carry the system, the utility value drops to 0.5. Another linear decrease occurs in the utility value (to 0.3) at the point where two helicopter trips are needed to transport the system. Above the two-trip point the utility value is zero. The reason for this shape

Fig. 25. Utility functions for efficiency attributes.

Fig. 26. Utility functions for technical risk attributes.

Fig. 27. Utility functions for size and weight attributes.

utility function is that if the power system can be carried in one load the RDPS weight has slight importance. However, if the system must be transported in two trips, this incurs considerably more cost and inconvenience, and it probably increases the on-site construction needed. There is a significant utility value decrease at the point where two trips are required to transport the system.

X. RESULTS

A. Rankings

The results of the rankings are given in Figs. 28-30. The details of the ranking for each of the six primary attributes are presented in Appendix C.

1. Case I--Teledyne Design with Strength Member Around Capsule. The top-ranked system was the Stirling engine, closely followed by the Sundstrand ORC and the Barber-Nichols/ORMAT ORC. The primary differences between the scores are that the Stirling had a larger score (lower cost) for the cost and the proven-efficiency portion of the ranking. This score is the result of the higher operating efficiency of the Stirling engine and the fact that a test Stirling engine has actually achieved the projected efficiency. The ORC systems have a larger score for reliability.

Fig. 28. Results of case I ranking. (STIRLING = Stirling, ORC = Organic Rankine Cycle, BN/O = Barber-Nichols Organic Rankine, TRW-OR = TRW-Organic Rankine, and BRTN = Brayton.)

Fig. 29. Results of case II rankings. (STIRLING = Stirling, ORC+TE = Organic Rankine Cycle + Thermoelectrics, and BRTN = Brayton.)

Fig. 30. Results of case III ranking. (STRLNG = Stirling, ORC+TE = Organic Rankine Cycle + Thermoelectrics, and BRTN = Brayton.)

2. Case II--WESF Capsule Inserted Directly in the Heat Accumulator Block.

The Stirling engine again is the top-ranked system, again slightly ahead of the ORC + TE system because of the lower cost and smaller differences between projected and achieved efficiencies. The lower cost for this case results from the lower engine cost for Stirling engines compared to the ORC + TE cost. The ORC + TE system again has a higher score for reliability.

3. Case III--Improved Capsule Design. The Brayton cycle was the top-ranked system, slightly ahead of the Stirling engine because of the perceived capability for higher reliability. Also the score for technical risk is higher because a test Brayton system has achieved the projected efficiency.

B. Sensitivities

The weight factors assigned to the attributes, and to some extent the attributes measures, are subjective. Therefore, some variation in the attribute weight factors should be run to demonstrate the sensitivity of the results to these subjective decisions. Two types of sensitivities were examined. First, the weight factors of two of the attributes were individually multiplied by 1.5,

and a new ranking was obtained with this change in attribute weight factors. The second sensitivity is a list of how the individual attribute contributions to the ranking change with a change in the attribute measure.

1. Sensitivity to Attribute Weight Factors. Two sensitivity runs were performed for the ranking. The first increased the weight factor of the reliability attribute by a factor of 1.5. The resulting change in attribute weight factors is shown in Fig. 31. The results for the three cases are given in Fig. 32-34.

For case I, the increase in the reliability weight factor changed the rankings by increasing the ranking of the ORC systems compared with the Stirling engine. However, the difference in the ranking between the Sundstrand, ORC Barber-Nichols/ORMAT-ORC and the Stirling engine is small.

The rankings for case II stayed the same. The difference between the Stirling and the ORC + TE was smaller than in the base case because of the higher perceived reliability of the ORC system.

The Brayton cycle remained the top-ranked system for case III. The difference between the Brayton cycle and Stirling engine increased because of the perceived higher reliability of the Brayton cycle system.

The second sensitivity run increased the weight factor of the technical risk attribute by a factor of 1.5. The resulting weight factors are given in Fig. 35 and the rankings are given in Figs. 36-38.

The rankings for the systems stayed the same for case I except the Barber-Nichols/ORMAT system exchanged with the Sundstrand-ORC. This exchange occurred because the difference between the achieved and projected efficiencies are large for the Sundstrand ORC and smaller for the Barber-Nichols/ORMAT ORC.

For Case II the increase in technical risk weight factors had no effect on the rankings. About the only change was to decrease the score for the ORC + TE system.

The case III ranking also did not change. The differences between the systems did increase because the difference between the projected and achieved efficiencies in the Stirling engine and ORC + TE is large.

2. Sensitivity to Attribute Measures. Tables VIII-X present the sensitivity of the score to the input values for the attribute measures. Also included in the table is the contribution for each attribute to the final ranking. Because the utility functions for 5.1--proven reliability, 3.2--man-

Fig. 31. Resulting weights for sensitivity analysis with reliability attribute weight multiplied by 1.5

Fig. 32. Case I ranking with reliability weight multiplied by 1.5. (BN/O = Barber-Nichols Organic Ranking, ORC = Organic Rankine Cycle, STRLNG = Stirling, TRW-OR = TRW organic Rankine, and BRTN = Brayton + Thermoelectric.)

Fig. 33. Case II ranking with reliability weight multiplied by 1.5. (BN/O = Barber-Nichols/ORMAT, ORC = Organic Rankine Cycle, STRLNG = Stirling, TRW-OR = TRW organic Rankine, BRTN = Brayton + Thermoelectric.)

Fig. 34. Case III ranking and reliability weight multiplied by 1.5. (BRTN = Brayton + Thermoelectric, STRLNG = Stirling, and ORC+TE = Organic Rankine cycle + Thermoelectric.)

Fig. 35. Resulting weights for sensitivity analysis with technical risk weight multiplied by 1.5.

Fig. 36. Case I ranking with technical risk weight multiplied by 1.5. STRLNG = Stirling, BN/O = Barber-Nichols/ORMAT, ORC = Organic Rankine Cycle, TRW-OR = TRW-Organic Rankine, and BRTN = Brayton Cycle.)

Fig. 37. Case II rankings for technical risk weight multiplied by 1.5.

Fig. 38. Case III ranking with technical risk weight multiplied by 1.5. (BRTN = Brayton, STIRLING = Stirling, and ORC+TE = Organic Rankine Cycle + Thermoelectric.)

TABLE VIII

CASE 1--SENSITIVITIES OF FIGURE-OF-MERIT CONTRIBUTIONS TO INPUT
VALUES AND TABULATION OF CONTRIBUTIONS TO FIGURE OF MERIT

Attribute	Units	Change in Attribute Measure Needed to Increase Contribution by 0.01	Attribute Weight Factor	Contribution to Ranking				
				Brayton	BN/O-ORC	ORC Sundstrand	TRW-ORC	Stirling
1.2 Lifetime Cost	\$/kWh	-0.65	0.160	0.000	0.076	0.119	0.115	0.121
2.2 Operational Failure	Cardinal ^c	0.8	0.125	0.100	0.112	0.112	0.112	0.100
2.1 Operational Degradation	Cardinal ^a	1.0	0.100	0.080	0.080	0.060	0.060	0.040
3.1 Maintenance Frequency	Visits/yr	-0.1	0.098	0.049	0.049	0.049	0.049	0.059
1.1 Development Cost	Cardinal ^a	1.0	0.096	0.058	0.058	0.038	0.019	0.058
4.1 Op. Temp. Eff.	%	2.3	0.066	0.050	0.014	0.039	0.047	0.036
1.3 Initial Cost	K\$	-90	0.064	0.011	0.036	0.049	0.048	0.050
5.1 Proven Reliab.	log (OP hr)	Indeterminate ^b	0.060	0.018	0.041	0.041	0.006	0.018
5.2 Proven Eff.	0%	-1.66	0.060	0.030	0.042	0.001	0.006	0.060
4.2 Low Temp. Eff.	%/°C	0.015	0.044	0.004	0.024	0.024	0.024	0.009
3.2 Man-hour Main.	hr	Indeterminate ^b	0.032	0.032	0.032	0.032	0.032	0.032
5.3 Cost Risk	Cardinal ^b	3.0	0.030	0.024	0.027	0.027	0.027	0.027
6.1 Weight	lbs	Indeterminate ^b	0.028	0.010	0.012	0.023	0.024	0.023
2.3 Start-up Failure	Cardinal ^b	4.0	0.025	0.022	0.022	0.022	0.022	0.020
6.2 Size	ft ³	Indeterminate ^b	0.012	0.006	0.010	0.011	0.011	0.011
Total				0.444	0.635	0.648	0.604	0.664

Note: Input values given in appendix B.

^aCardinal ranking used (scale 1-10).

^bDepends on attribute measure value (see utility functions).

TABLE IX
CASE II--SENSITIVITIES OF FIGURE-OF-MERIT CONTRIBUTIONS TO INPUT
VALUES AND TABULATION OF CONTRIBUTIONS TO FIGURE OF MERIT

Attribute	Units	Change in Attribute Measure Needed to Increase Contribution 0.01	Attribute Weight Factors	Contribution to Ranking		
				Brayton	ORC+TE	Stirling
1.2 Lifetime Cost	\$/kWh	-0.65	0.160	0.068	0.127	0.138
2.2 Operational Failure	Cardinal ^a	0.8	0.125	0.100	0.112	0.100
2.1 Operational Degradation	Cardinal ^a	1.0	0.100	0.080	0.050	0.040
3.1 Maintenance Frequency	Visits/yr	-0.1	0.098	0.049	0.049	0.059
1.1 Development Cost	Cardinal ^a	1.0	0.096	0.058	0.029	0.048
4.1 Op. Temp. Eff.	%	2.3	0.066	0.016	0.048	0.048
1.3 Initial Cost	K\$	-90	0.064	0.034	0.052	0.055
5.1 Proven Reliab.	log (OP hr)	Indeterminate ^b	0.060	0.018	0.041	0.018
5.2 Proven Eff.	0%	-1.66	0.060	0.042	0.008	0.045
4.2 Low Temp. Eff.	%/°C	0.015	0.044	0.004	0.026	0.016
3.2 Man-hour Main.	hr	Indeterminate ^b	0.032	0.032	0.032	0.032
5.3 Cost Risk	Cardinal ^a	3.0	0.030	0.024	0.024	0.027
6.1 Weight	lbs	Indeterminate ^b	0.028	0.013	0.024	0.024
2.3 Start-up Failure	Cardinal ^a	4.0	0.025	0.022	0.022	0.020
6.2 Size	ft ³	Indeterminate ^b	0.012	0.010	0.011	0.011
Total				0.527	0.656	0.683

Note: Input values given in appendix B.

^aCardinal ranking used (scale 1-10).

^bDepends on attribute measure value (see utility functions).

TABLE X
CASE III--SENSITIVITIES OF FIGURE-OF-MERIT CONTRIBUTIONS TO INPUT
VALUES AND TABULATION OF CONTRIBUTIONS TO FIGURE OF MERIT

Attribute	Units	Change in Attribute Measure Needed to Increase Contribution 0.01	Attribute Weight Factor	Contribution to Ranking		
				Brayton	ORC+TE	Stirling
1.2 Lifetime Cost	\$/kWh	-0.65	0.160	0.138	0.135	0.151
2.2 Operational Failure	Cardinal ^a	0.8	0.125	0.100	0.112	0.100
2.1 Operational Degradation	Cardinal ^a	1.0	0.100	0.080	0.050	0.040
3.1 Maintenance Frequency	Visits/yr	-0.1	0.098	0.049	0.049	0.059
1.1 Development Cost	Cardinal ^a	1.0	0.096	0.058	0.029	0.038
4.1 Op. Temp. Eff.	%	2.3	0.066	0.055	0.055	0.060
1.3 Initial Cost	K\$	-90	0.064	0.055	0.054	0.059
5.1 Proven Reliab.	log (OP hr)	Indeterminate ^b	0.060	0.018	0.018	0.018
5.2 Proven Eff.	Δ%	-1.66	0.060	0.060	0.010	0.034
4.2 Low Temp. Eff.	%/°C	0.015	0.044	0.009	0.026	0.027
3.2 Man-hour Main.	hr	Indeterminate ^b	0.032	0.032	0.032	0.032
5.3 Cost Risk	Cardinal ^a	3.0	0.030	0.024	0.021	0.024
6.1 Weight	lbs	Indeterminate ^b	0.028	0.024	0.024	0.024
2.3 Start-up Failure	Cardinal ^a	4.0	0.025	0.022	0.022	0.020
6.2 Size	ft ³	Indeterminate ^b	0.012	0.011	0.011	0.011
Total				0.735	0.648	0.697

Note: Input values given in appendix B

^aCardinal ranking used (scale 1-10).

^bDepends on attribute measure value (see utility functions).

hours required for maintenance, 6.1--weight, and 6.2--size are not linear, the change in the contribution produced by an attribute measure change depends on the value of the attribute measure. Therefore no values for the change in these were given in the tables.

Table XI presents the changes that occur in various attribute measures when the efficiency is multiplied by a factor of 1.1.

XI. CONCLUSIONS

A. System Needs

The capabilities of an RDPS can not really be determined until a detailed system design is done. Decisions are necessary on how to best integrate the heat source, heat transport, conversion system, and heat rejection system. The heat-rejection system probably represents the biggest uncertainty in an RDPS design. Design trade-offs between lower temperatures, the need for smaller size, and the use of forced convection will have a significant effect on the RDPS performance. These trade-offs and a more detailed examination of system losses will undoubtedly lower the performance and affect the reliability of many of the proposed systems. Thus what is reported here is an optimistic view of the RDPS performance.

TABLE XI
CHANGE IN ATTRIBUTE MEASURES WITH CHANGE IN EFFICIENCY

Attribute	System Characteristics	Attribute Measure Change Per 10% Efficiency Increase ^a
1.2 Lifetime Costs	Base system	- 0.44 \$/kWh
	With heat pipes	- 0.45 \$/kWh
	With thermoelectrics	- 0.47 \$/kWh
1.3 Initial Cost	Base system	- 29.2 k\$
	With heat pipes	- 29.5 k\$
	With thermoelectrics	- 30.7 k\$
6.1 Weight		- 690 lb
6.2 Size		16 ft ³

^aNew efficiency = 1.10 old efficiency.

The one integrated design done (Appendix A) shows some of the problems that appear with an integrated design. The use of forced convection requires an additional 100-W parasitic power loss and reduces the perceived reliability by introducing an additional piece of rotating machinery. However, the use of natural convection cooling would require a larger radiator area, larger pumping losses, and perhaps higher condenser temperature. This type of design has not been done for the other systems, it may be that reduced performance or additional parasitic losses will have to be accepted.

Another item that has not been included in the performance of the systems is the parasitic power loss resulting from electronic power controls. This was estimated to be about 100 W in Appendix A. Thus the useful output of all the systems should probably be reduced by this amount.

The heat source for use with dynamic systems also needs to be examined. A number of people have recommended that cast-melt strontium-fluoride be used for the fuel rather than the hot-pressed pellets now being used. The use of cast melt strontium-fluoride would increase the temperature and power of the fuel capsules, but new processing facilities might be necessary. Also the integration of heat transport and safety into the heat-source design requires some heat-source design effort.

B. Dynamic System Comparisons

The following comments can be made from the comparison and ranking of the systems:

1. The easiest and quickest dynamic system to integrate into the present Teledyne heat-source design would be an ORC system; however, efficiencies may be low.
2. The cascading of ORC systems and thermoelectrics is a promising system if the predicted thermoelectrics cost and performance is obtained.
3. The Stirling engine was ranked first in the two lower temperature cases (case I and II) and was a close second in the high temperature case III. Thus the Stirling engine appears to be the best all-round choice when the exact heat source capabilities are unclear. The primary uncertainty with Stirling systems is their reliability.
4. Brayton cycle systems are only reasonable for higher temperatures than can be achieved with present heat-source design. However, with the design of higher temperature capsules, Brayton cycle systems offer an efficient and potentially reliable system.

C. System Summaries

Because of the low maximum temperature of the ORC systems, they can readily be adapted to the existing Teledyne heat-source design--a hot-pressed strontium-fluoride capsule and an additional-strength member around the capsule. The ORC design presented in Appendix A would substitute a boiler consisting of tubes containing the working fluid in place of the thermoelectrics. A conventional ORC system would be used as the converter, and insulation would be put around the existing fins to reduce heat losses. However, this design predicts comparatively low efficiencies compared with the other systems. With the input of more development money, claims are made that a more efficient ORC system can be built. However, the companies with the most experience with the ORC predict the lowest system efficiencies.

Cascading thermoelectrics and ORC systems seem to be promising. The additional cost of the thermoelectrics is outweighed by the reduction in per-kilowatt-hour cost achieved with the higher efficiencies. The primary questions here are the cost of integrating the thermoelectrics into the design and the operation of the thermoelectrics in the high-temperature conditions.

Stirling engines have the highest efficiency of any of the systems for all three temperature ranges considered. The efficiencies predicted have been achieved, or are very close to being achieved, in test machines. If what is desired is a high efficiency system for whatever temperature heat source is designed, the Stirling engine is the best choice. The main uncertainty in the Stirling engine is in its reliability.

The following points tend to reduce the perceived reliability of a Stirling engine system compared to the other systems.

1. Stirling engines have sliding surfaces and require better seals between moving parts than the other systems.
2. The working fluid in the ORC and the Brayton cycle is used for lubrication. In the Stirling engine, lubricants other than the working fluid may be necessary.
3. The Stirling engine produces oscillations in a rod and generally linear alternators are used to convert these oscillations to electricity. These linear alternators have not had as much operating experience and generally have a lower efficiency than the conventional rotating alternators.

4. The Stirling engine does not have an integral pump or compressor to move the working fluid. If it is necessary to pump fluid through the heat-transport or heat-rejection system, an auxiliary pump must be included. Introduction of the pump would reduce reliability and increase parasitic losses for the system.

One method of eliminating the drawbacks presented in items 3 and 4 is to couple the output of the Stirling engine to a hydraulic system. This system would provide a rotating shaft for conventional alternators and for power takeoff requirements.

Questions on the reliability of Stirling engines are being addressed. Lifetime tests at NASA-Lewis on the proposed PNL Stirling engine radioisotope power source may answer some of these reliability questions.

If high temperature isotopic heat sources are designed, a Brayton cycle may be promising. A Brayton cycle machine has achieved good efficiencies at 750°C , and the design offers the probability of reliable operation. The problems with a Brayton cycle system is its sensitivity to the operating environment. The Brayton system has a single point design that assumes a single value for low temperature, compressor losses, turbine input temperature, etc. If one of these parameters is changed, the system efficiency drops. If tradeoffs must be made during the system design, the efficiency of the Brayton cycle will probably suffer more than the other systems because of this sensitivity to compressor loss and to the value of the heat rejection temperature.

D. Use of ^{238}Pu for Large Terrestrial Heat Sources

The use of ^{238}Pu for large terrestrial heat sources has few advantages and many disadvantages. On the advantage side, a ^{238}Pu source will be a smaller, lighter, heat source capable of operating at higher temperatures than ^{90}Sr . It will also have a longer half-life so power-flattening requirements are less. However, on the disadvantage side, it will be about twice as expensive per kilowatt-hour, even when the longer half-life allows operation for 17 years compared to the 5 years for ^{90}Sr . A 7500-W(t) ^{238}Pu source will contain more than a critical mass of ^{238}Pu mixture, so concerns such as safeguards and criticality must be addressed. Further the production of ^{238}Pu must compete in a reactor with the production of weapon materials, and because the chemical processing facilities for isolating ^{238}Pu are limited, it would only be possible to produce for civilian use two 7500-W(t) sources per year.

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APPENDIX A

BARBER-NICHOLS DESIGN FOR AN ORGANIC RANKINE RDPS USING EXISTING TELEDYNE 7500-(t) ⁹⁰Sr SOURCE

In Appendix A, a Rankine engine report prepared by R. E. Barber for Los Alamos National Laboratory has been reproduced in its entirety for the convenience of those who may wish to study the Barber-Nichols design in detail.

BARBER-NICHOLS

ENGINEERING

RANKINE ENGINE STUDY
FOR THE ASSESSMENT OF
DYNAMIC ENERGY CONVERSION SYSTEMS FOR
RADIOISOTOPES HEAT SOURCES

Subcontract No. 9-LC4-R8127-1

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1.0 INTRODUCTION

A study was made of a small organic Rankine engine to be coupled with the Strontium-90 isotope of Reference 1. This engine would yield a net power output of 710 watts at 28V DC. The system size would be approximately three feet in diameter by four feet high and weighs approximately 4,000 pounds, not including the isotope heat source. The estimated prototype development cost is approximately \$360,000 and the estimated 50-unit production cost is \$35,560 a unit.

2.0 SYSTEM DESCRIPTION

A Rankine engine was coupled with the Strontium-90 isotope heat source for a small 1 kW power supply. This system is made up of two major subassemblies, the isotope heat source and the Rankine engine which are described in more detail below. A sketch of the system is shown in Figure 1.

2.1 ISOTOPE HEAT SOURCE

This study considered the Strontium-90 heat source as described in Reference 1. It was specified in Reference 1 that the surface temperature of the heat source is 566°C (1050°F) and the design heat supply is 7200 watts. The Rankine engine was evaluated based on these values and the size of the isotope heat source as presented in Reference 1.

2.2 ORGANIC RANKINE CYCLE (ORC) SYSTEM

The ORC engine is a regenerative, hermetically sealed unit with a single stage axial flow turbine. Toluene is the working fluid with a maximum cycle temperature of 400°C (750°F) and maximum cycle pressure of 2.8 MPa (400 psia). A high speed, permanent magnet alternator is direct-coupled to the turbine and cooled by the working fluid. The high frequency AC power from the samarium cobalt alternator is converted to 28V DC output by means of a solid state rectifier. The upper operating temperature limit of toluene is 427°C (800°F) because of thermal decomposition. A maximum cycle temperature of 400°C (750°F) was chosen to lie below the decomposition limit.

The ORC utilizes a regenerator to improve cycle efficiency and an air-cooled condenser for engine cooling. Figure 2 shows a schematic diagram of the engine and is described in more detail in the following sections.

FIGURE 1

SKETCH OF POSSIBLE COMPONENT ARRANGEMENT

FIGURE 2
SCHEMATIC DRAWING OF RANKINE
POWER UNIT

3.0 GENERAL DESCRIPTION OF ORGANIC RANKINE CYCLE ENGINE

The organic Rankine cycle engine utilizes a cycle which was suggested in the 1950's by a Scottish engineer William Rankine and is the same as that used in the public utility steam generating plants. The heat input to the Rankine cycle is utilized to heat and vaporize the working fluid to obtain the desired conditions at the prime mover inlet. Work is extracted from the working fluid as it flows through the prime mover. The working fluid is condensed to liquid in the condenser and then pumped to a high pressure in the pump. When organic fluid is used as a working fluid, very often a regenerator is required as a thermal efficiency improvement device to transfer available heat at the prime mover exhaust to the liquid leaving the pump before it enters the boiler. As in the Carnot cycle, the thermal efficiency of the Rankine cycle increases and the minimum temperature decreased. Generally the maximum temperature allowable is limited by the heat source of the decomposition temperature of the fluid. The minimum temperature is generally chosen as low as possible and is limited by the available heat sink or heat rejection temperature.

There are many fluid properties which affect the desirability of a particular fluid for use in a particular application. Efficiency of any thermodynamic cycle increases when the cycle resembles the Carnot cycle as closely as possible. For a Rankine cycle this means:

- a) Heat addition at the temperature as close to the heat source as possible. For the isotope (isothermal heat source) the optimum fluid has a critical temperature above the heat source temperature, thereby maximizing the ratio of the boiling (isothermal heat addition) to preheating (non-isothermal heat addition).
- b) Highest possible efficiency of the prime mover (expander) to approach isentropic process.

- c) Heat rejection process as close to the heat sink temperature as possible. This means that working fluid should have a minimum amount of superheat at the expander exit or the regenerator exit to approach isothermal heat rejection.
- d) Maximum possible pump efficiency to approach isentropic compression.

There are some practical considerations to take into account:

- 1) It is desirable to keep the condensing pressure above 1 psia (7 kPa) to maintain a practical volume flow into the condenser.
- 2) For a small size engine (1 to 10 kW), it is desirable to use high molecular weight fluids to increase volume flow at the turbine exit and consequently, results in higher turbine efficiency.
- 3) Moderate pressure (less than 600 psia) at the maximum cycle temperature reduces pump work and heat exchangers/piping costs.
- 4) Large amounts of superheat at the turbine exhaust will require a regenerator and hence, increases the engine cost.

Other practical considerations such as the working fluid toxicity, flammability, freezing temperatures, lubrication and heat transfer properties influence the fluid selection for each particular application.

3.1 FLUID SELECTION AND CYCLE CONDITIONS

The heat source for this engine is an isothermal isotope as defined by Reference 1. The heat sink is the ambient air at a temperature of 27°C (80°F). Total heat input to the system is

7200 watts. As was mentioned above, maximum cycle efficiency is obtained when the highest temperature of the cycle approaches the heat source temperature. For this application, thermal stability of the working fluid is the limiting cycle temperature. Unfortunately only water and liquid metals are thermally stable at temperatures above 500°C. Since liquid metals have very high freezing temperatures along with the high condensing temperatures at moderately low pressures, only water was considered for the working fluid at the maximum temperature range. Most of the organic fluids are not thermally stable at the temperatures above 200°C. Working fluids, like halocarbons (Freons), were rejected as candidates for this system because they could not meet the performance requirement at their maximum allowable temperature. To obtain maximum performance, the maximum temperature in the cycle should not be less than 399°C (750°F) and this limits the choice of the working fluid to very few candidates. Methyl-benzene (toluene), methanol and some perfluorocarbons are thermally stable at a temperature of 425°C (797°F) and all of them were considered for this application.

3.1.1 Water as a Working Fluid

Water has excellent thermal stability and can be used at high cycle temperatures. Because of the desuperheating during the adiabatic expansion, steam at the turbine inlet has to be sufficiently superheated. This lowers the boiling pressure and hence, the average temperature of heat addition. Even with the high allowable cycle temperature for this small system (approximately 2 kW), water is a relatively poor choice. Because of the high heat of vaporization and small system size, the mass flow rate of the steam through the turbine is very low. This, together with the high adiabatic head (amount of energy available per unit mass during isentropic expansion), leads to a very low turbine efficiency. Cycle calculations using realistic performance predictions for major components in the system of this size show inferior overall cycle efficiency for the steam as the working fluid.

3.1.2 Perfluorocarbons as the Working Fluid

Based on the working fluid selection criteria, perfluoromethylcyclohexane (trade name FLUTEC PP2) was chosen as a candidate working fluid for this system. PP2 has a low freezing temperature, -38°C (-36°F), high molecular weight (350) and moderate condensing pressure at the ambient temperature. The primary problem with this fluid (common to every perfluorocarbon) is that the critical point is below the operating point; therefore, superheat at the turbine exit is very high and a relatively large regenerator is required. For a small size system, the large regenerator may not introduce a major cost size problem to the system. Cycle calculations show that systems using PP2 as the working fluid indicates that it could be used in this application; however, its primary advantage over toluene (the selected fluid) is that it is inflammable while it has several other disadvantages.

3.1.3 Toluene as the Working Fluid

Thermodynamically toluene is probably the superior candidate for the working fluid. The upper operating temperature for the toluene cycle could be as high as 425°C (800°F). The freezing temperature of the toluene is -95°C (-139°F) and the critical point is sufficiently high to provide good cycle efficiency. Flammability and toxicity of toluene are not considered a critical factor for this hermetically sealed system. The toluene cycle efficiency with a cycle temperature of 400°C (750°F), which is well below toluene decomposition temperature, is approximately equal to that of PP2 and the regenerator in the toluene system will be about half of the size of the regenerator for the PP2 system.

Considerable development work with toluene systems from 4 to 600 kW has been carried out over the past 10 to 15 years (i.e., Reference 2). This experience makes toluene the first choice for the working fluid in this application. The cycle conditions for this application are shown in Table 1.

TABLE 1
CYCLE CONDITIONS

	Temp. °F	Pressure psia	Volume ft ³ /lb	Enthalpy Btu/lb	Entropy Btu/lb-R
Boiler Exit	750.0	400.00	.2959	132.545	.12457
Turbine Inlet	749.8	395.00	.3003	132.545	.12479
Turbine Exit (Isentropic)	552.7	9.01	12.9928	46.984	.12479
Turbine Exit (Actual)	625.5	9.01	13.9480	82.919	.15905
Regenerator Exit (Vapor)	294.8	8.81	9.6722	-64.457	-.00128
Condenser Exit	200.0	8.81	.0197	-258.017	-.29115
Pump Exit (Isentropic)	201.7	435.00	.0197	-256.376	-.29115
Pump Exit (Actual)	212.2	435.00	.0198	-251.452	-.28376
Regenerator Exit (Liquid)	487.4	420.00	.0262	-104.075	-.10117
Turbine Flow Rate			103.9 lb/hr		
Pump Flow Rate			120.9 lb/hr		
Cycle Efficiency @ Turbine Shaft			17.8%		
Working Fluid			Toluene		
Turbine Efficiency			58.0%		
Pump Efficiency			25.0%		
Regenerator Effectiveness			80.0%		
Boiler Pressure Drop			20.0 psid		
Condenser Pressure Drop			.10 psid		
Regenerator Pressure Drop (Liquid)			15.0 psid		
Regenerator Pressure Drop (Vapor)			.10 psid		
Turbine Inlet Pressure Drop			5.0 psid		
Heat Input			24573.6 Btu/hr (7.2 kW)		
Bearing Flow Rate			17.0 lb/hr		

3.2 PREDICTED PERFORMANCE

The predicted performance of the Rankine engine is shown in Table 2. The turbine shaft power is 1.51 kW, the bearing loss (0.25 kW) and the pump power needed (0.23 kW) results in 1.03 kW into the alternator and an output of 0.91 kW at 28V DC.

The fan utilizes 100 watts and the controls also use 100 watts resulting in a net output of 710 watts from the Rankine engine. This is a substantial increase over the thermal electric output of 500 watts from Reference 1. However, this is less than desired because of the parasitic losses which are a big percentage of such a small machine.

3.3 ORC ENGINE CONFIGURATION

The ORC engine consists of six major components. These are:

- 1) Turbine-alternator-pump unit
- 2) Boiler
- 3) Regenerator
- 4) Condenser
- 5) Isotope cooling fins
- 6) Controls

Each component is described in more detail below.

3.3.1 Turbine-Alternator-Pump (TAP)

The hermetically sealed TAP assembly consists of a single-stage axial flow turbine, a Samarium-Cobalt permanent magnet alternator and a centrifugal pitot-type feed pump. These items are mounted on a single shaft rotating at 65,000 rpm and supported by fluid film hydrodynamic bearings. A similar unit is shown in the photograph of Figure 3.

The turbine is a partial admission, axial flow, impulse design with a pitch diameter of 2.7 inches, 100 blades, 12% admission,

TABLE 2
CYCLE PERFORMANCE

Project Component Efficiencies

Turbine Efficiency	58%
Alternator/Rectifier Efficiency	88%
Pump Efficiency	25%
Bearing Power Loss	.25 kW _m *
Condenser Fan Power	.10 kW _e **
Control Power	.10 kW ₃

Cycle Performance

Turbine Shaft Power	1.51 kW _m
Pump Power	.23 kW _m
Electrical Output of Rectifier	.91 kW _e
Electrical Output of System	.71 kW _e

* kW_m = Mechanical Power

** kW_e = Electrical Power

CUTAWAY MODEL OF TURBO ALTERNATOR

FIGURE 3

tip speed of 745 fps and a projected efficiency of 62.3%. The cycle calculations were made with a turbine efficiency of 58% to be conservative; therefore, an increase in cycle efficiency of seven may be obtained with development.

The alternator will employ six Samarium-Cobalt magnets of the rotor with a nine-tooth copper-wound, laminated iron stator. The alternator will produce 3-phase, 3250 Hz, 600 volt AC power at 65,000 rpm. The high frequency power is rectified to 28V DC at an overall alternator/rectifier efficiency of 88%.

The feed pump is a centrifugal pitot-type with a predicted 27% efficiency. The pump is designed to supply full system flow of 54.4 kg/hr (120 lb/hr) and peak pressure of 3.0 MPa (435 psi).

The entire TAP rotating unit is supported on two hydrodynamic tilting pad bearings, each comprised of five radial shoes with the working fluid (toluene) functioning as the lubricant. There are two hydrodynamic flat-pad thrust bearings with self-aligning supports of tin-antimony babbitt plating.

3.2.2 Boiler

The boiler consists of 0.125 inch ID tubes of stainless steel brazed into the base of the copper cooling fins. The tubes will make four to six wraps around the isotope heat source resulting in 20 to 30 feet of tubing. The average heat transfer coefficient in the tube is approximately $300 \text{ Btu/Ft}^2\text{-}^\circ\text{F}$ when combining the preheating, boiling and superheater coefficients.

3.3.3 Regenerator

The regenerator will be a conventional fin-tube cross counter-flow configuration of stainless steel tubing with brazed-on copper fins. The total heat transfer area is 29 ft^2 .

3.3.4 Condenser

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The air-cooled condenser consists of finned aluminum tubes in parallel, arranged in three concentric layers. Cooling air is drawn in by the motor-driven fan and exhausted upward across the condenser tubes.

A hotwell is located below the condenser liquid outlet. The hotwell liquid level is controlled by a float valve with the activated bypass shown in Figure 2.

3.3.5 Isotope Cooling Fins

Cooling fins are envisioned outboard of the boiler tubing. These fins are normally covered by insulation so that they will operate at near fluid temperature. In the event that the Rankine engine fails overtemperaturing the toluene, the insulation falls away and the cooling fins provide rejection of the 7700 watts of the isotope by natural convection. The cooling fins were sized to connect the 7200 watts at a root temperature of less than 700°F. The resultant fins are approximately 2.5 inches long based on the analysis presented in Reference 1. The fins would be constructed of nickel-plated copper.

3.3.6 Controls

An electronic control package will be provided to insure safety of the unit and provide electrical load control for the alternator. The combined load/voltage/TAP speed control will be accomplished by a solid state switching of excess power into an air-cooled load bank. This switch will handle the normal situation of varying loads of less than the peak power generation capability. If the engine is overloaded it will slow down below rated speed thereby dropping voltage and decreasing load.

3.4 SIZE AND WEIGHT

The total system size is shown in Figure 1 as 34 inches in

diameter and 48 inches high. The Rankine engine component size and estimated weight is shown in Table 3. The total Rankine engine weight is 3906 pounds. This weight could be cut to 2000 pounds by making the cooling fins of the more expensive titanium.

3.5 RELIABILITY AND MAINTENANCE

The Rankine engine is a hermetic sealed machine very similar to an air conditioning machine and hence, has the potential for long maintenance-free operation as demonstrated by hermetic air conditioners. Few Rankine engines have been put into operation for long periods of time; however, Barber-Nichols has accumulated over 30,000 hours of operation on a group of ten solar powered Rankine engines and a firm in Israel has demonstrated single unit operational times in excess of 40,000 hours on small (1 kW) gas-fired Rankine engines.

The critical components in the proposed system are:

- 1) TAP bearings and alternator
- 2) Fan motor and bearings
- 3) Condenser surfaces
- 4) Rankine plumbing leaks
- 5) Electrical controls
- 6) Rankine working fluid

Each of these is addressed below.

- 1) TAP bearings and alternator. The TAP bearings are fluid film hydrodynamic. This type of bearing has no metal-to-metal contact except during startup and shutdown, as the rotating shaft is supported by fluid films. Therefore, the bearing life is infinite for the present application since the machine runs continuously. However, the bearing life will be limited by unpredictable contamination in the fluid or a momentary loss of fluid. The alternator also has an infinite life prediction,

TABLE 3
COMPONENT SIZE AND WEIGHT

	Material	Size	Weight/lbs
TAP	Stainless Steel	9" dia. x 12"	40
Boiler	Stainless Steel/Copper	20" ID x 21" OD x 21" high	195
*Boiler	Copper	21" ID x 26" OD x 21" high	3000
Cooling Fins			
Regenerator	Stainless Steel	8" dia. x 12"	151
Condenser	Stainless Steel/Alum.	20" ID x 25" OD x 3" thick	220
Hotwell		8" dia. x 10" long	50
Insulation		3" thick	100
Miscellaneous			150
			3906 lbs

* Fin weight could be decreased to approximately 1000 pounds by use of titanium in place of the nickel-plated copper.

with the life being limited by the unpredictable life of the insulation of the field windings. Field tests and the resultant field failures will be the basis of a maintenance program to replace and refurbish the TAP before failure.

- 2) Fan motor and bearings. The fractional horsepower fan motor and bearings are expected to have a three to four year life and must be replaced to prevent failure.
- 3) Condenser surfaces. The condenser surfaces will be affected by atmospheric conditions and will require cleaning as in a conventional air conditioning machine.
- 4) Rankine plumbing leaks. Unpredictable plumbing leaks will cause shutdown of the unit and require maintenance. Care in manufacture and shipping must be taken to minimize these problems.
- 5) Electrical controls. The electrical controls will be solid state and will have a multi-year projected life. However, field tests will be required to determine the maintenance requirements.
- 6) Rankine working fluid. The fluid life will be greater than two years and field tests will determine the time period at which the system should be flushed and re-filled with new fluid.

It is anticipated that after field experience, a schedule will be determined for scheduled annual maintenance which will include:

- Annually cleaning the condenser surface
- Leak check
- Control checkout

- As determined by field tests, TAP replacement
- System flush and fluid replacement
- Fan motor replacement
- Electronic part replacement

3.6 COST

A cost estimate was made based on 50 units over a two-year period. The results are presented in Table 4. It is estimated that the 50-unit cost would be \$35,560/unit.

TABLE 4
RANKINE ENGINE COSTS
Assumes 50 Units over 2-Year Period
All Costs in 1984 Dollars

Turbine-Alternator-Pump (TAP)	\$ 6,750	
Boiler (Includes Cooling Fins)	8,500	
Regenerator	3,750	
Condenser/Motor/Fan	3,650	
Controls/Rectifier	1,200	
Assembly	2,800	
Checkout Test	<u>800</u>	
		\$27,450/unit
Tooling for 25-Units/Year	75,000	1,500/Unit
Project Management-Billing		<u>6,610/Unit</u>
		\$35,560/Unit

4.0 RECOMMENDED DEVELOPMENT APPROACH

The recommended development approach is:

- Phase I: Preliminary design of Rankine system. Make trade-offs of system parameters to optimize design. Predict performance, prototype cost and 100-unit cost, and provide final design specifications.
- Phase II: Final design. Complete design in accordance with specifications of Phase I. Estimate prototype cost and 100 unit cost.
- Phase III: Prototype fabrication and test. Build and test prototype. Analyze data, and modify design and hardware as required. Estimate 100-unit cost.
- Phase IV: Prototype test with isotope heat source. Test power system, and modify design and hardware as required. Estimate 100-unit cost.
- Phase V: Manufacture 100 units. Estimated costs for this program are:

Phase I	\$ 50,000	3 months ARO
Phase II	85,000	4 months ARO
Phase III	175,000	6 months ARO
Phase IV	50,000	3 months ARO
	<hr/> \$360,000	

5.0 REFERENCES

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APPENDIX B

SPECIFIC VALUES FOR MULTIATTRIBUTE RANKING

Tables B-I through B-V give the specific values used in the ranking analysis for case I, Tables B-VI through B-VIII give the ranking analysis for Case II, and Tables B-IX through B-XI give the ranking analysis for case III. Each table is broken into two parts: the first shows the decision analysis parameters (the input numbers to the MADT code), and the second shows the system characteristics.

TABLE B-I
CASE I: BRAYTON CYCLE

Decision Analysis Parameters

1.1 Development costs	<u>6 (ranking)</u>
1.2 Lifetime costs	<u>16.5 \$/kWh</u>
1.3 Initial costs	<u>1084.8 k\$</u>
2.1 Probability of output degradation	<u>8 (ranking)</u>
2.2 Probability of operational failure	<u>8 (ranking)</u>
2.3 Probability of startup & transportation failure	<u>8 (ranking)</u>
3.1 Frequency of maintenance	<u>0.5 visits/yr</u>
3.2 Man hours required for maintenance	<u>2 h</u>
4.1 Efficiency at operating temperatures	<u>10%</u>
4.2 Efficiency at lower than operating temperature	<u>0.045%/°C</u>
5.1 Technical risk in obtaining reliability	<u>3 log (h)</u>
5.2 Technical risk in obtaining efficiency	<u>5%</u>
5.3 Technical risk in obtaining cost	<u>8 (ranking)</u>
6.1 Weight	<u>18 000 lb</u>
6.2 Size	<u>420 ft³</u>

System Characteristics

System: Based On Mini-BRU, Xe-He mixture working fluid molecular wt 83.8

Heat transport: Heat pipes

Temperatures: Heat accumulator block	640°C BOL	610°C EOL
Top of heat pipe:	600°C BOL	570°C EOL
Turbine inlet temperature	575°C BOL	545°C EOL

Efficiencies:^a 11.4% BOL^b 10% EOL

BOL heat source size needed to achieve 1500 W(e) at EOL: 18 000 W(t)

Engine cost: \$75 000

Slope of efficiency vs temperature curve:^b 0.045%/°C

Operational hours:^c 1 000

Actual efficiency achieved:^d 5%

Size & weight based on Teledyne shipping configuration^e

^a Assuming 95% rectifier efficiency.

^b Ref. 2.

^c See Niel Footnote, p. 51.

^d Based on spread of efficiency vs temperature curve in Ref. 2.

^e See Vogt footnote, p. 2.

CASE I: ORGANIC RANKINE--BARBER NICHOLS/ORMAT

1.1	Development costs	<u>6 (ranking)</u>
1.2	Lifetime costs	<u>12.21 \$/kWh</u>
1.3	Initial costs	<u>802.0 k\$</u>
2.1	Probability of output degradation	<u>8 (ranking)</u>
2.2	Probability of operational failure	<u>9 (ranking)</u>
2.3	Probability of startup & transportation failure	<u>9 (ranking)</u>
3.1	Frequency of maintenance	<u>0.5 visits/yr</u>
3.2	Man hours required for maintenance	<u>2 h</u>
4.1	Efficiency at operating temperature	<u>14.3%</u>
4.2	Efficiency at lower than operating temperature	<u>0.023 %/°C</u>
5.1	Technical risk in obtaining reliability	<u>4.2 log (h)</u>
5.2	Technical risk in obtaining efficiency	<u>3%</u>
5.3	Technical risk in obtaining cost	<u>9 (ranking)</u>
6.1	Weight	<u>13 000 lb</u>
6.2	Size	<u>290 ft³</u>

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TABLE B-III

CASE I: ORGANIC RANKINE--SUNDSTRAND

Decision Analysis Parameters

1.1 Development costs	4 (ranking)
1.2 Lifetime costs	10.07 \$/kWh
1.3 Initial costs	661.8 k\$
2.1 Probability of output degradation	6 (ranking)
2.2 Probability of operational failure	9 (ranking)
2.3 Probability of startup & transportation failure	9 (ranking)
3.1 Frequency of maintenance	0.5 visits/yr
3.2 Man hours required for maintenance	3 h
4.1 Efficiency at operating temperature	21.7%
4.2 Efficiency at lower than operating temperature	0.023%/°C
5.1 Technical risk in obtaining reliability	4.2 log (h)
5.2 Technical risk in obtaining efficiency	9.8%
5.3 Technical risk in obtaining cost	9 (ranking)
6.1 Weight	8260 lb
6.2 Size	193 ft ³

System Characteristics

System: Sundstrand concept, toluene working fluid

Heat transport: Pumped fluids

Temperatures: Heat accumulator block 640°C BOL 620°C EOL
Boiler 400°C BOL 370°C EOL

Efficiencies:^a 22.4%^a BOL 21.7% EOL

BOL heat source size needed to achieve 1500 W(e) at EOL: 8260 W(t)

Engine cost: \$75 000

Slope of efficiency vs temperature curve:^c 0.023%/°C

Operational hours: 15 000

Actual efficiency achieved: 12.6%

Size and weight based on Teledyne shipping configuration^d

^aAssumes 95% rectifier efficiency.

^bSee Lacy footnote, p. 22.

^cR. E. Barber, "Potential of Rankine Engines to Produce Power From Waste Heat Systems," Ninth Intersociety Energy Conversion Engineering Conference, San Francisco, California, August 1974.

^dSee Vogt footnote, p. 2.

TABLE B-IV

CASE I: ORGANIC RANKINE--TRW

Decision Analysis Parameters

1.1 Development costs	<u>2 (ranking)</u>
1.2 Lifetime costs	<u>\$10.26 \$/kWh</u>
1.3 Initial costs	<u>674.2 k\$</u>
2.1 Probability of output degradation	<u>6 (ranking)</u>
2.2 Probability of operational failure	<u>9 (ranking)</u>
2.3 Probability of startup & transportation failure	<u>9 (ranking)</u>
3.1 Frequency of maintenance	<u>0.5 visits/yr</u>
3.2 Man hours required for maintenance	<u>2 h</u>
4.1 Efficiency at operating temperature	<u>24.3%</u>
4.2 Efficiency at lower than operating temperature	<u>0.023%/°C</u>
5.1 Technical risk in obtaining reliability	<u>2 log (h)</u>
5.2 Technical risk in obtaining efficiency	<u>9%</u>
5.3 Technical risk in obtaining cost	<u>9 (ranking)</u>
6.1 Weight	<u>6475 lb</u>
6.2 Size	<u>175 ft³</u>

System Characteristics

System: TRW concept, toluene working fluid

Heat transport: Pumper fluids

Temperatures: Heat accumulator block 640°C BOL 610°C EOL
Boiler 400°C BOL 370°C EOL

Efficiencies:^a 25% BOL 24.3% EOL

BOL heat source size needed to achieve 1500 W(e) at EOL: 7375 W(t)

Engine cost: \$125 000^b

Slope of efficiency vs temperature curve:^c 0.023%/°C

Operational hours:^d 100

Actual efficiency achieved:^d 16%

Size and weight based on Teledyne shipping configuration^e and TRW information^f

^aReg. 2.

^b"Organic Rankine Cycle Economic Radioisotope Plus Thermodynamic Generator Study," Development Plan d by TRW Inc., Redondo Beach, California, January 1974.

^cR.E. Barber, "Potential of Rankine Engines to Produce Power From Waste Heat Streams," Ninth Intersociety Energy Conversion Engineering Conference, San Francisco, California, August 1974.

^dJ. E. Boretz and R. Maresko, "Subsea Isotope Power System (STPS)," Proceedings of the 15th Intersociety Energy Conversion Engineering Conference, Seattle, Washington, August 18-22, 1980.

^eSee Vogt footnote, p. 2.

^fSee Boretz footnote, p. 32.

TABLE B-V
CASE I: STIRLING ENGINE

Decision Analysis Parameters

1.1 Development costs	<u>6 (ranking)</u>
1.2 Lifetime costs	<u>9.94 \$/kWh</u>
1.3 Initial costs	<u>652.8 k\$</u>
2.1 Probability of output degradation	<u>4 (ranking)</u>
2.2 Probability of operational failure	<u>8 (ranking)</u>
2.3 Probability of startup & transportation failure	<u>8 (ranking)</u>
3.1 Frequency of maintenance	<u>0.4 visits/yr</u>
3.2 Man hours required for maintenance	<u>3 h</u>
4.1 Efficiency at operating temperature	<u>21%</u>
4.2 Efficiency at lower than operating temperature	<u>0.04%/°C</u>
5.1 Technical risk in obtaining reliability	<u>2 log (h)</u>
5.2 Technical risk in obtaining efficiency	<u>0%</u>
5.3 Technical risk in obtaining cost	<u>9 (ranking)</u>
6.1 Weight	<u>8800 lb</u>
6.2 Size	<u>200 ft³</u>

System Characteristics

System: Sunpower RE-1000, Helium working fluid, free-piston design

Heat transport: Heat pipes

Temperatures:	Heat accumulator block	640°C BOL	610°C EOL
	Top of heat pipe	600°C BOL	570°C EOL
	Engine head	565°C BOL	535°C EOL

Efficiencies:^{a,b} 22% BOL 21% EOL

BOL heat source size needed to achieve 1500 W(e) at EOL: 8535 W(t)

Engine cost: \$50 000

Slope of efficiency vs temperature curve:^b 0.040%/°C

Operational hours:^c 1000

Actual efficiency achieved:^b 21%

Size and weight based on Teledyne shipping configuration^d

^a80% efficient alternator design; 95% efficient rectifier.

^bSee Berchowitz footnote, p. 29.

^cRef. 16.

^dSee Vogt footnote p. 2.

TABLE B-VI

CASE II: BRAYTON CYCLE

DECISION ANALYSIS PARAMETERS

1.1 Development costs	<u>6 (ranking)</u>
1.2 Lifetime costs	<u>12.5 \$/kWh</u>
1.3 Initial costs	<u>826.8 k\$</u>
2.1 Probability of output degradation	<u>8 (ranking)</u>
2.2 Probability of operational failure	<u>8 (ranking)</u>
2.3 Probability of startup & transportation failure	<u>9 (ranking)</u>
3.1 Frequency of maintenance	<u>0.5 visits/yr</u>
3.2 Man hours required for maintenance	<u>2 h</u>
4.1 Efficiency at operating temperature	<u>15%</u>
4.2 Efficiency at lower than operating temperature	<u>0.045%/°C</u>
5.1 Technical risk in obtaining reliability	<u>3 log (h)</u>
5.2 Technical risk in obtaining efficiency	<u>3%</u>
5.3 Technical risk in obtaining cost	<u>8 (ranking)</u>
6.1 Weight	<u>12 000 lb</u>
6.2 Size	<u>280 ft³</u>

System Characteristics

System: Based on Mini-Bru, xenon-hellium mixture working fluid molecular wt 83.8

Heat transport: Heat pipes

Temperatures: Heat accumulator block 745°C BOL 715°C EOL

Top of heat pipe: 705°C BOL 675°C EOL

Turbine inlet temperature: 650°C BOL 620°C EOL

Efficiencies:^{a,b} 17%^a BOL 15%^a EOL

BOL heat source size needed to achieve 1500 W(e) at EOL: 12 000 W(t)

Engine cost: \$75 000

Slope of efficiency vs temperature curve:^b 0.045:%/°C

Operational hours:^c 1000 h

Actual efficiency achieved:^d 12%

Size and weight based on Teledyne shipping configuration^e

^aRectifier efficiency 95%.

^bRef. 2.

^cSee Neil footnote, p. 51.

^dBased on spread of efficiency vs temperature curve in Ref. 2.

^eSee Vogt footnote, p. 2.

TABLE B-VII

CASE II: ORGANIC RANKINE PLUS THERMOELECTRICS

Decision Analysis Parameters

1.1 Development costs	<u>3 (ranking)</u>
1.2 Lifetime costs	<u>9.63 \$/kWh</u>
1.3 Initial costs	<u>632.6 k\$</u>
2.1 Probability of output degradation	<u>5 (ranking)</u>
2.2 Probability of operational failure	<u>9 (ranking)</u>
2.3 Probability of startup & transportation failure	<u>9 (ranking)</u>
3.1 Frequency of maintenance	<u>0.5 visits/yr</u>
3.2 Man hours required for maintenance	<u>3 h</u>
4.1 Efficiency at operating temperature	<u>24.6%</u>
4.2 Efficiency at lower than operating temperature	<u>0.020%/°C</u>
5.1 Technical risk in obtaining reliability	<u>4.2 log (h)</u>
5.2 Technical risk in obtaining efficiency	<u>8.7%</u>
5.3 Technical risk in obtaining cost	<u>8 (ranking)</u>
6.1 Weight	<u>7280 lb</u>
6.2 Size	<u>170 ft³</u>

System Characteristics

System: Sundstrand ORC, toluene working fluid, TAGS/Lead-telluride thermoelectrics

Heat transport: Thermoelectrics--heat conductor

ORC--Pumped fluids

Temperature: Heat accumulator block (hot shoe) 745°C BOL 715°C EOL
 Boiler (cold shoe) 400°C BOL 370°C EOL

Efficiencies:^a BOL ORC^b 21.5% TE 3.8% Total 25.3%

EOL ORC^b 20.8% TE 3.8% Total 24.6%

BOL heat source size needed to achieve 1500 W(e) at EOL: 7280 W(t)

Engine cost: \$75 000

TE cost: \$12 400

Slope of efficiency vs temperature curve:^c 0.020%/°C

Operational hours (ORC):^b 15 000

Actual efficiency achieved: ORC^c 12.1% TE 3.8% Total 15.9%

Size and weight based on Teledyne shipping configuration^c

^aSee Lacy footnote, p. 22.

^bOverall efficiency for energy input into hot shoe of thermoelectrics.

^cSee Vogt footnote, p. 2.

TABLE B-VIII

CASE II: STIRLING ENGINE

Decision Analysis Parameters

1.1 Development costs	5 (ranking)
1.2 Lifetime costs	9.08 \$/kWh
1.3 Initial costs	596.7 k\$
2.1 Probability of output degradation	4 (ranking)
2.2 Probability of operational failure	8 (ranking)
2.3 Probability of startup & transportation failure	8 (ranking)
3.1 Frequency of maintenance	0.4 visits/yr
3.2 Man hours required for maintenance	3 h
4.1 Efficiency at operating temperature	24.5%
4.2 Efficiency at lower than operating temperature	0.030%/°C
5.1 Technical risk in obtaining reliability	3 log (h)
5.2 Technical risk in obtaining efficiency	2.5%
5.3 Technical risk in obtaining cost	9 (ranking)
6.1 Weight	7315 lb
6.2 Size	170 ft ³

System Characteristics

System: Sunpower design, RE-1000 helium working fluid

Heat transport: Direct conduction

Temperatures: Heat accumulator block 745°C BOL 715°C EOL
 Engine heat^{b,c} 600°C BOL 570°C EOL

Efficiency: *^{B-11} 25.6% BOL 24.5% EOL

BOL heat source size needed to achieve 1500 W(e) at EOL: 7315 W(t)

Engine Cost: \$50 000

Slope of efficiency vs temperature curve:^c 0.03%/°C

Operational hours:^d 1000

Actual efficiency achieved:^c 22%

Size and weight based on Teledyne shipping configuration^e

^aSee Wheelright footnote, p. 2.

^b89% rectifier efficiency; 90% alternator efficiency.

^cSee Berchowitz footnote, p. 29.

^dSee Ref. 16.

^eSee Vogt footnote, p. 2.

TABLE B-IX

CASE III: BRAYTON CYCLE

Decision Analysis Parameters

1.1 Development costs	<u>6 (ranking)</u>
1.2 Lifetime costs	<u>9.12 \$/kWh</u>
1.3 Initial costs	<u>599.3 k\$</u>
2.1 Probability of output degradation	<u>8 (ranking)</u>
2.2 Probability of operational failure	<u>8 (ranking)</u>
2.3 Probability of startup & transportation failure	<u>9 (ranking)</u>
3.1 Frequency of maintenance	<u>0.5 visits/yr</u>
3.2 Man hours required for maintenance	<u>2 h</u>
4.1 Efficiency at operating temperature	<u>26.7%</u>
4.2 Efficiency at lower than operating temperature	<u>0.04%/°C</u>
5.1 Technical risk in obtaining reliability	<u>3 log (h)</u>
5.2 Technical risk in obtaining efficiency	<u>0%</u>
5.3 Technical risk in obtaining cost	<u>8 (ranking)</u>
6.1 Weight	<u>6700 lb</u>
6.2 Size	<u>156 ft³</u>

System Characteristics

System: Based on Min-Bru, xenon-helium mixture working fluid molecular wt 83.8

Heat transport: Heat pipes

Temperatures: Heat accumulator block	945°C BOL	915°C EOL
Top of heat pipe	905°C BOL	875°C EOL
Turbine inlet temperature	850°C BOL	820°C EOL

Efficiencies:^a 27.6%^b BOL 26.7%^b EOL

BOL heat source size needed to achieve 1500 W(e) at EOL: 6710 W(t)

Engine cost: \$75 000

Slope of efficiency vs temperature curve:^a 0.04 %/°C

Operational hours:^a 1000

Actual efficiency achieved: 27.6% BOL 26.7% EOL

Size and weight based on Teledyne shipping configuration^c

^aRef. 14

^b95% rectifier efficiency.

^cSee Vogt footnote, p. 2.

TABLE B-X

CASE III: ORGANIC RANKINE PLUS THERMOELECTRICS

Decision Analysis Parameters

1.1 Development costs	<u>3 (ranking)</u>
1.2 Lifetime costs	<u>9.24 \$/kWh</u>
1.3 Initial costs	<u>607.4 k\$</u>
2.1 Probability of output degradation	<u>5 (ranking)</u>
2.2 Probability of operational failure	<u>9 (ranking)</u>
2.3 Probability of startup & transportation failure	<u>9 (ranking)</u>
3.1 Frequency of maintenance	<u>0.5 visits/yr</u>
3.2 Man hours required for maintenance	<u>3 h</u>
4.1 Efficiency at operating temperature	<u>26.7%</u>
4.2 Efficiency at lower than operating temperature	<u>0.02%/°C</u>
5.1 Technical risk in obtaining reliability	<u>3 log (h)</u>
5.2 Technical risk in obtaining efficiency	<u>8.4%</u>
5.3 Technical risk in obtaining cost	<u>7 (ranking)</u>
6.1 Weight	<u>6710 lb</u>
6.2 Size	<u>157 ft³</u>

System Characteristics

System: Sundstrand ORC, toluene working fluid, SiGe (Gap) thermoelectrics

Heat Transport: Thermoelectrics--heat conduction

ORC--pumped fluids

Temperatures: Heat accumulator block (hot shoe) 945°C BOL 915°C EOL

Boiler (cold shoe) 400°C BOL 370°C EOL

Efficiencies:^a BOL ORC^b 20.9% TE 6.5% Total 27.4%

EOL ORC^b 20.2% TE 6.5% Total 26.7%

BOL heat source size needed to achieve 1500 W(e) at EOL: 6710 W(t)

Engine cost: \$75 000

TE cost: \$11 400

Slope of efficiency vs temperature curve:^b 0.02%/°C

Operational hours (ORC):^a 15 000

Actual efficiency achieved: ORC^b 11.8% TE 6.5% Total 18.3%

Size and weight based on Teledyne shipping configuration^c

^aSee Lacy footnote, p. 22.

^bOverall efficiency for energy input into hot shoe of thermoelectrics.

^cSee Vogt footnote, p. 2.

TABLE B-XI
CASE III: STIRLING ENGINE

Decision Analysis Parameters

1.1 Development costs	<u>4 (ranking)</u>
1.2 Lifetime costs	<u>8.44 \$/kWh</u>
1.3 Initial costs	<u>554.8 k\$</u>
2.1 Probability of output degradation	<u>4 (ranking)</u>
2.2 Probability of operational failure	<u>8 (ranking)</u>
2.3 Probability of startup & transportation failure	<u>8 (ranking)</u>
3.1 Frequency of maintenance	<u>0.4 visits/yr</u>
3.2 Man hours required for maintenance	<u>3 h</u>
4.1 Efficiency at operating temperature	<u>28.3%</u>
4.2 Efficiency at lower than operating temperature	<u>0.019%/°C</u>
5.1 Technical risk in obtaining reliability	<u>3 log (h)</u>
5.2 Technical risk in obtaining efficiency	<u>4.3%</u>
5.3 Technical risk in obtaining cost	<u>8 (ranking)</u>
6.1 Weight	<u>6330 lb</u>
6.2 Size	<u>148 ft³</u>

System Characteristics

System: SIPS concept helium working fluid

Heat transport: Direct conduction

Temperatures: Heat accumulator block 945°C BOL 915°C EOL

Engine head:^a 900°C BOL 970°C EOL

Efficiencies:^b 28.9% BOL 28.3% EOL

BOL heat source size needed to achieve 1500 W(e) at EOL: 6330 W(t)

Engine cost: \$50 000

Slope of efficiency vs temperature curve:² 0.019%/°C

Operational hours: 1000

Actual efficiency achieved: 24%

Size and weight based on Teledyne shipping configuration^c

^aSee Wheelright footnote, p. 2.

^bRef. 15.

^cSee Vogt footnote, p. 2.

APPENDIX C

DETAILS OF ATTRIBUTES CONTRIBUTIONS

The attributes covered for cases I, II, and III in the figures of Appendix C are costs, reliability, maintenance, efficiency, technical risk, and size/weight.

The meaning of system abbreviations used in the figures are given below.

STRLNG = Stirling

ORC = Organic Rankine Cycle

BN/O = Barber-Nichols Organic Rankine

TRW-OR = TRW-Organic Rankine

BRTN = Brayton

ORC+TE = Organic Rankine Cycle + Thermoelectric

Fig. C-1. Details of case I cost attribute contributions.

Fig. C-2. Details of case I reliability attribute contributions.

Fig. C-3. Details of case I maintenance attribute contributions.

Fig. C-4. Details of case I efficiency attribute contributions.

Fig. C-5. Details of case I technical risk attribute contributions.

Fig. C-6. Details of case I size/weight attribute contributions.

Fig. C-7. Details of case II cost attribute contributions.

Fig. C-8. Details of case II reliability attribute contributions.

Fig. C-9. Details of case II maintenance attribute contributions.

Fig. C-10. Details of case II efficiency attribute contributions.

Fig. C-11. Details of case II technical risk attribute contributions.

Fig. C-12. Details of case III size/weight attribute contributions.

Fig. C-13. Details of case III cost attribute contributions.

Fig. C-14. Details of case III reliability attribute contributions.

Fig. C-15. Details of case III maintenance attribute contributions.

Fig. C-16. Details of case III efficiency attribute contributions.

Fig. C-17. Details of case III technical risk attribute contributions.

Fig. C-18. Details of case III size/weight attribute contributions.