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ISOTOPES KILOWATT PROGRAM
TASK I - CONCEPTUAL DESIGN AND EVALUATION

R. A. Robinson

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ISOTOPES DEVELOPMENT CENTER

ISOTOPES KILOWATT PROGRAM

TASK I -- CONCEPTUAL DESIGN AND EVALUATION

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ABSTRACT

Conceptual designs have been prepared for several types of radioisotope electric power systems in order to determine which types of systems show the most promise for early development for terrestrial and undersea applications in the 2 kW(e) to 10 kW(e) power range. Strontium-90 titanate and ^{60}Co are considered as potential radioisotope fuels for the systems. The present development status of various energy conversion system components is reviewed.

ISOTOPES KILOWATT PROGRAM
TASK I - CONCEPTUAL DESIGN AND EVALUATION

1. INTRODUCTION

In 1968 the Oak Ridge National Laboratory started work on a program to evaluate various types of radioisotope energy-conversion systems for the production of 1 to 10 kW of electric power for terrestrial and undersea applications. This program is being carried out for the U. S. Atomic Energy Commission Division of Reactor Development and Technology and the Naval Facilities Engineering Command. The work is divided into two separate tasks. Task I, the subject of this report, is a parametric and engineering analysis of four radioisotope fuels, $^{144}\text{Ce}_2\text{O}_3$, ^{90}SrO , $^{90}\text{SrTiO}_3$, and ^{60}Co metal, and several types of energy-conversion devices. Task II will be concerned with a more detailed engineering study of two or three radioisotope energy-conversion system concepts that have been selected as a result of the information developed in Task I. The detailed Scope of Work for Task I is given in Appendix C. The major objectives of Task I were the following:

1. preparation of parametric data for comparing the four fuels as to fuel block volume, weight, shielding, and temperature limitations,
2. evaluation of heat-exchange problems in multikilowatt fuel arrays as related to energy-conversion system temperature and heat-flux requirements and safety considerations,
3. analysis of methods for converting thermal to electrical energy within the limitations of operating conditions achievable with radioisotope heat sources,
4. preparation of conceptual power-system design layouts in sufficient detail to provide preliminary comparisons of weight, volume, ease of assembly, and safety characteristics,
5. recommendations of system concepts for more detailed analyses based on current status of technology and near-future applications.

The use of radioisotope devices for the generation of electric power has thus far been confined to static converters (thermoelectrics) operating in the range of milliwatts of power to a few tens of watts. These systems have been successfully employed in a variety of hostile environments, such as the deep ocean, the Arctic region, and outer space, and they have been developed to the extent that they are now available from private industry suppliers. The success of these small systems has led to interest in long-lived radioisotopic systems generating electric power in the kilowatt range. A survey of U. S. Navy activities^{1,2} has indicated a large number of potential undersea and terrestrial applications of radioisotope power systems which are very likely to be cost competitive with conventional energy systems and which in some instances, particularly in deep-ocean applications, appear to be the only practical system to use.

The following types of energy conversion systems have been considered in this preliminary evaluation study:

1. Rankine cycles with steam, organic, potassium, and cesium working fluids,
2. supercritical Rankine (Feher) cycle with carbon dioxide as the working fluid,
3. Stirling cycle with helium as the working fluid,
4. Brayton cycle with a helium-xenon mixture as the working fluid,
5. thermoelectric junctions with lead telluride thermoelements.

The cost of the radioisotope fuel is a major portion of the cost of a developed isotope power system. A small change in conversion efficiency can have a significant effect on system cost, and therefore there is an economic incentive for considering dynamic conversion systems that are potentially capable of converting 25% or more of the thermal energy to electrical energy. However, in certain types of activity, where silent operation is necessary, thermoelectric converters may be the only acceptable type of system despite the inherent limitations on thermal-to-electrical power conversion efficiency. Also, thermoelectric generators have the advantage of being more nearly developed than small, unattended, dynamic conversion systems. For these reasons thermoelectric converters are also included in the preliminary study.

The upper limit on the operating temperature for the various conceptual systems was set by the study guideline that the radioisotope fuel capsules be constructed of alloys that could be exposed to air at operating temperature without undergoing serious oxidation. While there are several superalloys that can be exposed to air at 2000°F for fairly long periods of time, this requirement set the normal upper operating temperature of the fuel capsule surfaces at ~1800°F and preferably lower, with short-term excursions during startup, emergencies, etc., to 2000°F. A higher capsule operating temperature could be tolerated with fuel capsules which have a refractory-alloy inner load-bearing wall and a noble-metal outer wall for oxidation and corrosion resistance, but the cost would be considerably increased and the added advantage dubious in terms of near-future power conversion technology.

No particular application was specified as a basis for the study except that the operating environment be air or water. In order to estimate sizes of heat rejection equipment, air temperatures of 0, 40, and 130°F and a water temperature of 40°F were assumed in the conceptual system designs.

The power level range for the conceptual systems is 1 to 10 kW(e) and the operating lifetime is 1 to 5 years. There are undoubtedly some types of applications that would require power for only a few months, but in view of the assembly, testing, and transportation time a minimum life of

1 year was chosen as a more practical design goal. The power level range represents a substantial jump from that of presently operating systems (<100 W) and is probably representative of the lower end of the range for dynamic conversion systems and the upper end for thermoelectric converters. Although there has been some component development work in various laboratories for both dynamic and static closed-loop energy converters in this power range, a complete radioisotope-fueled system has never been built and operated.

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¹F. K. Kawahara et al., Present and Potential Navy Applications of Radioisotopic Energy - Preliminary Survey, USNRDL-LR-216 (Secret) (Sept. 29, 1966); Note: Unclassified summary (OUO) issued as Reviews and Lectures No. 181 (Nov. 8, 1966).

²Energy Systems of Extended Endurance in the 1-100 Kilowatt Range for Undersea Applications, Publication 1702, National Academy of Sciences, Washington, 1968.

2. SUMMARY AND RECOMMENDATIONS

RADIOISOTOPE FUELS

The isotopes $^{144}\text{Ce}_2\text{O}_3$, ^{90}SrO , $^{90}\text{SrTiO}_3$, and ^{60}Co were considered as possible fuels for multikilowatt heat sources; $^{144}\text{Ce}_2\text{O}_3$ was eliminated during the very early phases of the study primarily because of its short (0.78-y) half-life. Although large quantities of ^{144}Ce are obtained during reactor fuel reprocessing operations, it cannot be accumulated for later use, except for relatively short time periods, because of its rapid decay. Based on an optimistic estimate of six months for recovery, purification, and encapsulation of ^{144}Ce fuel followed by three months for converter assembly, testing, transport, and deployment, approximately half of the original fuel inventory would decay before the mission application could be started. These difficult scheduling problems, coupled with the relatively limited range of mission applications for which ^{144}Ce can be used, more than cancel out any speculative fuel-cost advantage.

Cobalt-60 has two advantages over the ^{90}Sr fuel form: it has a higher power density ($>15 \text{ W/cm}^3$) and, since it is produced by the irradiation of nonradioactive ^{59}Co in a nuclear reactor, its availability is not tied to reactor fuel reprocessing operations. However, there is neither enough excess reactor volume nor a high enough neutron flux in commercial power reactors to produce the high-specific-activity ^{60}Co (100-300 Ci/g) needed for isotopic power applications, and the supply would probably have to come from government reactors. If future commercial power reactors are built with sufficient excess neutrons for producing other products, the operators would very likely choose to produce more valuable products, such as plutonium, which would give a greater return than could be obtained from ^{60}Co if its price were to remain competitive with that of ^{90}Sr .

The major disadvantages of ^{60}Co are the shielding requirements and its short half-life (5.28 y compared with 28 y for ^{90}Sr). The weights of fuel and shielding for ^{60}Co and ^{90}Sr heat sources would be the same at a thermal power level of 100 kW for steel shields, 50 kW for nickel shields, 3.5 kW for tungsten shields, and 2 kW for uranium shields. At higher power levels ^{60}Co heat sources could be lighter than ^{90}Sr heat sources because of its higher power density. In large heat sources the fuel capsules must be arranged to provide adequate heat-transfer paths from the interior of the capsule array to the outer surface to prevent overheating and possible melt down of the capsules should any of the active heat-removal systems fail during transport or use. In the case of ^{60}Co this design philosophy of passive heat rejection under loss-of-coolant conditions results in the fuel capsules having to be spread out in a large volume and the effective power density not being much greater than that of ^{90}Sr fuels.

The half-life of ^{60}Co starts becoming a problem at conversion system lifetimes (from initial fueling) of about 2 years. The amount of excess fuel needed at the beginning of life in order to have the rated power level at the end of life creates significant excess heat discharge and

temperature control problems. In the case of present-day thermoelectric generators, which are very sensitive to hot-junction temperature, even 2-year-mission applications may be out of the range of ^{60}Co . New thermoelectric materials under development which can tolerate much wider temperature ranges without couple degradation may alleviate this problem in the future.

Of the two remaining fuels, ^{90}SrO and $^{90}\text{SrTiO}_3$, the oxide has a higher practical power density ($\sim 1.5 \text{ W/cm}^3$) than the titanate ($\sim 0.85 \text{ W/cm}^3$), but must be spread out in passively cooled multicapsule fuel block assemblies for the same reasons that apply to ^{60}Co . Also, it must be protected from the air during fuel pellet fabrication and encapsulation operations because it slowly reacts with moisture to form the hydroxide and with CO_2 to form the carbonate. Providing for this protection would add to the fuel cost. Of all the fuels, $^{90}\text{SrTiO}_3$ is the best characterized in isotopic power applications since it has been used in over 20 thermoelectric devices. The procedures for preparation and encapsulation have been well developed as a result of several AEC programs, and the fuel has been accepted by safety and regulatory groups for a wide variety of terrestrial and undersea applications.

Present AEC prices for both ^{60}Co and ^{90}Sr are not indicative of the prices that would prevail if multimegacurie quantities were produced routinely for isotopic power devices. It is expected that by 1970 the AEC will be routinely removing all the ^{90}Sr and ^{137}Cs from the fuel reprocessing wastes at its Hanford plant prior to solidification of the remaining wastes for long-term storage. This is being done as part of the Hanford Waste Management Program to minimize waste mobility, and, depending on accounting procedures, the separated ^{90}Sr could be considered as paid for up to that point. As a further extension of this Waste Management Program at Hanford, it is planned, by about 1975, to routinely convert the separated ^{90}Sr to solid form, probably SrF_2 , and encapsulate the solid in Hastelloy C-276 cans to further reduce waste handling costs. This step could also result in a reduction in fuel costs for isotopic power units. Likewise, if sufficient demand developed to keep one of the AEC production reactors at Savannah River operating for the production of ^{60}Co , very large quantities could be made available at a unit cost that would be expected to be very much lower than present selling prices for high-specific-activity ^{60}Co . At the present time, it can only be stated that neither ^{60}Co nor ^{90}Sr has a distinct cost advantage.

On the basis of this study it is recommended that major emphasis be placed on the use of $^{90}\text{SrTiO}_3$ in the Task II system designs because of its suitability for both short- and long-mission applications, its advanced state of development in isotopic power applications, and the lack of any firm technical or economic incentive favoring the other fuels. While it is not expected that there will be any significant differences between a system designed for a short-term mission ($\sim 1\text{-}2$ years) using ^{60}Co and one using ^{90}Sr , it is suggested that one of the Task II dynamic system designs for ^{90}Sr be repeated for a ^{60}Co heat source to study the feasibility of interchangeable fuels.

FUEL CAPSULE DESIGN

The parametric study of fuel capsule dimensions did not reveal any particular advantages for a specific size. For the purpose of preparing the fuel block and system concepts, the inside diameter of the ^{60}Co capsule was set at 0.75 in. to be consistent with the dimensions (0.745-in. dia) of the presently available wafer form of this fuel, and the inside diameter chosen for the capsules containing the ^{90}Sr fuel forms was 2.5 in., which is well within the demonstrated capabilities of the fuel pellet hot-pressing fabrication process. In both cases the length-to-diameter ratio was limited to 3 and 4 to provide a capsule that can be handled easily with remotely operated equipment during the capsule fueling, welding, and decontamination operations. Although the conceptual system designs considered both surface and undersea applications, all the capsules were designed to withstand a 10,000-psig external pressure because ocean transport is expected to be involved even in terrestrial applications and a unit might be dropped inadvertently to the ocean floor, the bulk of which is at depths of 20,000 ft or less.

Capsule wall thickness is conservatively based on the capsule itself being able to withstand the 10,000-psig pressure at a 20,000-ft depth without credit being taken for any support offered by the fuel inside. The calculated capsule wall thicknesses needed to satisfy this requirement are 0.095 and 0.220 in. respectively for the 0.75-in.-ID and 2.5-in.-ID capsules. This is well within the range of current remote welding capability. The range of capsule sizes used for preparing conceptual system designs provides thermal loadings of 545 to 757 W per capsule for $^{90}\text{SrTiO}_3$, 1027 to 1406 W for ^{90}SrO , and 908 to 1086 W for ^{60}Co .

The conceptual system designs are based on the use of superalloy fuel capsules. Many superalloys retain adequate strength properties for isotopic power applications at temperatures up to 1800°F and also have reasonably good oxidation resistance. Based on the available data and present state of development, Hastelloy C (or its new modification Hastelloy C-276) is recommended as the encapsulating material. Thus far Hastelloy C has been used almost exclusively in terrestrial and undersea isotopic power applications. Extensive compatibility data are available for both ^{60}Co and ^{90}Sr fuels in contact with Hastelloy C-276, long-term oxidation experiments have been conducted, and its corrosion behavior in seawater with and without the presence of radiation has been extensively studied as part of the AEC's various isotopic power development programs. The fabrication, welding, and inspection procedures have been thoroughly demonstrated through preparation of fuel capsules for existing isotopic power systems.

SHIELDING AND HEAT BLOCKS

The effects of many different configurations and materials on the size, weight, and temperature distribution of the shield-heat block assembly have been studied. From the standpoints of the handling and hazards problems during initial fabrication, assembly, shipping, installation, and normal operation the most promising configuration was found to be a

cylindrical billet of steel or nickel 3 to 4 ft in diameter and 3 to 8 ft tall (depending on the power output), with the isotope capsules installed in an array of axial holes close to the center of this large block. Axial fins on the outer surface of the billet would remove heat under shipping and emergency conditions simply by thermal radiation and convection to the surroundings. This approach to the design of the shield-heat block assembly permits the heat block to be fueled and shipped separately from the rest of the energy conversion system without the inclusion of any accessory cooling equipment. A single steel or nickel block provides good thermal conduction to remove the fission-product decay energy from the center of the block when it is insulated and brought up to temperature to produce electric power. Fusible metal plugs can be employed to provide heat conduction through the thermal insulation if the temperature of the block exceeds the permissible limit because of failure in the heat transport system of the electric power producing equipment.

Tungsten or uranium could also be used as shielding and heat block materials, but each has one or more drawbacks. The thermal conductivity and melting point of uranium are too low, and the cost of tungsten for the heat block-shield assembly of a 5-kW(e) power unit would run over \$100,000, compared with about \$3000 for a steel shield and heat block assembly or about \$25,000 for nickel. There is some question as to whether tungsten can be fabricated in the form of a huge block; even if it could, there would remain numerous difficulties associated with differential thermal expansion between the tungsten and other components in the system such as the Hastelloy fuel capsules. It may be necessary, however, to use tungsten for direct-conduction thermoelectric generator systems because of the relatively high hot-junction temperature required for efficient operation. If so, the block would probably be fabricated from several small subassemblies. Both tungsten and uranium are highly sensitive to attack by air, moist air, and water, and would require some sort of protective coating. There is no certainty that a satisfactory coating could be obtained, particularly because of the problem of matching the very low coefficient of thermal expansion of tungsten with that of a corrosion-resistant coating material. Either steel or nickel, on the other hand, is readily available, fabricable, and adequately corrosion-resistant in air and seawater.

A major problem in shielding a fission reactor for ocean use is prevention of activation of seawater. However, the gamma and bremsstrahlung radiation from an isotope heat source will not activate seawater; hence with some systems some or all of the shielding required for shipping an isotope heat source can be removed once the source is submerged in water. A cage or shell can be placed around it so that the enclosed water will serve as shielding. The heat source and shield assembly weight might be reduced from over 10 tons to less than 3 tons for a typical case. Thin shields of this type could be used with a steam Rankine cycle, but some shielding is necessary with an organic Rankine cycle because of radiation damage to the organic working fluid. Insufficient data are available to predict the extent to which various types of thermoelectric junctions will require shielding from gamma radiation.

HEAT-REMOVAL SYSTEMS

Various means of removing heat from the system have been investigated, including direct conduction from the heat block through thermoelectric junctions, use of alkali metal heat pipes to couple the heat block to either thermoelectric junctions or heaters for the dynamic cycles, and various arrangements incorporated in the heat block and shield assembly for transmitting the heat to the working fluid of the thermodynamic cycle. In choosing the heat-removal system, as much consideration must be given to emergency removal of heat from the radioactive material, should the power conversion system fail, as is given to requirements of the heater for the power unit. The upper temperature limit of 1800°F (2000°F for emergency conditions) imposed on the Hastelloy fuel capsules, coupled with the particular characteristics of the individual power conversion systems, makes the choice of heat-removal system very much dependent on the choice of power conversion system. Since safety under all credible accident conditions is the predominant design consideration, heat-removal systems which require that a fluid flow in passages within the heat block in order to keep the fuel capsule temperature within limits are not as attractive as systems which remove heat from the outside surfaces of the block, even though there is some weight penalty. Compliance with shipping regulations is easier with the large passively cooled blocks, and more reliable emergency heat-removal schemes are possible if the integrity of a secondary-heat-exchanger loop does not have to be guaranteed.

THERMODYNAMIC CYCLES

All the thermodynamic cycles that have been advocated for use in small isotope power plants were examined. The ideal cycles were examined, with the principal temperature and pressure limitations characteristic of each cycle defined, and were compared with the ideal Carnot cycle. The losses characteristic of reciprocating and turbine engines were reviewed, as were fundamental limitations on the operating life of the machinery as indicated by the background of operating experience that has been accumulated. It appears that a turbine engine could be developed in the near future to satisfy the desired requirements of more than one year of unattended operating life and at the same time give an overall thermal efficiency in excess of 10%. It is believed, however, that this goal is unlikely for a reciprocating engine because of difficulties with piston ring and cylinder lubrication.

Detailed analyses were carried out for cycles employing turbogenerators that appeared to be of most interest; various organic working fluids; steam, cesium, and potassium vapors in Rankine cycles; and a Brayton cycle that would use a mixture of xenon and helium. The high-temperature liquid-metal Rankine cycles do not look attractive for undersea applications because there is no advantage to the use of a high condenser temperature. In fact, a condenser temperature above the boiling point of seawater might cause film boiling on the seawater side of the condenser surface, resulting in noisy and possibly damaging operation. Consequently, the thermodynamic cycles of interest were the steam Rankine cycle, the organic Rankine cycle, and the Brayton cycle.

The principal developmental problem envisioned for the steam turbine-generator units is how to protect the windings against moisture penetrating the electrical insulation and causing electrical shorts. There could be difficulties with radiolytic decomposition of the water and as a consequence mass transfer, corrosion, and release of hydrogen, but a number of recent developments indicate that these problems can be resolved satisfactorily.

In the organic turbine-generator systems the electric insulation in the generator might be attacked by the organic working fluid, but there is less likelihood of this occurring than in the steam turbine-generator systems. The degree to which noncondensable gases will be released from the organic because of radiation and thermal damage poses the principal problem associated with organic turbines because these gases could seriously degrade the performance of the condenser and thus the output of the system. The permissible temperature and radiation levels can be determined only by operating a mockup system with the full temperature and radiation levels contemplated for the full-scale system.

The principal problems with the Brayton-cycle system are associated with the close clearances in the gas bearings and their sensitivity to small amounts of dirt or distortion from creep or other thermal effects. In addition, the Brayton-cycle system will require operation of the fuel capsules at a temperature not far from the desired upper limit, which leaves little margin for emergency cooling in the event of a failure in the gas circulation system.

A review of the development and operating experience with various 1- to 10-kW(e) turbine-generator units disclosed that at least one steam turbine and at least three organic turbines and one Brayton-cycle turbine have been built and operated for at least 500 hr. All five units represent reasonable approximations to the specialized requirements of an undersea power plant.

Currently available thermoelectric materials, notably, lead telluride, yield a very complex and expensive system with an overall thermal efficiency of perhaps 7%. New materials under development show promise of overall thermal efficiencies of 10% or more, but considerable developmental work must be carried out before they are validated. Although thermoelectric units would be expected to have greater long-term reliability than turbogenerators because of the absence of moving parts, the large number of couples required in kilowatt-size units presents a formidable reliability problem during manufacture and assembly of the converter module.

The results of the analysis of the various energy conversion methods coupled with the survey of the present status of materials and hardware development led to the recommendation that the Task II systems design effort be directed toward steam Rankine-cycle and organic Rankine-cycle converter systems. In addition, preliminary design for a 1-kW thermoelectric system based on the expected performance and operating requirements of advanced materials rather than present-day thermoelements

appears advisable if completely silent operation becomes a requirement for some applications. Although the Brayton cycle offers the highest potential thermal efficiency of all the energy conversion systems that were studied, it is recommended that it be dropped from the Task II effort because of the small margin between fuel capsule operating temperature and failure temperature that would exist if the high efficiency were to be realized in practice.

CONCEPTUAL DESIGNS

Conceptual designs have been developed for several thermoelectric power units, several organic turbine-generator units, a steam turbine-generator power unit, and a gas turbine-generator power unit which in each case uses thermoelectric junctions or turbine-generator units that have been known to operate for at least 500 hr. The size, weight, and estimated cost of the principal components for each power plant are summarized in Table 2.1.

An appraisal of the operating experience for these experimental power plants indicates that each involves a number of major uncertainties. First and foremost, no turbine-generator unit is available on which there is sufficient operating experience to assure reliable performance (except perhaps for units built by Ormat in Israel, for which, however, no detailed operating data have yet been obtained). There are not enough data available on the combined effects of elevated temperatures and radiation damage to organic working fluids for estimating the extent to which the life of a hermetically sealed organic Rankine-cycle system may be limited by the release of noncondensibles. The NASA Brayton-cycle unit looks attractive, but the close running clearances in the gas bearings make such a system inherently sensitive to small amounts of dirt and thermal distortion. Thus it is difficult to predict the time and cost of the development program, the cost of production units, and the reliability.

Although the effort to develop turbine-generator units for the nuclear space power program has not yet produced a demonstrably reliable unit in spite of large expenditures over more than 10 years, it is believed that a highly reliable unit could be developed at a modest cost to meet the less difficult requirements of an undersea power plant. The temperature regime is much lower and weight is not a major consideration, which is particularly important.

In view of the high cost and low thermal efficiency of thermoelectric junctions and the possibility of getting quiet high-speed turbine-generator units of high reliability, the latter appear more attractive. Further, if satisfactory turbine-generator units can be developed in the 1- to 10-kW(e) size range, the development of a larger unit should be easier; of course, as the size increases, the cost per unit will rise.

The thermal efficiency of an isotope power plant could be improved through the use of any of several binary cycles. For example, thermoelectrics could be used between the heat source and the boiler of an

Table 2.1. Principal Factors and Considerations Affecting the Choice of Power Conversion System for Isotope Power Units

Working Fluid	Thermoelectric Cycle	Rankine Cycle						Brayton Cycle		Stirling Cycle, Helium	Otto Cycle, Air and Fuel	Diesel Cycle, Air and Fuel
		Steam	Dowtherm A	Dowtherm E	Monochlorobenzene	Cesium	Potassium	Xe-He	CO ₂			
Performance Potential												
Practicable peak temperature, °F	1000	1050	~600	~375	~400	2000	2000	2000	1500	1500	4500	4500
Practicable peak pressure, psia		1000						100	2200	2000	1500	1500
Nature of temperature limitation	Stability of lead telluride	Corrosion	Thermal decomposition of fluid	Thermal decomposition of working fluid	Thermal decomposition of working fluid	Creep in turbine wheel and heater	Creep in turbine wheel and heater	Creep in turbine wheel and heater	Creep in turbine wheel and heater, dissociation of CO ₂	Creep in turbine wheel and heater	Chemical energy release per pound of working fluid	Chemical energy release per pound of working fluid
Estimated overall thermal efficiency, % [at 5 kW(e)]	7	12	18					24	22	25	20	30
Cycle bottom temperature, °F	200	70	225	115	70			80	80	200	70	70
Operating Experience												
On model delivering 1 to 10 kW(e), hr	None	1000	2500	~10 ⁶ hr on units of 0.2 to 3 kW(e)		None	None	1500	None	17,000	>10 ¹⁰	>10 ¹⁰
Total operating time on units roughly similar to above, hr	>10 ⁶ (for modules and small units)	>10 ⁶	3500			None	None	2000	None	23,000	>10 ¹¹	>10 ¹¹
Maximum operating time without a shutdown, hr	50,000 (for small units)			>10 ⁴						750	~10,000	~10,000
Limitations on Operating Life, Unattended												
Parts affected by wear	None	None	None	None	None		None	None	None	Piston, rings, cylinder	Piston, rings, cylinder	Piston, rings, cylinder
Parts affected by thermal decomposition	Thermoelectrics	None	Working fluid	Working fluid	Working fluid	None	None	None	None	None	Lube oil	Lube oil
Parts affected by corrosion or erosion	None	Generator windings	Generator windings	Generator windings	Generator windings	Generator windings	Generator windings	None	None	None	Spark plugs, valves, points	Valves
Parts affected by dirt or deposits	None	Possibly boiler	Possibly boiler and bearings	Possibly boiler and bearings	Possibly boiler and bearings	None	None	Gas bearings	Gas bearings	Heat transfer surfaces	Piston rings, bearings, valves	Injection nozzles, valves
Parts affected by thermal stresses	Thermal bonds	None	None	None	None	Turbine casing and rotor	Turbine casing and rotor	Turbine casing, rotor, and bearings	Turbine casing, rotor, and bearings	None	Valves	Piston rings, bearings
Suitability for 20,000 hr of operation on the sea floor	Excellent	Good	Good	Good	Good	Poor	Poor	Good	Good	Poor	Completely unsuitable	Completely unsuitable
Minimum shield weight submerged [for 5 kW(e) unit], lb		6,100	12,400	12,400	12,400			18,100			Completely unsuitable	Completely unsuitable
Major development problems	Differential thermal expansion and thermal stresses, fabrication cost, quality control	Generator windings, recombination of radiolytic gas, system control	Established upper limits on temperature and radiation for acceptable gas release, Generator windings, control	Same as for Dowtherm A	Same as for Dowtherm A			Establish reliability of gas bearings, startup and control	Feasibility of fabrication of small complex parts to extremely close tolerances, reliability of gas bearings			
Availability												
Time required to obtain a prototype unit		5 months	5 months					12 months				
Estimated cost of three 5 kW(e) prototype units (thermoelectric elements or turbine-generator only)	\$1,500,000 to \$2,500,000	\$30,000 to \$50,000	\$100,000 to \$200,000					\$750,000				

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organic Rankine cycle, two different sets of thermoelectric materials could be used in series over a wide temperature span, etc. However, both the binary system and the control problems of the dual electric power sources create a greater degree of complexity and hence lower reliability than could be obtained with a single system, and it is recommended that they not be considered further at this time.

3. RADIOISOTOPE FUELS AND CAPSULE DESIGN

RADIOISOTOPE FUELS

General Data

Four fuel compounds were considered in the initial phase of the program, $^{144}\text{Ce}_2\text{O}_3$, ^{90}SrO , $^{90}\text{SrTiO}_3$, and ^{60}Co metal. These fuels were selected for consideration because they are readily available, the processing and encapsulation technology of each is known and available, their properties and compatibility are known, and they are available at reasonable cost. Specific power of the fabricated fuel compound, half-life of the isotope, and shielding requirements were also considered in making the fuel form selection.

Physical and radiation property data for the candidate fuels are included as Appendix A to this report. These data represent the most recent literature values and were used for the calculations in this report.

Fuel volumes and weights were calculated for each of the candidate fuel forms for missions lives of 1/2, 1, 5, and 10 years for end-of-life (EOL) powers of 1, 10, 20, 50, and 100 kW(th). This information is given in Table 3.1 and presented graphically in Figs. 3.1-3.5. These data indicate that $^{144}\text{Ce}_2\text{O}_3$ is not a satisfactory fuel for multikilowatt devices because of its short half-life. The data in Table 3.1 do not include allowances for decay during processing and encapsulation, shipping to site, and loading the device. These allowances could easily increase the beginning-of-life (BOL) power requirement for ^{144}Ce fuel power by a factor of 2. Other factors making ^{144}Ce unattractive as a fuel are the problems that would be imposed by power flattening and excess-heat dumping and by the fact that it cannot be stockpiled for use at a later date (again because of its short half-life) and therefore would need to be produced on close schedules coinciding with and complementing loading, encapsulating, and fueling schedules.

Cobalt-60 also has the disadvantage of a half-life limitation. Figure 3.6 is a plot of normalized EOL fuel weight or volume vs mission life and may also be used as a normalized BOL power vs mission life. The curve shows that power-flattening problems will arise for long (5- to 10-year) ^{60}Co mission lives unless, as discussed in "Cost Analysis," fuel is added at interim periods.

In the case of at least the early models of turbogenerator systems, it is unlikely that they will be expected to operate for more than a year or two without major overhaul or replacement and the comparatively short half-life of ^{60}Co may not be a problem. However, as the technology evolves to the point where minor maintenance and field replacement of turbogenerator components are practical, the need to also replace the heat source would be an unnecessary added burden. In the case of thermoelectric generator systems, continuous operating lifetimes in excess of 5 years have already been demonstrated by the SNAP-7 series of generators in both terrestrial and undersea applications and there is reason to expect that large units will be capable of the same or better performance.

A potential fuel candidate, for which sufficient physical data were not developed in time for this study, is $^{90}\text{SrF}_2$, which is expected to become available in large quantities in the near future as a by-product of waste management programs. Its power density and physical density are very close to those of $^{90}\text{SrTiO}_3$, but it has a somewhat lower melting point¹ (2665°F) and greater solubility. Its physical and chemical properties are being studied as part of the ORNL supporting technology program on ^{90}Sr isotopic power fuels. For present purposes the data relating to fuel volume, weight, and shielding² presented for SrTiO_3 apply very closely to SrF_2 .

Table 3.1. Volume and Weight of ^{90}SrO , $^{90}\text{SrTiO}_3$, and $^{144}\text{Ce}_2\text{O}_3$ for Various Mission Lives and EOL Source Power

EOL Source Power [kW(th)]	Volume (in. ³)				Weight (lb)			
	^{90}SrO	$^{90}\text{SrTiO}_3$	$^{144}\text{Ce}_2\text{O}_3$	$^{60}\text{Co}^a$	^{90}SrO	$^{90}\text{SrTiO}_3$	$^{144}\text{Ce}_2\text{O}_3$	$^{60}\text{Co}^a$
<u>6-Month Mission Life</u>								
1	41.50	73.24	5.96	1.6	6.00	11.93	1.30	0.5
10	415.00	732.40	59.60	15.8	60.00	119.30	13.00	5.1
20	830.20	1465.10	119.10	31.7	120.00	238.60	25.80	10.2
50	2075.60	3663.00	298.00	79.2	300.00	596.50	64.60	25.5
100	4151.20	7324.00	596.00	158.3	600.00	1193.00	129.20	50.9
<u>1-Year Mission Life</u>								
1	41.90	74.00	9.30	1.7	6.10	12.04	2.00	0.55
10	419.00	740.00	93.00	17.0	61.00	120.40	20.00	5.5
20	838.80	1480.30	186.10	34.0	121.20	240.80	40.30	10.9
50	2097.00	3700.80	465.10	84.8	303.00	601.90	100.80	27.3
100	4194.00	7400.00	930.20	169.7	610.00	1203.80	201.60	54.6
<u>5-Year Mission Life</u>								
1	46.20	81.62	317.80	2.9	6.70	13.28	68.90	0.92
10	462.00	816.20	3178.00	28.7	67.00	132.75	689.00	9.2
20	924.60	1631.70	6356.70	57.5	133.60	265.50	1377.90	18.5
50	2311.40	4079.20	15891.60	143.7	334.00	663.75	3444.70	46.2
100	4622.80	8162.00	31780.00	287.4	670.00	1327.50	6889.30	92.4
<u>10-Year Mission Life</u>								
1	52.90	93.24	b	5.6	7.60	15.19	b	1.8
10	529.00	932.40	b	55.7	76.00	151.90	b	17.9
20	1056.70	1864.80	b	111.5	152.70	303.80	b	35.8
50	2651.20	4662.00	b	278.7	381.70	759.38	b	89.6
100	5302.40	9324.00	b	557.4	760.00	1518.30	b	179.2

^aSpecific activity, 300 Ci/g.

^bNot considered because half-life too short.

Fig. 3.1 EOL Power and Fuel Volume for ^{90}SrO .

Fig. 3.2. EOL Power vs Fuel Volume for $^{90}\text{SrTiO}_3$.

Fig. 3.3. EOL Power vs Fuel Volume and Weight for Ce_2O_3 (5-Year Life).

Fig. 3.4. EOL Power vs Fuel Volume and Weight for Ce_2O_3 (1/2- and 1-Year Lives).

Fig. 3.5. EOL Power vs Fuel Volume and Weight for ^{60}Co Metal (300 Ci/g).

Fig. 3.6. Normalized EOL Weight or Volume of ^{60}Co vs Mission Life.

Cost Analysis

The present price of $^{90}\text{SrTiO}_3$ and ^{90}SrO is \$0.20 per curie, and it is not unreasonable to project a price of \$0.06 to \$0.08 per curie (exclusive of encapsulation cost) for the very large quantities that would be needed for kilowatt-size power systems in the future. Lesser quantities than the amounts considered here would cost more. There is no cost advantage for ^{90}SrO over $^{90}\text{SrTiO}_3$, even though the oxide is an intermediate in the production of the titanate, because present oxide pellet fabrication techniques are more costly than in the case of the titanate.

Fuel costs were calculated based on both the current price and projected price for mission lives of 1, 5, and 10 years for ^{90}Sr and ^{60}Co fuels. The initial cost of fuel for both a 10- and a 30-kW(th) EOL source was calculated (Table 3.2). None of the costs include any factor associated with the recovery and reuse of the fuel at the end of the mission life. It is interesting to note the sharp decrease in unit cost as a function of mission life. These data are presented graphically in Figs. 3.7 and 3.8. Somewhat lower costs per kilowatt hour result when all the available heat is utilized in some manner.

Table 3.2. Fuel Costs for Various Mission Lives
at Current and Projected Fuel Costs

Mission Life (years)	Initial Cost		Total Initial Cost [kWhr(th)]	
	10 kW(th) EOL	30 kW(th) EOL	EOL Power Only Usable	All Power Usable
Current Cost of ^{90}Sr : \$0.20 per Ci; \$29.60 per W				
1	\$305,176	\$ 915,528	\$3.48	\$3.44
5	337,440	1,012,320	0.77	0.73
10	384,800	1,154,400	0.44	0.39
Projected Cost of ^{90}Sr : \$0.08 per Ci; \$11.84 per W				
1	\$122,070	\$ 366,211	\$1.39	\$1.38
5	134,976	404,928	0.31	0.29
10	153,920	461,760	0.18	0.16
Projected Cost of ^{90}Sr : \$0.06 per Ci, \$8.88 per W				
1	\$ 91,553	\$ 274,658	\$1.05	\$1.04
5	101,232	303,696	0.23	0.22
10	115,440	346,320	0.13	0.12
Projected Cost of ^{60}Co : \$0.185 per Ci; \$12.00 per W				
1	\$136,368	\$ 409,104	\$1.56	\$1.47
5	233,000	699,000	0.53	0.38
10	449,436	1,348,308	0.51	0.24

Fig. 3.7. Cost of ^{90}Sr Fuel for Various Mission Lives at Present and Projected Costs of ^{90}Sr Fuel. Curves based on EOL power.

Fig. 3.8. Cost of ^{60}Co Fuel per kWhr(th) for Various Mission Lives Based on Projected Cost of Fuel and on Only EOL Power Utilized.

Estimated production costs for large quantities of ^{60}Co are shown in Fig. 3.9 as a function of irradiation time.³ Costs range from ~\$0.10 per Ci [\$6.40 per W(th)] at 100 Ci/g to ~\$0.20 per Ci at 400 Ci/g, and they do not change rapidly with length of irradiation time if at least 1 to 1.5 years can be allowed. Costs increase sharply for specific activities above ~450 Ci/g due to burnup and decay of ^{60}Co in the reactor. These estimates include all direct expenditures (e.g., reactor operation, components, fuel burnup) but exclude depreciation and other AEC factors.

Estimated costs are based on one reactor producing ^{60}Co as its major product. Quantities of <10 MCi/yr of ^{60}Co would be considered incidental production and would be included in reactor loadings designed for other products. The irradiation time would depend on the neutron flux for those loadings, but the ^{60}Co costs would not be substantially higher than

those shown by the curves in Fig. 3.9. For quantities significantly >10 MCi/yr, the design of reactor loadings would be influenced by the need to minimize irradiation time for the ^{60}Co product.

Based on an estimated cost of \$12 per W(th) for capsules containing 300 Ci/g of cobalt, fuel costs were calculated for sources having a thermal power of 10 kW after a mission of 5 years. A total of 19.4 kW is required for a source that is sealed at beginning of life. The initial ^{60}Co cost would be \$233,000 (costs for other powers can be scaled directly). The power cost would be \$0.53 per kWhr(th) if only the 10 kW were usable over the life of the source, or \$0.38 per kWhr(th) if the excess energy above 10 kW were also usable.

Fig. 3.9. Cost Estimates for Production of Large Amounts of ^{60}Co .

For a heat source to which fuel can be added each year, an initial loading of 11.4 kW(th) is required, with 1.4 kW being added at 1-year intervals during the first 4 years. The cost of the total requirement, 17.0 kW, is \$204,000. The power cost would be \$0.47 per kWhr(th) if 10 kW were usable and \$0.44 per kWhr(th) if all the energy were usable over the 5-year period.

It is apparent that a significant savings in ^{60}Co fuel cost may be achieved with annual additions of fuel and that power conditioning problems may be less difficult. Little savings in heat-source size and weight would be expected because space must be available for inserting the fresh fuel.

The preceding discussion of prices for ^{60}Co and ^{90}Sr pertains only to the fuel costs and does not include the encapsulation costs which, for both fuels, depend primarily on the fuel capsule specifications. The cost per capsule should be about the same for each fuel since the same quality assurance and structural requirements would be needed irrespective of the radioisotope used. Encapsulation costs stated on a cents/curie basis are misleading because this number depends primarily on the size of the capsule and successively larger capsules will have a lower unit cost. Encapsulation costs for $^{90}\text{SrTiO}_3$ heat sources at ORNL during the last 2 years have ranged from ~\$4000 to ~\$10,000 per capsule. In the case of the higher figure, the encapsulation costs included a complete dimensional, dye penetrant and ultrasonic inspection of capsule components, calorimetric measurement of each fuel pellet, electron beam closure weld, ^{85}Kr leak testing of the completed capsule, and a remote ultrasonic inspection of the closure weld to verify soundness and weld penetration. The lower figure includes dye penetrant inspection of capsule components, tungsten-inert gas welding, vacuum leak testing of the closure weld, and calorimetric measurement of the heat output of the completed source.

There is no comparable cost data for ^{60}Co sources encapsulated to isotopic power system specifications since it has not been used in systems presently in operation. For equivalent thermal power, a ^{60}Co capsule will be smaller than a ^{90}Sr capsule and the cost of the capsule itself would be expected to be less; however, this cost is not significant in comparison to the costs for the remote encapsulation operations.

CAPSULE CONFIGURATION STUDY

A study was conducted to parameterize fuel block and capsule configurations to obtain minimum weight, minimum volume, and maximum surface heat-flux heat-source configurations consisting of groups of capsules. The study was conducted for $^{144}\text{Ce}_2\text{O}_3$, ^{90}SrO , and $^{90}\text{SrTiO}_3$ fuels since these fuels were the only ones over which much control of capsule and/or fuel pellet diameter could be exercised. As will be seen on reviewing the information derived in this study, no one source capsule evolved as the most practical or desirable, and for the conversion systems of interest the calculated surface heat-flux values ($>100 \text{ W/in.}^2$) for the large source arrays are too high for heat transfer to a working fluid at reasonable fuel block operating temperatures (2000°F). For these cases the fuel block was spread out over more than the minimum required volume in order to provide more area for heat transfer (Chapter 4). In low-thermal-power output source arrays the minimum required volume results in heat fluxes that are too low for good heat transfer at the temperatures of interest, and part of the fuel block surface will have to be insulated and not used for heat transfer to the working fluid for the heat flux on the rest of the surface to be improved.

A 360/75 computer program was written to obtain the capsule dimensions for a total cylindrical source configuration with a length-to-diameter (L/D) ratio of 1.0. The program varies capsule diameters from 1 to 5 in.

in 0.2-in. increments and calculates the number of capsules required, the diameter of the smallest circle into which they can be grouped, the length of the capsules, and the L/D ratio for each configuration. The configuration with an L/D ratio nearest to 1 is used to calculate the maximum heat flux over the entire surface area of cylindrical geometry. In order for the program to provide the optimum capsule dimensions which will give the minimum fuel block surface area for a given thermal power level, it was necessary for capsule wall thicknesses and capsule L/D ratios to be inserted as parameters. In this series of calculations the wall thickness was set at 0.100 in. and the L/D ratio at 4. These values were selected as being consistent with expected capsule structural and handling requirements for under-sea and terrestrial applications. Computer data for the three source configurations (A, C, and D) shown in Fig. 3.10 which have L/D ratios nearest

Fig. 3.10. Source Configurations A, C, and D.

to 1.0 were obtained for $^{90}\text{SrTiO}_3$, ^{90}SrO , ^{60}Co metal, and $^{144}\text{Ce}_2\text{O}_3$. Configuration A has a single capsule in the center, configuration C has three capsules in the center, and configuration D has four capsules in the center. The source dimensions that will give the highest fuel block surface heat flux for each configuration are listed in Tables 3.3-3.6 for 1-, 10-, 20-, 50-, and 100-kW(th) sources for 1- and 5-year mission lives.

Tables 3.7-3.10 are compilations of the source dimensions giving the highest heat flux without regard to a particular configuration. The surface temperatures of these optimum source configurations, based on the assumption that the sources are suspended in air, are given in Tables 3.11 and 3.12. The surface temperatures for the sources assumed to be sitting in air on a 1-ft-thick concrete block are given in Tables 3.13 and 3.14. The EOL heat fluxes have been plotted against the EOL power for these sources and are given in Figs. 3.11-3.18.

It is necessary to include some shielding for the ^{60}Co calculations since most of the heat is generated in the shielding rather than in the source capsule. The indicated shield thickness is designed to reduce the radiation level to 100 rads/hr at the surface of the shield.

Equations have been derived for the straight-line portion of the curves of the plots of EOL power versus EOL heat flux for $^{90}\text{SrTiO}_3$, ^{90}SrO , ^{60}Co , and $^{144}\text{Ce}_2\text{O}_3$. These equations, in which A = maximum EOL heat flux in W/in.² and B = EOL power in kW, and the conditions that apply are listed below:

$^{90}\text{SrTiO}_3$

$$\text{1-year mission life: } \ln A = 0.361 \ln B + 1.9601 \quad (1)$$

$$\text{5-year mission life: } \ln A = 0.367 \ln B + 1.8810. \quad (2)$$

^{90}SrO

$$\text{1-year mission life: } \ln A = 0.353 \ln B + 2.3076. \quad (3)$$

$$\text{5-year mission life: } \ln A = 0.356 \ln B + 2.2773. \quad (4)$$

^{60}Co

$$\text{1-year mission life: } \ln A = 0.7432 \ln B + 1.4469. \quad (5)$$

$$\text{5-year mission life: } \ln A = 0.5834 \ln B + 1.5581. \quad (6)$$

$^{144}\text{Ce}_2\text{O}_3$

$$\text{1-year mission life: } \ln A = 0.375 \ln B + 6.3069. \quad (7)$$

$$\text{5-year mission life: } \ln A = 0.376 \ln B + 0.9416. \quad (8)$$

Table 3.3. $^{90}\text{SrTiO}_3$ Power Source Dimensions and Heat Flux Data for Various Power Levels

Source Power [kW(th)]		Number of Capsules	Diameter (in.)			Source Length (in.)	Heat Flux (W/in. ²)	
EOL	BOL		Capsule	Pellet	Source		EOL	BOL
<u>1-Year Mission Life</u>								
Configuration A								
1	1.031	1	4.8	4.6	4.8	5.17	8.76	9.03
10	10.310	7	3.8	3.6	11.4	11.39	16.34	16.84
20	20.619	7	4.8	4.6	14.4	13.84	21.00	21.65
50	51.546	19	3.8	3.6	19.0	20.56	27.87	28.73
100	103.093	19	4.8	4.6	24.0	25.07	35.78	36.89
Configuration C								
1	1.031	3	2.7	2.5	5.818	5.776	6.30	6.49
10	10.310	12	2.9	2.7	11.759	11.809	15.30	15.77
20	20.619	12	3.6	3.4	14.598	14.766	19.76	20.37
50	51.546	12	4.8	4.6	19.464	19.985	27.52	28.37
100	103.093	27	4.1	3.9	24.735	24.595	34.82	35.9
Configuration D								
1	1.031	4	2.4	2.2	5.794	5.610	6.46	6.66
10	10.310	4	4.8	4.6	11.587	12.188	15.28	15.75
20	20.619	114	1.2	1.0	14.846	14.243	19.79	20.40
50	51.546	114	1.6	1.4	19.795	18.030	28.79	29.68
100	103.093	114	2.0	1.8	24.740	21.710	37.75	38.92
<u>5-Year Mission Life</u>								
Configuration A								
1	1.136	1	5.0	4.8	5.0	5.24	8.23	9.35
10	11.364	7	3.8	3.6	11.4	12.51	15.33	17.42
20	22.727	7	4.8	4.6	14.4	15.20	19.74	22.43
50	56.818	19	4.0	3.8	20.0	20.34	26.23	29.81
100	113.636	19	5.0	4.8	25.0	25.37	33.62	38.20
Configuration C								
1	1.36	3	2.8	2.6	6.034	5.893	5.92	6.73
10	11.364	12	3.0	2.8	12.165	12.098	14.39	16.35
20	22.727	12	3.7	3.5	15.003	15.340	18.58	21.11
50	56.818	12	5.0	4.8	20.275	20.225	25.85	29.37
100	113.636	27	4.2	4.0	25.338	25.748	32.70	37.16
Configuration D								
1	1.136	4	2.45	2.25	5.914	5.901	6.08	6.91
10	11.364	4	5.0	4.80	12.070	12.340	14.35	16.31
20	22.727	114	1.20	1.00	14.846	15.649	18.59	21.12
50	56.818	114	1.60	1.40	19.795	19.822	27.05	30.74
100	113.636	114	2.00	1.80	24.744	23.878	35.49	40.33

Table 3.4. ^{90}SrO Power Source Dimensions and Heat Flux Data for Various Power Levels

Source Power [kW(th)]		Number of Capsules	Diameter (in.)			Source Length (in.)	Heat Flux (W/in. ²)	
EOL	BOL		Capsule	Pellet	Source		EOL	BOL
<u>1-Year Mission Life</u>								
Configuration A								
1	1.031	1	4.0	3.8	4.0	4.380	12.47	12.85
10	10.310	7	3.2	3.0	9.6	9.392	23.36	24.08
20	20.619	7	4.0	3.8	12.0	11.585	30.17	31.10
50	51.546	7	5.0	4.8	15.0	17.883	41.80	43.09
100	103.093	19	4.0	3.8	20.0	20.919	51.47	53.06
Configuration C								
1	1.031	3	2.25	2.05	4.849	4.944	8.91	9.18
10	10.310	12	2.45	2.25	9.935	9.722	21.81	22.48
20	20.619	12	3.00	2.80	12.165	12.410	28.30	29.18
50	51.546	12	4.05	3.85	16.423	16.262	39.60	40.82
100	103.093	12	5.00	4.80	20.275	20.763	50.81	52.38
Configuration D								
1	1.031	4	2.00	1.80	4.828	4.823	9.11	9.39
10	10.310	4	4.05	3.85	9.777	9.49	21.94	22.62
20	20.619	4	5.00	4.80	12.070	12.658	28.22	29.09
50	51.546	114	1.30	1.10	16.084	16.590	40.17	41.41
100	103.093	114	1.60	1.40	19.795	20.349	53.16	54.80
<u>5-Year Mission Life</u>								
Configuration A								
1	1.136	1	4.2	4.0	4.2	4.360	11.732	13.33
10	11.364	7	3.2	3.0	9.6	10.302	21.960	24.95
20	22.727	7	4.0	3.8	12.0	12.729	28.330	32.19
50	56.818	19	3.4	3.2	17.0	16.383	37.620	42.75
100	113.	19	4.2	4.0	21.0	20.813	48.410	55.01
Configuration C								
1	1.136	3	2.35	2.15	5.064	4.953	8.400	9.54
10	11.364	12	2.50	2.30	10.137	10.228	20.53	23.33
20	22.727	12	3.10	2.90	12.570	12.748	26.61	30.24
50	56.818	12	4.15	3.95	16.828	17.005	37.21	42.28
100	113.636	27	3.50	3.30	21.115	21.502	47.02	53.43
Configuration D								
1	1.136	4	2.05	1.85	4.949	5.011	8.59	9.76
10	11.364	4	4.20	4.00	10.139	10.149	20.63	23.44
20	22.727	14	2.90	2.70	12.719	12.611	26.38	29.98
50	56.818	114	1.35	1.15	16.702	16.727	38.00	43.18
100	113.636	114	1.65	1.45	20.414	20.897	50.13	56.96

Table 3.5. $^{144}\text{Ce}_2\text{O}_3$ Power Source Dimensions and Heat Flux Data for Various Power Levels

Source Power [kW(th)]		Number of Capsules	Diameter (in.)			Source Length (in.)	Heat Flux (W/in. ²)	
BOL	BOL		Capsule	Pellet	Source		EOL	BOL
<u>1-Year Mission Life</u>								
Configuration A								
1	2.439	1	2.6	2.4	2.6	2.660	30.92	75.41
10	24.390	7	2.0	1.8	6.0	5.985	59.04	144.
20	48.780	7	2.4	2.2	7.2	7.843	77.27	188.46
50	121.951	7	3.2	3.0	9.6	10.371	109.28	266.53
100	243.900	7	4.0	3.8	12.0	12.807	141.04	344.
Configuration C								
1	2.439	3	1.45	1.25	3.125	3.154	26.60	64.88
10	24.390	3	2.90	2.70	6.249	6.188	54.70	133.41
20	48.780	12	1.90	1.70	7.704	7.674	71.69	174.85
50	121.951	12	2.50	2.30	10.137	10.296	102.19	249.24
100	243.900	12	3.10	2.90	12.57	12.826	132.50	323.17
Configuration D								
1	2.439	4	1.30	1.10	3.138	3.070	21.87	53.34
10	24.390	4	2.55	2.35	6.156	6.131	56.15	136.95
20	48.780	4	3.20	3.00	7.725	7.411	73.10	178.29
50	121.951	4	4.20	4.00	10.139	10.217	102.69	250.46
100	243.900	14	3.00	2.80	13.158	11.833	131.39	320.46
<u>5-Year Mission Life</u>								
Configuration A								
1	83.333	7	3.0	2.8	9.0	8.242	2.78	231.67
10	833.333	19	3.8	3.6	19.0	17.755	6.15	512.5
20	1666.667	19	4.6	4.4	23.0	23.603	7.89	657.5
50	4166.667	37	4.6	4.4	32.2	30.159	10.68	890.
100	8333.333	61	4.4	4.2	39.6	39.986	13.44	1120.
Configuration C								
1	83.333	12	2.3	2.1	9.326	8.529	2.59	215.83
10	833.333	12	4.6	4.4	18.653	18.789	6.07	505.83
20	1666.667	27	3.9	3.7	23.529	23.491	7.67	639.16
50	4166.667	48	4.0	3.8	32.096	31.152	10.51	875.83
100	8333.333	48	5.0	4.8	40.120	38.919	13.45	1120.83
Configuration D								
1	83.333	4	3.80	3.60	9.173	8.696	2.61	217.5
10	833.333	114	1.50	1.30	18.558	17.959	6.30	525.
20	1666.667	114	1.85	1.65	22.888	22.177	8.27	689.16
50	4166.67	114	2.40	2.20	29.693	30.983	11.70	975.
100	8333.333	114	3.00	2.80	37.116	38.135	15.13	1260.83

Table 3.6. ^{60}Co Power Source Dimensions and Heat Flux Data for Various Power Levels

Source Power [kW(th)]		Number of Capsules	Diameter (in.)			Shield Thickness (in.)	Source Length (in.)	Heat Flux (W in. ²)	
EOL	BOL		Capsule	Pellet	Source			EOL	BOL
<u>1-Year Mission Life</u>									
<u>Configuration A</u>									
1	1.140	1	1.55	1.35	7.630	3.04	7.593	3.66	4.17
10	11.402	1	2.95	2.75	9.750	3.40	9.741	22.32	25.45
20	22.805	1	3.65	3.45	10.630	3.49	10.581	37.65	42.93
50	57.012	1	4.85	4.65	12.010	3.58	11.928	73.93	84.30
100	114.025	7	2.25	2.05	14.130	3.69	14.152	106.20	121.09
<u>Configuration C</u>									
1									
10	11.402	3	1.65	1.45	10.356	3.40	10.227	19.98	22.78
20	22.805	3	2.00	1.80	11.290	3.49	11.278	33.40	38.08
50	57.012	3	2.65	2.45	12.871	3.58	12.784	64.30	73.32
100	114.025	12	1.75	1.55	14.476	3.65	14.279	102.26	116.60
<u>Configuration D</u>									
1									
10	11.402	4	1.45	1.25	10.300	3.40	10.254	20.06	22.87
20	22.805	4	1.80	1.60	11.325	3.49	11.085	33.57	38.28
50	57.012	4	2.35	2.15	12.833	3.58	12.650	65.21	74.36
100	114.025	4	2.90	2.70	14.381	3.69	14.207	103.43	117.94
<u>5-Year Mission Life</u>									
<u>Configuration A</u>									
1	1.931	1	1.80	1.60	7.76	2.98	7.684	3.55	6.85
10	19.305	1	3.45	3.25	10.09	3.32	10.105	20.81	40.17
20	38.610	1	4.30	4.10	11.06	3.38	10.979	34.86	67.30
50	96.525	7	2.15	1.95	13.51	3.53	13.434	58.31	112.57
100	193.050	7	2.65	2.45	15.11	3.58	15.102	92.92	179.38
<u>Configuration C</u>									
1	1.931	3	1.00	0.80	8.115	2.98	8.091	3.23	6.24
10	19.305	3	1.90	1.70	10.734	3.32	10.753	18.38	35.48
20	38.610	3	2.35	2.15	11.824	3.38	11.768	30.42	58.73
50	96.525	3	3.10	2.90	13.740	3.53	13.757	56.10	108.30
100	193.050	12	2.05	2.90	15.473	3.58	15.274	89.36	172.51
<u>Configuration D</u>									
1									
10	19.305	4	1.70	1.50	10.744	3.32	10.620	18.52	35.75
20	38.610	4	2.10	1.90	11.850	3.38	11.589	30.67	59.21
50	96.525	4	2.75	2.55	13.698	3.53	13.571	56.94	109.92
100	193.050	4	3.40	3.20	15.368	3.58	15.295	90.23	174.19

Table 3.7. $^{90}\text{SrTiO}_3$ Optimum Power Source Dimensions and Heat Flux Data

Source Power [kW(th)]		Capsule Array	Diameter (in.)			Source Length (in.)	Heat Flux (W/in. ²)	
EOL	BOL		Capsule	Pellet	Source		EOL	BOL
<u>1-Year Mission Life</u>								
1	1.031	A-1	4.8	4.6	4.800	5.17	8.76	9.03
10	10.310	A-7	3.8	3.6	11.400	11.39	16.34	16.84
20	20.619	A-7	4.8	4.6	14.400	13.84	21.00	21.65
50	51.546	D-114	1.6	1.4	19.795	18.03	28.79	29.68
100	103.093	D-114	2.0	1.8	24.740	21.71	37.75	38.92
<u>5-Year Mission Life</u>								
1	1.136	A-1	5.0	4.8	5.000	5.240	8.23	9.35
10	11.364	A-7	3.8	3.6	11.400	12.510	15.33	17.42
20	22.727	A-7	4.8	4.6	14.400	15.200	19.74	22.43
50	56.818	D-114	1.6	1.4	19.795	19.822	27.05	30.74
100	113.636	D-114	2.0	1.8	24.744	23.878	35.49	40.33

Table 3.8. ^{90}SrO Optimum Power Source Dimensions and Heat Flux Data

Source Power [kW(th)]		Capsule Array	Diameter (in.)			Source Length (in.)	Heat Flux (W/in. ²)	
EOL	BOL		Capsule	Pellet	Source		EOL	BOL
<u>1-Year Mission Life</u>								
1	1.031	A-1	4.0	3.8	4.000	4.380	12.47	12.85
10	10.310	A-7	3.2	3.0	9.600	9.392	23.36	24.08
20	20.619	A-7	4.0	3.8	12.000	11.585	30.17	31.10
50	51.546	A-7	5.0	4.8	15.000	17.883	41.80	43.09
100	103.093	A-114	1.6	1.4	19.795	20.349	53.16	54.80
<u>5-Year Mission Life</u>								
1	1.136	A-1	4.2	4.0	4.200	4.360	11.73	13.33
10	11.364	A-7	3.2	3.0	9.600	10.302	21.96	24.95
20	22.727	A-7	4.0	3.8	12.000	12.729	28.33	32.19
50	56.818	D-114	1.35	1.15	16.702	16.727	38.00	43.18
100	113.636	D-114	1.65	1.45	20.414	20.897	50.13	56.96

Table 3.9. $^{144}\text{Ce}_2\text{O}_3$ Optimum Power Source Dimensions and Heat Flux Data

Source Power [kW(th)]		Capsule Array	Diameter (in.)			Source Length (in.)	Heat Flux (W/in. ²)	
EOL	BOL		Capsule	Pellet	Source		EOL	BOL
<u>1-Year Mission Life</u>								
1	2.439	A-1	2.6	2.4	2.6	2.660	30.92	75.41
10	24.390	A-7	2.0	1.8	6.0	5.985	59.04	144.
20	48.780	A-7	2.4	2.2	7.2	7.843	77.27	188.46
50	121.951	A-7	3.2	3.0	9.6	10.371	109.28	266.53
100	243.900	A-7	4.0	3.8	12.0	12.807	141.04	344.
<u>5-Year Mission Life</u>								
1	83.333	A-7	3.00	2.80	9.000	8.242	2.78	231.67
10	833.333	D-114	1.50	1.30	18.558	17.959	6.30	525.
20	1666.667	D-114	1.85	1.65	22.888	22.177	8.27	689.16
50	4166.667	D-114	2.40	2.20	29.693	30.983	11.70	975.
100	8333.333	D-114	3.00	2.80	37.116	38.135	15.13	1260.83

Table 3.10. ^{60}Co Optimum Power Source Dimensions and Heat Flux Data

Source Power [kW(th)]		Capsule Array	Diameter (in.)			Shield Thickness (in.)	Source Length (in.)	Heat Flux (W/in. ²)	
EOL	BOL		Capsule	Pellet	Source			EOL	BOL
1-Year Mission Life									
1	1.140	A-1	1.55	1.35	7.63	3.04	7.593	3.66	4.17
10	11.402	A-1	2.95	2.75	9.75	3.40	9.741	22.34	25.47
20	22.805	A-1	3.65	3.45	10.63	3.49	9.921	39.31	44.82
50	57.012	A-1	4.85	4.65	12.01	3.58	11.928	73.90	84.26
100	114.025	A-7	2.25	2.05	14.13	3.69	14.152	106.17	121.06
5-Year Mission Life									
1	1.931	A-1	1.80	1.60	7.76	2.98	7.684	3.55	6.85
10	19.305	A-1	3.45	3.25	10.09	3.32	10.105	20.82	40.19
20	38.610	A-1	4.30	4.10	11.06	3.38	10.979	34.87	67.32
50	96.525	A-7	2.15	1.95	13.51	3.52	13.434	58.35	112.64
100	193.05	A-7	2.65	2.45	15.11	3.58	15.102	92.99	179.52

Table 3.11. EOL Surface Temperatures of Various Isotopic Power Sources for 1- and 5-Year Mission Lives^a

EOL Power [kW(th)]	⁹⁰ SrO		⁹⁰ SrTiO ₃		¹⁴⁴ Ce ₂ O ₃		⁶⁰ Co	
	EOL	Surface	EOL	Surface	EOL	Surface	EOL	Surface
	Heat Flux (W/in. ²)	Temp (°F)	Heat Flux (W/in. ²)	Temp (°F)	Heat Flux (W/in. ²)	Temp (°F)	Heat Flux (W/in. ²)	Temp (°F)
<u>1-Year Mission Life</u>								
1	12.47	867	8.76	746	30.92	1217	3.66	516
10	23.36	1116	16.34	975	59.04	1563	22.34	1090
20	30.17	1230	21.00	1076	77.27	1735	39.31	1350
50	41.80	1390	28.79	1220	109.28	2000	73.90	1720
100	53.16	1520	37.75	1340	141.04	2220	106.17	1970
<u>5-Year Mission Life</u>								
1	11.73	850	8.23	734	2.78	450	3.55	508
10	21.96	1106	15.33	952	6.30	665	20.82	1070
20	28.33	1207	19.74	1047	8.27	755	34.87	1290
50	38.00	1328	27.05	1192	11.70	870	58.35	1570
100	50.13	1490	35.49	1310	15.13	965	92.99	1900

^aSources assumed to be suspended in air.

Table 3.12. BOL Surface Temperatures of Optimum Isotopic Power Sources for 1- and 5-Year Mission Lives^a

EOL Power [kW(th)]	⁹⁰ SrO		⁹⁰ SrTiO ₃		¹⁴⁴ Ce ₂ O ₃		⁶⁰ Co	
	EOL	Surface	EOL	Surface	EOL	Surface	EOL	Surface
	Heat Flux (W/in. ²)	Temp (°F)	Heat Flux (W/in. ²)	Temp (°F)	Heat Flux (W/in. ²)	Temp (°F)	Heat Flux (W/in. ²)	Temp (°F)
<u>1-Year Mission Life</u>								
1	9.03	763	12.85	882	75.41	1713	4.17	548
10	16.84	991	24.08	1132	144.00	2233	25.47	1156
20	21.65	1092	31.10	1244	188.46	2507	44.82	1419
50	29.68	1225	43.09	1399	266.53	2918	84.26	1816
100	38.92	1353	54.80	1535	344.00	3262	121.06	2090
<u>5-Year Mission Life</u>								
1	9.35	775	13.33	896	231.67	2736	6.85	685
10	17.42	1005	24.95	1147	525.00	3940	40.19	1332
20	22.43	1106	32.19	1261	689.16	4463	67.32	1649
50	30.74	1242	43.18	1401	975.00	5235	112.64	2030
100	40.33	1372	56.96	1558	1260.83	5825	179.52	2457

^aSources assumed to be suspended in air.

Table 3.13. EOL Surface Temperatures of Optimum Isotopic Power Sources for 1- and 5-Year Mission Lives^a

EOL Power [kW(th)]	⁹⁰ SrO		⁹⁰ SrTiO ₃		¹⁴⁴ Ce ₂ O ₃		⁶⁰ Co	
	EOL Heat Flux (W/in. ²)	Surface Temp (°F)	EOL Heat Flux (W/in. ²)	Surface Temp (°F)	EOL Heat Flux (W/in. ²)	Surface Temp (°F)	EOL Heat Flux (W/in. ²)	Surface Temp (°F)
<u>1-Year Mission Life</u>								
1	8.76	800	12.47	924	30.92	1305	3.66	549
10	16.34	1041	23.36	1192	59.04	1672	22.34	1172
20	21.00	1150	30.17	1310	77.27	1857	39.31	1442
50	28.79	1295	41.80	1460	109.28	2144	73.90	1843
100	37.75	1433	53.16	1622	141.04	2387	106.17	2128
<u>5-Year Mission Life</u>								
1	8.23	781	11.73	904	2.78	484	3.55	541
10	15.33	1012	21.96	1160	6.30	711	20.82	1141
20	19.74	1118	28.33	1275	8.27	800	34.87	1378
50	27.05	1260	38.00	1423	11.70	920	58.35	1686
100	35.49	1394	50.13	1584	15.13	1024	92.99	2015

^aSources assumed to be resting on 1-ft-thick concrete.Table 3.14. BOL Surface Temperatures of Optimum Isotopic Power Sources for 1- and 5-Year Mission Lives^a

EOL Power [kW(th)]	⁹⁰ SrO		⁹⁰ SrTiO ₃		¹⁴⁴ Ce ₂ O ₃		⁶⁰ Co	
	EOL Heat Flux (W/in. ²)	Surface Temp (°F)	EOL Heat Flux (W/in. ²)	Surface Temp (°F)	EOL Heat Flux (W/in. ²)	Surface Temp (°F)	EOL Heat Flux (W/in. ²)	Surface Temp (°F)
<u>1-Year Mission Life</u>								
1	9.03	811	12.85	936	75.41	1832	4.17	584
10	16.84	1054	24.08	1205	144.00	2405	25.47	1227
20	21.65	1162	31.10	1322	188.46	2693	44.82	1514
50	29.68	1308	43.09	1478	266.53	3141	84.26	1937
100	38.92	1450	54.80	1642	344.00	3520	121.06	2248
<u>5-Year Mission Life</u>								
1	9.35	824	13.33	951	231.67	2970	6.85	729
10	17.42	1063	24.95	1215	525.00	4284	40.19	1450
20	22.43	1173	32.19	1335	689.16	4861	67.32	1764
50	30.74	1320	43.18	1492	975.00	5689	112.64	2181
100	40.33	1462	56.96	1668	1260.83	6445	179.52	2655

^aSources assumed to be resting on 1-ft-thick concrete.

Fig. 3.11. EOL Heat Flux vs EOL Power for $^{90}\text{SrTiO}_3$ - 1-Year Mission Life.

Fig. 3.12. EOL Heat Flux vs EOL Power for $^{90}\text{SrTiO}_3$ - 5-Year Mission Life.

Fig. 3.13. EOL Heat Flux vs EOL Power for ^{90}SrO - 1-Year Mission Life.

Fig. 3.14. EOL Heat Flux vs EOL Power for ^{90}SrO - 5-Year Mission Life.

Fig. 3.15. EOL Heat Flux vs EOL Power for ^{60}Co - 1-Year Mission Life.

Fig. 3.16. EOL Heat Flux vs EOL Power for ^{60}Co - 5-Year Mission Life.

Fig. 3.17. EOL Heat Flux vs EOL Power for $^{144}\text{Ce}_2\text{O}_3$ - 1-Year Mission Life.

Fig. 3.18. EOL Heat Flux vs EOL Power for $^{144}\text{Ce}_2\text{O}_3$ - 5-Year Mission Life.

FUEL CAPSULE DESIGN

Design Criteria

The applications considered in this program require that the fuel capsule be able to withstand temperatures of 1832°F at a pressure of 26 psia, as in a Brayton-cycle heat exchanger, and pressures of up to 10,000 psi at a temperature of 392°F , as in an undersea unit exposed to the seawater at a depth of about 20,000 ft. Another, and very important, requirement was that the material be resistant to oxidation at elevated temperature. In view of the first requirement, only nickel- or cobalt-based superalloys and refractory metal alloys were considered; the refractories were eliminated because of cost, handling, welding, and oxidation problems associated with their use.

Compositions and melting points of the various materials under consideration are given in Table 3.15, normal and galvanic corrosion rates in seawater in Table 3.16, and short-term tensile properties in Table 3.17. Figures 3.19 and 3.20 show the creep properties and Figs. 3.21 and 3.22 the stress rupture properties of the materials.

Table 3.15. Compositions and Melting Points of Oxidation-Resistant Alloys^a

Alloy	Composition (wt %)											Melting Point (°C)
	Ni	Cr	Fe	Co	Ti	Al	Mo	Mn	Si	W	Cb	
<u>Nickel-Based Alloys</u>												
Incoloy 825	42	21	30		1	0.2	3	1	0.5			1360
Inconel	72	15.5	8					1				1395
Inconel 600	77	15.5	7					0.2	0.2			1370
Rene 41	55	19		11	3	1.5	10					1315
Udimet 700	53	15	1	18.5	3	4	5					1220
Hastelloy N	69	7	5		0.5		16.5	1	1	0.5		1326
Hastelloy X	47	22	18	1.5			9	1	1	1		1260
Hastelloy C	55	15.5	5.5	2.5			16	1	1	4		1270
TD Nickel (2% ThO ₂)	98											1450
TD Nickel- Chromium	78	20										1430
RA-333	45	25	18	3			3	2	1	3		1300
<u>Cobalt-Based Alloys</u>												
Haynes 25	10	20	3	50				1.5	1	15		1330
Haynes 152	1	21	1	64						11	2	1315
Haynes Stellite 21	2.5	27.5	2	61			5.5	1	1			1350

^aFrom: C. P. Ross, C. L. Angerman, and F. D. R. King, Development of ⁶⁰Co Capsules for Heat Sources, DP-1096 (June 1967).

Table 3.16. Normal and Galvanic Corrosion Rates of SNAP-21 Materials in Seawater^a

Material	Seawater ^b Condition	Normal Corrosion Rate (mpy)	Galvanic Corrosion Effect ^c (mpy)
Al	A	9.6×10^{-2}	1.1×10^2
	D	3.1×10^{-2}	1.1
Cu	A	2.9	1.4
	D	8.3×10^{-2}	9.7×10^{-2}
Hastelloy C	A	4.3×10^{-3}	3.8×10^{-1}
	D	4.2×10^{-3}	8.0×10^{-5}
Hastelloy X	A	8.0×10^{-4}	
	D	4.0×10^{-3}	
Nickel	A	9.8×10^{-4}	2.8×10^{-3}
	D	2.2×10^{-3}	1.6×10^{-4}
Type 304 stainless steel	A	2.3×10^{-3}	5.7×10^{-1}
	D	4.0×10^{-3}	1.9×10^{-5}
Ta	A	9.6×10^{-4}	6.2×10^{-3}
	D	4.8×10^{-3}	1.6×10^{-4}
Ti 621	A	7.5×10^{-4}	5.0×10^{-4}
	D	7.1×10^{-3}	2.0×10^{-4}
U-8% Mo	A	1.2	7.7
	D	6.8×10^{-1}	6.4×10^{-3}

^aFrom: D. A. Kubose and H. I. Cordova, Electrochemical Corrosion Studies of Galvanically Coupled SNAP-21 Materials, USNRDL-TR-68-26 (Jan. 15, 1968).

^bA = aerated; D = deaerated.

^cMeasured with unequal areas of anode and cathode.

Table 3.17. Tensile Properties of Encapsulating Materials^a

Alloy	Temp (°C)	Short-Term Tensile Properties		
		Ultimate Tensile Strength (psi)	Yield Strength (psi)	Elongation (%)
Inconel 600	760	28,000	17,000	46
	982	8,000	4,000	118
Hastelloy C	815	51,000		36
	1037	14,000		36
Hastelloy X	871	36,000	25,000	51
	1093	13,000	8,000	40
	1204	5,000	4,000	31
Haynes 25	815	50,000		15
	982	23,000		16
	1260	5,000		19
Haynes 152	760	94,000	49,000	10
	1037	26,000	17,000	
	1149	15,000	10,000	
Haynes Stellite 21	760	66,000		11
	982	32,000		36
TD Nickel	871	23,000	21,000	0.6
	1037	16,000	15,000	0.5
	1315	11,000	10,000	0.4

^aFrom: D. A. Kubose and H. I. Cordova, Electrochemical Corrosion Studies of Galvanically Coupled SNAP-21 Materials, USNRDL-TR-68-26 (Jan. 15, 1968).

Fig. 3.19. Conditions for Secondary Creep Rate of $10^{-4}\%$ /hr. [From: C. P. Ross, C. L. Angerman, and F. D. R. King, Development of ^{60}Co Capsules for Heat Sources, DP-1096 (June 1967).]

Fig. 3.20. Conditions for Secondary Creep Rate of 10^{-5} %/hr. [From: C. P. Ross, C. L. Angerman, and F. D. R. King, Development of ^{60}Co Capsules for Heat Sources, DP-1096 (June 1967).]

Fig. 3.21. Stress Rupture Properties of Capsule Alloys. [From: C. P. Ross, C. L. Angerman, and F. D. R. King, Development of ^{60}Co Capsules for Heat Sources, DP-1096 (June 1967).]

Fig. 3.22. 1000°C Stress Rupture Data Extrapolated Through One Half-Life of ^{60}Co . [From: C. P. Ross, C. L. Angerman, and F. D. R. King, Development of ^{60}Co Capsules for Heat Sources, DP-1096 (June 1967).]

Hastelloy C (or Hastelloy C-276) was selected as the encapsulating material because of its good strength characteristics and excellent oxidation resistance at elevated temperatures, its excellent resistance to normal corrosion and corrosion by galvanic effect in both aerated and deaerated seawater, and good short-term tensile properties and acceptable creep properties at elevated temperatures. In addition, considerable experience has been gained in the handling, machining, welding, and inspection of this alloy, and actual operating knowledge has been acquired in the SNAP-7, SNAP-21, and SNAP-23 programs. Development problems associated with its use for either ^{90}Sr or ^{60}Co are equal.

A typical fueling procedure involves hot-pressing the SrTiO_3 powder at a pressure of 1400 psig in a furnace heated to 2237°F. A nickel wire screen is inserted in the powder to reinforce the pressed pellet. As many pellets as are necessary to fill the container are inserted in a thin metal liner. The liner is used for ease of handling and contamination control. It is inserted in the Hastelloy C capsule and the lid is weld-sealed by either gas tungsten arc (GTA) or electron beam (EB) methods. The surface of the capsule is decontaminated and the welded area is leak checked and inspected for penetration and voids. The capsule is then ready for insertion in a heat exchanger. In the case of ^{60}Co the fuel

is produced by irradiation of ^{59}Co metal wafers. These wafers are removed from their irradiation cans and loaded directly into fuel capsules without the need of any pellet pressing operation.

Stress Analysis

It is seen from the power source dimension and heat flux data given in Tables 3.3 through 3.6 that no one capsule diameter is "best" from a heat flux standpoint for all of the ^{90}Sr cases. Capsule designs which have the maximum surface heat flux will give the minimum volume source. However, as is discussed in Chapter 4, it is necessary to deviate from a minimum volume capsule array when designing the heat block to ensure that capsule surface temperatures do not exceed operating limits of construction materials. Therefore inside diameters of 1.0 to 2.5 in. with L/D ratios of 3 and 4 were arbitrarily selected for ^{90}Sr fuel capsules. Experience in the SNAP programs has indicated that 4 in. is the maximum practical limit for inside diameter, which is consistent with existing ^{90}Sr fuel preparation, encapsulation, and inspection technology.

The ^{60}Co capsule was designed with an inside diameter of 0.75 in. and lengths of 3.4 to 4.0 in. This capsule size was selected to match the 0.745-in.-dia ^{60}Co wafers produced at the SRO plant. Other sizes and shapes of ^{60}Co can be produced if desired; however, the individual pieces that are placed in the reactor must be thin ($\sim \leq 1/16$ in.) if a fairly uniform specific activity throughout the metal is to be achieved.

The maximum amount of thermal power that could be contained in each of the proposed source capsules was calculated for $^{90}\text{SrTiO}_3$, ^{90}SrO , and ^{60}Co . These data are presented in Table 3.18.

Table 3.18. Capsule Thermal Power

Inside Diameter (in.)	L/D Ratio	Inside Length (in.)	Power [W(th)]		
			$^{90}\text{SrTiO}_3$	^{90}SrO	^{60}Co
1.0	3	3.396	37	70	
	4	4.636	51	95	
1.5	3	5.010	123	232	
	4	6.850	167	317	
2.0	3	6.716	294	553	
	4	9.176	402	756	
2.5	3	7.978	454	1027	
	4	10.928	747	1406	
0.75	3	3.06			908
	4	3.66			1086

A stress analysis was made to determine the cap and wall thickness requirements for the capsules. Four stress points were examined: the center of the cap (S_1), the edge of the cap (S_2), the top edge of the cylinder (S_3), and a point on the cylinder wall midway between the ends (S_4). The formulas used are as follows:

$$S_1 = \frac{6M}{t_c^2} - \frac{V}{t_c^2} - \frac{3W}{8\pi m t_c^2} (3m + 1) ,$$

$$S_2 = \frac{6M}{t_c^2} - \frac{V}{t_c} ,$$

$$S_3 = \frac{2V}{t_w} \lambda R - \frac{2M}{t_w} \lambda^2 R - \frac{pR}{t_w} ,$$

$$S_4 = \frac{2V}{t_w} (\lambda R e^{-\lambda x} \cos \lambda x) - \frac{pR}{t_w} - \frac{2\lambda^2 R m e^{-\lambda x} (\cos \lambda x - \sin \lambda x)}{t_w} ,$$

where

M = moment, in.-lb/linear in.

$$= \frac{\frac{2pR^2\lambda^3 Et_c D_w}{Et_w(1 - \nu/2) [Et_c + 2RD_w\lambda^3(1 - \nu)]} + \frac{pR^3\lambda^2 D_w}{4D_c(1 + \nu)}}{2\lambda + \frac{2R\lambda^2 D_w}{D_c(1 + \nu)} - \frac{\lambda Et_c}{Et_c + 2D_w\lambda^3 R(1 - \nu)}} ,$$

E = modulus of elasticity, psi,

D_c, D_w = flexibility parameter for end cap and for wall, respectively

$$= Et^3/12(1 - \nu^2) ,$$

m = inverse of Poisson's ratio,

p = external pressure, psi,

R = capsule radius, in.,

t_c = thickness of end cap, in.,

t_w = cylinder wall thickness, in.,

V = transverse shear, lb/linear in.

$$= M\{2\lambda + [2R\lambda^2 D_w/D_c(1 + \nu)]\} - pR^3\lambda^2 D_w/4D_c(1 + \nu) ,$$

x = distance from top edge of cylinder to midpoint of its length, in.,

W = total applied load, lb

= pressure $\times$ area,

λ = cylinder flexibility parameter

$$= [3(1 - \nu^2)/R^2 t^2]^{1/4} ,$$

ν = Poisson's ratio.

The above equations were programmed for the IBM 360/75 computer and used to establish the cylinder wall and cap thicknesses for the ^{90}Sr and ^{60}Co capsules.

Figure 3.23 is a schematic of the fuel capsule showing all the pertinent dimensions. The trepan in the end cap for GTA welding would be eliminated if EB welding were used. For comparison purposes, the tensile yield strength of Hastelloy C was decreased from 60,000 psi, which was used in the 2.5-in.-dia case, to 50,000 psi, which was used in the 1.0-, 1.5-, and 2.0-in. cases. In all cases the structural support provided by the fuel pellets was ignored.

FUEL	I.D.	A	B	C	D	E	F	G	I	J	K	L	M
Sr-90	1"	1.0955	.95	.302	.191	.163	4.96	.163	1.241	1.0965	1.001	1.020	.036
		1.0950			.189	.161	3.72	.161	1.239	1.0955	.999		.035
	1 1/2"	1.6355	1.41	.412	.301	.256	7.36	.256	1.841	1.6365	1.501	1.520	.058
		1.6350			.299	.254	5.52	.254	1.839	1.6355	1.499		.056
	2"	2.1835	1.89	.502	.391	.333	9.84	.333	2.461	2.1845	2.001	2.030	.076
		2.1830			.389	.331	7.38	.331	2.459	2.1835	1.999		.073
	2 1/2"	2.7235	2.36	.602	.491	.417	11.76	.417	2.941	2.7245	2.501	2.540	.095
		2.7230			.489	.415	8.81	.415	2.939	2.7235	2.499		.091
*Co-60	3"	.8785	.762	.312	.201	.171	4.00	.171	.941	.8795	.751	.818	.040
		.8780			.199	.169	3.40	.169	.939	.8785	.749		.038

*BASED ON CAPSULE DESIGN SHOWN IN SRO REPORT DP-1096.

MAT'L:
HASTELLOY "C"

Fig. 3.23. Fuel Capsule Design.

3. REFERENCES

- ¹B. Porter and E. Brown, "Melting Point of Inorganic Fluorides," J. Am. Ceram. Soc. 45, 49 (1962).
- ²S. J. Rimshaw and E. E. Ketchen, Strontium-90 Data Sheets, ORNL-4358 (March 1969).
- ³J. W. Joseph et al., Radioisotope Cobalt for Heat Sources, DP-1012 (1965).

4. SHIELDING AND HEAT BLOCK DESIGN

SHIELDING ANALYSIS

Calculations were made to determine¹ the thickness required to reduce the shield surface dose rate to 200 mR/hr and to determine the weights of the fuel block, which includes the shield, and of the fuel capsules for ^{90}SrO , $^{90}\text{SrTiO}_3$, and ^{60}Co for various fuel block power levels. The arrangements of fuel capsules in the fuel blocks and shields were limited to configurations that would be capable of dissipating their heat to the air by natural convection without exceeding the temperature limitation of the capsule alloy. For all the fuels this requirement resulted in a larger fuel-block volume than that of the minimum-volume capsule arrays discussed in Chapter 3. The fourth fuel candidate in the study, $^{144}\text{Ce}_2\text{O}_3$, was not included in these calculations since the data discussed in Chapter 3 show it to be an unattractive fuel form for the power levels and applications of interest. Calculations were also made for a mixed-fission-product heat source to obtain an indication of the effects of a relatively low-power-density fuel on the heat block and shield weight for comparative purposes. For the purpose of these calculations it is assumed that the mixed fission products are in the form of a phosphate glass, which has been proposed as one possible way of disposing of reactor fuel-reprocessing wastes in the future. Uranium, tungsten, nickel, and steel were selected for study as shielding materials; a comparison of a few of their characteristics is given below:

- Uranium: high radiation attenuation, low thermal conductivity, relatively expensive
- Tungsten: high radiation attenuation, high thermal conductivity, very expensive
- Nickel: low radiation attenuation, good thermal conductivity, moderately inexpensive
- Steel: low radiation attenuation, good thermal conductivity, inexpensive

For the ^{90}SrO and $^{90}\text{SrTiO}_3$ calculations the fuel pellet was assumed to have a diameter of 2.5 in. and to be encapsulated in Hastelloy C-276; the fuel capsule was assumed to have an outside diameter of 2.94 in., a length-to-diameter ratio of between 3 and 4, and an end cap thickness of about 1/2 in. The capsules would be inserted in 3-in.-dia holes and have the triangular pattern shown in Fig. 4.1, with the center-line spacing of the capsules varied from 3.5 to 4.5 in. Calculations were made for 7, 13, 19, 31, and 37 fuel positions, the number of which will vary with power level.

The ^{60}Co source was assumed to be made up of 0.745-in.-OD wafers of ^{60}Co encapsulated in a 0.94-in.-OD Hastelloy C-276 container, with the capsules inserted into 1-in.-dia holes having a triangular pitch as shown in Fig. 4.1. The center-line spacing of the capsule was varied from 1.5 to 2.5 in., and calculations were made for 3, 7, 13, 19, and 31 fuel positions.

Fig. 4.1. Fuel and Shield Arrangement.

Fig. 4.2. Fuel and Shield Configuration for Mixed-Fission-Product Fuel.

The configuration used for the mixed-fission-product fuel is shown in Fig. 4.2. The source diameter was varied from 16 to 24 in. and the power density from 0.10 to 0.25 W/cm³. In order that these power densities and diameters would not create excessive internal temperatures, provision was made for adding radial steel ribs to the outer diameter of the fueled region. The ribbed structure would be inserted into the shield before the source material is to be added. The temperature gradient in the ribs requires that they be slotted every 3 to 6 in. along the length to prevent bowing.

For shielding calculations the gamma emission rates (photons W⁻¹ sec⁻¹) for ⁹⁰SrTiO₃, ⁹⁰SrO, and ⁶⁰Co were taken directly from ref. 1 and are shown in Table 4.1. For the mixed fission products it was assumed that the reactor fuel would experience a burnup of 33,000 MWd/MT at a rate of 24.5 MW/MT and would be allowed to decay for 1 year. For shielding purposes the source term was assumed to be only the ¹⁴⁴Ce and its daughter product, ¹⁴⁴Pr. These isotopes account for about 60% of the thermal output and are the major sources of high-energy gammas. The source terms are then 60% of those given in ref. 1 for ¹⁴⁴Ce. An alternative calculation was made based on a nomograph² for shipping-cask shield requirements for irradiated fuels. The nomograph gives the requirement in inches of lead, and this was converted to steel by multiplying by 1.67. For a surface dose rate of 100 mR/hr the converted shield thickness obtained for steel was 15 to 15.4 in., while the calculation gives a thickness of 15.2 to 15.9 in.

Table 4.1. Gamma Emission Rates

Energy (MeV)	Gamma Emission Rate (photons W ⁻¹ sec ⁻¹)			
	SrO	SrTiO ₃	Co	Mixed Fission Products
0.25	1.923(11) ^a	1.705(11)		
0.35				4.92(11)
0.50	8.695(10)	6.612(10)		
0.70				4.49(10)
0.80	2.649(10)	1.958(10)		
0.95				3.69(10)
1.10	8.255(9)	6.103(9)		
1.172			2.385(12)	
1.333			2.385(12)	
1.40	2.229(9)	1.650(9)		
1.5				7.02(9)
1.55				5.79(9)
1.70	4.086(8)	3.027(8)		
2.00	2.806(7)	2.082(7)		
2.15				5.79(8)
2.20				2.24(10)
2.60				1.20(7)

^aRead: 1.923 × 10¹¹, etc.

The power density, capsule heat flux, and capsule length for SrO, SrTiO₃, and Co, and mixed fission products are given in Table 4.2. The capsule length is assumed to be 10% longer than the fueled length to allow for the end caps and internal spacing requirements. For the mixed fission products the capsule heat flux is based on the outside diameter of the cylindrical fuel region and the length per kilowatt is for the cylindrical region only. It does not include any allowance for the hemispherical bottom.

Table 4.2. Fuel Capsule Parameters

Fuel	Fuel Power Density		Capsule Heat Flux (Btu hr ⁻¹ ft ⁻²)	Capsule Length (in./kW)
	Btu/in. ³	W/cm ³		
SrO	78	1.40	6000	9.77
SrTiO ₃	45	0.85	3450	17.0
Co	2320	41.5	12,300–24,600 ^a	3.75
Mixed fission products	5.6–14	0.10–0.25	4,100 ^b –6,050 ^c	1.59 ^b –1.61 ^c

^aOnly about 25% of the cobalt energy is deposited in the capsule in which it is generated; however, the contribution from the adjacent capsule may increase this by a factor of 2.

^bFor 0.10 W/cm³ and 24-in.-dia fueled region.

^cFor 0.25 W/cm³ and 16-in.-dia fueled region.

Two calculational methods were used in determining dose rate as a function of thickness for the uranium and steel shields. In one method the dose rate was determined for a homogeneous fuel and shield material in the fueled region, which was treated as a cylindrical volume source; in the other method the dose rate was determined at the surface of the shield for a single element. The total dose rate at the surface was found by summing the contribution of the individual elements. For the case of the end shields the upper-limit value was used. The calculation procedures used are described elsewhere.³

The two methods were in good agreement for the steel shield. For the uranium shields the very high internal self-shielding by the source region caused the method treating the homogenized mixture as a cylindrical volume source to predict dose rates which were too low. The macroscopic cross section for the uranium shield is so large that most of the surface dose comes from the single fuel position nearest the surface. For the strontium fuels the element nearest the surface accounts for essentially all of the dose, whereas for the cobalt fuel it accounts for 80% of the total dose.

The density used for the uranium shields was that of pure uranium. The high temperature involved in the applications being considered here would require the uranium to be alloyed with molybdenum and the density consequently reduced by 3 or 4%; therefore the shield thickness shown for uranium should be increased by 3 or 4%. Although the theoretical density of tungsten (19.05 g/cm^3) is greater than that of uranium, the actual density for the large tungsten shields being considered here is 18.3 g/cm^3 . The dose rates with tungsten shielding were determined by summing the rates from single-element calculations as was done for the uranium shields.

The dose rate as a function of shield thickness for nickel was obtained from the calculations made for steel. Because the ratio of the adsorption coefficient of iron to nickel is independent of the gamma-ray energy over the range of interest, the value of the product of the macroscopic cross section and shield thickness can be made the same as for a case calculated for steel. The dose is the same as that obtained for the steel but at a point equivalent to the outside diameter of the steel shield. This point was moved back to the new shield surface to obtain the surface dose rate with nickel. The increased self-shielding within the fueled region that is due to the ~16% difference in the adsorption cross sections of the two materials is neglected in the method. This error is very small and within an accuracy with which one can read the curves used in the detailed calculations. A correction factor was calculated for the strontium fuel with 10- to 12-in.-thick nickel and steel shields and for cobalt with 15- to 19-in.-thick nickel and steel shields. This correction factor was found to be between 0.865 and 0.870 for all cases. A detailed calculation was made for a nickel shield for a 7-element SrO fuel case, and the true correction factor was 0.860. This factor was used for the nickel shield dimensions given in Tables 4.3-4.5. It is emphasized that the accuracy implied by this correction factor and the results shown in these tables and in Tables 4.6-4.8 and in Figs. 4.3-4.12 are for purposes of internal consistency and comparison purposes.

Table 4.3. Nickel and Tungsten Shield Dimensions and Total Weight for SrO Fuel

Power (kW)	Number of Elements	Capsule Length (in.)	Shield Length (in.)	3.5-in. Spacing		4.0-in. Spacing		4.5-in. Spacing	
				Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)
Nickel									
10	7	13.95	32.1	28.7	6,500	29.6	7,000	30.5	7,400
	13	7.51	25.9	33.7	7,300	35.4	8,100	37.1	8,900
30	13	22.53	40.9	33.7	11,300	35.4	12,500	37.1	13,800
	19	15.42	33.9	36.1	10,800	37.6	11,700	39.5	12,900
	31	9.45	27.9	40.8	11,300	42.8	12,500	45.3	14,000
50	19	25.70	44.1	36.1	13,800	37.6	15,100	39.5	16,700
	31	15.75	34.2	40.8	13,700	42.8	15,100	45.3	17,000
	37	13.20	31.6	43.3	14,300	45.6	15,900	48.5	18,000
Tungsten									
10	7	13.95	22.2	18.4	3,500	19.4	4,000	20.4	4,400
	13	7.51	15.8	23.5	4,200	25.3	4,900	27.0	5,600
30	13	22.53	30.8	23.5	7,700	25.3	9,100	27.0	10,600
	19	15.42	23.7	25.4	6,800	27.4	8,100	29.4	9,500
	31	9.45	17.7	29.9	7,100	32.6	8,700	35.2	10,300
50	19	25.70	34.0	25.4	9,600	27.4	11,400	29.4	13,400
	31	15.75	24.0	29.9	9,300	32.6	11,400	35.2	13,600
	37	13.20	21.5	32.4	9,900	35.4	12,200	38.4	14,600

Table 4.4. Nickel and Tungsten Shield Dimensions and Total Weight for SrTiO₃ Fuel

Power (kW)	Number of Elements	Capsule Length (in.)	Shield Length (in.)	<u>3.5-in. Spacing</u>		<u>4.0-in. Spacing</u>		<u>4.5-in. Spacing</u>	
				Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (in.)
<u>Nickel</u>									
10	13	13.07	30.0	32.4	7,700	34.1	8,600	35.8	9,500
	19	8.94	25.9	34.6	7,600	36.3	8,400	38.2	9,300
30	19	26.82	43.8	34.6	12,600	36.3	13,900	38.2	15,500
	31	16.44	33.4	39.3	12,400	41.5	13,900	44.0	15,700
	37	13.77	30.8	41.8	12,900	44.3	14,600	47.2	16,700
50	19	44.70	61.7	34.6	17,500	36.3	19,400	38.2	21,600
	31	27.39	44.4	39.3	16,200	41.5	18,200	44.0	20,600
	37	22.95	40.0	41.8	16,500	44.3	18,700	47.2	21,400
<u>Tungsten</u>									
10	13	13.07	20.7	22.9	5,000	24.7	5,900	26.4	6,900
	19	8.94	16.6	24.8	4,700	26.8	5,600	28.8	6,500
30	19	26.82	34.5	24.8	9,100	26.8	11,000	28.8	13,000
	31	16.44	24.1	29.3	8,900	32.0	11,000	34.6	13,100
	37	13.77	21.4	31.8	9,400	34.8	11,600	37.8	14,000
50	19	44.70	52.4	24.8	13,600	26.8	16,400	28.8	19,400
	31	27.39	35.0	29.3	12,500	32.0	15,500	34.6	18,600
	37	22.95	30.6	31.8	12,900	34.8	16,100	37.8	19,600

Table 4.5. Nickel and Tungsten Shield Dimensions and Total Weight for Co Fuel

Power (kW)	Number of Elements	Capsule Length (in.)	Shield Length (in.)	1.5-in. Spacing		2.0-in. Spacing		2.5-in. Spacing	
				Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)
Nickel									
10	3	12.50	41.6	33.9	12,000	34.5	12,500	35.0	12,800
	7	5.36	35.8	36.0	11,700	37.0	12,400	38.1	13,100
30	7	16.07	46.5	36.0	15,200	37.0	16,000	38.1	17,000
	13	8.65	40.0	38.0	14,600	39.7	15,900	41.5	17,400
	19	5.92	37.7	39.3	14,700	41.3	16,200	43.3	17,800
50	13	14.42	45.7	38.0	16,600	39.7	18,200	41.5	19,800
	19	9.87	41.7	39.3	16,200	41.3	17,900	43.3	19,700
	31	6.05	38.5	41.5	16,700	44.2	19,000	46.8	21,300
Tungsten									
10	3	12.50	26.9	17.2	4,100	17.8	4,400	18.4	4,700
	7	5.36	20.3	18.5	3,600	19.5	4,000	20.5	4,400
30	7	16.07	31.0	18.5	5,500	19.5	6,100	20.5	6,700
	13	8.65	23.8	20.7	5,300	22.4	6,200	24.2	7,200
	19	5.92	21.2	21.5	5,100	23.5	6,000	25.5	7,100
50	13	14.42	29.5	20.7	6,500	22.4	7,600	24.2	8,900
	19	9.87	25.1	21.5	6,000	23.5	7,100	25.5	8,400
	31	6.05	21.3	23.4	6,000	26.1	7,500	28.7	9,100

Table 4.6. Steel and Uranium Shield Dimensions and Total Weight for SrO Fuel

Power (kW)	Number of Elements	Capsule Length (in.)	Shield Length (in.)	3.5-in. Spacing		4.0-in. Spacing		4.5-in. Spacing	
				Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)
Steel									
10	7	13.95	35.1	31.8	7,810	32.6	8,220	33.5	8,680
	13	7.51	28.9	36.7	8,580	38.4	9,400	40.1	10,300
30	13	22.53	43.9	36.7	12,900	38.4	14,100	40.1	15,400
	19	15.42	36.9	39.2	12,300	40.6	13,300	42.5	14,600
	31	9.45	30.9	43.9	13,000	45.8	14,100	48.3	15,800
50	19	25.70	47.1	39.2	15,600	40.6	16,800	42.5	18,400
	31	15.75	37.2	43.9	15,500	45.8	16,900	48.3	18,800
	37	13.20	34.6	46.4	16,100	48.6	17,700	51.5	19,900
Uranium									
10	7	13.95	21.2	17.3	2,990	18.3	3,390	19.3	3,810
	13	7.51	14.8	22.4	3,560	24.1	4,180	25.9	4,890
30	13	22.53	29.8	22.4	6,810	24.1	8,060	25.9	9,480
	19	15.42	22.7	24.3	5,990	26.3	7,200	28.3	8,520
	31	9.45	16.7	28.8	6,220	31.5	7,660	34.1	9,170
50	19	25.70	32.9	24.3	8,350	26.3	10,100	28.3	12,000
	31	15.75	23.0	28.8	8,170	31.5	10,200	34.1	12,200
	37	13.20	20.4	31.3	8,650	34.3	10,800	37.3	13,100

Table 4.7. Steel and Uranium Shield Dimensions and Weight for SrTiO₃ Fuel

Power (kW)	Number of Elements	Capsule Length (in.)	Shield Length (in.)	3.5-in. Spacing		4.0-in. Spacing		4.5-in. Spacing	
				Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)
Steel									
10	13	13.07	32.8	35.2	8,900	36.9	9,790	38.6	10,700
	19	8.94	28.7	37.5	8,830	39.1	9,620	41.0	10,600
30	19	26.82	46.6	37.5	14,100	39.1	15,400	41.0	17,000
	31	16.44	36.2	42.2	13,900	44.3	15,300	46.8	17,200
	37	13.77	33.6	44.7	14,500	47.1	16,100	50.0	18,200
50	19	44.70	64.5	37.5	19,400	39.1	21,100	41.0	23,300
	31	27.39	47.2	42.2	17,900	44.3	19,800	46.8	22,200
	37	22.95	42.8	44.7	18,200	47.1	20,326	50.0	23,000
Uranium									
10	13	13.07	19.7	21.8	4,320	23.5	5,130	25.3	6,050
	19	8.94	15.6	23.7	4,010	25.7	4,820	27.7	5,710
30	19	26.82	33.5	23.7	8,060	25.7	9,810	27.7	11,700
	31	16.44	23.1	28.2	7,820	30.9	9,770	33.5	11,800
	37	13.77	20.4	30.7	8,270	33.7	10,400	36.7	12,600
50	19	44.70	51.4	23.7	12,100	25.7	14,800	27.7	17,700
	31	27.39	34.0	28.2	11,100	30.9	14,000	33.5	17,000
	37	22.95	29.6	30.7	11,600	33.7	14,600	36.7	17,900

Table 4.8. Steel and Uranium Shield Dimensions and Weight for Co Fuel

Power (kW)	Number of Elements	Capsule Length (in.)	Shield Length (in.)	1.5-in. Spacing		2.0-in. Spacing		2.5-in. Spacing	
				Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)	Shield OD (in.)	Total Weight (lb)
Steel									
10	3	12.50	46.3	39.0	15,700	39.6	16,200	40.1	16,600
	7	5.36	40.7	41.2	15,400	42.2	16,200	43.2	16,900
30	7	16.07	51.4	41.2	19,500	42.2	20,400	43.2	21,400
	13	8.65	45.1	43.2	18,800	44.9	20,300	46.7	22,000
	19	5.92	42.9	44.5	18,900	46.5	20,700	48.5	22,500
50	13	14.42	50.8	43.2	21,100	44.9	22,800	46.7	24,700
	19	9.87	46.9	44.5	20,700	46.5	22,600	48.5	24,600
	31	6.05	43.8	46.8	21,400	49.5	23,900	52.1	26,500
Uranium									
10	3	12.50	25.0	15.3	3,100	15.9	3,300	16.5	3,600
	7	5.36	19.1	16.6	2,800	17.6	3,100	18.6	3,500
30	7	16.07	28.8	16.6	4,200	17.6	4,700	18.6	5,200
	13	8.65	21.6	18.8	4,000	20.5	4,800	22.3	5,700
	19	5.92	18.9	19.6	3,800	21.6	4,600	23.6	5,500
50	13	14.42	27.3	18.8	5,100	20.5	6,000	22.3	7,100
	19	9.87	22.9	19.6	4,600	21.6	5,600	23.6	6,700
	31	6.05	19.2	21.5	4,600	24.2	5,900	26.8	7,300

Fig. 4.3. Dose Rate as a Function of Side-Shield Thickness for Steel Shield with SrO and SrTiO₃ Fuels.

Fig. 4.4. Dose Rate as a Function of End-Shield Thickness for a Steel Shield with SrO and SrTiO₃ Fuels.

Fig. 4.5. Effect of Fuel Center-to-Center Spacing on Dose Rate as a Function of Side-Shield Thickness for Steel Shield and 7-Fuel-Position Array with SrO and SrTiO₃ Fuels.

Fig. 4.6. Dose Rate as a Function of Side-Shield Thickness for Uranium and Tungsten Shields with SrO and SrTiO₃ Fuels.

Fig. 4.7. Dose Rate as a Function of End-Shield Thickness for Uranium and Tungsten Shields with SrO and SrTiO₃ Fuels.

Fig. 4.8. Dose Rate as a Function of Side-Shield Thickness for a Steel Shield with Co Fuel.

Fig. 4.9. Dose Rate as a Function of End-Shield Thickness for a Steel Shield with Co Fuel.

Fig. 4.10. Dose Rate as a Function of Side-Shield Thickness for Uranium and Tungsten Shields with Co Fuel.

Fig. 4.11. Dose Rate as a Function of End-Shield Thickness for Uranium and Tungsten Shields with Co Fuel.

Fig. 4.12. Dose Rate as a Function of Side-Shield Thickness for Steel Shield with Mixed-Fission-Product Fuel.

The gamma-ray energy source terms and method of calculations used here are identical to those of ref. 1, in which the reliability of the dose rates was estimated to be $\pm 50\%$, with the probability that the dose rates were overestimated rather than underestimated. The only difference is that ref. 1 assumes a single fuel region (solid cylinder with height equal to diameter) rather than one in which the fuel is dispersed in a central part of the shield. Except for the length-to-diameter effect, this method is identical to the single-element calculation discussed previously. The accuracy of the results which follow should be similar to that for the results of ref. 1.

The results of the calculations for steel, uranium, and tungsten are shown in Figs. 4.3-4.12. When using these curves it is important to remember that they are based on the capsule diameters previously given (2.5 in. for both ^{90}Sr fuels, 0.94 in. for ^{60}Co metal). For each capsule array the shield thickness is valid for any power level above the minimum for that array since only the fuel block would be lengthened if higher power levels were required. In the case of the two ^{90}Sr fuels in tungsten or uranium shields (Fig. 4.6) the number of capsules in the fuel block and the spacing between capsules have essentially no effect on the shielding thickness since most of the dose rate is determined by only the outermost capsule of any array. The shield thickness for the strontium and cobalt is the metal thickness outside the outermost hole in which the fuel is located. In all cases the dose rate is given at the surface of the shield. The results in terms of shield size and total weight, including fuel, are summarized in Tables 4.3-4.8 for the various combinations of fuel, shield material, power, and fuel array for a surface dose rate of 200 mR/hr. The values for SrO and SrTiO_3 are given for a center-to-center fuel spacing of 3.5, 4.0, and 4.5 in. and those for cobalt for spacings of 1.5, and 2.0, and 2.5 in.

The overall weights of the configurations with the lowest weights for each shield material were plotted for $^{90}\text{SrTiO}_3$ and ^{60}Co metal for various EOL powers as functions of mission lives (Figs. 4.13-4.16). These data were taken from Tables 4.3-4.8 and were extrapolated to 100 kW(th) on the upper end and 1 kW(th) on the lower end. The curves indicate that the weight for a particular size source is dependent on the shielding material selected. [Although retained here for comparative purposes, the uranium-shielded ^{60}Co fuel blocks are not practical devices because of excessive temperature (see Table 4.14 in "Thermal Analysis"). Excessive temperatures are also experienced in the larger steel-shielded ^{60}Co fuel blocks.] The points at which the shielded fuel block weights are equal ("crossover point") were taken from Figs. 4.13-4.16 and are plotted in Fig. 4.17 as functions of mission life and EOL power for each shielding material. A point falling below one of these lines would indicate that a $^{90}\text{SrTiO}_3$ -fueled heat block would be advantageous from a weight standpoint. A point above would indicate that ^{60}Co is the desired fuel material from the weight standpoint. The final selection of fuel and shielding material would, of course, be made on the bases of economics, availability of fuel, and maximum allowable weight of the system.

Fig. 4.13. Comparative Weights of $^{90}\text{SrTiO}_3$ and ^{60}Co Metal Fuel Blocks Shielded with Steel.

ORNL DWG. 69-14072

Fig. 4.14. Comparative Weights of $^{90}\text{SrTiO}_3$ and ^{60}Co Metal Fuel Blocks Shielded with Nickel.

ORNL DWG. 69-14073

Fig. 4.15. Comparative Weights of $^{90}\text{SrTiO}_3$ and ^{60}Co Metal Fuel Blocks Shielded with Tungsten.

Fig. 4.17. EOL Power at Which the Weight of a $^{90}\text{SrTiO}_3$ Fuel Block Is Equal to That of a ^{60}Co Metal Fuel Block for Mission Lives of Up to 10 Years for Steel, Nickel, Tungsten, and Uranium Shields.

The variation in dose rates for the mixed fission products of different diameters (see Table 4.9) probably reflects the effect of the steel ribs rather than the larger diameter. The amount of steel in the outer 3 to 4 in. of the fueled region is 19% for the 24-in.-dia case, 25% for the 20-in.-dia case, and 30% for the 16-in.-dia case. Thus the ribs have the effect of both decreasing the power density and increasing the amount of self-shielding in the source. The dose rate for a given shield thickness is directly proportional to the source strength and therefore the heat-generation rate, so that one can directly determine the required shielding for heat-generation rates other than the 0.1 W/cm^3 shown in Fig. 4.12. For example, if the heat-generation rates were 0.2 W/cm^3 , the dose rate for a given shield thickness would be twice that shown in Fig. 4.12.

Table 4.9. Shield Dimensions and Total Weight for Mixed-Fission-Product Fuel and a Steel Shield

Fuel Power Density (W/cm ³)	Shield OD (in.)	10-kW Power		30-kW Power		50-kW Power	
		Level		Level		Level	
		Shield Length (in.)	Total Weight (lb)	Shield Length (in.)	Total Weight (lb)	Shield Length (in.)	Total Weight (lb)
<u>16-in.-dia Fueled Region</u>							
0.10	44.6	71.7	27,400	152.5	61,000	233.3	94,600
0.15	45.7	59.3	23,000	113.2	45,800	167.1	68,700
0.20	46.4	53.3	20,800	93.7	37,900	134.1	55,100
0.25	46.9	49.7	19,400	82.1	33,100	114.4	46,700
<u>20-in.-dia Fueled Region</u>							
0.10	49.30	55.5	24,500	105.2	49,200	155.0	74,000
0.15	50.40	48.3	21,500	81.5	38,100	114.6	54,600
0.20	51.10	44.0	20,100	69.7	32,300	94.6	44,600
0.25	51.70	43.0	19,400	62.9	29,000	82.8	38,700
<u>24-in.-dia Fueled Region</u>							
0.10	53.90	45.6	22,600	77.0	40,700	108.4	58,800
0.15	54.90	41.4	20,600	62.3	32,400	83.2	44,200
0.20	55.70	a		55.3	28,500	70.6	36,800
0.25	56.40	a		47.6	23,800	60.1	30,400

^aFor these cases the total fuel volume required is less than that of the lower hemispherical region.

THERMAL ANALYSIS

The large number of combinations of fuel array, shield thicknesses, and shield surface heat fluxes involved in this study necessitated that the thermal analysis for the fuel blocks be separated into three parts: radiation and conduction in the inner fueled region, conduction through the shield and radiation, and convection from the shield surface. The temperature within the fuel block may be limited by either the shield material, the fuel, or the fuel capsule material. For strontium and cobalt fuels the maximum allowable temperature for the Hastelloy C-276 cladding is assumed to be 1800°F for normal operations and 2000°F for emergency conditions. For the mixed-fission-product fuel the maximum operating temperature is assumed to be 1650°F.

The temperature distribution within the inner fueled region is quite complex and depends to a large extent on the manner in which the heat is transferred from the fuel to the cladding and from the cladding to the shield block. These calculations were made (Appendix B) using a two-dimensional, Rθ (radial and angular), computer code. The code calculates the steady-state temperature within the fuel, cladding, and shield block, taking into consideration the internal heat generation in the fuel (and also in the shield block for Co fuel), the radiation of heat from the fuel to the cladding, both radiation and conduction of heat from the cladding to the shield block, and conduction through the shield block. The conductivity of the various materials may be treated as a temperature-dependent variable. The transfer of heat from the fuel to the cladding is assumed to be by radiation only, with the emissivity of each surface equal to 0.6. The transfer of heat from the cladding to the shield block was assumed to be by both radiation and conduction. The emissivity of both surfaces was assumed to be 0.6 and the radial gap was assumed to be 0.030 in. and filled with air.

The thermal conductivity, k , of the various materials in the inner fueled area was treated as a variable when sufficient data were available. The thermal conductivity values, in $\text{Btu hr}^{-1} \text{ft}^{-2} (\text{°F/ft})^{-1}$, are given below, with the temperature, t , in °F:

$$\frac{\text{SrO}}{k} = 1.81 - 5.67 \times 10^{-4} t + 8.57 \times 10^{-8} t^2 .$$

$$\frac{\text{SrTiO}_3}{k} = 3.15 - 1.11 \times 10^{-3} t + 1.75 \times 10^{-7} t^2 .$$

$$\frac{\text{Hastelloy C}}{k} = 4.69 + 4.63 \times 10^{-3} t .$$

$$\frac{\text{Nickel}}{k} = -38.78 + 0.150 t - 1.149 \times 10^{-4} t^2 + \\ + 3.984 \times 10^{-8} t^3 - 5.208 \times 10^{-12} t^4 .$$

Steel

$$k = 4.879 + 9.009 \times 10^{-2} t - 1.225 \times 10^{-4} t^2 + 5.908 \times 10^{-8} t^3 - 9.470 \times 10^{-12} t^4 .$$

Uranium

$$k = 10.$$

Tungsten

$$k = 50.$$

Air

$$k = 1.89 \times 10^{-2} + 1.44 \times 10^{-5} t .$$

A 30° section of a 19-element fuel block is shown in Fig. 4.18, and a typical temperature distribution through the region is given in Table 4.10. In this case the fuel is SrTiO_3 , the shield material is steel, and the center-to-center distance between elements is 4.5 in. The temperatures shown in Table 4.10 are based on an assumed temperature of 1000°F at the outer diameter of the fueled region.

The heat-transfer coefficient between the fuel elements and the block affects not only the temperature difference between the element and the block but also the distribution of heat flow from the element to the block. Referring to the data of Table 4.10, the temperature gradient around the holes containing the outer elements is large; with such gradients the heat flow from the element will be greater toward the outer, or lower, temperature region. The interesting point here is that, as the heat-transfer coefficient increases, the amount of heat flowing from the outer part of the element increases and the temperature gradient in the shield block decreases.

Fig. 4.18. Typical Section of a 19-Element Fuel Block.

Table 4.10. Temperature Distribution^a for SrTiO₃
 Fuel and a Steel Shield with a 4.5-in.
 Center-to-Center Spacing

Line No. ^b	Temperature Distribution (°F) for Radial Positions a-e ^c				
	a	b	c	d	e
1	1780	1660	1499	1400	1391
2	1780	1660	1499	1398	1386
3	1716	1620	1465	1391	1391
4	1716	1617	1458	1380	1379
5	1716	1609	1441	1349	1345
6	1716	1598	1417	1306	1295
7	1716	1587	1395	1267	1255
8	1716	1579	1378	1234	1214
9	1716	1577	1372	1227	1212
10	1603	1517	1339	1272	1280
11	1603	1513	1329	1251	1255
12	1603	1502	1302	1200	1200
13	1603	1488	1266	1132	1126
14	1603	1475	1231	1061	1040
15	1603	1466	1208	1025	1005
16	1603	1463	1201	1020	1007
17	1571	1480	1283	1205	1212
18	1571	1476	1273	1186	1193
19	1571	1467	1247	1129	1126
20	1571	1456	1216	1059	1040
21	1571	1448	1193	1023	1004
22	1571	1442	1182	1015	1005
23	1571	1441	1179	1013	1002

^aThe temperature at the OD of the fueled region is assumed to be 1000°F.

^bCorresponds to numbers shown on Fig. 4.18.

^ca = center of fuel element; b = OD of fuel;
 c = ID of cladding; d = surface of shield block;
 e = 3/4 in. into shield block.

The heat-transfer rate between the fuel container and the shield block for several cases is shown in Figs. 4.19-4.21. In Fig. 4.19 radiation only is assumed with an emissivity of 0.6 for both surfaces, in Fig. 4.20 both radiation and conduction through air are assumed, and in Fig. 4.21 both radiation and conduction through helium are assumed. The emissivity for both Figs. 4.20 and 4.21 is again taken as 0.6. A liquid-metal bond was also considered. The use of a liquid metal would complicate the fabrication procedures, require an expansion volume within the shield block to accommodate the rather large coefficient of expansion of these materials, and possibly cause material compatibility problems; also, if the liquid metal were lost, excessive fuel capsule temperatures might result.

Fig. 4.19. Heat Flux from Fuel Element to Shield Block for Thermal Radiation Only.

Fig. 4.20. Heat Flux from Fuel Element to Shield Block for Thermal Radiation and Conduction Through Air.

Fig. 4.21. Heat Flux from Fuel Element to Shield Block for Thermal Radiation and Conduction Through Helium.

The choice of thermal conductivity for the gap between the fuel container and the shield block will depend on the type of accident or failure that may be hypothesized. From the results shown in Figs. 4.19-4.21 helium would certainly be the choice. However, if for any reason the helium were to be diluted with air, the air would very rapidly reduce the thermal conductivity to the point that the results shown in Fig. 4.20 would apply. It should be realized that the concern here is not the leakage or reduction of helium pressure but its dilution since the thermal conductivity is essentially independent of pressure until the mean free path length of the molecules approaches that of the gap. However, a 50% mixture of air and helium will reduce the thermal conductivity to about 35% of the original value for helium. For a liquid metal the problems are somewhat different. In this case, if the liquid metal is lost, there will be some time period during which the gap will be filled with the metal vapor, which in most cases has a very low value of thermal conductivity. Thermal conductivity values of several gases and saturated vapors⁴ at 1400°F are given below:

Thermal Conductivity <u>[Btu hr⁻¹ ft⁻² (°F/ft)⁻¹]</u>	
Helium	0.212
Air	0.039
Mercury	0.009
Cesium	0.0035
Rubidium	0.005
Potassium	0.008
Sodium	0.015
Lithium	0.035

The only vapor with a thermal conductivity approaching that of air is lithium, but the use of lithium would lead to severe materials problems for the temperatures of interest. The effect of a metal vapor in the gap would, for most of the elements shown above, be to reduce the heat-transfer rate to the extent that it would approach that for radiation only as shown in Fig. 4.19.

Although helium is more desirable, the present analysis uses the more conservative case in which the gap is filled with air. The use of the thermal conductivity of air for the gap is conservative only if the block is operating in air. For applications requiring vacuum insulation in which the fuel block is operated under a vacuum, the use of the air conductivity may be optimistic.

The results of the calculations for the fueled region of the shield block are shown in Tables 4.11-4.13. The temperature differences given are the temperatures above that outside the fueled region. Because of the variation in both conductivity and thermal radiation with temperature, the calculations were carried out for the case in which the temperature just outside the fueled region was 1000°F. Although the calculations are valid for only this temperature, the results may be used for other temperatures varying by 100 to 200°F with very little error.

Table 4.11. Temperature Difference Between Outer Surface of Fueled Region and the Maximum Fuel, Cladding, and Block Temperature Position for SrTiO₃ Fuel

Number of Elements	Temperature Difference (°F)					
	3.5-in. Spacing			4.5-in. Spacing		
	Fuel	Cladding	Block	Fuel	Cladding	Block
<u>Nickel</u>						
13	724	429	323	638	317	195
18	688	411	317	625	319	211
19	789	510	419	669	358	241
<u>Steel</u>						
7	641	322	206	587	247	117
13	849	583	496	715	418	309
18	824	605	544	690	419	331
19	1009	767	695	780	499	400
<u>Uranium</u>						
7	742	451	352	657	343	227
12	822	623	575	741	498	426
13	1040	802	734	886	626	540
18	1021	849	813	850	632	572
<u>Tungsten</u>						
13	659	345	229	597	260	131
19	730	437	332	618	290	165

Table 4.12. Temperature Difference Between Outer Surface of Fueled Region and the Maximum Fuel, Cladding, and Block Temperature Position for SrO Fuel

Number of Elements	Temperature Difference (°F)					
	3.5-in. Spacing			4.5-in. Spacing		
	Fuel	Cladding	Block	Fuel	Cladding	Block
<u>Nickel</u>						
7	1057	420	227	1007	347	133
12	1114	571	441	1050	444	271
13	1259	683	538	1120	506	327
18	1252	747	639	1100	511	350
19	1378	826	702	1170	571	405
<u>Steel</u>						
7	1138	531	365	1051	410	211
12	1228	755	663	1122	565	425
13	1516	983	880	1281	710	568
18	1502	1071	1006	1224	699	580
19	1804	1297	1220	1406	858	736
<u>Uranium</u>						
7	1289	720	587	1163	563	396
12	1459	1053	1000	1304	829	739
13	1831	1325	1252	1567	1040	938
<u>Tungsten</u>						
12	1056	472	314	1012	378	184
13	1161	559	393	1057	419	222
18	1163	618	487	1044	424	240
19	1276	704	562	1093	469	283

Table 4.13. Temperature Difference Between Outer Surface of Fueled Region and the Maximum Fuel, Cladding, and Block Temperature Position for Co Fuel

Number of Elements	Temperature Difference (°F)					
	1.5-in. Spacing			2.5-in. Spacing		
	Fuel	Cladding	Block	Fuel	Cladding	Block
<u>Nickel</u>						
6	1030	625	250	970	560	185
7	1210	800	325	1060	670	265
12	1250	900	600	1090	740	450
13	1435	1090	710	1210	880	555
18	1385	1080	820	1145	820	550
<u>Steel</u>						
6	1060	720	480	1000	650	370
7	1355	1020	650	1215	880	570
12	1440	1230	1125	1280	1030	890
<u>Tungsten</u>						
6	1020	610	200	960	550	135
7	1190	780	285	1030	640	215
12	1175	820	465	1055	690	330
13	1320	970	570	1125	780	425
18	1270	945	635	1085	740	395

The configurations used in the calculations are as follows: The 19-element case is the same as that shown in Fig. 4.18, in the 13-element case only those elements in positions A, B, and C are used, and in the 7-element case only the elements in positions A and B are used. The 12- and 18-element cases are the same as the 13- and 19-element cases except that the central element A is eliminated.

The temperature gradient through the outer (unfueled) region of the shield and the temperature drop between the outer shield and its surroundings are not independent, as the temperature gradient in the shield depends on the outer boundary conditions and thus on the manner in which the heat source is being used. The largest temperature difference to be expected would be for the case in which the heat flow is restricted to radial flow over the fueled length. For this case the temperature difference is

$$\Delta t = \frac{q}{2\pi kL} \ln \frac{D_2}{D_1},$$

where

q = total heat generated, Btu/hr,

k = thermal conductivity of the shield material,
Btu hr⁻¹ ft⁻² (°F/ft)⁻¹,

L = fueled length, ft,

D_2 = outside diameter of the shield,

D_1 = outside diameter of the fueled region.

The temperature drop across the side shield using only the fueled length is shown in Table 4.14. The fuel and shield dimensions for these cases are given in Tables 4.3-4.8. The thermal conductivity values used to determine these temperatures are 24, 30, 10, and 50 Btu hr⁻¹ ft⁻² (°F/ft)⁻¹ respectively for steel, nickel, uranium, and tungsten.

Table 4.14. Temperature Difference Across Shield for ⁹⁰SrO, ⁹⁰SrTiO₃, and ⁶⁰Co Fuels

Number of Elements	Center-Line Spacing (in.)	Temperature Difference (°F) for			
		Steel	Nickel	Uranium	Tungsten
<u>⁹⁰SrO</u>					
7	3.5	226	164	256	58
	4.0	211	154	238	56
	4.5	200	145	222	50
13	3.5	321	232	341	77
	4.0	298	215	311	70
	4.5	278	200	288	66
19	3.5	441	318	453	102
	4.0	401	289	412	93
	4.5	373	267	378	85
<u>⁹⁰SrTiO₃</u>					
7	3.5	124	90	138	30
	4.0	116	85	127	29
	4.5	110	80	119	27
13	3.5	176	127	183	42
	4.0	163	117	166	38
	4.5	152	109	154	35
19	3.5	240	173	242	54
	4.0	219	157	220	50
	4.5	203	146	202	46
<u>⁶⁰Co</u>					
3	1.5	642	487	990	213
	2.0	599	453	910	195
	2.5	563	424	838	181
7	1.5	1314	991	1925	414
	2.0	1202	902	1702	368
	2.5	1112	833	1530	333
13	1.5	2032	1518	2786	606
	2.0	1812	1349	2386	512
	2.5	1649	1220	2101	461
19	1.5	2829	2111	3780	824
	2.0	2512	1864	3214	704
	2.5	2269	1677	2802	618

These temperature differences are true only for such applications as the direct-conduction thermoelectric system in which the surface heat flux from the shield is deliberately kept high by insulating the ends and the upper and lower parts of the side shield. For the Rankine-cycle systems in which heat is removed from or near the outer surface of the shield, the temperature drop will be much less. This is especially true for the steel and nickel shields in which the total length is 2 to 3 times the fueled length and the thickness of the side shield relatively large. Also, during fueling and during shipment of the heat blocks the entire surface of the shield could be made available for heat loss and the gradients would be about 1/2 to 1/4 those shown in Table 4.14.

The temperature of the outer surface of the shield is also dependent on the manner in which the heat is removed and thus on the specific application. Because of this dependence on a specific application a compilation of the heat flux from the shield sizes given in Tables 4.3-4.9 is presented in Tables 4.15-4.18 for heat loss from various parts of the shield surface. The heat flux from the surface as a function of surface temperature is given in Figs. 4.22-4.25. Figures 4.22 and 4.23 show typical curves for the heat loss from the shield blocks during shipping or handling when the unit is exposed to air at 100°F. Figure 4.22 is for an unfinned surface with emissivities of both 0.6 and 1.0, while Fig. 4.23 is for a unit with 1/2-in.-thick fins, 2-1/2 in. high located on 2-1/2-in. centers. By treating the surfaces, an emissivity of 0.9 to 1.0 can be obtained, and, when applied to a finned surface, the additional effect of the fins will give an emissivity very close to 1.0. The emissivity in Fig. 4.23 was assumed to be 1.0.

Table 4.15. Surface Heat Flux for Mixed Fission Products^a

Power Density (W/cm ³)	Heat Flux (Btu hr ⁻¹ ft ⁻²)		
	16-in.-dia Fueled Region	20-in.-dia Fueled Region	24-in.-dia Fueled Region
0.10	869	1277	1828
0.15	1271	1873	2693
0.20	1670	2464	3537
0.25	2065	3044	4367

^aBased on side area of fueled length.

Table 4.16. Shield Surface Heat Flux for Steel, Uranium, Nickel, and Tungsten with SrO Fuel

Power (kW)	Number of Elements	Heat Flux (Btu hr ⁻¹ ft ⁻²)								
		3.5-in. Spacing			4.0-in. Spacing			4.5-in. Spacing		
		Fueled Length	Total Length	Total Area	Fueled Length	Total Length	Total Area	Fueled Length	Total Length	Total Area
Steel										
10	7	3,527	1,402	965	3,440	1,367	934	3,348	1,330	901
	13	5,676	1,475	902	5,425	1,410	847	5,195	1,350	797
30	13	5,676	2,913	2,054	5,425	2,784	1,937	5,195	2,666	1,830
	19	7,764	3,245	2,119	7,497	3,133	2,021	7,161	2,993	1,899
	31	11,313	3,460	2,023	10,844	3,316	1,905	10,282	3,145	1,765
50	19	7,764	4,237	2,992	7,497	4,090	2,858	7,161	3,908	2,693
	31	11,313	4,790	3,012	10,844	4,591	2,842	10,282	4,353	2,640
	37	12,771	4,872	2,917	12,193	4,652	2,733	11,506	4,390	2,517
Uranium										
10	7	6,482	4,265	3,029	6,128	4,032	2,817	5,811	3,823	2,627
	13	9,300	4,719	2,686	8,644	4,386	2,418	8,043	4,081	2,177
30	13	9,300	7,031	5,110	8,644	6,535	4,653	8,043	6,081	4,239
	19	12,525	8,508	5,542	11,573	7,861	4,978	10,755	7,306	4,500
	31	17,244	9,758	5,240	15,766	8,922	4,591	14,564	8,241	4,078
50	19	12,525	9,784	7,145	11,573	9,040	6,458	10,755	8,401	5,875
	31	17,244	11,809	7,262	15,766	10,796	6,408	14,564	9,973	5,727
	37	18,932	12,250	6,932	17,276	11,179	6,073	15,887	10,280	5,370
Nickel										
10	7	3,907	1,698	1,173	3,789	1,646	1,127	3,677	1,598	1,083
	13	6,181	1,792	1,086	5,884	1,706	1,014	5,615	1,628	949
30	13	6,181	3,405	2,412	5,884	3,241	2,262	5,615	3,093	2,128
	19	8,431	3,835	2,503	8,095	3,682	2,368	7,705	3,505	2,215
	31	12,172	4,123	2,382	11,604	3,930	2,224	10,963	3,713	2,050
50	19	8,431	4,913	3,486	8,095	4,717	3,307	7,705	4,490	3,101
	31	12,172	5,606	3,511	11,604	5,344	3,287	10,963	5,049	3,037
	37	13,685	5,717	3,392	12,995	5,428	3,153	12,218	5,104	2,888
Tungsten										
10	7	6,095	3,830	2,708	5,781	3,632	2,528	5,497	3,454	2,367
	13	8,864	4,213	2,416	8,234	3,914	2,173	7,715	3,667	1,978
30	13	8,864	6,484	4,694	8,234	6,023	6,269	7,715	5,644	3,924
	19	11,983	7,796	5,076	11,108	7,227	4,580	10,352	6,736	4,157
	31	16,610	8,868	4,807	15,234	8,134	4,234	14,109	7,533	3,777
50	19	11,983	9,057	6,594	11,108	8,396	5,985	10,352	7,825	5,463
	31	16,610	10,900	6,716	15,234	9,997	5,954	14,109	9,259	5,342
	37	18,289	11,229	6,404	16,739	10,277	5,637	15,432	9,474	5,005

Table 4.17. Shield Surface Heat Flux for Steel, Uranium, Nickel, and Tungsten with SrTiO₃ Fuel

Power (kW)	Number of Elements	Heat Flux (Btu hr ⁻¹ ft ⁻²)								
		3.5-in. Spacing			4.0-in. Spacing			4.5-in. Spacing		
		Fueled Length	Total Length	Total Area	Fueled Length	Total Length	Total Area	Fueled Length	Total Length	Total Area
<u>Steel</u>										
10	13	3,400	1,355	882	3,244	1,293	827	3,101	1,236	778
	19	4,666	1,454	879	4,475	1,394	829	4,268	1,329	776
30	19	4,666	2,686	1,915	4,475	2,576	1,815	4,268	2,456	1,706
	31	6,765	3,072	1,941	6,444	2,927	1,816	6,100	2,770	1,683
	37	7,625	3,125	1,877	7,236	2,966	1,744	6,817	2,794	1,602
50	19	4,666	3,234	2,506	4,475	3,102	2,380	4,268	2,958	2,244
	31	6,765	3,927	2,714	6,444	3,741	2,546	6,100	3,541	2,367
	37	7,625	4,089	2,686	7,236	3,880	2,503	6,817	3,655	2,307
<u>Uranium</u>										
10	13	5,491	3,643	2,345	5,093	3,379	2,117	4,731	3,139	1,911
	19	7,384	4,231	2,405	6,809	3,902	2,140	6,317	3,620	1,918
30	19	7,384	5,911	4,366	6,809	5,451	3,940	6,317	5,058	3,578
	31	10,123	7,205	4,474	9,239	6,575	3,940	8,525	6,065	3,516
	37	11,102	7,494	4,276	10,114	6,827	3,739	9,287	6,269	3,300
50	19	7,384	6,421	5,219	6,809	5,921	4,737	6,317	5,494	4,328
	31	10,123	8,158	5,767	9,239	7,445	5,119	8,525	6,867	4,601
	37	11,102	8,608	5,668	10,114	7,841	4,997	9,287	7,200	4,445
<u>Nickel</u>										
10	13	3,694	1,609	1,045	3,510	1,529	975	3,343	1,457	912
	19	5,057	1,746	1,047	4,821	1,664	978	4,581	1,581	910
30	19	5,057	3,097	2,220	4,821	2,952	2,087	4,581	2,805	1,953
	31	7,264	3,575	2,251	6,879	3,386	2,088	6,488	3,194	1,925
	37	8,154	3,645	2,172	7,694	3,440	2,001	7,221	3,228	1,828
50	19	5,057	3,664	2,862	4,821	3,492	2,699	4,581	3,319	2,534
	31	7,264	4,483	3,107	6,879	4,245	2,893	6,488	4,004	2,677
	37	8,154	4,678	3,073	7,694	4,414	2,841	7,221	4,143	2,606
<u>Tungsten</u>										
10	13	5,227	3,330	2,125	4,846	3,060	1,916	4,534	2,863	1,748
	19	7,056	3,800	2,175	6,530	3,516	1,946	6,076	3,272	1,752
30	19	7,056	5,485	4,035	6,530	5,076	3,656	6,076	4,723	3,333
	31	9,743	6,646	4,134	8,921	6,086	3,657	8,251	5,628	3,276
	37	10,718	6,896	3,957	9,794	6,302	3,476	9,017	5,802	3,081
50	19	7,056	6,019	4,867	6,530	5,570	4,436	6,076	5,183	4,066
	31	9,743	7,628	5,377	8,921	6,984	4,793	8,251	6,459	4,323
	37	10,718	8,038	5,290	9,794	7,346	4,683	9,017	6,763	4,180

Table 4.18. Shield Surface Heat Flux for Steel, Uranium, Nickel, and Tungsten with Co Fuel

Power (kW)	Number of Elements	Heat Flux (Btu hr ⁻¹ ft ⁻²)								
		1.5-in. Spacing			2.0-in. Spacing			2.5-in. Spacing		
		Fueled Length	Total Length	Total Area	Fueled Length	Total Length	Total Area	Fueled Length	Total Length	Total Area
<u>Steel</u>										
10	3	3,209	866	610	3,160	853	598	3,121	843	588
	7	7,084	933	619	6,916	911	600	6,756	890	581
30	7	7,084	2,216	1,582	6,916	2,164	1,534	6,756	2,114	1,488
	13	12,559	2,409	1,629	12,084	2,318	1,547	11,618	2,228	1,468
	19	17,815	2,458	1,619	17,049	2,353	1,526	16,346	2,256	1,441
50	13	12,559	3,564	2,501	12,084	3,429	2,378	11,618	3,297	2,259
	19	17,815	3,748	2,542	17,049	3,587	2,398	16,346	3,439	2,267
	31	27,626	3,816	2,487	26,119	3,608	2,305	24,816	3,428	2,149
<u>Uranium</u>										
10	3	8,180	4,090	3,132	7,871	3,936	2,986	7,585	3,793	2,852
	7	17,583	4,934	3,439	16,584	4,654	3,186	15,693	4,404	2,962
30	7	17,583	9,817	7,621	16,584	9,259	7,092	15,693	8,761	6,623
	13	28,861	11,557	8,053	26,467	10,599	7,188	24,331	9,743	6,426
	19	40,448	12,669	8,343	36,703	11,496	7,316	33,593	10,522	6,478
50	13	28,861	15,241	11,337	26,467	13,977	10,162	24,331	12,849	9,123
	19	40,448	17,427	12,204	36,703	15,814	10,746	33,583	14,474	9,552
	31	60,137	18,949	12,147	53,427	16,835	10,327	48,244	15,202	8,953
<u>Nickel</u>										
10	3	3,692	1,109	788	3,628	1,090	771	3,576	1,074	756
	7	8,107	1,214	808	7,888	1,181	779	7,661	1,147	749
30	7	8,107	2,804	2,021	7,888	2,728	1,951	7,661	2,649	1,879
	13	14,278	3,088	2,093	13,667	2,955	1,975	13,074	2,827	1,862
	19	20,172	3,168	2,082	19,195	3,014	1,948	18,309	2,875	1,826
50	13	14,278	4,504	3,181	13,667	4,311	3,006	13,074	4,124	2,836
	19	20,172	4,773	3,244	19,195	4,542	3,038	18,309	4,332	2,852
	31	31,154	4,896	3,181	29,251	4,597	2,920	27,626	4,341	2,700
<u>Tungsten</u>										
10	3	7,276	3,381	2,562	7,031	3,267	2,455	6,802	3,161	2,355
	7	15,777	4,166	2,862	14,968	3,952	2,670	14,238	3,759	2,498
30	7	15,777	8,183	6,303	14,968	7,764	5,906	14,238	7,385	5,550
	13	26,212	9,526	6,639	24,222	8,803	5,986	22,421	8,149	5,402
	19	36,874	10,297	6,832	33,735	9,420	6,061	31,089	8,682	5,421
50	13	26,212	12,809	9,483	24,222	11,837	8,580	22,421	10,957	7,770
	19	36,874	14,495	10,148	33,735	13,261	9,033	31,089	12,221	8,104
	31	55,254	15,694	10,130	49,538	14,070	8,725	45,050	12,796	7,645

Fig. 4.22. Heat Flux by Thermal Radiation and Natural Convection to 100°F Ambient — No Fins.

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Fig. 4.23. Heat Flux by Thermal Radiation and Natural Convection to 100°F Ambient — With Fins.

Fig. 4.24. Heat Flux from Shield to Boiler for an Area Ratio of 2.

Fig. 4.25. Heat Flux from Shield to Boiler for an Area Ratio of 4.

Figures 4.24 and 4.25 are for a specific application in which the shield block is surrounded by boiler tubes. Figure 4.24 is for the case in which the surface area of the boiler tubes is 2 times that of the side-shield area, and Fig. 4.25 is for the case in which this area ratio is 4. The entire unit is insulated and the heat is transferred from the shield block to the boiler tubes by radiation and convection. The calculations were simplified by assuming the emissivity of the surfaces to be 1.0 and the fin efficiency to be 100%. The fin efficiency will depend on both the geometry and the thermal conductivity of the fin material. The effective fin efficiency in this case is complicated by the high percentage of heat transferred by thermal radiation. Over the range of values shown in Figs. 4.24 and 4.25 the fin efficiency for a surface with steel fins 1/2 in. wide and 2-1/2 in. high will be between 85 and 95%. Thus the heat fluxes shown in the figures are too high, possibly by as much as 15%. The air temperature shown in the figures will be affected very little by the fin efficiency and will be reduced by a few degrees.

In the current development work on the solidification of mixed-fission-product solids, 8- and 12-in.-dia pipes are used for storage. No exact data are available for determination of the maximum allowable internal heat-generation rate; the present practice is to limit the heat-generation rate so that the internal temperature does not exceed 1652°F or the wall temperature exceed 800°F. In order to make this material useful for a power source, the effective thermal conductivity, and thus the amount of heat per unit of length, must be increased. With a thermal conductivity of 1 Btu hr⁻¹ ft⁻² (°F/ft)⁻¹, the maximum linear rating for a unit will be about 2 or 3 kW/ft. By adding radial steel ribs the heat output can be increased by a factor of about 3 for the same temperature rise within the fuel region. This increase in heat output will add very little, if any, thickness to the shield.

For the thermal analysis of the mixed fission products the thermal conductivity of the glassy material was assumed to be 0.75 Btu hr⁻¹ ft⁻² (°F/ft)⁻¹, which is below the value reported for this material, but some conservatism is needed to allow for the present stage of development and also for the effects of irradiation and long-time high-temperature operation. It is assumed that outside the shield there will be longitudinal ribs or fins 1/2 in. thick and 2-1/2 in. high located about 2-1/2 in. apart.

The internal rib design was different for each case with the exception that the rib thickness was held constant at 1/4 in. The configuration used for each case was as follows:

- 16-in. outside diameter: 50 ribs, twenty-five 3 in. long and twenty-five 5 in. long
- 20-in. outside diameter: 50 ribs, twenty-five 4 in. long and twenty-five 7 in. long
- 24-in. outside diameter: 52 ribs, twenty-six 3 in. long, thirteen 6 in. long, and thirteen 9 in. long

The results of the analysis are shown in Figs. 4.26-4.28. In all cases the units were assumed to be located so that heat could be lost by both radiation and convection to 100°F surroundings. Although the required shield thickness will be greater than the 12 in. that was assumed, a thickness of 12 in. will give higher internal temperatures than a thicker shield since it will have a larger surface area and thus smaller temperature difference between the shield surface and the air.

Fig. 4.26. Temperature Through Fuel and Shield for 16-in.-dia Mixed-Fission-Product Source.

Fig. 4.27. Temperature Through Fuel and Shield for 20-in.-dia Mixed-Fission-Product Source.

Fig. 4.28. Temperature Through Fuel and Shield for 24-in.-dia Mixed-Fission-Product Source.

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5. HEAT REMOVAL SYSTEMS

The systems that were examined for removing the heat from the isotope fuel capsules to satisfy the various requirements during assembly, shipping, normal operation, and emergency conditions included the following: thermal and forced convection of the coolant directly over the surface of the isotope capsules, heat flow from the capsule through a gas- or liquid-filled gap to the heat block and through it to the coolant, and transport of the heat from either the fuel capsules or the heat block to the coolant stream by heat pipes. Results are given here of some generalized studies showing the relations between major parameters such as the temperature drop, the heat flux, and the size of the heat transfer elements required for typical systems.

INTERMEDIATE-COOLANT CIRCUITS

In the beginning of the design program it was thought that systems similar to those employed for space power units would be attractive — that is, systems employing an intermediate coolant such as NaK or pressurized water for transporting the heat from the isotope fuel capsules to the boiler for a Rankine-cycle system or to a heat transfer matrix for a Brayton-cycle system. However, a recent ORNL study¹ on the reliability of components such as pumps, power supplies, switch gear, and instrument sensors shows that such intermediate circuits would seriously detract from the reliability of the power unit, particularly if a pump were required for the fluid circuit. This study indicates that the mean time to failure for the coolant system would be ~200,000 hr, which means that there would be a 10% probability of a system failure in 20,000 hr just because of the intermediate-coolant system. Considering the failure probabilities of other elements in the power plant, there is a strong incentive for eliminating any intermediate-coolant circuit which requires a pump. This includes electromagnetic pumps in liquid-metal systems employing thermoelectric junctions as the electric power supply for the pumps. Pumps of this type are inherently subject to severe thermal stresses and to our knowledge have not been operated for more than 7000 hr without a failure, which in most cases occurred because of thermal stresses. The intermediate-coolant systems considered were those in which circulation would be maintained by natural thermal convection or, in the case of heat pipes, by boiling, condensing, and capillary action. Heat pipes using boiling to induce natural thermal convection were favored initially.

HEAT PIPES

The heat pipe² is potentially attractive as a heat removal device because of both its high effective thermal conductance and its lack of dependence on any electrical or mechanical device for continued operation. A schematic cross section through a typical heat pipe is shown in Fig. 5.1. The capillary structure (shown in the lower portion of the figure) consists of a series of longitudinal grooves machined in the cylinder wall. Attention is called to the fact that at the vapor radius, r_v , there is no land between the grooves. The insulated length shown is provided to allow for the necessary shield penetration.

Fig. 5.1. Schematic of Cylindrical Heat Pipe Structure.

Heat conducted through the evaporator wall into the liquid film in the capillary grooves generates vapor in the evaporator section. Removal of vapor by condensation in the condenser section results in vapor flow down the pipe. Capillary pumping returns the liquid to the evaporator, where the process is repeated. The required condition for heat pipe operation is, then, that the available capillary pumping just balance the losses in the vapor and liquid circuits. If we write an equation in words rather than symbols, steady operation requires that

$$\left(\begin{array}{c} \text{pressure rise} \\ \text{from capillary} \\ \text{forces} \end{array} \right) = \left(\begin{array}{c} \text{liquid-flow} \\ \text{pressure drop} \end{array} \right) + \left(\begin{array}{c} \text{vapor-flow} \\ \text{pressure drop} \end{array} \right) + \left(\begin{array}{c} \text{hydrostatic pressure} \\ \text{difference due to} \\ \text{gravity} \end{array} \right).$$

Cotter³ has developed a theory for predicting the various terms in the above equation. A partial check on its validity has been provided by Kemme,⁴ who found rather good experimental agreement with calculations for potassium as the working fluid. Cotter's theory is applied in the following discussion. The equation for the above statement is

$$\frac{2g_c \sigma \cos \theta}{r_c} = \frac{3\eta_\ell (Z_e + Z_c)}{4\pi r_v (r_w - r_v) r_c^2 \rho_\ell} \left(\frac{Q}{h_{fg}} \right) + \frac{3\eta_\ell Z_i}{2\pi r_v (r_w - r_v) r_c^2 \rho_\ell} \left(\frac{Q}{h_{fg}} \right) + \frac{1 - \frac{4}{\pi^2}}{8r_v^4 \rho_v} \left(\frac{Q}{h_{fg}} \right)^2 + \frac{8\eta_v Z_i}{\pi r_v^4 \rho_v} \left(\frac{Q}{h_{fg}} \right) + \rho_\ell Z g \sin \phi, \quad (1)$$

where

g = local acceleration, ft/hr²,

g_c = gravitational conversion constant, 4.17×10^8 lb_m·ft/lb_f·hr²,

h_{fg} = latent heat of vaporization, Btu/lb_m,

Q = heat transport, Btu/hr,

r_c, r_v, r_w = capillary, vapor, wall radii, ft,

Z_c, Z_e, Z_i, Z = condenser, evaporator, insulated section, and total lengths, ft,

η_ℓ, η_v = liquid, vapor viscosity, lb_m/ft·hr,

ϕ = angle from local horizontal, deg,

$\rho_\ell \rho_v$ = liquid, vapor density, lb_m/ft³,

σ = surface tension, lb_f/ft,

θ = contact angle, deg.

The left-hand side of Eq. (1) is simply the contribution of the capillary driving force when the principal radii of curvature are equal. The terms of the right-hand side are respectively the liquid pressure drop in the evaporator and condenser sections for uniform heat addition and removal, the liquid pressure drop in the insulated section, the vapor pressure drop in the insulated section, and the hydrostatic contribution.

Equation (1) relates the heat transport performance of a heat pipe to its geometric properties, to fluid properties, and to orientation in a gravitational field. One would, of course, like to maximize this performance, working within the constraints imposed by a specific application. Where these constraints do not involve the capillary radius, r_c , its optimum value may be found by rewriting Eq. (1) in the form

$$A Q^2 + B \frac{Q}{r_c^2} - \frac{C}{r_c} + D + E Q = 0, \quad (2)$$

and maximizing Q with respect to r_c :

$$r_c = \frac{2BQ}{C}, \quad (3)$$

where the constants do not involve r_c . As is pointed out by Cotter, this is equivalent to setting the viscous contribution to the liquid pressure drop equal to one-half the capillary term, that is, with the optimum r_c ,

$$B \frac{Q}{r_c^2} = \frac{1}{2} \frac{c}{r_c} . \quad (4)$$

For no static head, that is, a horizontal pipe or a zero-gravity environment, and the above optimal capillary radius, the vapor radius for maximum heat transport is given by

$$r_v = \frac{5}{6} r_w . \quad (5)$$

This result differs from that of Cotter, who found the optimum value of r_v/r_w to be $\sqrt{2/3}$. The difference is small, however, and can be shown to arise from variations in the details of the capillary structures.

A short computer program was written to determine the heat pipe performance from Eq. (1). Some results of calculations with this program are presented in Figs. 5.2 and 5.3. Evaporator and condenser lengths were set equal, the fluid was potassium, and the pipe material was stainless steel.

Figure 5.2 shows the heat pipe performance as a function of vapor temperature for several pipe diameters and evaporator lengths. In each case the output reaches a maximum for some temperature, which generally increases with pipe diameter but decreases with evaporator length. If the temperature is limited to, say, 1300°F, it appears that the output will be limited to 1 to 2, 3 to 4, or 6 to 8 kW for the 0.5-, 0.75-, or 1.0-in. pipes respectively. While the axial Q/A improves somewhat for the larger pipes, thus allowing a lesser total cross section of shield penetration, the larger holes required would be disadvantageous.

Figure 5.3 is a cross-plot of the curves for the 1-in.-OD pipe, with superimposed lines of constant Q/A in the evaporator. As can be noted, the output at the lower temperatures is extremely insensitive to the evaporator length. The previously mentioned maximum output as a function of temperature shows up here in the crossing of the higher temperature curves.

Fig. 5.2. Heat-Pipe Performance as a Function of Vapor Temperature for Several Pipe Diameters and Evaporator Lengths.

Fig. 5.3. Performance Curves for a 1-in.-OD Heat Pipe as a Function of Evaporator Length for Several Vapor Temperatures.

BOILERS

In the design of boilers the heat transfer mechanism can vary from thermal- or forced-convection heating of the liquid to subcooled boiling, to nucleate boiling with bubbly flow, to annular flow with boiling from the surface of the annular film, to film boiling, to mist flow, and finally to relatively simple forced-convection heat transfer to the superheated vapor.⁵ The characteristic heat transfer coefficients depend not only on the physical properties of the fluid, the mass flow rate, passage diameter, temperature, and Reynolds numbers but also on the flow regime, surface conditions, vapor quality, wetting characteristic, surface texture, and other factors. There are boiling flow stability problems associated with the effects of changes in the flow conditions on the pressure drop in the two-phase flow regions. The detailed techniques for handling these problems were worked out and are presented in a report⁶ on the design of boilers for cesium and potassium vapor systems. Using a somewhat similar technique, a new approach to the design of steam boilers for molten salt reactors was also worked out.⁷ Since large thermal stresses are likely to develop in most heat exchangers if an attempt is made to introduce water into a steam boiler when it is red hot, a special design approach is required. Fortunately, the approach proposed⁷ for molten salt reactors also appears to be applicable to isotope heat sources.

Two types of boilers were considered in the design study for this program: the once-through type and the recirculating type. In each case the heat load was based on the thermal efficiency given for the cycle by the detailed analyses presented in Chapter 9. The calculational procedures that were used followed the procedures in the above referenced reports very closely.

The results gave a good indication of the space required for incorporation of these units in the heat block and shield assembly. It should be noted that for isotope heat sources the boiler may be placed in or close to the fuel matrix because the development of voids in the coolant flow has no effect on the rate of heat generation, a problem that has been troublesome in fission reactors. Therefore for Rankine-cycle systems there is no incentive for using an intermediate-coolant circuit.

Once-Through Boilers

The characteristics of the once-through boilers designed for 5-kW(e) steam, cesium, and potassium power plants are summarized in Table 5.1. For convenience in the calculations these studies were carried out by estimating the thermal power input to a single tube for each of the different fluids. The number of tubes required was then calculated by dividing the desired thermal power input by the boiler power capacity per tube. In each case the design was intended to produce a high-performance boiler tube that would give an overall boiler design as compact as possible while maintaining conservative design values for such factors as thermal stress, fluid flow velocity and pressure drop, and good conditions for flow stability. The cesium and potassium boilers have a tapered tube in the boiling region with a constant-diameter section through the dryer region.

Table 5.1. Reference Designs for Once-Through Boilers

	Steam	Potassium	Cesium		
Boiler outlet temperature, °F	1050	1540	2000	1540	2000
Boiler outlet pressure, psia	1000	26	152	50	235
Boiler power, kW(th)	50	50	42.5	50	40
Boiler output per tube, kW(th)	50	30	21.3	7.1	8
Number of boiler tubes	1	2	2	7	5
Tube inlet OD, in.	0.25	0.20	0.20	0.20	0.20
Tube outlet OD, in.	0.66	0.50	0.50	0.50	0.50
Tube wall thickness, in.	0.050/0.080	0.025	0.025	0.025	0.025
Tube length, ft	18 ^a	4.8	4.8	2.1	2.1
Length of boiling region, ft	18	3.7	3.7	0.95	0.95

^aThe inner and outer tubes of the re-entrant tube are of the same length.

Fig. 5.4. Layout for a Potassium Boiler.

This approach is advantageous from the standpoints of flow stability, pressure drop, and compactness of the completed boiler. The boilers were designed to maintain relatively low velocities through the boiling region and thus maintain the favorable heat transfer coefficients associated with annular flow up to vapor qualities of about 95%. In view of the difficulty in drying out the droplets suspended in the vapor at the boiler outlet, a 180° bend should be introduced as indicated in Fig. 5.4 so that the droplets will be centrifuged to the wall and evaporated in a short distance. This will minimize the length of the dryer region.

The once-through steam boiler tube configuration contemplated is shown in Fig. 5.5. In this design the water enters in the bottom of the inner tube, where the cold feed water is isolated from the high-temperature heat source by an annulus

of superheated steam. This, coupled with the use of thermal sleeves, makes it possible to avoid serious thermal stresses under startup conditions, when the unit would be red hot before the feed water could be introduced. The characteristics of the unit are such that, as the feed water flow rate is reduced from the full-power design condition, a progressively shorter portion of the central tube is required for the boiling region; therefore the design length of the double-tube configuration is determined by the full-power design conditions. This comes about because the heat flux near the feed water inlet and throughout the boiling zone remains nearly constant irrespective of load.

Part-load design studies of the potassium, cesium, and steam once-through boilers described above indicate that, unlike most once-through boilers, these units should operate stably throughout the load region from zero to full power provided that they are coupled directly to a turbine with no throttle valve between the boiler and turbine; control can be accomplished by varying the feed flow rate. As a consequence, the boiler discharge pressure and turbine output will both be approximately directly proportional to the boiler feed flow rate.

Recirculating Boilers

Experience derived with the MPRE (Medium Power Reactor Experiment) and TLPR (Terrestrial Low Power Reactor) programs indicates that recirculating boilers have some outstanding advantages that make them well suited to the kilowatt isotope power source program. In the basic design shown in Fig. 5.6 the isotope would be encapsulated in long cylindrical rods

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Fig. 5.5. Diagram of a Section Through a Tube for a Re-entry-Tube Boiler.

Fig. 5.6. Recirculating Boiler with Fuel Capsules Cooled Directly by the Boiling Liquid.

that would be cooled directly by boiling of the working fluid for the thermodynamic cycle. The coolant would be recirculated by thermal convection between the boiling zone and a vapor separator-expansion tank located above it. Extensive tests with water have shown that this configuration provides a simple, reliable means of removing the heat from a heat transfer matrix of rods and that large differences in radial power distribution across the heat transfer matrix can be tolerated. The advantage here is that a heat source such as a set of cobalt rods can be mounted in the center of a heat transfer matrix with tungsten rods in the inner shield region immediately around them, followed by a tungsten shield that need be cooled only on its inner surface. Cooling of the various hot surfaces will be satisfactory in spite of large differences in the local heat flux. The dimensions of a series of boilers based on this configuration are given in Table 5.2 for power outputs corresponding to 1 to 10 kW(e). The boiler proportions are well suited to the use of water, Dowtherm A, cesium, or potassium. The mean temperature drop from the capsule surface, with a heat flux of $2660 \text{ Btu hr}^{-1} \text{ ft}^{-2}$, to the boiling liquid is given below:

Fluid	Temperature Drop (°F)
Water	15
Cesium	15
Potassium	15
Dowtherm A	135

Table 5.2. Proportions for Recirculating Boilers with Heat Transfer Directly from Isotope Capsules to Boiling Liquid
(Capsules on equilateral triangular pitch
with 0.0625-in. spacing)

Power [kW(th)]	Capsule OD (in.)	Capsule Length (in.)	Number of Capsules Required	Capsule Bundle Diameter (in.)	Isotope Volume (in. ³)	Isotope Volume per Capsule (in. ³)
<u>SrTiO₃</u>						
10	1.0	12	91	11.7	785	8.69
50	1.0	18	300	20.2	3,925	13.04
100	1.0	24	450	24.6	7,850	17.38
<u>SrO</u>						
10	1.0	12	49	8.5	429	8.69
50	1.0	18	165	15.1	2,145	13.04
100	1.0	24	247	18.4	4,290	17.38
<u>Co</u>						
10	0.30	6	24	2.18	14.9	0.638
50	0.30	6	117	4.4	74.5	0.638
100	0.30	6	234	6.15	149	0.638

Since the boiling heat transfer coefficient for Dowtherm A is lower than that for steam by a factor of about 9, either the surface temperature drop or the heat transfer surface area should be increased by a factor of $9 \times 37/50$ (the factor 37/50 allows for the reduced power required because of the difference in thermal efficiency implied by Table 9.1). The mass flow rate is higher by a factor of about 6 for the Dowtherm, which makes it necessary to increase the flow passage area by roughly the same factor if the pressure drop is to be kept the same.

In all cases the vapor leaving the vapor separator would be in essentially the dry saturated condition. The thermodynamic characteristics of Dowtherm A are such that the vapor expansion process through the turbine would be entirely in the superheated region. In the steam, cesium, and potassium turbines the vapor would expand into the saturated region, and the percentages of moisture would run as much as 20 or 30%. Vapor bleed-off passages from the turbine are required to remove the moisture in order to avoid turbine bucket erosion. Extensive experience indicates that, with proper design, thoroughly satisfactory operation can be obtained without any difficulty with turbine bucket erosion.⁸

GAS HEAT EXCHANGERS

A parametric analysis of heat exchangers was made for a 5-kW Brayton-cycle system which uses external fins to dump excessive heat in an emergency situation. The heat exchangers must raise the temperature of a helium-xenon working fluid from 1000 to 1500°F at a flow rate of 3456 lb/hr. The gas is heated by flowing over tubes which contain one of the candidate fuels. The heat exchangers were designed with the primary consideration given to minimizing the pressure drop. The pressure drop was held to less than 3% of the entrance gas pressure (26 psia).

Two basic types of heat exchangers were examined: cross flow and parallel flow. In the cross-flow type the fuel tubes are perpendicular to the gas stream both with and without radial fins. This design provides good heat-transfer characteristics and low pressure drop; however, when fin density increases, decontamination of the tubes after loading becomes a problem. In the parallel-flow type the tubes are parallel to the gas stream, both with and without longitudinal fins. This type is more easily handled in a hot cell and provides a more convenient geometry for shielding and emergency cooling. The low fin density on these tubes facilitates decontamination procedures. The design characteristics for the cross-flow heat exchangers are given in Table 5.3.

The heat exchangers using ⁶⁰Co fuel sources were all of the cross-flow type and used fuel material with a specific activity of 300 Ci/g. The ⁶⁰Co capsules are contained in tungsten shields to absorb and convert to heat more than 99% of the gamma radiation from the capsules. Each 5-kW(th) tungsten shield contains six ⁶⁰Co capsules (0.94-in. OD, 3.4 in. long) arranged on a 2.3-in.-dia circle with a spacing of 0.2 in. between capsules. The tungsten shield is 7.2 in. in diameter and 7.3 in. long. Three different design geometries were considered for the ⁶⁰Co-fueled heat

Table 5.3. Cross-Flow Heat Exchanger Characteristics

Spacing (in. on center)	Inside Dia (in.)	Source Tubes					Total Heat Exchanger Area (ft ²)	Overall Heat Trans Coeff [Btu hr ⁻¹ ft ⁻² (°F) ⁻¹]	Outlet Gas Temp (°F)	Max Clad Temp (°F)	Press. Drop (psia)	Heat Input (Btu/hr)		
		Length (ft)	No. per Row	No. of Rows	No. of Fins ^a	Fin Eff.								
<u>60Co</u>														
8.8	7.2 ^b	0.61	1	5	0	1.0	5.7	10.2	1554	2495	0.03	95,168		
					4	0.325	56.0	3.8		1786	0.2			
					12	0.377	156	3.2		1661	0.5			
			2	3	0	1.0	6.9	6.2	1647	3209	0.004	114,201		
					4	0.433	67.2	2.8		1964	0.03			
					12	0.526	188	2.1		1774	0.04			
		4	2	0	1.0	9.2	7.1	1833	3106	0.005	152,268			
				4	0.470	89.6	2.5		2116	0.01				
				12	0.625	250	1.6		1955	0.008				
		<u>SrO</u>												
		2.75	2.0	1.0	4	7	12	0.651	181	8.7	1562	1596	0.3	96,761
							4	0.604	70.7	10.3		1664	0.1	
0	1.0						15.3	15.2		1943	0.03			
8	2				12	0.836	207	4.0	1630	1786	0.007	110,584		
					4	0.804	80.8	4.8		2074	0.003			
					0	1.0	17.5	5.4		3854	<0.001			
3.0	13			1	12	0.893	252	2.6	1748	2098	<0.001	134,774		
					4	0.871	98	3.1		2758				
					0	1.0	21	3.2		7348				
2.25	1.5			1.0	5	9	12	0.657	224	8.8	1517	1542	0.3	87,474
							4	0.605	87	10.5		1590	0.1	
							0	1.0	18.9	16.4		1777	0.03	
		2.0	10		3	12	0.832	298	4.2	1659	1767	0.008	116,631	
						4	0.796	116	5.2		1960	0.003		
						0	1.0	25	6.1		3092	0.001		
		3.0	16	1	12	0.904	238	2.4	1545	1844	<0.001	93,305		
					4	0.881	93	3.0		2348				
					0	1.0	20.1	3.1		5747				
		1.75	1.0	1.0	6	17	12	0.679	363	8.2	1520	1538	0.3	88,121
							4	0.620	141	10.1		1570	0.1	
							0	1.0	30	16.8		1684	0.04	
2.0	13				4	12	0.841	370	4.0	1528	1603	0.007	89,849	
						4	0.802	144	5.1		1725	0.002		
						0	1.0	30.2	6.5		2390	<0.001		
3.0	20			2	12	0.904	427	2.4	1596	1792	<0.001	103,672		
					4	0.878	166	3.1		2089				
					0	1.0	34.8	3.6		3958				
4.0	27			1	12	0.940	384	1.5	1545	1992	<0.001	93,305		
					4	0.923	149	1.9		2619				
					0	1.0	31.3	2.2		6834				
1.25	0.5	2.0	19	11	12	0.841	931	3.7	1530	1565	0.009	90,281		
					4	0.794	356	4.9		1614	0.003			
					0	1.0	67.4	8.1		1843	0.002			
			3.0	28	5	12	0.906	935	2.2	1532	1625	<0.001	90,713	
						4	0.875	358	3.0		1748			
						0	1.0	67.7	4.4		2403			
		4.0	38	3	12	0.936	1015	1.5	1570	1758	<0.001	98,489		
					4	0.914	388	2.0		1995				
					0	1.0	73.5	2.9		3352				

Table 5.3. (Continued)

Source Tubes							Total Heat Exchanger Area (ft ²)	Overall Heat Trans Coeff [Btu hr ⁻¹ ft ⁻² (°F) ⁻¹]	Outlet Gas Temp (°F)	Max Clad Temp (°F)	Press. Drop (psia)	Heat Input (Btu/hr)
Spacing (in. on center)	Inside Dia (in.)	Length (ft)	No. per Row	No. of Rows	No. of Fins ^a	Fin Eff.						
SrTiO ₃												
2.75	2.0	1.0	4	12	12	0.637	310	9.1	1537	1555	0.6	91,729
					4	0.589	121	10.7		1591	0.2	
					0	1.0	26.3	16.3		1734	0.05	
		2.0	8	3	12	0.825	310	4.2	1537	1616	0.01	
					4	0.792	121	5.1		1765	0.004	
					0	1.0	26.3	5.8		2674	0.001	
2.25	1.5	1.0	5	16	12	0.643	398	9.2	1509	1522	0.6	85,996
					4	0.591	155	11.0		1548	0.2	
					0	1.0	33.5	17.4		1644	0.06	
		2.0	10	4	12	0.827	398	4.3	1509	1566	0.01	
					4	0.790	155	5.3		1670	0.004	
					0	1.0	33.5	6.3		2270	0.001	
		3.0	16	2	12	0.888	477	2.8	1593	1723	0.002	103,195
					4	0.862	186	3.4		1961	<0.001	
					0	1.0	40.2	3.7		3531		
		4.0	21	1	12	0.932	417	1.7	1530	1853		90,296
					4	0.915	163	2.1		2372		
					0	1.0	35.2	2.2		6058		
1.75	1.0	1.0	6	30	12	0.669	640	8.5	1509	1519	0.5	85,996
					4	0.609	249	10.4		1536	0.2	
					0	1.0	52.2	17.7		1596	0.07	
		2.0	13	7	12	0.830	647	4.3	1514	1551	<0.01	86,952
					4	0.788	252	5.4		1614		
					0	1.0	52.8	7.1		1949		
		3.0	20	3	12	0.898	640	2.5	1509	1608	<0.001	85,996
					4	0.870	249	3.3		1761		
					0	1.0	52.2	3.9		2717		
		4.0	27	2	12	0.930	768	1.8	1593	1790		103,195
					4	0.909	298	2.3		2085		
					0	1.0	62.7	2.6		4036		
1.25	0.5	2.0	19	19	12	0.834	1608	3.9	1510	1528	<0.02	86,235
					4	0.786	615	5.1		1555		
					0	1.0	116	8.5		1673		
		3.0	28	9	12	0.899	1683	2.4	1530	1577	<0.002	90,296
					4	0.866	644	3.2		1640		
					0	1.0	122	4.8		1971		
		4.0	38	5	12	0.931	1692	1.6	1533	1626	<0.001	90,774
					4	9.08	647	2.2		1747		
					0	1.0	122	3.1		2434		

^aPer inch of length.^bOutside diameter of fuel block.

exchanger: one, two, and four capsules per row. In addition, the fin density was varied from 0 to 12 fins per inch. Figures 5.7-5.9 are schematics of the three geometries considered. The pressure drops indicated on the figures do not include expansion and contraction losses caused by the transition piece on each end of the heat exchanger.

The most attractive of the ^{60}Co -fueled cross-flow heat exchangers is a continuous fin (12 fins per inch) model with one fuel tube per row for five rows. This model produces an exit gas temperature of 1554°F with a maximum cladding temperature of 1661°F and a pressure drop of 0.5 psia. The models having two and four tubes per row all had excessively high exit gas temperatures, which would cause the limitation on the cladding temperature to be exceeded.

The heat exchangers using $^{90}\text{SrTiO}_3$ and ^{90}SrO fuel sources were of both cross-flow and parallel-flow types. Most of the cross-flow heat exchangers provided satisfactory results except for ease of cooling in the event of a loss of gas flow. Only one model, the $^{90}\text{SrTiO}_3$ -fueled case with 30 rows containing 6 tubes per row (each tube 1.0-in. ID by 1 ft long), had an acceptable cooling capacity by use of external fins. Figures 5.10-5.13 are schematics of the ^{90}Sr -fueled cross-flow heat exchangers with varying numbers of tubes. As in the cross-flow cases, the pressure drops indicated on the figures do not include expansion and contraction losses caused by the transition piece on each end of the heat exchanger.

Fig. 5.7. ^{60}Co -Fueled Heat Exchanger with One Capsule per Row.

Fig. 5.8. ^{60}Co -Fueled Heat Exchanger with Two Capsules per Row.

Fig. 5.9. ^{60}Co -Fueled Heat Exchanger with Four Capsules per Row.

$T_{out} = 1554^{\circ}\text{F}$

Fig. 5.10. Schematic of $^{90}\text{SrTiO}_3$ -Fueled Cross-Flow Heat Exchanger with 60 Tubes.

 $T_{out} = 1514^{\circ}\text{F}$

Fig. 5.11. Schematic of $^{90}\text{SrTiO}_3$ -Fueled Cross-Flow Heat Exchanger with 91 Tubes.

Fig. 5.12. Schematic of $^{90}\text{SrTiO}_3$ -Fueled Cross-Flow Heat Exchanger with 54 Tubes.

Fig. 5.13. Schematic of $^{90}\text{SrTiO}_3$ -Fueled Cross-Flow Heat Exchanger with 32 Tubes.

The design characteristics of the parallel-flow heat exchangers are given in Table 5.4, and a schematic is shown in Fig. 5.14. The parallel-flow cases consider both longitudinally finned (six fins per tube) and unfinned fuel tubes parallel to the gas flow. The low fin density aids decontamination operations. The tubes are bundled in cylindrical arrangements of 7, 19, 37, and 61 tubes with inside diameters of 1.0, 1.5, 2.0, and 2.5 in. per tube. The presence of fins does not significantly lower the cladding temperature in the parallel-flow cases. The most attractive case for the $^{90}\text{SrTiO}_3$ -fueled heat exchangers uses nineteen 1.5-in.-ID tubes in a tube bundle 0.94 ft in diameter by 4.5 ft long. This model has a maximum cladding temperature of 1647°F with a pressure drop of 0.2 psia. The best ^{90}SrO -fueled heat exchanger uses nineteen 1-in.-ID tubes in a tube bundle 0.73 ft in diameter by 5.6 ft long. In this case the maximum cladding temperature is 1628°F and the pressure drop is 0.45 psia.

Table 5.4. Parallel-Flow Heat Exchanger Characteristics
 Outlet Gas Temperature: 1500°F
 Heat Input: 92,497 Btu/hr

Spacing (in. on center)	Inside Dia (in.)	Source Tubes				Total Heat Exchanger Area (ft ²)	Overall Heat Trans Coeff [Btu hr ⁻¹ ft ⁻² (°F) ⁻¹]	Max Clad Temp (°F)	Press. Drop (psia)
		Length (ft)	No. of Tubes	Bundle Dia (ft)	Fin Eff.				
SrTiO ₃									
1.25	0.5	110	7	0.31	0.398	242	27	1515	184
		41	19	0.52	0.525		19	1521	10
		21	37	0.73	0.615		15	1528	1.5
		12.7	61	0.94	0.680		12	1533	0.4
1.75	1.0	28	7	0.44	0.463	85.8	21	1552	15
		10.2	19	0.73	0.601		14	1576	0.8
		5.2	37	1.02	0.690		11	1602	0.1
		3.2	61	1.31	0.752		8	1630	0.03
2.25	1.5	12.3	7	0.56	0.512	49.4	17	1609	3.1
		4.5	19	0.94	0.653		11	1666	0.2
		2.3	37	1.31	0.740		8	1728	0.02
		1.4	61	1.7	0.798		6.3	1799	0.005
2.75	2.0	6.9	7	0.69	0.549	34.1	14	1687	1.0
		2.5	19	1.10	0.675		9.8	1775	0.07
		1.3	37	1.6	0.776		6.6	1913	0.007
		0.79	61	2.1	0.829		4.9	2050	0.002
1.25	0.5	110	7	0.31	1.0	127	64	1512	184
		41	19	0.52			35	1522	10.2
		21	37	0.73			24	1533	1.5
		13	61	0.94			17	1544	0.4
1.75	1.0	28	7	0.44		57	44	1538	15.4
		10	19	0.73			23	1570	0.8
		5.2	37	1.02			15	1606	0.1
		3.2	61	1.31			11	1647	0.03
2.25	1.5	12.3	7	0.56		37	33	1577	3.1
		4.5	19	0.94			17.2	1647	0.2
		2.3	37	1.31			11.1	1728	0.02
		1.4	61	1.69			7.8	1822	0.005
2.75	2.0	6.9	7	0.69		27	26	1632	1.0
		2.5	19	1.10			14.5	1737	0.07
		1.3	37	1.60			8.4	1907	0.007
		0.8	61	2.06			5.9	2078	0.002
SrO									
2.25	1.5	6.8	7	0.56	0.516	27	17	1700	1.7
		2.5	19	0.94	0.660		11	1806	0.09
		1.3	37	1.31	0.748		8	1926	0.01
		0.78	61	1.69	0.807		6	2066	0.003
2.75	2.0	3.8	7	0.69	0.555	18.9	14	1844	0.5
		1.4	19	1.10	0.684		10	2014	0.04
		0.7	37	1.60	0.786		6	2283	0.004
		0.4	61	2.06	0.840		5	2565	<0.001
1.25	0.5	61	7	0.31	0.399	134	26	1527	102
		23	19	0.52	0.527		19	1538	5.6
		11.6	37	0.73	0.618		14	1549	0.8
		7.0	61	0.94	0.683		12	1562	0.2

Table 5.4. (Continued)

Spacing (in. on center)	Inside Dia (in.)	Source Tubes		Bundle Dia (ft)	Fin Eff.	Total Heat Exchanger Area (ft ²)	Overall Heat Trans Coeff [Btu hr ⁻¹ ft ⁻² (°F) ⁻¹]	Max Clad Temp (°F)	Press. Drop (psia)
		Length (ft)	No. of Tubes						
1.25	0.5	61	7	0.31	<u>SrO</u> 0.399	134	26	1527	102
		23	19	0.52	0.527		19	1538	5.6
		11.6	37	0.73	0.618		14	1549	0.8
		7.0	61	0.94	0.683		12	1562	0.2
1.75	1.0	15	7	0.44	0.466	47	21	1594	8.5
		5.6	19	0.73	0.606		14	1640	0.5
		2.9	37	1.02	0.695		10.4	1687	0.07
		1.8	61	1.31	0.759		8.1	1741	0.02
1.25	0.5	61	7	0.31	1.0	70	63	1522	102
		22.5	19	0.52			35	1540	5.6
		11.6	37	0.73			23	1558	0.8
		7.0	61	0.94			17	1579	0.2
1.75	1.0	15.3	7	0.44		31.5	43	1568	8.5
		5.6	19	0.73			23	1628	0.45
		2.9	37	1.02			15	1697	0.07
		1.8	61	1.31			11	1776	0.02
2.25	1.5	6.8	7	0.56		20.2	32	1642	1.7
		2.5	19	0.94			17	1774	0.09
		1.3	37	1.31			11	1931	0.01
		0.8	61	1.69			7.4	2117	0.003
2.75	2.0	3.8	7	0.69		14.9	25	1745	0.5
		1.4	19	1.10			14	1948	0.04
		0.7	37	1.60			7.95	2281	0.004
		0.4	61	2.06			5.5	2635	<0.001

Fig. 5.14. Typical Parallel-Flow Heat Exchanger.

EMERGENCY HEAT REMOVAL

It is of the utmost importance that the isotope be cooled sufficiently at all times so that under no circumstances will the temperature of the fuel capsules exceed about 2000°F. Also, the heat loss to the environment should be small when the power unit is operating under normal conditions. To meet the second of these conflicting requirements, a blanket of thermal insulation must be provided to reduce the heat losses for normal operation to a low level — perhaps 5%. At the same time, for assured heat removal under emergency conditions, a standby cooling mechanism is required. Several means for accomplishing this were considered. For example, electric-motor driven pumps or blowers could be used to force air or water through cooling passages, but such systems would require a power supply, switch gear, and vulnerable sensing equipment to activate them. The reliability of any such sensing would fall far short of that required.¹

The heat transport capacity of a heat pipe increases rapidly with an increase in temperature. Hence with a proper choice of heat pipe proportions the heat transmitted from a heat block through the heat pipe to an alternate coolant would be only a modest fraction of the power output under normal conditions but under emergency conditions, when the temperature of the heat block would rise several hundred degrees, the heat pipe would remove all the heat. Unfortunately, for most of the systems considered the permissible margin between the normal operating temperature and the permissible temperature under emergency conditions is usually not sufficient to permit the use of heat pipes for emergency cooling without an excessive loss under normal operating conditions. As can be seen from Fig. 5.2, for example, increasing the temperature of a typical potassium heat pipe from 1000°F to 1300°F will increase the heat transport rate by a factor of 4. This would yield a heat loss of 25% under normal operating conditions — an excessive amount.

Cooling circuits making use of boiling and condensing circuits operating by thermal convection could be employed, but they, too, would be very sensitive to attitude and could not be depended upon for heat removal under all possible accident conditions, including tipping or inversion of the assembly.

The most promising arrangement evolved for emergency cooling of units to be operated in air makes use of natural thermal convection of air over the outer surface of the shield. The heat block and shield assembly would be mounted inside a thermally insulated enclosure fitted with a set of openings at the bottom and a stack at the top. The opening from the inside of the enclosure up into the stack would be closed by a plate held in place by a fusible alloy plug. If the temperature in the enclosure exceeded the permissible limit, the fusible alloy plug would melt, the valve would open under the force of gravity, and air would then circulate freely, flowing in through the openings at the bottom, up over the finned outer surface of the shield assembly, and out through the stack at the top.

The most promising arrangement considered for emergency cooling of units to be operated under the sea makes use of a layer of thermal insulation between the outer perimeter of the shield assembly and the inside of the pressure vessel enclosing the unit. This thermal insulation would consist of an array of ceramic blocks, of high-temperature thermal insulation such as Min-K-2000, designed to fit snugly over the outer surface of the shield assembly and fill the gap between it and the surrounding pressure vessel. The inner end of a hole through the center of each block would be filled with an aluminum alloy plug and the outer end of the hole would be filled with Kaowool, a type of rock-wool insulation. If the temperature of the outer surface of the shield exceeds the permissible limit, the melting point of the aluminum alloy would let the plug melt and flow outward through the Kaowool thermal insulation to form a high-conductivity fin that would conduct heat from the shield to the pressure vessel. Analyses indicate that, if the aluminum plug melts and flows out to the cold wall of the containment vessel, it will freeze almost instantly and begin cooling the shield. The number of such thermal bonds required is indicated by Fig. 5.15, which shows the temperature drop through the aluminum bonds for a range of bond areas and heat loads. With only about 200 in.² of total cross section in these aluminum fins traversing the gap between the shield and the pressure vessel, the heat would be removed from a 50-kW(th) shield and heat block assembly and the temperature kept well below the maximum permissible level.

Some notion of the proportions of the system is given by a typical case. The effect of temperature on the thermal conductivity of both Min-K-2000 thermal insulation and Kaowool operated in air is shown in Fig. 5.16. For a typical heat block assembly designed to provide a heat source of 50 kW(th) in a shield assembly 40 in. in diameter and 70 in. high, the external surface area would be about 70 ft². If 90% of the surface area were covered with a 1-in.-thick layer of Min-K-2000 blocks having a thermal conductivity of 0.002 Btu hr⁻¹ ft⁻² (°F/ft)⁻¹, with a shield external surface temperature of 1000°F, the heat loss through the thermal insulation would be approximately 4 kW. If the remaining 10% of the surface were covered with Kaowool, the thermal conductivity would be about 0.05 Btu hr⁻¹ ft⁻² (°F/ft)⁻¹ and the heat loss through the Kaowool would be about 1 kW, giving a total heat loss of 5 kW, or 10% of the total heat output.

For some applications it may be desirable to employ reflective insulation made of many layers of thin sheet aluminum spaced about 0.1 in. apart. If this were done, the gap between the shield and the surrounding vessel should be divided into segments by ceramic block insulation so that fusible plugs installed against the shield will melt to form a distributed pattern of heat paths through the thermal insulation to the surrounding vessel. With this distributed pattern good cooling will be provided irrespective of the attitude of the assembly at the time of melting.

Fig. 5.15. Relation Between the Temperature Drop from Shield to Outer Pressure Vessel, the Heat Load, and the Cross-Sectional Area of Aluminum Thermal Bonds Bridging a 2.0-in. Gap.

Fig. 5.16. Apparent Thermal Conductivity vs Temperature for Various Materials in Air. [From: J. M. Ryan et al., Lightweight Thermal Protection System Development, Vol. II, Insulation Material Data and Test Apparatus, AFML-TR-65-26 (January 1965).]

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6. THERMODYNAMIC CYCLES

In order to determine the thermodynamic efficiency possible in an actual power plant, a review is given here of the characteristics of various ideal and actual cycles, together with the basic limitations on both the cycles and the machinery required to implement them. This includes the effects of the principal parameters on the efficiency of the ideal cycles, the various losses associated with heat transfer, fluid flow, and mechanical friction, the size, weight, and cost of the machines, and design details such as clearances and bearing loads.

IDEALIZED CYCLES

A good indication of the overall thermodynamic cycle efficiencies that might be realized in an actual power plant can be obtained by establishing the thermodynamic efficiency of the ideal cycle and the effects on it of the major operating parameters, and then establishing the losses in the actual machine that will detract from the efficiency of the ideal cycle. The standard for comparison of the characteristics of the principal actual cycles is the Carnot cycle.

Carnot Cycle

One of the major engineering achievements of all time is the classical demonstration of Carnot that the highest overall thermal efficiency one can hope to obtain from an ideal thermodynamic cycle is given by the expression $(T_2 - T_1)/T_2$. Thus the upper limit on the thermal efficiency of any cycle is imposed by the above simple relation that involves only the peak and minimum temperatures of the cycle. This provides an excellent criterion for comparing the potential of the various cycles for which practical machinery has been developed.

Rankine Cycle

The Rankine cycle, the first thermodynamic cycle to be put to practical use, is one of the most important of all thermodynamic cycles. The characteristics of the Rankine cycle can be seen by comparing the Carnot cycle temperature-entropy and pressure-volume diagrams of Fig. 6.1 with those of Figs. 6.2 and 6.3 for a series of Rankine cycles. By definition, the temperature-entropy diagram for the Carnot cycle is rectangular. The pressure-volume diagram, while geometrically more complex, is more helpful because the net work from the cycle is directly proportional to the net area of the diagram.

The effects of temperature on the efficiencies of the ideal Carnot cycle are compared in Fig. 6.4 with those of a series of steam Rankine cycles. For a given peak temperature, superheating in a Rankine cycle leads to a loss in cycle efficiency which becomes progressively greater as the amount of superheat is increased. Thus the superheated steam cycle would be less attractive than a saturated steam cycle except for the introduction of secondary effects such as turbine bucket erosion by droplets of moisture, which may change the picture. These effects are discussed later, but it is pointed out here that a major consideration in the design of the equipment is the pressure that it

Entropy (Btu/lb °R)

ORNL DWG. 69-14093

Fig. 6.1. Temperature-Entropy and Pressure-Volume Diagrams for a Carnot Cycle with Peak Temperature of 1000°F and Pressure of 1000 psia and an Initial Temperature of 60°F.

Fig. 6.2. Temperature-Entropy and Pressure-Volume Diagrams for a Simple Ideal Rankine Cycle with Saturated Steam Supplied at 1000 psia and 545°F and with a Condenser Temperature of 100°F.

Entropy (Btu/lb °R)

ORNL DWG. 69-14095

Fig. 6.3. Temperature-Entropy and Pressure-Volume Diagrams for a Simple Ideal Rankine Cycle with Superheated Steam Supplied at 1000 psia and 1000°F and with a Condenser Temperature of 100°F.

Fig. 6.4. Effects of Temperature on the Efficiencies of the Carnot Cycle and a Series of Steam Rankine Cycles with Various Amounts of Superheat. (From: A. P. Fraas, "Application of Modern Heat Transfer and Fluid Flow Experience to the Design of Boilers for Automotive Steam Power Plants," presented at International Automotive Engineering Congress, Detroit, Jan. 13-17, 1969, SAE Paper 690047.)

must withstand; therefore if a design is limited by the peak pressure in the system rather than by the peak temperature, it may be advantageous to employ superheat in order to increase the efficiency of the cycle.

Superheating in a steam cycle eliminates moisture from the lower stages of the turbine, but causes the temperature-entropy diagram to deviate substantially from the ideal Carnot cycle and yields a loss in cycle efficiency for a given set of temperature limits. This loss can be avoided through the use of an organic working fluid such as Dowtherm A, for which the slope of the saturation line is such that the vapor expands into the superheated region as shown in Fig. 6.5. The problems associated with this effect are discussed later in connection with the losses experienced in actual cycles.

The general effects of condenser temperature on a simple steam cycle can be inferred from the ideal Carnot cycle. An explicit indication of these effects is shown in Fig. 6.6 for the ideal Carnot cycle and for both an ideal Rankine cycle and an actual Rankine cycle in which the turbine has an efficiency of 80%. These curves show vividly the importance of reducing the condenser temperature in a cycle once the peak temperature and pressure are established by practical design considerations. This effect is important, for example, if the system must be designed for use in a hot desert environment where allowances for temperature differences in the heat transfer equipment will make it difficult to obtain condenser temperatures below perhaps 160°F, whereas in a deep ocean environment with a 40°F ambient seawater temperature it may not be at all difficult to get a condensing steam temperature of 80°F.

The efficiency of a Rankine cycle can be improved by modifying the cycle to make use of regenerative feedwater heating. Figure 6.7 shows temperature-entropy diagrams for both two and five stages of regenerative feed heating. The heat in the steam bled off from each stage supplies a portion of the heat required to raise the temperature of the feedwater from the condenser temperature to the boiling point, and thus reduces the heat input required per pound of steam leaving the boiler. In the limit, the efficiency of this cycle with an infinite number of feedwater heating stages reaches the efficiency of the Carnot cycle. In practice, the size, weight, cost, and complexity of the heat exchangers and piping required for the feedwater heating system are such that six to eight stages of feed heating represent the upper limit of what has been found practicable to employ even in very large plants. Figure 6.8 shows the cycle efficiency ideally obtainable for a 1000 psia, 900°F steam cycle as a function of the temperature of the feedwater supplied to the boiler and the number of stages employed in the feedwater heating system. This shows that the incentive to increase the number of feedwater heating stages is quite limited in practice, even for idealized cycles.

Another means for modifying the Rankine cycle to more closely approach the ideal Carnot cycle, particularly for a system employing superheat, is the use of reheat. Figure 6.9 shows a temperature-entropy diagram for a system with one stage of reheat. This arrangement not only improves the efficiency of the cycle but also increases the output of the cycle for a given size of feed pump, boiler, condenser, and regenerative feed heater complex, thus reducing the capital costs in a well-proportioned system. The improvements in cycle efficiency obtainable in a steam system through the use of reheat are indicated in Fig. 6.10.

Fig. 6.5. Temperature-Entropy and Pressure-Volume Diagrams for a Simple Ideal Rankine Cycle with Saturated Dowtherm A Supplied at 650°F and 45.5 psia and Condensed at 250°F and 0.20 psia.

Fig. 6.6. Effects of Condenser Temperature on the Thermal Efficiency of a Series of Rankine Cycles. (From: A. P. Fraas, op. cit., SAE Paper 690047.)

Fig. 6.7. Temperature-Entropy Diagrams for 2 and 5 Stages of Regenerative-Feedwater Heating for the Rankine Cycles of Fig. 6.2.

Fig. 6.8. Effects of Number of Stages of Feedwater Heating and Boiler Inlet Temperature on Thermal Efficiency of a Typical Rankine Cycle. (From: A. P. Fraas, op. cit., SAE Paper 690047.)

ORNL DWG. 69-14101

Fig. 6.9. Temperature-Entropy Diagram for a Rankine-Cycle System with 1 Stage of Reheat.

Fig. 6.10. Increase in Thermal Efficiency Due to Reheat. [From: R. L. Reynolds, "Reheating in Steam Turbines," Trans. ASME (Am. Soc. Mech. Engrs.) 71, 701 (1949).]

Fig. 6.11. Temperature-Entropy and Pressure-Volume Diagrams for a Simple Ideal Brayton Cycle with Air Supplied at 14.7 psia and 520°R. (From: A. P. Fraas, Combustion Engines, copyright 1948, McGraw-Hill Book Co., New York; used with permission of McGraw-Hill Book Co.)

Brayton Cycle

The temperature-entropy and pressure-volume diagrams for a simple Brayton cycle (also known as the Joule cycle) are shown in Fig. 6.11. A simple derivation for the thermal efficiency gives

$$\eta_{th} = 1 - \frac{1}{\left(\frac{P_1}{P_2}\right)^{(k-1)/k}}$$

Thus the thermal efficiency of the Brayton cycle depends on the pressure ratio rather than on the peak and minimum temperatures, although these factors are implicitly involved in any actual cycle. The effects of the pressure ratio on the efficiency of ideal air-gasoline cycles with stoichiometric mixtures for the Brayton and Otto cycles together with the Rankine cycle are shown in Fig. 6.12. The Brayton cycle inherently gives a poorer approximation to the ideal Carnot cycle than is given by the Rankine cycle. On the other hand, practical design considerations may permit operation with a higher peak temperature than could be obtained in a Rankine cycle, which might give a higher efficiency for the Brayton cycle. The effects of the peak cycle temperature on the efficiency of the simple Brayton cycle depend on the efficiencies of the compressor and turbine. These effects are discussed later in connection with the actual cycle.

Fig. 6.12. Effects of Pressure Ratio on Thermal Efficiency of Brayton, Otto, and Rankine Cycles. (From: A. P. Fraas, *op. cit.*, SAE Paper 690047.)

As in the Rankine-cycle system, a marked improvement can be obtained in thermodynamic cycle efficiency through the use of a regenerative heating system that employs a substantial portion of the heat in the exhaust gas leaving the turbine to heat the gas leaving the compressor before it goes to the heat source. Figure 6.13 shows the temperature-entropy diagram illustrating the process, and Fig. 6.14 shows the effect of recuperator effectiveness on the efficiency of the idealized cycle. (The recuperator effectiveness is the ratio of the temperature rise in the gas being heated to the temperature difference between the two gas streams entering the heat exchanger.) In practice, the recuperator effectiveness is ordinarily 75 to 85%, but in special cases might be as high as 95%.

Fig. 6.13. Temperature-Entropy Diagram for a Brayton Cycle with a Recuperator. [From: A. P. Fraas and M. N. Ozisik, A Comparison of Gas-Turbine and Steam-Turbine Power Plants for Use with All-Ceramic Gas-Cooled Reactors, ORNL-3209 (July 1965).]

Fig. 6.14. Effects of Recuperator Effectiveness on Efficiency of an Ideal Brayton Cycle. Reactor outlet temperature, 1350°F; turbine outlet temperature, 975°F; compressor inlet and outlet temperatures, 100 and 275°F; compressor outlet temperature, 1000 psia. (From: A. P. Fraas and M. N. Ozisik, op. cit., ORNL-3209.)

Otto Cycle

Although no one has proposed the use of an Otto cycle in conjunction with an isotope power plant, this cycle is of interest for comparative purposes because it is used in virtually all modern high-speed gasoline and diesel engines. (Modern diesel engines actually follow an Otto cycle more closely than they do the constant pressure-combustion classical "diesel" cycle employed in the large, old, low-speed diesel engines.) The familiar pressure-volume diagram for an Otto cycle is presented in Fig. 6.15, along with the temperature-entropy diagram.

Fig. 6.15. Temperature-Entropy and Pressure-Volume Diagrams for an Ideal Otto Cycle.

The overall thermal efficiency of the ideal Otto, or constant-volume, cycle making use of an ideal working fluid with constant specific heats is given by

$$\eta_{th} = 1 - \left(\frac{1}{V_1/V_2} \right)^{k-1}.$$

Thus it is evident that the thermal efficiency of the Otto cycle is a function of the volumetric compression ratio rather than the peak and minimum temperatures, although these are implicitly dependent on the compression ratio for a given heat addition per unit weight of working fluid. This means that changing the power output by changing the amount of heat added at constant volume for each cycle changes the peak temperature, but has little effect on the thermal efficiency. The effects of compression ratio on the thermal efficiency of the ideal Otto cycle are shown in Fig. 6.12, along with the similar curves for the Rankine and Brayton cycles.

Stirling Cycle

Interest in the Stirling cycle has flared up every few decades since the cycle was first proposed in 1815.¹⁻³ Many engines operating on the Stirling cycle have been built; generally they have been notable for their quiet operation, but have been heavy and bulky in comparison with other types of engines of the same power output. Temperature-entropy and pressure-volume diagrams for the Stirling cycle are shown in Fig. 6.16. The cycle requires two pistons which operate 90° out of phase in the same or adjacent connected cylinders, one controlling the volume of a hot region and the other controlling the volume of a cold region. Referring to the diagrams of Fig. 6.16, the compression portion of the cycle begins with the bulk of the fluid in the cold region. The piston in that region moves to compress the gas at constant temperature (from 1 to 2 in Fig. 6.16). This is followed by a constant-volume increase in pressure and temperature from 2 to 3 as the working fluid is forced through a heat transfer matrix as it flows from the cold to the hot region of the engine. Addition of heat to the gas in the hot region continues during the subsequent expansion so that it is carried out isothermally from 3 to 4, and then the working fluid is forced from the hot region to the cold region at constant volume from 4 to 1 so that the temperature is reduced to the initial value. The thermal efficiency of the ideal Stirling cycle is equal to that of the Carnot cycle and is independent of the volumetric compression ratio.

Supercritical CO₂ (Feher) Cycle

The supercritical CO₂ cycle was given serious consideration at ORNL in 1958 as one of the more promising thermodynamic cycles suited to the direct coupling of gas-cooled reactors to turbine-generators for central station applications. This cycle has a thermodynamic advantage over the Brayton cycle in that the compression operation is carried out in the region close to the triple point so that the compression work is much less than it would be for a Brayton cycle, and, in fact, falls roughly midway between the compression work for a Brayton cycle and that for a Rankine cycle.

Fig. 6.16. Temperature-Entropy and Pressure-Volume Diagrams for an Ideal Stirling Cycle.

This effect can be deduced from Fig. 6.17, which shows pressure-volume and temperature-entropy diagrams for the supercritical CO_2 cycle.⁴ To take full advantage of this characteristic it is necessary to operate with heat rejection from the cycle at 100°F or less, a condition that is easy to satisfy for an undersea power plant but difficult for a terrestrial power plant that must operate in desert or tropical conditions.

There has been a recent surge of interest in this cycle for use in small portable power units as the result of the efforts of E. G. Feher at the McDonnell Douglas Astronautics Company. Designs have been evolved for units of 10 to 150 kW(e) for terrestrial applications,^{4,5} but none have been built.

Fig. 6.17. Temperature-Entropy Diagram for an Ideal Supercritical CO_2 Cycle. [From: E. G. Feher et al., System Analysis and Design Concept of a 150 KWE Supercritical Thermodynamic Cycle Power Conversion Module, DAC-60998-F (January 1969).]

Dissociating Gas Cycles

A number of cycles have been proposed that are somewhat similar to the supercritical CO_2 cycle in that they fall somewhere between a Brayton cycle and a Rankine cycle. Some of them employ a working fluid that dissociates during the heating phase of the cycle to give a large increase in the volume of gas to be expanded through the engine, and then they reassociate in the cooler to reduce the volume of gas that must be compressed, thus increasing the net work from the cycle. This effect is shown in Fig. 6.18 for an aluminum chloride cycle in which the gas in the compressor is Al_2Cl_6 , whereas that leaving the turbine for the expansion engine is AlCl_3 . Somewhat similar cycles can be obtained with aluminum bromide, SO_2 , etc. No systems have been built and operated on any of these cycles.

Fig. 6.18. Temperature-Entropy and Pressure-Volume Diagrams for an Ideal AlCl_3 - Al_2Cl_6 Cycle.
 [From: M. Blander et al., Aluminum Chloride as a Thermodynamic Working Fluid and Heat Transfer Medium, ORNL-2677 (September 21, 1959.)]

ACTUAL CYCLES EMPLOYING RECIPROCATING ENGINES

An actual cycle differs from the corresponding idealized cycle as a consequence of many effects, mostly small but cumulatively substantial. The major losses in reciprocating engines are those stemming from mechanical friction, heat losses, fluid pumping power requirements, and rounding of the corners of the pressure-volume diagram as a consequence of valving and other effects associated with going from one phase to another in the cycle. While these losses are common to all thermodynamic cycles employing a reciprocating engine, their relative importance differs substantially from one cycle to another.

Friction, particularly that between the piston and cylinder wall, represents a major loss, especially in high-speed engines. To avoid large losses associated with gas blow-by past the piston, it is necessary to use piston rings which are forced firmly outward against the cylinder wall by the pressure of the gas acting behind the ring in the bottom of the ring groove. Only a limited amount of lubricant can be made available on the cylinder wall or the amount of oil picked up and carried off by the working fluid will be excessive. Further, in a closed-cycle system (such as that of a Brayton or Stirling engine) oil deposits on the hot surfaces of the heater will lead to carbon formation and serious fouling of the heat transfer surfaces. The effects of engine speed on the total frictional losses in a typical automobile engine are shown in Fig. 6.19. Tests of many types of engine have shown that the principal parameter is not the crankshaft rpm but the nominal piston speed, which is calculated by multiplying the rpm by twice the stroke in feet. The frictional losses increase as nearly the square of the piston speed, and become excessive for piston speeds above about 3000 fpm.

Fig. 6.19. Effects of Engine Speed on Frictional Losses in Reciprocating Engines. (From: A. P. Fraas, Combustion Engines, copyright 1948, McGraw-Hill Book Co., New York; used with permission of McGraw-Hill Book Co.)

The life of a reciprocating engine is often limited by the rate at which the piston rings and cylinder wall wear during operation. Under favorable conditions, reciprocating steam engines have been known to operate almost continuously for as much as 20 years with scarcely any detectable wear; under unfavorable conditions, in air-cooled aircraft engines operated with cylinder wall temperatures above 300°F, severe wear has been known to occur in only 150 hr of operation. If there is no problem either with abrasion

by dirt in the oil or with corrosion (e.g., by moisture-containing sulfurous acid from combustion products), the most important parameter influencing wear is cylinder wall temperature; under otherwise favorable conditions the wear rates may be undetectable at cylinder wall temperatures of 200°F, substantial at 300°F, and catastrophic at 325°F.

Rankine Cycle

Perhaps the most important factor limiting the thermal efficiency of an actual Rankine-cycle reciprocating engine is the practicable expansion ratio. If adequate allowances are made for clearances over the top of the piston and for recesses for valves and ports, it is difficult to get a clearance volume less than about 2% of the displacement volume of the cylinder; hence the expansion ratio available in a cylinder can hardly exceed about 40 to 1. For a steam engine cylinder this would correspond to an expansion from 50 psi at 280°F down to 1 psi at 100°F, or from 500 psi at 467°F down to 10 psi at 193°F. A double or a triple expansion engine with two or three cylinders in series can be used, and many such engines have been built, but the blowdown losses between successive cylinders are large (see Fig. 6.20), the differences in cylinder size and output are very large, and the upper limit on the peak temperature imposed by cylinder wall lubrication problems makes it hardly practicable to use a steam pressure higher than about 500 psi, for which the saturation temperature is 467°F. As discussed in the following paragraph, the high heat transfer coefficient for condensing steam will heat the cylinder wall to an excessive temperature from the lubrication standpoint if a steam pressure above about 500 psi is employed. Thus the thermal efficiency of a reciprocating steam engine cycle is inherently limited to a modest value — around 15%.

Heat losses from the gas to the cylinder wall during the high-temperature portion of the cycle are a problem in any reciprocating engine, and particularly in a steam engine if the steam supply pressure corresponds to a saturation temperature higher than the temperature of the cylinder head and the upper portion of the cylinder wall. This comes about because steam engines present a very different problem from gasoline or diesel engines in that the condensing heat transfer coefficient for steam is extremely high — of the order of $10,000 \text{ Btu hr}^{-1} \text{ ft}^{-2} (\text{°F/ft})^{-1}$ compared with around $100 \text{ Btu hr}^{-1} \text{ ft}^{-2} (\text{°F/ft})^{-1}$ for combustion gases. Consequently, steam will condense during the first portion of the cycle and then, when the pressure of the cylinder drops to a low value toward the latter part of the cycle, the film of condensed steam will re-evaporate, thus chilling the inner surfaces of the cylinder head and cylinder wall. Heat losses can easily run 15% of the heat in the steam being admitted to the cylinder. To avoid these losses, the engine should be designed so that the heat losses from the cylinder head and upper portion of the cylinder barrel are kept to a minimum and the steam admission pressure is no higher than the value that corresponds to the normal cylinder-head operating temperature.

Fig. 6.20. Pressure-Volume Diagram for a Double Expansion Steam Engine Showing Losses Associated with Blowdown from the High-Pressure to the Low-Pressure Cylinder for Cycle of Fig. 6.3.

Otto Cycle

In addition to frictional losses, Otto cycle engines are subject to substantial losses in the area of the pressure-volume diagram (see Fig. 6.21). The most important is the heat loss from the high-temperature combustion gases to the cylinder wall. The cylinder wall temperature can be kept low enough to assure good lubrication for the piston rings with a total heat loss from the engines corresponding to about a third of the total heat input to the cycle, but, since much of this heat loss is from the exhaust port, the consequent reduction in cycle efficiency is more like 10%.

A substantial period of time is required to obtain smooth combustion in Otto-cycle engines; as a consequence, the left corners of the pressure-volume diagram are rounded and several percent of the available energy is lost (see Fig. 6.21). The corners at the right of the pressure-volume diagram are rounded because the exhaust blowdown process requires considerable time — the exhaust valve starts to open well before bottom dead center and the blowdown process is not completed until the piston has returned a substantial distance on the exhaust stroke. An additional loss (not evident in Fig. 6.21), usually about 1% of the net power output, is associated with the pumping power required to push the exhaust gas out of the cylinder and draw in a fresh charge.

Fig. 6.21. Difference Between Ideal and Actual Pressure-Volume Diagrams for an Otto Cycle. (From: A. P. Fraas, Combustion Engines, copyright 1948, McGraw-Hill Book Co., New York; used with permission of McGraw-Hill Book Co.)

Stirling Cycle

The factor causing the greatest departure of the actual Stirling cycle from the ideal is that the actual crank mechanism and heat transfer surfaces do not yield the ideal constant-temperature or constant-volume compression and expansion processes, but instead tend to become mixed. Therefore the actual pressure-volume diagram looks quite different from the ideal. These effects can be visualized by comparing Fig. 6.22 with Fig. 6.16.

As in other reciprocating engines, frictional losses detract from the thermal efficiency of the actual cycle. The problems in the Stirling engine are aggravated by the fact that a seal around the piston rod for the displacer piston represents an appreciable frictional loss in addition to that of the piston rings on the main power piston. Further, the work per cycle is small unless the pressure in the cylinder is kept high,

Fig. 6.22. Pressure-Volume Diagram for an Actual Cycle in a Stirling Engine. (Reproduced through courtesy of Research Laboratories, General Motors Corp.)

and this means high pressures throughout the cycle because the volumetric compression ratio is necessarily low. (A relatively large residual gas volume is inherently present in the heat transfer matrices for heating, regenerating, and cooling the gas as it flows between the hot and cold regions of the cylinder.) To keep the pumping power losses within acceptable limits as the gas is forced back and forth through the heat transfer matrices, the operating speed of the engine must be kept to about one quarter of that for a conventional internal combustion engine. Inasmuch as the high cylinder pressures require a massive construction and the relatively low piston speeds reduce the number of cycles obtainable per minute, the Stirling engine generally tends to be larger and heavier for a given output than a diesel engine.

The peak cycle temperature in the Stirling engine is limited in part by materials considerations in the heater; in practice a peak temperature of $\sim 1300^{\circ}\text{F}$ appears to be the maximum practicable. Similarly, substantial heat losses to the cylinder walls are inherent because, for good lubrication of these walls, the metal temperature must be kept to below about 300°F .

ACTUAL CYCLES EMPLOYING TURBINE ENGINES

The deviations between ideal and actual cycles in turbine engines are relatively insensitive to the choice of working fluid for a given turbine blade geometry and fluid Reynolds number. Thus a much greater portion of the losses discussed below can be made without reference to the type of cycle or working fluid than is the case for reciprocating engines.

The greatest single source of loss in turbine engines commonly stems from aerodynamic effects. With refined aerodynamic passages and well-designed turbine blades the bulk of these losses stem from fluid friction in the boundary layer, although some stem from secondary flow effects. In practice, it has been found that by far the most important parameter affecting these losses is the blade Reynolds number. As shown in Fig. 6.23, the aerodynamic efficiency can be raised to 90% or higher if the machine is large enough for a blade Reynolds number of 10^6 or more to be obtained. However, in the smaller size units this is not ordinarily possible, and the smaller the unit the lower the Reynolds number. Therefore the working fluid chosen should have a high molecular weight because this will usually give a higher Reynolds number in the turbine than would be obtained for the same power output with a lower molecular weight fluid. It is for this reason that mercury, organic vapors, and inert gases such as krypton and xenon have looked attractive as working fluids for small turbines.

A portion of the flow tends to bypass the blade through the clearance between the blade tip and the casing. This leads to a loss not only because of the mass of fluid bypassed but also as a consequence of changes in the flow pattern over the tip of the blade. The latter effect can be reduced by shrouding the blade tips, and this is sometimes done. If it is not, the loss is approximately proportional to twice the ratio

Fig. 6.23. Effects of Blade Reynolds Number on Aerodynamic Efficiency of a Compressor Stage. [From: B. Eckert, The Influence of Physical Dimensions (Such as Hub: Tip Ratio, Clearance, Blade Shape) and Flow Conditions (Such as Reynolds Number and Mach Number) on Compressor Characteristics, Vol. 3, BUSHIPS-338 (Wright Aeronautical Corp. Engineering Translation No. 22) (May 1946).]

of the radial tip clearance divided by the blade height. For large blade heights this loss is likely to be only 1 or 2%, but for small-diameter turbines with small blade heights it may run as much as 10 or 15%.

A small but nontrivial loss occurs from the flow bypassing the blade rows through the seals both between stages and at the ends of the shaft, particularly at the high pressure end. This loss commonly runs about 1% but may run substantially more in a small turbine.

In small turbines the flow may not be sufficient to permit a reasonable blade height with the flow uniformly distributed around the entire perimeter of the turbine wheel. When this is the case, a partial admission turbine is usually used, that is, one in which the high-velocity gas stream from the nozzles enters only a fraction of the total perimeter of the wheel. This reduces the tip losses from that portion of the wheel, but there is a loss associated with churning of the gas in the casing around the balance of the perimeter. The loss commonly runs about 10%, but of course varies widely with the proportions.

The expansion process in vapor turbines is often carried out well into the saturated region, where as much as 30% of the vapor admitted to the turbine will have condensed. This moisture will appear initially as droplets approximately a few microns in diameter, but the droplets tend to be centrifuged out and deposited on the surfaces of rotor and stator blades from which they are thrown off as much larger droplets. The large droplets may cause severe erosion of the turbine buckets (depending on detail design

effects), and they always cause a loss in efficiency because of churning of the liquid, that is, the momentum interchange between the rotor and the liquid droplets when they impinge on or are thrown off the rotor represents a substantial fraction of the momentum interchange between the vapor and the rotor and hence represents a substantial loss. This loss commonly amounts to about 1% per 1% of moisture (see Fig. 6.24).

Fig. 6.24. Effects of Moisture Churning Losses on Turbine Efficiency; C_o = Nozzle Exit Velocity, U = Turbine Wheel Tip Speed, and $1 - g$ = Amount of Moisture. (From: G. T. Csanady, Theory of Turbomachines, copyright 1964, McGraw-Hill Book Co., New York; used with permission of McGraw-Hill Book Co.)

is concerned. In addition, essentially the same types of fluid machine losses appear in the compression portion of the cycle, but here their importance is very much greater than in the Rankine cycle because the compression work ordinarily represents about $2/3$ of the turbine work, whereas the feed-pump work in the Rankine cycle ordinarily represents about 1% of the turbine work. As a consequence, the Brayton cycle is very sensitive to the aerodynamic design of the turbine and compressor, and it is very difficult to achieve high efficiency in these components in small sizes. However, through the use of highly refined design and fabrication techniques and high-molecular-weight gases, attractive overall thermal efficiencies can be achieved in small units provided that high gas temperatures can be used at the inlet to the turbine.

Rankine Cycle

The effects of the various losses on a steam turbine cycle are shown in Fig. 6.25 for a range of net output from 5 kW(e) to 2 MW(e) for a fixed set of turbine inlet and exhaust steam conditions. These data were plotted from parametric studies in which the overall turbine and system design was reasonably well proportioned to give close to the best possible efficiency within the limits of practical design considerations. Figure 6.25 indicates the pronounced effect of size on the cumulative effects of the various losses, and suggests the advantages associated with the use of a high-molecular-weight fluid which would permit an increase in the Reynolds number and thus reduce the aerodynamic losses in the smaller size units. In practical terms the advantage of organic fluids over steam largely disappears for units designed to deliver 300 kW(e) or more.

Brayton Cycle

The aerodynamic, tip leakage, and seal leakage losses are substantially the same in a Brayton cycle as in the Rankine cycle as far as the turbine

Fig. 6.25. Effects of Design Power Output on Overall Thermal Efficiency of Steam Turbine Cycle Designed for Dry Saturated Steam at 640 psia and 493°F Turbine Inlet with Condenser at 150°F and 3.7 psia and a Dowtherm-A System Designed for 650°F with a Condenser Pressure of 1 psia. [Plotted from data in: A. P. Fraas, Design Considerations and Proposed Designs for Steam Turbine-Generator Units for Terrestrial Power Plants of 30 KW(E) to 2 MW(E), ORNL-TM-1837 (June 1967) and Sundstrand Aviation, Organic Rankine Cycle Power Conversion Systems, B358 A.]

The relative importance of the various losses in the Brayton cycle is indicated by Figs. 6.26-6.28, which show respectively the effects of turbine and compressor efficiency, turbine inlet temperature, and compressor inlet temperature on the overall thermal efficiency of the cycle.

Fig. 6.26. Effects of Turbine and Compressor Aerodynamic Efficiency on the Efficiency of a Simple Brayton Cycle. (From: A. P. Fraas, Combustion Engines, copyright 1948, McGraw-Hill Book Co., New York; used with permission of McGraw-Hill Book Co.)

Fig. 6.28. Effects of Compressor Inlet Temperature on the Efficiency of a Simple Brayton Cycle. (From: A. P. Fraas, Combustion Engines, copyright 1948, McGraw-Hill Book Co., New York; used with permission of McGraw-Hill Book Co.)

Fig. 6.27. Effects of Turbine Inlet Temperature on the Efficiency of a Simple Brayton Cycle. (From: A. P. Fraas, Combustion Engines, copyright 1948, McGraw-Hill Book Co., New York; used with permission of McGraw-Hill Book Co.)

Supercritical CO₂ Cycle

The losses in the supercritical CO₂ cycle are essentially the same as those in the Brayton cycle except that the relative importance of compressor efficiency is reduced as a consequence of deviations from the perfect-gas laws in the vicinity of the triple point. Unfortunately, this advantage is offset by the very small size of such components as the turbine and compressor wheels in machines designed to deliver only about 10 kW(e). In a typical machine, for example, the turbine blade height will be only about 0.080 in., which makes it very difficult to maintain the elegantly precise turbine blade profiles required for good aerodynamic efficiency. In addition, tip leakage losses tend to become a large fraction of the total flow through the wheel. Consequently, a machine with useful turbine and compressor efficiencies is much more difficult to fabricate than a Brayton-cycle machine of the same net power output if the latter is designed for a high-molecular-weight gas. It is largely for this reason that no one has yet built a system that would actually operate on the supercritical CO₂ cycle.

THERMOELECTRIC CONVERSION

Thermoelectric Performance Parameters

A brief review of the parameters governing thermoelectric power conversion is given to aid in the discussions on thermoelectric system design. To simplify the development of the pertinent equations, the temperature-dependent material properties (Seebeck coefficient, α ; thermal conductivity, k ; and electrical resistivity, ρ) are expressed as average parameters over a given temperature range. For a more rigorous development of these equations, the reader is referred to the literature.⁶⁻⁸

An ideal thermoelectric couple, one which has no extraneous resistance from contacts and connecting straps, is illustrated in Fig. 6.29.

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Fig. 6.29. Thermoelectric Circuit.

The potential difference generated by a thermoelectric material in a temperature gradient results from the diffusion of charge carriers from hot to cold areas. An N-type semiconductor having conductance electrons free to diffuse then becomes positive at the hot junction. This property is produced in lead telluride (PbTe) by adding 0.01 to 0.06 mole % of lead iodide. The opposite effect is produced by adding 0.1 to 1 mole % of sodium to PbTe; then the positive carriers diffuse to the cold areas. There is also an opposing flow of charged carriers resulting from the thermal gradient. Thus the maximum potential is a function of the thermal transport properties.

The open-circuit Seebeck voltage, E , or "thermoelectric power" of a thermoelectric element operating in a temperature gradient, Δt , is given by

$$E = \bar{\alpha} \Delta t, \quad (1)$$

where $\bar{\alpha}$ is the average Seebeck coefficient over the operating temperatures of interest (100-1100°F). When the element is connected to an external load and current flows, I , the voltage across the load is

$$V = \bar{\alpha} \Delta t - IR \quad (2)$$

and the power output is

$$P = I(\bar{\alpha} \Delta t - IR), \quad (3)$$

where R is the resistance of the thermoelectric material.

The efficiency of a thermoelectric element, η , is given by

$$\begin{aligned} \eta &= \frac{P}{Q_S} \\ &= \frac{I(\bar{\alpha} \Delta t - IR)}{Q_S}, \end{aligned} \quad (4)$$

where P is the electrical power output and Q_S is the thermal power input. From the energy balance at the hot junction of a thermoelectric element shown in Fig. 6.30,

$$Q_S = Q_C + Q_{\pi} - \frac{1}{2} (Q_J + Q_T), \quad (5)$$

where

$$\begin{aligned} Q_C &= \int_{t_h}^{t_c} k dt, \text{ in which } t_c \text{ and } t_h = \text{cold and hot junction temperatures} \\ &= k dt = \text{heat conducted,} \\ Q_{\pi} &= I \alpha_h t_h \\ &= \text{Peltier heat effect,} \end{aligned}$$

$$Q_J = I^2 R$$

= Joule heat generated,

$$Q_T = IT \frac{d}{dt} dT$$

= Thompson heat generated.

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Fig. 6.30 Heat Balance in a Thermoelectric Element.

Using the equations in Fig. 6.30 and the Kelvin relations, Eq. (5) can be expressed in terms of the average thermoelectric material properties and the temperature difference $\Delta t = t_h - t_c$ as

$$Q_s = \bar{\alpha} t_h I + \bar{k} \Delta t - \frac{1}{2} I^2 R . \quad (6)$$

Substituting into Eq. (4),

$$\eta = \frac{I(\alpha \Delta t - RI)}{\bar{\alpha} t_h I + \bar{k} \Delta t - \frac{1}{2} RI^2} . \quad (7)$$

The current for maximum efficiency is obtained when $\partial \eta / \partial I = 0$.

For the ideal thermoelectric element having no extraneous resistance

$$I(\max \eta) = \frac{\alpha \Delta t}{R(1 + B)} , \quad (8)$$

where

$$B = \left(1 + \frac{\alpha^2}{\rho k} \frac{t_h + t_c}{2} \right)^{1/2} .$$

The thermoelectric figure of merit, Z , which is a function of only the material properties, is defined as

$$Z = \frac{\bar{\alpha}^2}{\rho k} . \quad (9)$$

The maximum efficiency is then

$$\eta_{\max} = \frac{t_h - t_c}{t_h} \frac{B - 1}{B + t_c/t_h} . \quad (10)$$

Therefore the maximum efficiency of a thermoelectric element having no extraneous resistance is a product of the Carnot efficiency (the first term) and a second factor containing the material parameters and is independent of the geometry of the element.

The extraneous electrical resistance of a thermoelectric converter due to contact interface and connector straps can be an important factor in design optimization. Since all the thermoelectric performance parameters are a function of the extraneous resistance, it must be included in performance calculations. Richards and Nystrom⁹ conveniently accomplished this by defining an extraneous resistance multiplying factor, M , as

$$\left(R \frac{A}{L}\right)_M = M \left(R \frac{A}{L}\right) = \left(R \frac{A}{L}\right) + \left(R_{\text{ext}} \frac{A}{L}\right) , \quad (11)$$

where $\left(R \frac{A}{L}\right)$ is the average resistivity of the thermoelectric material and R_{ext} is the extraneous resistance. Their treatment of the problem is unique in that a common geometric factor L/A (length/area) is included as an integral part of some of the parameters, enabling the performance parameters to be calculated as a function of the material properties only, independent of the element geometry. For example, the power output parameter is

$$\left(P \frac{L}{A}\right)_M = \left(I \frac{L}{A}\right)_M \left[E - \left(I \frac{L}{A}\right)_M \times \left(R \frac{A}{L}\right)_M \right] , \quad (12)$$

where I , R , and P are functions of the extraneous resistance.

Richards and Nystrom also defined a depreciation factor, d , which is a function of M , so that the performance parameters with an extraneous resistance present can be calculated from the ideal performance parameters. For example, the efficiency of a thermoelectric element having extraneous resistance can be calculated from the theoretical efficiency by

$$(\eta)_M = \eta d_{\eta} , \quad (13)$$

where d_{η} is a depreciation for efficiency. By repeating the calculations for various assumed values of the extraneous resistance multiplier, they found that the fractional changes in the performance parameters with extraneous resistance differ only slightly between N- and P-type materials and are relatively independent of the junction temperatures. The depreciation factor for several performance parameters is shown in Fig. 6.31 as a function of the extraneous resistance multiplier.

Fig. 6.31. Effect of Extraneous Resistance Multiplier, M , on Thermo-electric Performance Parameters. (From: J. D. Richards and T. L. Nystrom (3M Co.), "Procedure for the Design of a Thermoelectric Converter and Comparison with Test Results Obtained on a High Power Density Device," presented at Mid-Year Meeting of the Society of Automotive Engineers, Detroit, May 20-24, 1968, SAE Paper 680395.)

The extraneous resistance multiplier is shown in Fig. 6.32 as a function of the design current for various leg diameters. These values of M can be used to determine d_η from Fig. 6.31. From these relations it is clearly seen that large leg diameters are desirable for maximum efficiency, particularly in high-current devices.

Fig. 6.32. Effect of Current and Element Diameter, D , on Extraneous Resistance Multiplier, M . (From: J. D. Richards and T. L. Nystrom, op. cit., SAE Paper 680395.)

Thermoelectric Materials

The figure of merit of a thermoelectric material is given in Eq. (9) in terms of α , ρ , and k , which are functions of the electron concentration of the material. The electron concentration of some semiconductors, including PbTe, silicon, and germanium, is in the proper range for useful thermoelectric devices. By doping and alloying these materials, their figure of merit can be optimized. For the operating temperature range of interest in this study (100 to 1100°F), the PbTe thermoelectric materials have a much higher Z value than the Si-Ge alloys.

Figures 6.33 through 6.35 give α , ρ , and k for two N-type PbTe materials, and Fig. 6.36 gives the dimensionless figure of merit, Zt . The 3N material is more heavily doped with PbI₂ than is the 4N material, which changes the temperature dependence of α , ρ , and k and causes the maximum figure of merit to occur at a higher temperature. A thermoelectric element with a higher Z value can be obtained by making a segmented 3N/4N element in which the 3N material is operated at the temperature range where its Z value is the highest (100 to 500°F) and the 4N material is operated at higher temperatures. The calculated material efficiency of segmented 3N/4N legs is shown in Fig. 6.37 as a function of the junction temperatures.

Fig. 6.33. Seebeck Coefficient of 3N- and 4N-Type Thermoelectric Materials. [From: 3M Company, SNAP-21 Program, Phase II. Thermoelectric Generator Performance Evolution and Design Description, MMM-3691-34 (June 27, 1968).]

Fig. 6.34. Electrical Resistivity of 3N- and 4N-Type Thermoelectric Materials. (From: 3M Company, MMM-3691-34.)

Fig. 6.35. Thermal Conductivity of 3N- and 4N-Type Thermoelectric Materials. (From: 3M Company, MMM-3691-34.)

Fig. 6.36. Dimensionless Figure of Merit of 3N- and 4N-Type Thermoelectric Materials. (From: 3M Company, MMM-3691-34.)

The dimensionless figure of merit of P-type PbTe materials is shown in Fig. 6.38. Segmented 3P/10P legs have been tested extensively in systems having cold junction temperatures below about 300°F. The 2P material has a significantly higher figure of merit and efficiency (Fig. 6.39) than 3P material; however, higher sublimation rates and contact incompatibility have limited the use of the 2P material to relatively short-life devices. The use of refractory metal contacts appears to have solved the compatibility problems, and sublimation can be limited by the use of a xenon gas overpressure or eliminated by completely encasing the material. The design aspects of systems using these materials are discussed in Chapter 10.

Fig. 6.37. Material Efficiency of Segmented 3N/4N Legs. (From data supplied by 3M Company, St. Paul, Minn.)

Fig. 6.38. Dimensionless Figure of Merit of P-Type Thermoelectric Materials. (From data supplied by 3M Company, St. Paul, Minn.)

Fig. 6.39. Material Efficiency of Segmented 3P/10P and 2P Legs. (From data supplied by 3M Company, St. Paul, Minn.)

Advanced thermoelectric materials that have significantly higher figures of merit are under development. The figure of merit of an N-type 95% PbTe-5% Ge-Te alloy is 28% greater than that of 3N material (Fig. 6.40). A P-type alloy, 95% PbTe-5% PbSe, that has a figure of merit 20% greater than 2P material has also been developed (Fig. 6.41). Fabrication development, contacting, and mechanical property studies and life testing of these alloys are in progress.

A thermoelectric material having higher temperature capability and a figure of merit as much as 50% greater than materials used in present-day devices has been developed and is being tested in a classified AEC program.

Fig. 6.40. Figure of Merit of $(\text{PbTe})_{0.95} - (\text{Ge-Te})_{0.05}$ N-Type Material Compared with 3N PbTe. [From: J. P. Dismukes, I. Kudman, and H. I. Moss, Improved Compatible Materials for Thermoelectric Power Generation, NYO-3886-2 (November 1968).]

Fig. 6.41. Figure of Merit of $(\text{PbTe})_{0.95} - (\text{Ge-Te})_{0.05} + 0.7$ mole % Na P-Type Material Compared with 2P PbTe. (From: J. P. Dismukes, I. Kudman, and H. I. Moss, NYO-3886-2.)

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7. CHEMICAL STABILITY AND COMPATIBILITY OF RANKINE-CYCLE WORKING FLUIDS

Dowtherm A, monochlorobenzene, and similar organic working fluids are attractive for small Rankine-cycle turbine-generator systems because they give better turbine proportions, lower turbine speeds, and better turbine efficiencies than any other working fluid, for example, water or mercury. Further, it appears possible to use an organic material as the electrical insulation in the generator stator without loss of dielectric strength after long exposure to the working-fluid vapor.

The chemical stability of the organic working fluid determines the operating life of a Rankine-cycle system, and the most important problem associated with its use in a small nuclear power plant is its tendency to decompose and/or polymerize under the combined influence of elevated temperature and nuclear radiation. Gummy or tarry products of polymerization may foul heat-transfer surfaces, clog bearings, or partly plug small passages such as the nozzles of jet pumps. The gaseous decomposition products include hydrogen, methane, ethane, etc., which tend to accumulate as noncondensable gas in the condenser and may eventually cause a loss in system output. The gas has ordinarily been vented at intervals in the systems that have been built, but this requirement would introduce a serious complication in the operation of power plants designed for operation on the sea floor and hence should be avoided if at all possible. Unfortunately, there is little experience with hermetically sealed Rankine-cycle systems operating in the temperature range of interest here, and there is essentially no experience with such systems operating in both the temperature range and the radiation-dose-level range of interest.

Sundstrand has gained substantial experience with organic working fluids in a series of test programs for the Air Force, NASA, and the AEC. Their current program is sponsored by the AEC. The more significant experiments at Sundstrand are summarized in Table 7.1. These studies were of hermetically sealed systems, with purified fluid used for the initial loading. However, none of the tests included radiation effects.

Aerojet-General Corporation of Azusa, California, in a recent program emphasized the possibilities of employing a jet condenser in an organic Rankine-cycle system. Their operating experience did not include extended endurance tests.

An Israeli concern, Ormat Turbines, Ltd., has built and operated a number of small Rankine-cycle systems employing monochlorobenzene and o-dichlorobenzene. Tables 7.2 and 7.3 summarize their operating experience, which, although substantial, does not indicate either the amount of attention required during operation or the operating conditions.

The Bureau of Standards¹ is systematically investigating the effects of temperature on the formation of both gaseous and solid decomposition products in organic materials suitable as working fluids in small Rankine-cycle systems. The experimental technique entails the use of hermetically sealed glass capsules.

Table 7.1. Summary of Sundstrand Experience with Hermetically Sealed^a
Dowtherm A - A System Containing Boiler, Condenser, and Feed Pump

Type of system: fluid stability test loop simulating
power system
Structural material: type 316 stainless steel
Condenser temperature: 320°F

Boiler Power Input (kW)	Boiler Temper- ature (°F)	Total Operating Time (hr)	Decomposition Products		Longest Run With- out Vent- ing (hr) ^b	Liquid Inventory (lb)		
			Gases (cm ³ /hr)	Solids (lb/hr)		At High Temper- ature	At High Radia- tion	Total
8	650	6450	0.033 ^c 0.038 ^g 0.067 ^h	3.4×10^{-7} ^d	500	1.1 ^e	0	22 ^f
13	700	5435 ⁱ	0.033 ^c 0.0475 ^g 0.137 ^h	3×10^{-6} ^d	500	1.1 ^e	0	22 ^f

^aLoops had minute air inleakage.

^bVapor samples (300 m at 1 psia) taken from hot well at 500-hr intervals.

^cIncludes hydrogen, methane, ethane, and propane found in hot-well vapor.

^dIncludes terphenyls, terphenyl ethers, benzene, and phenol.

^eEstimated inventory above 500°F.

^fInitial inventory - liquid samples withdrawn at rate of 3 lb/10,000 hr.

^gIncludes (c) plus an estimate of the H₂ leaking through a palladium window on hot well.

^hIncludes (c) and (g) plus O₂, N₂, CO, and CO₂ in condenser vapor, which are gases associated with air.

ⁱTest time as of January 16, 1969, continuing.

Table 7.2. Operating Experience with Organic Vapor Turbines
Built by Ormat Turbines, Ltd.

Power [kW(e)]	Type of Unit	Year Built	Hours Operated	Number of Starts
3	Solar	1961	450	320
3	Kerosene	1962	820	430
0.2	Propane	1964	3000	
0.3	Solar	1964	3450	973
0.8	Solar	1966	710	160
			<u>1050</u>	<u>273</u>
			9480	2156+

Table 7.3. Operating History of Ormat
Propane-Heated Hermetically Sealed Units
0.2 - 1.2 kW(e)

Number of Units	Total Starts	Integrated Endurance (hr)	Longest Single Run (hr)	Status August 1968
1	23	15,000	15,000	Still running
3	50	36,000	12,500	All units running
5	120	30,000	6,400	Four units running
3	45	5,000	2,000	All units running
<u>12</u>	<u>238</u>	<u>86,000</u>		

A substantial amount of experience has been acquired by irradiating small samples of organic working fluids as part of the general investigation of radiation damage problems. This work indicates that it is difficult to separate the effects of temperature and radiation; irradiation of an organic fluid at high temperatures seems to yield higher decomposition rates than those resulting from irradiation or high temperature alone.

RADIOLYTIC AND PYROLYTIC DECOMPOSITION

Gamma radiation breaks the hydrogen-carbon bonds in aromatic hydrocarbons such as Dowtherm A; the hydrogen atoms released tend to combine to give molecular hydrogen, and the unsaturated aromatics tend to polymerize. The reaction coefficients for the formation of hydrogen gas and the formation of higher polymers from gamma irradiation are given in Fig. 7.1 as a function of the exposure temperature for biphenyl. Figure 7.2 gives the hydrogen gas formation rate from the pyrolysis of biphenyl as a function of the exposure temperature. Figure 7.3 indicates the total amount of hydrogen gas produced from the combined effects of radiolysis and pyrolysis on biphenyl as a function of temperature at a constant exposure time of 5×10^4 hr. The reactions from which the hydrogen was formed were assumed to be equivalent to zero-order reactions. This assumption allows direct scaling of the values given in Fig. 7.3 linearly with time for the range of the variables shown.

The pyrolysis of Dowtherm A follows a first-order reaction based upon its disappearance. The first-order rate constant is shown in Fig. 7.4 as a function of temperature. The extent of the pyrolytic degradation of Dowtherm A is shown in Fig. 7.5 as a function of temperature for various exposure times. The hydrogen gas for biphenyl as shown in Fig. 7.3 is assumed to apply to Dowtherm A.

A value of approximately 0.92 for G_{HCl} was determined from the data given in Ref. 2 for the gamma irradiation of monochlorobenzene. The temperature at which the experiments were performed was not given. Figure 7.6 gives the hydrogen chloride formation rate from the pyrolysis of monochlorobenzene as a function of the exposure temperature. Figure 7.7 indicates the total amount of hydrogen chloride gas produced from the combined effects of radiolysis and pyrolysis on monochlorobenzene as a function of the exposure temperature at a constant exposure time of 5×10^4 hr and a constant G_{HCl} of 0.92. The reactions from which the hydrogen chloride was formed were assumed to be equivalent to zero-order reactions, which allows direct scaling of the values given in Fig. 7.7 linearly with time for the range of variables shown.

The pyrolysis of monochlorobenzene follows a first-order reaction based upon its disappearance. The first-order rate constant is shown in Fig. 7.8 as a function of temperature. The extent of pyrolytic degradation of monochlorobenzene is shown in Fig. 7.9 as a function of the temperature for various exposure times.

Fig. 7.1. Temperature Dependence in Biphenyl Radiolysis with Gamma Rays.
 [From: K. L. Hall, Chap. 3, p. 54, in Organic Coolant Summary Report,
 R. F. Makens, ed., IDO-11401 (December 1964).]

ORNL DWG. 69-14113

Fig. 7.2. Volume of Hydrogen Produced from Pyrolysis of Biphenyl.

Fig. 7.3. Volume of Hydrogen Produced from Addition of Pyrolytic Effects to Radiolytic Effects of Gamma Rays on Biphenyl for 5×10^4 hr.

Fig. 7.4. First-Order Rate Constant for Pyrolysis of Dowtherm A. [From: Aerojet-General Corporation, Organic Rankine-Cycle Power System Demonstration Program, AGC-3486 (December 1967), p. 24.]

Fig. 7.5. Pyrolytic Degradation of Dowtherm A.

Fig. 7.6. Volume of Hydrogen Chloride Produced from Pyrolysis of Monochlorobenzene. (Calculated from data in Ref.2.)

Fig. 7.7. Volume of Hydrogen Chloride Produced from Addition of Pyrolytic Effects to Radiolytic Effects of Gamma Rays on Monochlorobenzene for 5×10^4 hr with $G_{\text{HCl}} = 0.92$.

ORNL DWG. 69-14119

Fig. 7.8. First-Order Rate Constant for Pyrolysis of Monochlorobenzene.
(Calculated from data in Ref. 2.)

ORNL DWG. 69-14120

Fig. 7.9. Pyrolytic Degradation of Monochlorobenzene.

Bordonali et al.² report: "Attapulgius clays, while not capable of absorbing the organic chlorinated compounds, do absorb both the hydrochloric and the ferrous chloride, and therefore could interfere with the formation of fouling films through the removal of the mineral chlorine and of the soluble iron." This characteristic of Attapulgius clays could be used to reduce the level of hydrogen chloride produced from the degradation of organic chlorine working fluids.

The total amount of working-fluid degradation for Dowtherm A and monochlorobenzene is shown in Fig. 7.10 as a function of a Rankine-cycle efficiency for various operating conditions. Figures 7.11 and 7.12 give the volume of gas formed from biphenyl and monochlorobenzene respectively as a function of a Rankine-cycle efficiency for the operating conditions used to determine total decomposition. The biphenyl data for gas formation were used for Dowtherm A because it is chemically similar to biphenyl and there is a lack of information on Dowtherm A.

The values given in Figs. 7.10-7.12 are best estimates determined by available data. The bulk of the data considered was determined in static glass capsule tests. The effects of a dynamic system on the materials of construction have not been included, as their magnitude is indeterminate from the available data. To determine the total effects for a real system, the time-temperature history of the fluid must be integrated as it travels through the fluid system, with the materials of construction of the containing system taken into account since they have an influence on the thermochemistry of the fluid. Thus the only way to evaluate the suitability of these fluids for the purposes intended is to run them in a system that is as close to the real system as possible.

CORROSION AND MASS TRANSFER IN BOILING-WATER SYSTEMS

While corrosion and mass transfer in a properly designed and operated water system are probably so small that attack and penetration of the metal surfaces will be less than 0.001 in. in a 3-year period of operating, even very small amounts of surface attack may lead to excessive local deposits in such critical regions as the boiler, bearings, or turbine buckets (where dynamic unbalance may prove to be a problem). The first step in avoiding these difficulties is the choice of the structural material of the system.³ Stainless steel is an obvious candidate, but is sensitive to chloride corrosion. The high-nickel alloys, Inconel, Inco 800, and Inco 600, have properties similar to those of stainless steel but are not subject to chloride corrosion and hence would be more attractive for use in oceanographic work, where chloride contamination of the cycle working fluid might be a problem. Perhaps the most promising candidate of all is titanium, which has shown exceptional resistance to attack by seawater, uranyl sulfate, and other corrosive water solutions. Further, titanium turbine buckets have shown about the same resistance to erosion as Stellite, the best material available heretofore. The resistance of titanium to oxidation stems from the formation of a dense, hard, adherent oxide film which can be induced by an anodizing process similar to that used for aluminum.

ORNL DWG. 69-14121

Fig. 7.10. Pyrolytic Degradation of Monochlorobenzene and Dowtherm A as a Function of Cycle Efficiency After 5×10^4 hr Exposure.

Fig. 7.11. Volume of Hydrogen Evolved from Biphenyl per Pound of Boiler Inventory as a Function of Cycle Efficiency After 5×10^4 hr Exposure.

ORNL DWG. 69-14123

Fig. 7.12. Volume of Hydrogen Chloride Evolved from Monochlorobenzene per Pound of Boiler Inventory as a Function of Cycle Efficiency After 5×10^4 hr Exposure.

Water Chemistry

High-temperature attack of some of the structural metals (but not titanium) by water depends primarily on the oxygen concentration, which will depend on the extent to which decomposition of the water occurs under irradiation. This matter is treated in some detail in an ORNL report.³ Briefly, as long as the hydrogen and oxygen are maintained in solution with a small excess of hydrogen, recombination will occur under the influence of a strong radiation field. A small excess of hydrogen will suppress the free-oxygen concentration in the system.

In common with boiling-water reactors, in radioisotope systems radiolytic oxygen will be present in the steam that is evolved in the boiler as it passes through the turbine into the condenser. However, the concentrations of oxygen will be low because of the small volume of water from which the oxygen is evolved. In the steam boilers considered in this study, only a very small amount of radiation energy is deposited in the water in the boiler where the free surfaces permit the dissociated hydrogen and oxygen to readily evolve from solution. Radiolytic gas ($2\text{H}_2 + \text{O}_2$) formation in the boiler of a 5-kW(e) power unit is estimated to be $0.01 \text{ cm}^3/\text{sec}$ at STP, or about 0.4 ppm of oxygen in the steam if gas stripping is so rapid that no recombination of radiolytic gases takes place. No accelerated attack due to this oxygen would be expected on titanium at the peak temperature in the system, or on titanium or stainless steel at the condenser temperature.

Control of Radiolytic Gases

The pressure of radiolytic gases within the condenser will be controlled at a low value by returning them to the boiler along with the feed water. Recombination occurs while the dissolved gases are traversing the boiler inlet-flow passages. This recombination method is feasible with this system because of special design features of the condenser, feed pump, and connecting piping and because of the boiler design.

Experience with once-through condensers in the MPRE program demonstrated that, with a system designed to operate with a dry sump condenser in the manner employed in the MPRE and proposed for the power plants described in this report, the fluid leaving the condenser tends to occlude bubbles to the extent of about 50% by volume. Subcooling of the stream leads to the condensation of the vapor and the collapse of these bubbles so that the only gas bubbles in the liquid stream entering the feed pump will consist of noncondensable gases. Experience with the Intermediate Water System, an electrically heated test rig at ORNL which simulates a full-scale space power plant, showed that the pump performance is not seriously affected by having as much as 50% by volume of vapor bubbles in the fluid stream leaving the condenser. For an undersea power unit in which the outer pressure shell would serve as the condenser, a baffle could be installed in the vessel as a liner just inside the inner wall so that the steam leaving the turbine would flow into the gap between

the baffle and the shell at the top and then flow down along the wall of the cooled pressure shell. The space between the baffle and the pressure shell would run about $3/4$ in. at the top, and would taper down to about $1/4$ in. near the bottom. In addition, to maintain a moderately high velocity of the vapor in the condenser passages and thus promote entrainment of vapor bubbles in the liquid, the baffle could be corrugated so that the vapor flow would have to spiral through the latter portion of the condensing region.

Recombination in the boiler inlet-flow passages can be utilized because boiling does not occur in this region even though the water there is subjected to high-intensity gamma radiation. In order to ensure that recombination will occur rapidly and that the steady-state concentrations of oxygen will be very low, hydrogen will be added to the system to make the concentration within the feed water about $30 \text{ cm}^3/\text{kg}$. This added hydrogen will also dilute the oxygen in the condenser, making the mixture non-flammable even without steam dilution.

Test data indicate that the gas entrainment rate could be about 4 times the $30 \text{ cm}^3/\text{kg}$ proposed above. Since the gas overpressure required to hold the hydrogen and oxygen in solution is only 2 atm, the pressure leaving the feed pump will run about 200 psia, and the gas bubbles will be very finely divided (judging from experience with the Intermediate Water System), the gas should go back into solution quickly — and should be in solution before it reaches the boiler. The amount of hydrogen that would have to be added to the system to give $30 \text{ cm}^3/\text{liter}$ of feed water would be ~6 liters for the system proposed later in this report. In practice, some hydrogen will be generated by corrosion, and this will be taken into account in the initial hydrogen additions.

The above indicates a reasonable possibility of avoiding the use of special water treatment equipment in order to avoid difficulty with corrosion or deposits in a hermetically sealed steam system of the type proposed. If a corrosion-resistant material such as titanium is used when oxygen is present in the higher temperature regions and if around 30 cm^3 of H_2 per liter of water can be entrained at the condenser outlet and can be recirculated through the system, the large complex of instrumentation, valves, and auxiliary equipment normally employed in pressurized and boiling reactor plants for water cleanup purposes will not have to be used.

REFERENCES

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8. AVAILABILITY OF EQUIPMENT FOR CLOSED THERMODYNAMIC-CYCLE SYSTEMS DESIGNED TO PRODUCE 1 TO 10 kW(e)

A major obstacle to the development of isotope power units of 1 to 10 kW(e) is the lack of a commercially available small closed-cycle power unit. Further, a review of efforts to develop such small systems for military and space applications shows that they have generally been expensive and time-consuming.

In surveying the field for heat engines that might be adapted to meet the Navy requirements for an undersea power source at a reasonable cost, the status of the machine development work under way was discussed with the engineers responsible. These discussions were carried out initially by phone and continued in detail in trips to SAE and ASME meetings and to the laboratories of Aerojet, NASA, Rocketdyne, Sundstrand, Swenson Research, and Thompson-Ramo-Woolridge Equipment Co. (TRW). This section summarizes the information obtained on the status of the development of the various pieces of experimental equipment that appear to be possible candidates for a small isotope-power-plant prototype unit, together with the costs and time periods for their respective development programs.

The principal problem in the development program for any thermodynamic cycle system concerns the turbine-generator or engine-generator unit. The heat exchangers and pumps present much less difficult problems; in general they can be designed so that a good degree of reliability can be obtained with a modest amount of shakedown testing. The turbine- or engine-generator units, however, must operate with close clearances, and can be damaged if there are deposits, warping, growth, or thermal distortion. The rubbing surfaces probably should be lubricated with the working fluid — which generally is a poor lubricant — in order to avoid contamination of the working fluid by the lubricant and the formation of deposits on such regions as the heater. For good efficiency the speeds of small turbines must be high, but these high speeds lead to bearing and shaft dynamic stability problems. Further, the windings in the generator may lose dielectric strength from gradual permeation by the working fluid.

SUMMARY OF TURBINE-GENERATOR DEVELOPMENT EFFORTS

The difficulties that have been experienced in the past, the magnitude of efforts that have been expended, and the degree of success achieved are indicated by the information summarized in Table 8.1. Each effort is discussed in some detail below.

Mercury Vapor Turbine Development at TRW

Historically the first item is the mercury-vapor Rankine-cycle development program that was carried out at TRW mainly for the AEC. Something over 30,000 hr of operating experience was accumulated with small turbine-generator units¹ at a total program cost of ~\$30,000,000. The major obstacles encountered were the difficulties inherent because of the mass transfer and corrosion of mercury in iron-chrome-nickel alloy systems, the poor wetting characteristics of mercury which make it reluctant to

enter the small clearances in bearings and also lead to boiling instabilities, and the relatively high temperatures which make thermal distortion and differential thermal expansion problems important.

Mercury Turbine Development at Aerojet

Work on a mercury-vapor Rankine-cycle system was initiated at Aerojet in 1960 under NASA contract.² The objective was to develop a 60-kW(e) Rankine-cycle system that could be coupled to a NaK-cooled fission reactor under development by Atomics International. The system design power output was subsequently reduced to 30 kW(e). Aerojet encountered the same troubles arising from the poor wetting characteristics, mass transfer, and corrosion that were experienced by TRW. To ease the bearing lubrication problem the unit was redesigned in 1963 to employ an organic fluid as the bearing lubricant, and a Visco seal system was installed to isolate the lubricant from the mercury.³ Tests of turbine-generator units have been under way since 1960, and around 2000 hr of turbine-generator operation has been obtained.

Organic Turbine Development at Aerojet

The prime parameter in the aerodynamic design of the turbine is the square root of the molecular weight of the working fluid, and Dowtherm A vapor with a molecular weight of 165 has much the same fluid dynamic properties as mercury vapor with a molecular weight of about 200, whereas steam with a molecular weight of 18 is a completely different fluid as far as vapor turbines are concerned. As a consequence, it is possible to operate turbines designed for mercury on Dowtherm A provided that the turbine inlet temperature is reduced from perhaps 1100°F for mercury to about 600°F for Dowtherm A. Changing the working fluid from mercury to Dowtherm A greatly relieves the thermal stress and distortion problems, and — even more important — should greatly relieve the bearing lubrication problems.

In view of this, work on adapting the SNAP-8 turbine to an organic Rankine-cycle system was initiated at Aerojet in 1966.⁴ Initially the emphasis on the work with organic fluid systems was on jet condensers and the advantages of jet condensers under zero-g conditions in space. The work has been carried out with both AEC and Aerojet funds. (There seems to be little advantage to the use of jet condensers for terrestrial applications other than a means of entraining and redissolving at least a part of the noncondensibles.) Aerojet has designed and built a 12-kW(e) unit, which they operated for about 80 hr, and a 6-kW(e) unit (see Fig. 8.1), which they operated for about 50 hr. Their present program is concerned with the latter and is funded by the AEC. The prime objective will be to carry out extensive endurance testing.

The Aerojet unit makes use of ball bearings lubricated with special-grade mineral oil. Aerojet employs Visco seal pumps on the shaft on either side of the bearings, which not only act as seals between the lubricant and the Dowtherm A working fluid but also as a means of recirculating the mineral oil lubricant. They have had some difficulty in balancing the pressures between their lubricating oil and Dowtherm A circuits, but

Table 8.1. Summary of Power Units of 1 to 10 kW(e) Capacity

Thermodynamic Cycle	Engine	Manufacturer and Sponsor	Power (kW)	Overall Thermal Efficiency (%)	Total Operating Experience (hr)	Size of Engine Unit		Investment in Development Program (to 1968)	Heat Source	RPM	Voltage	Frequency (cps)	Number of Phases
						Dimensions (in.)	Weight (lb)						
<u>Rankine</u>													
Mercury vapor	Turbine CRU-V	TRW, AEC	6		~25,000			~\$20,000,000 (in series of related programs)	Electric	36,000	115 ac	1800	3
	Turbine CSU-I		3		6700	5 x 12			Electric	40,000	110 ac	2000	2
Dowtherm A	Turbine	Sundstrand, USAF, Navy, and NASA	1.5	17	2700	9 x 9 x 12		965,000 (+5,000,000 in related programs)		24,000		3200	3
	Turbine	Aerojet	12	17.7		12 x 15	130			24,000		1600	3
<u>o</u> -Dichlorobenzene	Turbine	Ormat	0.2 to 1.2		86,000				Propane				
Steam	Reciprocating	Thermo Electron, Corps of Engineers	0.1	3.7	1500	16 x 16 x 12.5	29	350,000+	Kerosene	1800	28 dc		
			0.3	4.8	900	14 x 14 x 14	42		Kerosene	1800	28 dc		
			1.5	7.7	3000	24 x 24 x 24 (complete plant)	125 (plant)		Kerosene	3600	28 dc		
	Turbine	Swenson Research Lab, AGA	3.7		500	6 x 10			Gas	64,000	60 ac	1100	3
<u>Brayton (He-Xe)</u>	Turbine	AiResearch, NASA	3	20	160	4.6 x 4.6 x 21			Electric	64,000	115 ac	2133	3
	Turbine	Pratt & Whitney, NASA	10 to 30		1000			\$20,000,000	Electric	36,000	115 ac		3
	Turbine	AiResearch, NASA	2 to 10	25	100	15 x 20			Electric	36,000	115 ac	1200	3
<u>Stirling</u>	Reciprocating	General Motors	3 to 350	35	20,000				Kerosene				
			3	35	3000			Kerosene					

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Fig. 8.1. ORACLE 6-kW(e) Turbine Alternator Pump Assembly (TAPA).
(Reproduced through courtesy of Sundstrand Aviation - Denver.)

believe that some modifications in the Visco seals will handle this satisfactorily. The frictional losses in the bearings and seals represent about 2.5 kW of the power input from the turbine, a serious shortcoming of the two-fluid arrangement. Aerojet chose this arrangement because they could virtually duplicate the parts used so successfully in the SNAP-8 mercury vapor turbines and therefore minimize their developmental effort. Unfortunately, this arrangement wastes power in friction, complicates the system, and offers possibilities for mixing of the lubricating oil and Dowtherm A under transient and other adverse conditions.

Organic Turbine Development at Sundstrand

Since 1962 Sundstrand has been engaged in developing a series of Rankine-cycle turbine-generator units designed to use organic materials such as Dowtherm A. They built a 1.5-kW(e) turbine-generator unit and operated it for 2700 hr under a Navy contract in 1964; it is still in good running condition.⁵ A small program involving a 6-kW(e) unit⁶ is continuing under AEC sponsorship. The first unit has been built and preliminary shakedown tests are now under way. The rotary unit is approximately 18 in. long and the turbine wheel has a diameter of approximately 7.5 in. The unit employs hybrid bearings lubricated by the working fluid (see Fig. 8.2).

Sundstrand is working on a number of larger systems, one of which comes under a program funded jointly by Sundstrand and the American Gas Association (the program cost is about \$750,000 per year). The objective of this work is to produce a 40-kW(e) system by 1972 which would provide electric power

Fig. 8.2. Components of Sundstrand Aviation 6-kW(e) Combined Rotating Unit. (Reproduced through courtesy of Sundstrand Aviation - Denver.)

for shopping centers and institutions. They plan to use Dowtherm A at 700°F, expanding it down to 0.2 psi and about 280°F. Waste heat from the cycle would be used in an absorption refrigeration system. The power produced would be at high frequency and would be used for fluorescent lighting. The heat content of the gas burned would amount to 860,000 Btu/hr; the system would produce waste heat amounting to 550,000 Btu/hr at 270°F for the absorption refrigeration system. The efficiency of the system as a producer of electricity is expected to fall between 16 and 18%. The size of the unit would be 6-1/2 ft high × 3 ft wide × 8 ft long.

They have also designed a larger system, which would use pyridine and yield an overall thermal efficiency of 27%. It would fit in a casing 4 ft in diameter by 17 ft high and would weigh about 22,000 lb. The peak temperature would be 700°F and the heat would be rejected to 70°F water.

Organic Turbine Development at Rocketdyne

In 1966 Rocketdyne set out to design and build a 1.5-kW(e) power unit to be run on chemical fuel.⁷ The unit was designed to meet the Ft. Belvoir requirements for a portable power unit that would operate very quietly in the field. They chose to use monoisopropylbiphenyl with a turbine inlet temperature of 700°F and a turbine inlet pressure of 57 psia. The system was built of aluminum and stainless steel, with most of the joints welded but with some of them having O-ring and Swagelok fittings to facilitate access for maintenance. The turbine-generator unit was built with ball bearings to operate at 24,000 rpm (see Fig. 8.3). A Barsky

Fig. 8.3. 1.5-kW(e) Organic Turboalternator Pump. (Reproduced through courtesy of Rocketdyne.)

impeller was employed and was found to give a pump efficiency of 61% even though a wheel from another machine was used which was not designed for this particular application. The overall thermal efficiency of the power unit was 7.2%, with the stack losses representing 25% of the energy in the fuel. They estimated that with a low condenser temperature such as would be available in a naval application and with a better boiler and burner design they could manage to get as much as a 13% overall thermal efficiency.

It took Rocketdyne one year to design and build the prototype unit, and about another year to carry it through an extensive series of tests. The unit was then sent to Ft. Belvoir, where it was successfully started and run several times before being shipped to Rocketdyne. The program was not supported by any governmental agency. Altogether the unit was operated for about 350 hr with 500 starts and stops. The principal noise emitted from the unit came from the burner, but they felt that this could be corrected. The unit is in a package about 2 ft on a side and is in one of the laboratories at Rocketdyne. It has not been operated for about a year but is in operating condition. Rocketdyne does not have any programs under way or in sight in which the unit will be used or any similar unit designed and built.

Steam Engine Development at Thermo Electron

The small steam engine under development by Thermo Electron for the Army Corps of Engineers at Ft. Belvoir has entailed the development of three units of, respectively, 0.25, 0.67, and 3 bhp.⁸ A total of 5400 hr of operation has been obtained, of which 3000 hr has been obtained on the 3-bhp unit. However, many operating difficulties were experienced, principally in connection with the lubrication of the packing glands separating the cylinder from the crank case. It is difficult to say at this time when or how well these problems will be resolved. Current efforts at Thermo Electron are being directed toward operating the unit with organic working fluids.

Organic Turbines Developed by Ormat

A series of small turbine-generator units have been built, tested, and marketed by Ormat Turbines, Ltd., in Yavne, Israel.⁹ They operate on either Dowtherm E or monochlorobenzene and deliver from 400 W to 3 kW(e) with a turbine inlet temperature of about 350°F, the heat source usually being a propane burner. Ormat reports that two 400-W units have operated for 15,000 hr each on nonstop, unattended operation. These units have operated for an impressive 80,000-plus hours, the bulk of it in field operation.

Steam Turbine Development at Swenson Research

Insofar as can be determined, only one small hermetically sealed steam turbine-generator unit with water-lubricated bearings has been built and operated (although many thousands with oil-lubricated bearings and shaft seals have been run for many years). This unit was developed by Swenson Research, Inc., in Bedford Heights, Ohio, for the Consolidated Gas Service Corporation as a power source for air-conditioning units. It is a

3.7-kVA 3-phase machine with a single-stage 3.5-in. Terry turbine wheel designed to operate at 64,000 rpm to produce 1100 cps at 60 V per phase (to ground).¹⁰ The inlet-steam pressure is 185 psig at 800°F, the discharge pressure is 2 psia (120°F), and the turbine efficiency is 41%. The unit has been operated for about 1000 hr. The generator windings are potted in epoxy resin to protect them from the steam. The unit is supposed to operate for at least 3 years unattended. The overall thermal efficiency of the steam power plant is 9.5%. A section through the unit is shown in Fig. 8.4.

Fig. 8.4. Cross Section Through Turboalternator Rated at 3.5 kW(e) (shown in Fig. 10.11); Length: 9.5 in. (Reproduced through courtesy of Swenson Research, Inc.)

The turbine-generator unit and the feed pump are mounted inside an aluminum bell jar about 1 ft in diameter and 16 in. high. The unit is mounted on rubber in a partial vacuum so that it will be remarkably quiet. During a test a barely audible whine could be heard and the unit was so quiet that the low hiss of the steam exhaust from the reciprocating feed pump, which operates at about 30 cycles/min, could be heard. If the steam condenser temperature is reduced from 140°F to 60°F (as would be the case in a unit immersed in 40°F water), the noise level is greatly reduced. There is no sign of bearing noise, because the turbine-generator unit is fitted with tilting-pad bearings; the entire installation is designed to give a minimum noise level, which would make it well suited for a home installation.

Variable-Speed Generator Built by Westinghouse (Lima, Ohio)

Westinghouse has built a 30-kW, 28-V, dc generator for use on gas turbine engines in Army tanks. About 30 of these units have been built for the Corps of Engineers at Ft. Belvoir, and about 10 have been subjected to a substantial amount of testing. These units have the attractive characteristic that they are homopolar generators with a set of voltage control and conversion equipment built into them so that they can be operated at any speed between 6000 and 18,000 rpm and still deliver a constant voltage. The unit is also attractive in that it was designed for flange mounting and can be converted to a turbine drive by using a single-stage turbine wheel overhung at the end of the shaft on the splined stub that is used for driving the generator from the tank engine. The stator was designed to be cooled with oil in passages that operate in parallel with the oil supply for the bearings. It is believed that it would not be too difficult to design a turbine casing and wheel that could be fitted on the drive end of this generator and to install a new cover plate incorporating a feed pump on the antidrive end of the generator. The biggest problem would probably be associated with the bearings. Ball bearings are used in the unit now and they would have a limited life — probably not more than a few thousand hours. Further, ball bearings have shown high wear rates when lubricated with Dowtherm. Thus the bearing region should probably be redesigned for stepped-pad or tilting-pad bearings.

Brayton-Cycle Turbine Under Development by NASA

Work on machinery for a small Brayton cycle has been under way at AiResearch¹¹ and NASA Lewis Research Center¹² for about 8 years. The operating experience obtained to date is limited — about 1000 hr on an early turbine-generator unit of a two-shaft machine in which the compressor was to have been driven by a separate turbine. The latter unit, a 2- to 10-kW(e) machine, was started up about February 1, 1969, with tilting-pad gas bearings, and completed 600 hr of testing at about 5 kW(e) by mid-May 1969.

The principal difficulty experienced during the 600-hr test was gas leakage from the system. The first 375 hr was run with argon and the last 225 hr with krypton. Leakage from the system increased continuously throughout the test, reaching a value of 80 lb/day with argon and 40 lb/day with krypton. Much of the leakage apparently stemmed from a Viton seal in the parting surface between the turbine and compressor casing. The temperature of the flange at this seal was 380°F, which, while somewhat higher than originally anticipated, was still somewhat lower than the seal manufacturer had indicated would be acceptable. Figure 8.5 shows a section through the unit. Four additional units of this new model are being built, and one of them will be installed in a new environmental test facility at Plumbrook, for which initial operation is expected in the summer of 1969. An overall thermal efficiency of 25% is anticipated for this system. No difficulty was experienced with the gas bearings in either the 1000- or 600-hr test, but the difficult problems presented by gas bearings make it hard to predict how long it will take or how much it will cost to develop a reliable system.

Fig. 8.5. Cutaway of Brayton Rotating Unit; Length of Unit: 29.5 in. (Reproduced through courtesy of AiResearch Manufacturing Company of Arizona.)

Stirling Engine Development at General Motors

The principal efforts in America on the development of the Stirling engine have been carried out at General Motors Research Laboratories and at the Allison Division of General Motors. A total of 23,000 hr of testing has been accumulated on a number of different models since the program began in 1958. The unit of greatest interest for the purposes of this study delivers 11 bhp at 3600 rpm with a heater wall temperature of 1400°F and a cooling water temperature of 100°F. This unit could be coupled to a generator to give a net electrical output of about 7 kW(e) with an overall thermal efficiency of about 25%. The total operating experience with this model is 3000 hr and the longest run has been 750 hr, with piston ring and piston rod seal wear principally limiting the endurance life. This implies the importance of avoiding radiation damage to the lubricant in order that the lubrication problems not be aggravated.

AVAILABILITY OF TURBINE-GENERATOR UNITS FOR SYSTEM EVALUATION

The first step in the evaluation of the feasibility of a proposed thermodynamic-cycle system is determination of the compatibility of the materials in the system and of the stability and control characteristics of the plant. A turbine-generator unit will be an essential part of such a system inasmuch as the effects of deterioration in the working fluid on the bearings, possible attack of the magnetic materials in the generator by the working fluid, and possible deterioration of the electrical insulation as a result of exposure to the working fluid are all vital problems. A review of Table 8.1 and discussions with representatives of the various development organizations involved indicate that the principal candidates for a preliminary test unit are as follows:

1. The Sundstrand 1.5-kW(e) unit that has been run for approximately 2500 hr on Dowtherm A is available and can be obtained on loan from Sundstrand in a relatively short time.
2. A 6-kW(e) unit designed by Sundstrand or Aerojet for operation on Dowtherm A might be employed. The time required to secure this unit would probably be about 8 months and the cost probably \$30,000 to \$60,000.
3. A production version of a monochlorobenzene turbine-generator unit of about 1 kW(e) could probably be obtained from Ormat in a few months for \$10,000 to \$20,000.
4. A 3-kW(e) steam turbine-generator unit could probably be obtained from Swenson Research, Inc., in a few months for \$10,000 to \$20,000. The greatest uncertainties would probably be the life of the feed pump and the life of the electrical insulation in the generator.

In summary, it appears that the most promising approach would be to employ the Sundstrand 1.5-kW turbine-generator unit in a Dowtherm-A system, an Ormat unit in a monochlorobenzene system, or a small Swenson unit in a

steam system. Endurance tests of two or three of these units in hermetically sealed systems with both the temperature and radiation conditions anticipated in the field would give a much better basis for judging the material compatibility problems. Particular attention should be given to possible losses in performance as a consequence of the buildup of non-condensibles, to possible bearing troubles from deposits of tars and resins, and to losses in the dielectric strength of the windings in the electric generator from permeation by the working fluid. If properly designed, these systems could also permit an investigation of one or more fresh approach to the problems of system control with the objective of validating a control scheme that would give a very simple and reliable system.

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9. DETAILED ANALYSES OF CYCLES FOR ISOTOPE POWER UNITS

The information presented in Chapters 6, 7, and 8, coupled with practical design considerations, indicates that the field of interest may be narrowed to a number of Rankine cycles and the Brayton cycle. In view of the naval objectives for about 3 years of unattended operation on the sea floor and in view of the definite life limitations imposed by piston ring and cylinder wear in all but low-temperature low-efficiency reciprocating steam engines, it further appears that the field should be narrowed to turbines.

In choosing the working fluids for the Rankine cycles of this portion of the study, steam and organic working fluids were logical choices. Since organic working fluids are limited by chemical stability considerations to temperatures of not more than 700°F and steam to not more than 1100°F, whereas with refractory metal fuel capsules it is possible to go to 2000°F, cesium and potassium were considered as high-temperature Rankine-cycle working fluids to determine to what extent these fluids would make possible a higher thermal efficiency. The Brayton cycle was included because it, too, would permit the use of a high peak temperature in the cycle and consequently make possible a higher thermal efficiency, at least for the ideal case.

The loss in strength of structural materials at elevated temperatures and the corrosion and mass-transfer problems limit the turbine inlet temperature that is practicable to employ. In addition, turbine aerodynamic design considerations impose important limitations on turbine inlet and outlet design conditions because there is little point in obtaining a high efficiency for the ideal cycle if it entails operation in a fluid flow regime that yields a low efficiency in the turbine. The effects of the choice of working fluid and design conditions on the overall thermal efficiency of the complete power plant were studied, using a common set of basic design criteria to obtain essentially comparable degrees of optimization in the aerodynamic design of the turbine for each fluid and each design condition considered.

The study of turbine cycles for isotope power units of this report represents an extrapolation and extension of two studies^{1,2} that were carried out previously at ORNL. The two studies were the development of a series of proposed steam turbine designs for terrestrial low-power reactor plants of 30 kW(e) to 2 MW(e),¹ and a design comparison of cesium and potassium as working fluids for Rankine-cycle space power plants.²

REFERENCE DESIGN CONDITIONS

The fluids considered for the Rankine cycles were potassium, cesium, steam, and Dowtherm A, and a mixture of helium and xenon was considered for a Brayton cycle. Two turbine inlet conditions were considered for the steam cycle: 1050°F at 1000 psia and 493°F at 640 psia. The 1050°F steam temperature condition was chosen as representative of a coal-fired steam power plant, while the 493°F saturated-vapor condition was chosen as

representative of current pressurized-water reactor power plants. The two condenser temperatures considered for each steam cycle were 150°F at 3.7 psia and 70°F at 0.36 psia, representing respectively a unit designed for operation with air cooling under hot day conditions and design conditions for power plants that might be submerged in the sea. For the potassium and cesium vapor cycles the two turbine inlet temperatures were 1540°F and 2000°F, which represent the maximum practicable temperature for stainless steel and refractory metal systems respectively. In both cases the turbine discharge pressure was taken as 0.5 psia, because the vapor density falls off so rapidly at lower condenser temperatures that the turbine proportions become badly distorted and a severe loss in efficiency results.

The turbine inlet temperature for the Dowtherm-A cycle was chosen as 650°F because a substantial amount of operating experience indicates that Dowtherm A will thermally decompose under higher temperatures. A condenser temperature of 290°F was chosen to give the same 0.5-psia turbine outlet pressure that was used for the cesium and potassium vapor cycles.

A turbine inlet temperature of 1540°F was chosen for the Brayton cycle because of turbine bucket strength limitations and heat transfer and hot spot problems in the isotope heat source. The heat sink temperatures employed for the steam cycle were also chosen for the Brayton cycle because they seemed equally applicable.

THERMODYNAMIC CALCULATIONS

Rankine Cycles

Most of the thermodynamic calculations for the Rankine cycles were carried out with conventional techniques and the thermodynamic data of Refs. 3-7. In the cesium calculations the Mollier diagram in Ref. 2 had to be extrapolated from the thermodynamic relations and data given in Ref. 4. The results of these thermodynamic calculations are summarized in Table 9.1.

Dowtherm-A Cycle

If the vapor from Dowtherm A is expanded from a dry saturated condition, the expansion line goes into the superheated region rather than into the saturated region as occurs in most types of vapor turbines. As a consequence, for the design case presented here, the vapor at the turbine exit is superheated about 200°F. Thus there is a strong incentive to make use of a regenerative heat exchanger to employ this heat for regenerative feed heating prior to condensing the vapor. An 80% effective regenerator was assumed and the calculations were made accordingly.

Brayton Cycle

In choosing a Brayton-cycle working fluid for an isotope power plant it is advantageous from the corrosion standpoint to employ an inert gas. The first point to consider in choosing between the various inert gases is the effect on cycle efficiency. The thermodynamic efficiency of the ideal

Table 9.1. Summary of Thermodynamic Cycle and Aerodynamic Performance Calculations for a Series of Rankine Cycles Designed to Produce 5 kW(e)

Item	Parameter	K		Cs		H ₂ O				Dowtherm A	
1	T ₁ , turbine inlet temperature, °F	2000	1540	2000	1540	1050	493	1050	493	650	650
2	P ₁ , turbine inlet pressure, psia	152.3	29.6	234.5	57.5	1000	640	1000	640	71.3	71.3
3	V ₁ , vapor inlet specific volume, ft ³ /lb	3.726	16.7	0.6774	2.45	0.8633	0.720	0.8633	0.720	0.902	0.902
4	Vapor inlet enthalpy, Btu/lb	1216.5	1195	315.3	309.5	1533.7	1203	1533.7	1203	393.3	393.3
5	S ₁ , vapor inlet entropy, Btu/lb·°F	1.0237	1.0867	0.3223	0.3373	1.6721	1.4396	1.6721	1.4396	0.842	0.842
6	X ₁ , vapor quality, %	100	100	100	100	100+	100	100+	100	100	100
7	T ₂ , turbine outlet temperature, °F	(1380) 920	920	(1250) 790	790	150	150	70	70	490 ^a	470
8	P ₂ , turbine outlet pressure, psia	0.5	0.5	0.5	0.5	3.7	3.7	0.363	0.363	0.50	0.25
9	V ₂ , turbine outlet specific volume, ft ³ /lb	688	688	186.7	186.7	97.1	97.1	868.4	868.4	123	234
10	h ₂ , turbine outlet enthalpy, Btu/lb	822	911	215.2	235.4	1006	864	884	761	323	310
11	X ₂ , turbine outlet quality, %	63	72	63	72	88	74	80.3	68.5	100+	100+
12	Δh _{is} , isentropic enthalpy drop, Btu/lb	394.5	284	100.1	74.1	527.7	339	650	442	70.3	75.3
13	h _{l2} , enthalpy of liquid at condenser temperature, Btu/lb	269	269	68	68	118	118	38	38	97.9	84.4
14	h _{lc} , enthalpy of liquid, Btu/lb	251	251	62	62	108	108	33	33	86.6	73.4
15	Amount of subcooling, °F	109	100	100	100	10	10	5	5	25	25
16	Δh _b , enthalpy rise in boiler, Btu/lb	965.5	944	253.3	247.5	1525.7	1095	1500.7	1170	246.7 ^b	211.7
17	Ideal Rankine-cycle thermal efficiency, %	40.9	30.15	39.6	29.9	34.6	31.0	43.1	37.8	28.5 ^b	35.6
18	Vapor in-flow for 50% conversion efficiency, lb/sec [5 × 3413 × 2]/(3600 × 16)°]	0.00984	0.01007	0.0375	0.0383	0.00621	0.00665	0.00631	0.0081	0.0384	0.0266
19	Vapor in-flow for 50% conversion efficiency, ft ³ /sec (18 × 3)	0.0366	0.168	0.0254	0.0939					0.0384	0.0240
20	Speed, rpm	36,000	36,000	36,000	24,000	48,000	48,000	48,000	48,000	14,500	48 000
21	H _{ad} , adiabatic head (total), ft-lb/lb (778 × 12)	307,000	221,000	77,900	57,600	410,000	263,500	505,000	344,000	54,600	58.500
22	First-stage H _{ad}	50,000	55,000	37,900	27,600	130,000	130,000	160,000	170,000	18,200	19 500
23	H _{ad} ^{1/4}	15	15.3	14	12.5	18.5	18.5	19	19.2	11.6	11.8
24	V ₃ , first-stage exit volume flow rate, ft ³ /sec	0.060	0.32	0.18	0.53	0.013	0.04	0.017	0.094	0.14	0.115
25	First-stage √V ₃	0.245	0.566	0.425	0.73	0.106	0.205	0.030	0.307	0.374	0.340

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Table 9.1. (Continued)

Item	Parameter	K		Cs		H ₂ O				Dowtherm A	
26	N _s , first-stage specific speed	2.6	5.5	6	8	0.8	1.25	0.78	1.8	11.4	9.6
27	D _s , first-stage specific diameter	15	9	8	6.3	27	22	28	17	4.2	4.6
28	First-stage diameter, in.	2.94	4.0	2.91	4.41	1.86	2.92	2.32	3.26	1.64	1.60
29	First-stage aerodynamic efficiency, %	47	58	59	62	30	35	29	41	72	71
30	Second-stage H _{ad}	50,000	55,000	40,000	30,000	140,000	133,500	170,000	174,000	18.200	19.500
31	H _{ad} ^{1/4}			14.2	13.2	18.7	18.5	19.2	19.3	11.6	11.8
32	V ₃ , second-stage exit volume flow rate, ft ³ /sec	0.10	0.75	7.00	7.15	0.069	0.645	0.174	7.03	0.70	0.75
33	Second-stage $\sqrt{V_3}$			2.65	2.69	0.263	0.805	0.418	2.65	0.838	0.866
34	N _s , second-stage specific speed	3.4	8.7	34	34.3	1.76	5.6	2.4	16.8	25.2	25.2
35	D _s , second-stage specific diameter	13	6	2	2	18	8.5	5	4	2.4	2.4
36	Second-stage diameter, in.			4.48	4.48	2.53	3.67	3.26	6.59	2.08	2.12
37	Second-stage aerodynamic efficiency, %	52	63	77	77	41	58	46	68	79.5	79.5
38	Third-stage H _{ad}	50,000	55,000			140,000		175,000		18,200	19.500
39	H _{ad} ^{1/4}					18.7		19.3		11.6	11.8
40	V ₃ , third-stage exit volume flow rate, ft ³ /sec	0.25	2.00			0.604		5.46		4.3	5.7
41	Third-stage $\sqrt{V_3}$					0.775		2.34		2.08	2.39
42	N _s , third-stage specific speed	5.3	14			6		13		64	70
43	D _s , third-stage specific diameter	8.5	4.5			8		4.3		1.3	1.2
44	Third-stage specific diameter, in.					3.98		6.25		2.80	2.92
45	Third-stage aerodynamic efficiency, %	58	66			59		67		80.5	80.5
46	Fourth-stage H _{ad}	50,000	56,000								
47	H _{ad} ^{1/4}		15.4								
48	V ₃ , fourth-stage exit volume flow rate, ft ³ /sec	0.60	6.88								
49	Fourth-stage $\sqrt{V_3}$		2.62								
50	N _s , fourth-stage specific speed	8.4	26								
51	D _s , fourth-stage specific diameter	6	2.6								
52	Fourth-stage specific diameter, in.		5.29								
53	Fourth-stage aerodynamic efficiency, %	62.5	74.0								

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Table 9.1. (Continued)

Item	Parameter	K	Cs		H ₂ O				Dowtherm A		
54	Fifth-stage H _{ad}	50,000									
55	H _{ad} ^{1/4}										
56	V ₃ , fifth-stage exit volume flow rate, ft ³ /sec	2.00									
57	Fifth-stage√V ₃										
58	N _s , fifth-stage specific speed	15									
59	D _s , fifth-stage specific diameter	4.6									
60	Fifth-stage specific diameter, in.										
61	Fifth-stage aerodynamic efficiency, %	68									
62	Sixth-stage H _{ad}	57,000									
63	H _{ad} ^{1/4}	15.5									
64	V ₃ , sixth-stage exit volume flow rate, ft ³ /sec	6.88									
65	Sixth-stage√V ₃	2.62									
66	N _s , sixth-stage specific speed	26									
67	D _s , sixth-stage specific diameter	2.6									
68	Sixth-stage specific diameter, in.	5.28									
69	Sixth-stage aerodynamic efficiency, %	74.0									
70	Overall turbine aerodynamic efficiency, %	60.2	65.2	68	69.5	43.3	47	47.3	77.4	77.1	
71	Last-stage spouting velocity, ft/sec [√64.4(H _{ad}) _{last st}]	1800	1880	1610	1390	3010	2940	3320	3350	1082	1120
72	Last-stage vapor viscosity, lb/hr·ft	0.047	0.047	0.054	0.054	0.027	0.027	0.025	0.025	0.0247	0.0247
73	Last-stage nominal blade Reynolds number [(12 × D × (71))/(9 × (72))]	3520	3700	8600	7400	34,700	49,500	11,800	12,200	12,000	8,000
74	Next-to-last-stage nominal blade Reynolds No.	12,000	12,600	335,000	98,000	193,000	635,000	193,000	450,000	53,000	37,000
75	Vapor exit quality, %	79.5	83	77	79	100	95	100	88	100+	100+
76	Efficiency factor for generator, bearing, seal, and feed pump losses and regenerative feed heating	70	70	70	70	70	70	70	70	64.7	64.7
77	Overall thermal efficiency, excluding heat losses, %	14.5	11.9	15.6	11.5	10.5	10.2	14.3	13.1	14.1	17.8
78	Estimated heat losses, kW	8	5	8	5	2.5	1	2.5	1	1.3	1.3

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Table 9.1. (Continued)

		K		Cs		H ₂ O			Dowtherm A	
79	Heat input requirements, kW	42.5	47	40	48.5	48.5	50	35.9	39.2	36.8 30.2

^aCorresponds to a Dowtherm-A condenser temperature of 290°F.

^bIn the Dowtherm-A cycle the superheat in the vapor leaving the turbine makes it highly advantageous to employ a regenerator. As a consequence an 80% effective regenerator was assumed, and the enthalpy rise in the boiler was calculated for the cycle with the regenerator. However, this precluded any advantage from feed heating and hence gave a lower value for the efficiency factor for generator, etc., than for the other cycles.

^cEncircled numbers are item numbers.

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Brayton cycle is given by the relation

$$\text{thermal efficiency} = 1 - \frac{1}{(P_1/P_2)^{(k-1)/k}},$$

where P_1 and P_2 are the turbine inlet and outlet pressures and k is the ratio of the specific heat at constant pressure to that at constant volume. Since all the inert gases are monatomic, they have the same ratio of the specific heats. Consequently, as far as the ideal Brayton-cycle efficiency is concerned, it makes no difference which inert gas is employed. However, the efficiency of small turbines and compressors falls off quite rapidly as the Reynolds number is reduced to below $\sim 100,000$ (Refs. 8,9) and therefore it is desirable to employ a gas that will give as high a Reynolds number as possible for operation between a given set of temperature limits for a given power plant output.

Table 9.2 summarizes the physical properties of typical gases, together with some calculational steps made to obtain nominal Reynolds numbers for a consistent set of assumptions with respect to power plant output and cycle operating conditions. In designing turbines the gas velocity entering the turbine buckets (relative velocity) is normally considered to be about 125% of the turbine wheel tip speed, and may be limited either by sonic velocity considerations or by turbine-bucket stress limitations.

Table 9.2. Effects of Choice of Brayton-Cycle Working Fluid on the Reynolds Number in the Compressor and Turbine and on a Heat Transfer Parameter at 1200°F

	He	Ar	Ne	Xe	Xe + He ^a
Molecular weight	4	39.94	20.183	131.3	80
Specific heat at constant pressure, c_p , Btu lb ⁻¹ (°F) ⁻¹	1.25	0.124	0.246	0.0381	0.0625
Thermal conductivity, k , Btu hr ⁻¹ ft ⁻² (°F/ft) ⁻¹	0.194	0.0234	0.0585	0.008	0.032
Viscosity, lb hr ⁻¹ ft ⁻¹	0.109	0.1225	0.1585	0.1075	0.1323
$\frac{0.023 (c_p)^{0.4} k^{0.6}}{\mu^{0.4}}$	5.89	0.629	1.28	0.215	0.556
$c_p - c_v$, Btu lb ⁻¹ (°F) ⁻¹	0.497	0.0498	0.0986	0.01513	0.0247
c_v , Btu lb ⁻¹ (°F) ⁻¹	0.753	0.0746	0.147	0.0230	0.0376
Relative Reynolds No. for same power, pressure, and diameter, $1/c_p \mu$	7.34	65.9	25.65	244	120
Nominal Reynolds No. for the same power, limiting velocity, temperature, and pressure in a 5-kW unit	8,000	65,000	27,800	181,400	101,800
Exit specific volume at 10 psia, ft ³ /lb	445	44.6	88.2	13.6	22.6
Velocity of sound at exit, fps	5260	1665	2340	920	1180
Stress-limited tip speed, fps	1500	1500	1500	1500	1500

^a59.6 + 40.4 mole %.

For high-molecular-weight fluids it was assumed that the relative velocity between the fluid and the wheel should not exceed sonic velocity, and hence the sonic velocity was employed in computing a nominal Reynolds number. For the lower molecular weight working fluids (He and Ne), experience with turbine-bucket stress problems indicates that the corresponding gas velocity relative to the turbine buckets should not exceed 2000 fps. (This gives a turbine-bucket tip speed of about 1500 fps.)

In comparing two gases for use in a machine of a given power output

$$W_1 c_{p1} \simeq \text{power} \simeq W_2 c_{p2} .$$

Hence

$$W_1/W_2 = c_{p2}/c_{p1} ,$$

where W is gas weight flow rate, lb/sec, and c_p is gas specific heat, Btu lb⁻¹ (°F)⁻¹. Thus for a given power and rotor size the Reynolds numbers for the different inert gases can be related by

$$\frac{Re_1}{Re_2} = \frac{G_1 D_1 \mu_2}{G_2 D_2 \mu_1} = \frac{c_{p2} \mu_2}{c_{p1} \mu_1} ,$$

where G is the mass velocity and μ is the gas viscosity. For a given limiting relative velocity in the rotor, the weight flow rates for two inert-gas Brayton cycles can be related to the axial flow passage area, A , the rotor diameter, D , the relative velocity in the rotor, V , and the specific volume, v , as follows:

$$\frac{W_1}{W_2} = \frac{v_2 V_1 A_1}{v_1 V_2 A_2} = \frac{c_{p2}}{c_{p1}} ,$$

$$\frac{A_1}{A_2} = \frac{c_{p2} v_1 V_2}{c_{p1} v_2 V_1} = \left(\frac{D_1}{D_2} \right) .$$

For the inert gases the product of c_p/v is constant; therefore

$$\left(\frac{D_1}{D_2} \right)^2 = \frac{V_2}{V_1} .$$

Thus for the same power, limiting gas velocity, and pressure the Reynolds numbers for two inert gases can be related by

$$\frac{Re_1}{Re_2} = \frac{G_1 D_1 \mu_2}{G_2 D_2 \mu_1} = \frac{v_2 \mu_2}{v_1 \mu_1} \left(\frac{V_1}{V_2} \right)^{1/2} .$$

The relative Reynolds numbers obtained for the above two cases are given in Table 9.2.

Based on the relations and values obtained in this way and a turbine-bucket chord of 0.25 in., the nominal Reynolds number was found to increase from 8000 for helium to 65,000 for argon and 181,000 for xenon. Krypton with a molecular weight of about 80 would give a Reynolds number of around 100,000, which would be good from the Reynolds number standpoint.

Another important factor is the required heat-transfer performance of the working fluid in the heat exchangers. Not only do the regenerator and cooler tend to become large and bulky but also the pressure losses through them tend to become large unless the heat-transfer coefficients for the working fluid are reasonably good. A good basis for comparing working fluids is provided by the familiar Dittus-Boelter relation:

$$h = 0.023 \frac{c_p^{0.4} k^{0.6}}{\mu^{0.4}} \frac{G^{0.8}}{D^{0.2}} .$$

The first portion of this relation is purely a function of the fluid physical properties and therefore can be computed independently of the mass flow rate and passage diameter. This has been done, and the resulting parameter is tabulated in item 5 of Table 9.2. The heat-transfer performance parameter falls off with increasing atomic weight by a factor of around 25 in going from helium to xenon. The value for the heat-transfer parameter for a mixture of helium and xenon is nearly three times that obtained for xenon alone, and yet the nominal Reynolds number in the turbine is reduced relatively little. The last column in Table 9.2 presents data for a mixture of xenon and helium which has been found to give a nearly optimum combination of characteristics. This working fluid has been chosen by NASA for use in their space power plant Brayton-cycle systems, and appears to be well suited for use as the reference design fluid in this study.

In reviewing previous studies on Brayton cycles, data were taken from two studies of several years ago^{8,10} and are given in Table 9.3. The first set is for an AiResearch study of 1965 and the other is for an ORNL study of 1962. The problem was discussed with J. McCormack and G. Sheppard of AiResearch, who stated that AiResearch had recently run a 3-kW(e) argon system with a 60,000-rpm turbine generator (2-pole) unit running on gas bearings and had obtained an overall thermal efficiency of 20%, not including the heat losses from the electrically heated heat source.¹¹ Further, they have completed construction and testing of a 36,000-rpm compressor turbine 4-pole generator unit running on gas bearings and the heat exchangers for a more elegantly designed 2.25-kW(e) to 10.5-kW(e) system that is being assembled at the Lewis Research Center. This system uses a mixture of helium and xenon having a molecular weight of 83, and is expected to give an overall thermal efficiency of about 25%. In view of these data, the information shown in column 3 of Table 9.3 was obtained to represent the reference design system for this report. No attempt was made to justify the detailed efficiency of the individual components in view of the AiResearch and NASA work that validates the choice of 25% for the overall thermal efficiency.

Table 9.3. Performance Parameters for Some Typical Brayton Cycles

	AiResearch	ORNL	Reference Design
Working fluid	Xe + He	Argon	Xe + He
Molecular weight of working fluid	60	40	80
Turbine inlet temperature, °F	1540	1400	1540
Compressor inlet temperature, °F	65	60	70
Gas system pressure drop/peak pressure	0.12	0.10	0.12
Compressor efficiency, %	81.3	78	
Turbine efficiency, %	87.8	84	
Generator efficiency, %	92.5	90	
Regenerator effectiveness, %	92	90	
Design power output, kW(e)	7.25	10	5
System thermal efficiency (no heat losses)	30.2	24.9	25
Estimated heat losses, kW(th)	1.0	4	4
Total thermal input required, kW	25	44.1	24

There is little incentive to increase the peak cycle temperature over the 1540°F used for the reference design system of Table 9.3 as long as it is easy to provide a heat dump that will hold the compressor inlet temperature to around 70°F. (The situation is quite different for space power plants, for which it is difficult indeed to dissipate the waste heat at a low temperature.) The single-stage centrifugal compressor and radial turbine units employed by AiResearch are operating at close to the optimum pressure ratio for that type of machine, and increasing the peak temperature would not increase the cycle efficiency very much.

AERODYNAMIC CALCULATIONS

The aerodynamic design of turbines is exceedingly complex, because it is influenced not only by such fluid properties as fluid density, viscosity, thermodynamic properties, and design conditions, including the inlet and outlet temperatures and pressures, but also by the 18 major geometric parameters listed below:

- Speed
- Rotor diameter
- Blade height
- Blade inlet angle
- Blade exit angle
- Blade chord
- Blade pitch
- Blade profile (airfoil shape)

Blade twist
 Tip clearance
 Sonic velocity
 Inlet nozzle angle
 Number of stages
 Head per stage
 Head distribution (between stages)
 Diameter distribution
 Degree of reaction
 Fraction admission

A number of attempts have been made to group these parameters so as to facilitate turbine design analysis work. The best approach is one developed by Balje,^{9,12} who succeeded in reducing the 18 geometric parameters to 7 dimensionless groups that can be plotted for a series of 7 major cases. Figures 9.1-9.3 cover the range of conditions of prime interest for this particular study. Balje drew these charts from test data obtained with representative turbines, and included aerodynamic losses in the blades, tip leakage losses, disk friction losses, losses from three-dimensional flow effects, and kinetic energy losses at the turbine exit. In working with Balje's charts we found that a somewhat higher efficiency could be obtained from the 1540°F cesium turbine by reducing its rpm to 24,000. When we did this, the turbine efficiency was increased by a few points. It is possible, in fact probable, that further refinements in the aerodynamic design of turbines will make possible somewhat higher efficiencies than those indicated by Balje's charts. However, studies at ORNL, at NASA Lewis Research Center, and at Wright Patterson Field indicate that, for comparative studies of the sort covered in this report, any such improvements will be equally applicable to any working fluid and that the relative performance should be unaffected in comparisons of working fluids.²

A review of parametric studies^{1,2} indicated that the number of stages should be kept to a minimum consistent with compressibility loss considerations. This, in turn, meant the use of a high turbine speed in a small-diameter unit. A review of both generator design considerations and operating experience with small high-speed generators that have been built indicated that a speed of 48,000 rpm might be used for the steam system and that because of turbine-bucket stress or sonic velocity considerations the speed ought to be reduced to 36,000 rpm or less for the potassium-, cesium-, and Brayton-cycle systems. The maximum adiabatic head per stage can be increased as the molecular weight of the working fluid is reduced. The data reviewed indicate that the adiabatic head per stage ought to be kept to around 150,000 ft-lb/lb for steam, 60,000 for potassium, and 40,000 for cesium.^{1,2}

As mentioned previously, the efficiency of turbines and compressors falls off slowly at first and then more rapidly as the Reynolds number is reduced to below about 100,000.^{13,14} The decrease becomes quite rapid if the Reynolds number is reduced to below about 10,000. Balje's charts^{9,12} were drawn on the premise that the Reynolds number in the turbine blades would be kept above 100,000. This Reynolds number is normally computed using the relative velocity between the gas and blade, either the blade chord or

Fig. 9.1. $N_s D_s$ Diagram for Single-Disk Turbines Showing Total Efficiency. [From: O. E. Baljé, "A Study on Design Criteria and Matching of Turbomachines: Part A — Similarity Relations and Design Criteria of Turbines," J. Eng. Power A84, 83-102 (January 1962).]

Fig. 9.2. $N_s D_s$ Diagram for Single-Stage, Full-Admission, Axial-Impulse Turbines. [From: O. E. Baljé, "A Study on Design Criteria and Matching of Turbomachines: Part A — Similarity Relations and Design Criteria of Turbines," J. Eng. Power A84, 83-102 (January 1962).]

Fig. 9.3. $N_s D_s$ Diagram for Single-Stage, Partial-Admission, Axial Turbines with $a^*/D = 0.01$. [From: O. E. Baljé, "A Study on Design Criteria and Matching of Turbomachines: Part A — Similarity Relations and Design Criteria of Turbines," J. Eng. Power A84, 83-102 (January 1962).]

the blade spacing, and the local gas density and viscosity. Nominal Reynolds numbers were computed on this basis (items 70-73 of Table 9.1) based on a blade chord of 0.25 in. and the relation

$$Re = \rho \frac{VD}{\mu},$$

where

ρ = gas density leaving the blade, lb/ft³,
 V = gas velocity entering blade (relative), fps,
 D = blade chord, ft,
 μ = gas viscosity, lb sec⁻¹ ft⁻¹ .

The relative velocity entering the blade was taken as 60% of the spouting velocity (item 71), which was calculated from the adiabatic head for the stage. Factors for converting inches to feet and hours to seconds were also included to give the factor 86.5 used in computing item 73.

The Reynolds numbers were computed for both the last stage, n , and the next-to-last stage, $n-1$. In reviewing these, it is evident that the steam turbines operating with a condenser temperature of 150°F and the Dowtherm-A cycle yield Reynolds numbers in the last stage sufficiently high that the overall turbine aerodynamic efficiency need not be corrected. However, the low-pressure exhaust steam cycles with the 70°F condenser temperature and the two cesium cycles both yield Reynolds numbers of around 14,000 in the last stage. The Reynolds numbers in the next-to-last stage, $n-1$, of the steam cycles run close to 10⁶, whereas they run between 185,000 and 454,000 for the two cesium cycles. Thus the aerodynamic efficiency of the turbines for these four cycles would probably be reduced somewhat — perhaps 5 or 10%. The aerodynamic efficiency of the potassium cycles would be even more severely affected since the last-stage Reynolds numbers run about 5,000 and the Reynolds numbers of the next-to-last stages run around 17,000 in both cases. Here the reduction in turbine aerodynamic efficiency would probably run around 10% from the nominal average value given in Table 9.1. However, there appears to be no generally accepted basis for applying such corrections and therefore no attempt was made to apply them here. It must be emphasized, however, that for Reynolds numbers below 100,000 some correction would be necessary, and, if a Reynolds number runs below 10,000, this correction would be quite large.

Balje's charts include a fixed allowance for gas leakage losses through the tip clearance between the rotor and the casing. These allowances are probably adequate for the steam and Dowtherm-A turbines, but the clearances required to allow for differential thermal expansion at the much higher temperatures of the cesium and potassium vapor cycles would probably entail greater losses than those implicit in Balje's charts, and this would probably reduce the aerodynamic efficiency by from three to six points from the values given in Table 9.1.

OVERALL CYCLE EFFICIENCY

A number of factors tend to degrade the efficiency of an actual power plant with respect to the value obtained by multiplying the efficiency of the ideal thermodynamic cycle by the aerodynamic efficiency of the turbine. These include the moisture churning losses in wet vapor turbines, the seal leakage losses both between stages and between the first stage and the adjacent low-pressure region around the bearing, the power required to drive the feed pump, the electrical losses in the generator, and the bearing and seal frictional losses. On the other hand, some improvement in cycle efficiency over the simple ideal cycle can be obtained through the use of regenerative feed heating. Time did not permit the detailed evaluation of each of these factors in the present study, but the conditions are sufficiently close to those of the detailed studies of Refs. 1 and 2 for it to seem reasonable to apply a single factor of 0.7 (derived from Refs. 1 and 2) to take care of all the factors except the moisture churning loss. Again, by taking advantage of the earlier work, it appears that a good approximation to the moisture churning loss can be obtained by taking it as equal to three-quarters of the moisture content in the turbine at the exit from the last stage if the vapor entering the first stage is saturated. The moisture content at the exit from the last stage should not be taken for the ideal cycle but rather for the enthalpy condition indicated by applying the aerodynamic efficiency to the ideal enthalpy drop for an isentropic expansion across the turbine. Thus, if the heat losses to the surroundings are neglected, the overall power plant efficiency can be determined by multiplying the efficiency of the ideal cycle by the turbine aerodynamic efficiency, a factor of 0.7 to allow for generator, bearing, seal, and feed pump losses, and a factor equal to $1 - \frac{3}{4}$ of the actual turbine vapor exit moisture content. Results of these calculations are included at the end of Table 9.1.

Heat losses to the surroundings through thermal insulation are ordinarily substantial, particularly for 1000°F and higher. Not only does the driving force for such heat losses increase with an increasing turbine inlet temperature, but also the thermal conductivity of thermal insulation tends to increase with increasing temperature. Data from test rig operating experience are plotted in Fig. 9.4 to show the magnitude of the losses to be expected through 3 in. of conventional high-temperature thermal insulation as a function of the temperature of the hot components. Preliminary estimates indicate that the surface area of the high-temperature equipment in a very compact isotope power unit would be at least 2 ft². Based on this value and the data in Fig. 9.4, the heat loss for each cycle listed in Table 9.1 was estimated. These values, coupled with the overall thermal efficiency of the plant as estimated immediately above, give the total heat input required for each cycle considered, and are given as the last item of Table 9.1.

The magnitude of the heat losses through conventional thermal insulation is sufficiently great to merit investigation of the very effective (and very expensive) new reflective insulations recently developed.^{15,16} These reflectors would have to be operated in a vacuum, but would probably make it possible to reduce the heat losses to around 10% of the values given as item 78 of Table 9.1. This would help the cesium and potassium vapor cycles substantially.

Fig. 9.4. Effects of Temperature on Heat Losses from Hot Surfaces Lagged with 3 in. of Thermal Insulation.

RESULTS OF PERFORMANCE ESTIMATES

A number of points emerge from the study. Perhaps the most remarkable is that, in spite of the high turbine inlet temperatures of the potassium and cesium cycles, the overall power plant thermal efficiency differs little from that of the steam cycles. The reason for this is that the volumetric expansion ratio that it is practicable to obtain in a turbine is limited to about 1000. Attempts to go beyond this range lead to major complications which while practical in a central station would not be practical in a small isotope power plant. Some further insight into this is given by the curves in Fig. 9.5 for vapor pressure as a function of temperature. As can be seen, the vapor pressure of potassium and cesium drops to only 0.1 mm Hg, or 100 μ m Hg, at about 500°F. At a factor of 10 above this pressure the vapor is so tenuous that the turbine wheel size becomes excessive, and the rotor weight and bearing and seal frictional losses would roughly equal the power input to the turbine wheel. Also, a further drop to 400°F gets one into the region where the mean free path of a molecule becomes so large relative to the blade spacing¹⁷ that conventional aerodynamic flow relations no longer apply. As a consequence, the practicable Carnot-cycle temperature ratio for a potassium or cesium cycle is little different from that for a steam cycle.

The effect of an excessive pressure ratio on turbine aerodynamic efficiency is shown quite strikingly in Table 9.1 (item 79) by the relatively small difference in thermal efficiency between the 1050°F and 493°F steam cycles. The reason becomes apparent if one notes the aerodynamic efficiency of the individual stages; these tend to be low at the turbine inlet, where the specific volume is small and the proportions of the first stage can be kept satisfactory only through the use of partial admission. As a matter of fact, in these small units maximum efficiency is obtained by using partial admission not only in the first stage but also in most of the stages.

Balje's charts indicate that a partial-admission turbine should be employed if the turbine specific speed has a value less than about 18, whereas full admission will give a higher efficiency for specific speeds above about 18. Also, for full-admission turbines, Fig. 9.2 can be used for intermediate stages where the kinetic energy in the exhaust can be used to good advantage in the subsequent stage, whereas Fig. 9.3 must be used for the last stage because the kinetic energy in the exhaust cannot be recovered.

The Rankine-cycle studies were based on axial flow turbines, and the Brayton-cycle work was based on radial flow turbines. Radial flow units give a little better efficiency in small machines if only a single stage is required but do not lend themselves well to the design of the multistage machines considered for the Rankine cycles of this study.

Comparison of the cycle efficiencies for all the various cycles studied indicates that the Brayton cycle is approximately 1.5 to 2.0 times as efficient as any of the Rankine cycles, because a Brayton-cycle system can be operated over a wide temperature ratio without the serious penalties associated with Reynolds-number effects in the turbine.

Fig. 9.5. Vapor Pressure as a Function of Temperature for Typical Rankine-Cycle Working Fluid.

CONCLUSIONS

In comparing the Rankine cycles it is apparent that fundamental thermodynamic and aerodynamic limitations on turbine design for potassium or cesium make it impractical to expand the vapor to a pressure below about 0.5 psia; consequently the high-peak cycle temperature obtainable with potassium or cesium does not lead to any real advantage in cycle efficiency over a steam cycle at a modest temperature. In fact, the results of this study indicate that where the condenser temperature can be kept below about 100°F, — for practically any terrestrial application except in a hot desert area — a steam cycle will actually give a higher overall thermal efficiency than a 2000°F cesium or potassium vapor cycle. Thus a steam cycle appears to be preferable, first because the equipment should be much less expensive to develop and second because the heat source can be designed as a recirculating boiler which could be operated submerged in a body of water to give a high degree of assurance that the isotopic heat source would always be adequately cooled so that there would never be any dispersion of radioactive material. There seems to be no way to get anything like the same degree of system simplicity and reliability,¹⁸ and hence the same degree of assurance of maintaining the integrity of the isotopic heat source for any of the high-temperature cycles, that is, the cesium, potassium, or Brayton cycles.

The overall thermal efficiency obtainable from the Dowtherm-A cycle is not much better than that obtainable with steam except for desert applications, where heat would have to be dumped to air at a temperature well over 100°F. Dowtherm A would have the advantage that for a hermetically sealed system troubles with the electrical insulation in the generator would probably be less difficult to avoid. This advantage may be offset by the difficulties associated with radiation damage to the Dowtherm in a single fluid system with a lightly shielded isotope heat source serving as the boiler. The complexities associated with a multiloop system and the associated degradation in system reliability¹⁸ would entail even more serious penalties.

An interesting check on the preceding analytical work is provided by Fig. 9.6, which consists of a set of curves for thermal efficiency as a function of peak cycle temperature for a series of heat sink temperatures for a Carnot cycle. Plotted on the same coordinates are points for a wide variety of actual heat engines operating on different thermodynamic cycles. In general, increasing the peak cycle temperature leads to an increase in thermal efficiency, but the deviation between the efficiency of the ideal Carnot cycle and that of the actual cycle increases as the peak cycle temperature is increased. The increase in this deviation results from the increasing importance of losses at the higher temperatures and the greater difficulty in making effective use of these higher temperatures.

The data plotted in Fig. 9.6 were for power plants having outputs generally much higher than that of the 1- to 10-kW(e) isotope power plant of this study. In view of the almost complete lack of good experimental data on overall thermal efficiency for small power plants suitable for use with an isotope heat source, it was necessary to make use of estimated values

in preparing Fig. 9.7. Generally, the resulting scatter band of points is similar to that in Fig. 9.6 and the cycles with the higher peak temperatures tend to give less of an increase in thermal efficiency than might be expected from the ideal Carnot cycle.

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Fig. 9.6. Comparison of Thermal Efficiencies of Large Power Plants with Ideal Efficiencies.

Fig. 9.7. Comparison of Thermal Efficiencies of Small Power Plants with Ideal Efficiencies.

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10. CONCEPTUAL DESIGN OF COMPLETE POWER UNITS

Many different requirements must be met in the design of complete isotope power units for marine and terrestrial service, and many approaches have been considered in an effort to devise systems that would satisfy all the requirements. The advantages and disadvantages of each approach are reflected in the conceptual designs that were prepared in an effort to satisfy all the difficult boundary conditions. Four power conversion systems were chosen for study: lead telluride thermoelectric junctions, a steam Rankine cycle, an organic Rankine cycle, and a Brayton cycle. A number of conceptual designs have been prepared for each system, and the more promising are presented in this section.

In developing the conceptual designs, a uniform set of design precepts, listed below, was established:

1. The isotope heat source should be shielded for shipping to meet ICC standards.
2. The shielded isotope heat source should be surrounded by a shipping cage that would permit free circulation of air for removal of the decay heat but would prevent accidental contact with the hot surfaces of the shield.
3. The shield and heat block should be designed so that under normal conditions the temperature of the Hastelloy C-276 cans encasing the isotope would not exceed 1800°F, and no foreseeable set of conditions will cause the temperature of the Hastelloy cans to exceed 2000°F.
4. The shield and heat block assembly should be able to withstand a 10-m drop onto a heavy concrete pier without adversely affecting the integrity of the shield as a radiation shield or the positive containment of the isotopic fuel.
5. To prevent the possibility of the isotope capsule jamming when it is inserted into the heat block, the diametrical clearance between the capsule and the heat block should be at least 0.020 in.
6. For purposes of this study the diameter of the isotope capsules was taken as 2.94 in. for ^{90}Sr fuel and 0.94 in. for ^{60}Co fuel.
7. Partly to avoid difficulties with capsule bowing as a consequence of a nonuniform temperature distribution under transient conditions and partly because of hot cell limitations, the maximum length of individual isotope capsules should be held to 12 in. If a greater length is required, two or more capsules should be employed end to end.
8. Heat transfer from the isotope capsule to the heat block should be taken as that given by thermal conduction through the gas filling the gap plus that by thermal radiation. (Filling the gap with a liquid metal such as sodium would give an excellent thermal bond but in case of leaks could lead to a problem of conduction.)

9. With the exception of the direct-conduction thermoelectric units, for undersea applications the isotope heat source should be surrounded by a layer of thermal insulation, which in turn would be protected from the surrounding high-pressure water by a suitable pressure vessel.

10. Because of their complexity the direct-conduction thermoelectric units should be assembled in a well-staffed and equipped shop.

11. The Rankine- and Brayton-cycle systems should be assembled with the heat source at dockside and installed in the pressure vessel for submergence in the sea. The installation should be designed to facilitate startup and thorough checkout of the unit before the outer pressure vessel is sealed.

12. In transferring the heat from the heat source to the working fluid of the thermodynamic cycle, not more than 10% of the heat should be wasted through the thermal insulation under normal full-power conditions, and the temperature of the isotope capsule should be sufficiently below 1800°F that it will not exceed 2000°F during an emergency such as might be induced by a cessation in the flow of the thermodynamic-cycle working fluid.

13. Provision should be made for heat removal from the isotope in the event that the thermodynamic-cycle working fluid stops circulating because of loss of fluid, a failure in the pump, or the like.

14. The heat from the isotope capsule should be transmitted to the thermodynamic working fluid in such a way that neither the integrity nor the effectiveness of the shield is adversely affected by a failure of the system.

15. The emergency heat-removal system should be able to function satisfactorily with the unit in any attitude, including on its side or inverted.

16. To obtain the highest possible reliability, the system should be designed so that the number of components is minimized and a very simple and reliable set of instrumentation and control equipment will suffice for startup, routine operations, and shutdown.

17. The maximum temperature in the heat block should be kept within acceptable limits; this is the most important function of the control system. (Regulation of the output of the turbine-generator is, of course, also important.)

18. The output of the generator either should be 28-V dc current or be converted to 28-V dc current so that it can be used in conjunction with a 24-V set of storage batteries that will handle heavy power demands for short periods, as well as provide emergency power.

The principal objective in developing each conceptual design was to meet the design precepts outlined above. Performance specifications and layout

drawings for existing thermoelectric and turbine-generator units were employed as a basis for establishing the size of the heat source required, the space requirements for the power conversion units, and the heat-transfer capacity of the heat exchanger equipment required. The connecting piping was arranged to provide good drainage with a minimum amount of heat losses and acceptable pressure drops. Consideration was given to the problems of assembly and startup, but no attempt was made to work out a completely detailed set of procedures for each concept.

In view of the high cost and long periods of time that have been required in the past to develop rotating machinery, the design studies were based on existing machines or relatively small modifications of existing machines which are well along in the development process. ORNL experience with heat exchanger design and development, however, indicates that new heat exchangers tailored to a particular application can be designed, built, and tested to yield satisfactory units at relatively little cost; hence no attempt was made to employ existing heat exchanger equipment.

THERMOELECTRIC POWER SYSTEMS

Two types of thermoelectric terrestrial radioisotope power systems were examined for the production of electrical power in the range of 2 to 10 kW(e): a system in which the thermoelectric converter is located in direct contact with the radiation shield and the shield material is used to transport the decay heat and a system in which the converters are remote from the radiation shield and an intermediate heat-transfer medium is used to transport the decay heat from the isotope source to the converters. The analyses were made primarily from the standpoint of the overall system. All individual component assemblies are of a preliminary design status and were pursued only to the point of determining feasibility.

Both systems were assumed to have either water or air cooling, depending on whether they were to be located under water, in air, or on land. The basic design problems are essentially the same, that is, transport of the heat from the source to the thermoelectric converter hot frame and transport of the waste heat from the thermoelectric converter cold frame to the ambient fluid. With the conversion efficiency of the thermoelectric converters approximately 8%, the thermal power required to produce 2 to 10 kW(e) would be 25 to 125 kW(th). For this study a 30-kW(th) source with an 8% conversion efficiency was used. The designs proposed to satisfy operating conditions are discussed below.

Direct-Conduction Thermoelectric System

The direct-conduction system imposes difficult requirements on the design of the radiation shield: the shield must transport the heat radially from the fuel to the thermoelectric converter hot frame, and the shield temperature adjacent to the hot frame must be maintained at approximately 1200°F in order to have a hot-junction temperature of 1050-1075°F. It is

difficult to meet the second requirement because the maximum allowable normal operating temperature of the Hastelloy C-276 fuel cladding is 1800°F. These temperature restraints coupled with the radiation shielding requirements drastically limit the choice of shield material for a particular fuel and fuel geometry. Of the shield materials investigated, tungsten appears to be the only one with a sufficiently high thermal conductivity to meet the temperature requirements of both the fuel and the thermoelectric converter for the power densities required.

A design for a 30-kW(th) direct-conduction thermoelectric power system is shown in Fig. 10.1. Dimensional and performance data for this design are given in Table 10.1.

Fig. 10. 1. Direct-Conduction Thermoelectric Power System.

Table 10.1. Design Characteristics of a 2-kW(e)
Thermoelectric Terrestrial Radioisotope Power Plant (BOL)

<u>Power Output</u>	
Net electric, kW	2.27
Isotope heat, kW(th)	30
<u>Fuel Capsules</u>	
Fuel material	⁹⁰ SrO
Fuel cladding material	Hastelloy C-276
Fuel outside diameter, in.	2.50
Cladding outside diameter, in.	2.94
Capsule length, in.	11.27
Number of capsules in-line	2
Number of capsules in plane of array	13
Total number of capsules	26
Heat output per capsule, kW(th)	1.154
Maximum fuel temperature, °F	2280
Maximum cladding temperature, °F	1640
<u>Heat Block Shield</u>	
Shield material	Tungsten
Capsule hole diameter, in.	3.0
Capsule hole center-line spacing, in.	4.5
Capsule hole pattern envelope diameter, in.	18.59
Fuel length, in.	22.53
Shield outside diameter, in.	27
Shield overall height, in.	30.8
Assembly weight, lb	10,600
Maximum heat block temperature, °F	1490
Temperature of heat block at outer surface of capsule array, °F	1270
<u>Freon-114 Boiler</u>	
Temperature, °F	160
Pressure, psia	113
Heat flux, W/in. ²	6.8
Temperature drop, boiling film, °F	8
Number of tubes	90
Tube inside diameter, in.	0.657
Tube minimum wall thickness, in.	0.109
Effective length, in.	23
<u>Freon-114 Condenser</u>	
Temperature, °F	160
Pressure, psia	113
Heat flux, inside, W/in. ²	3
Temperature drop, condensing film, °F	13
Number of tubes	80
Tube inside diameter, in.	1.732
Tube outside diameter, in.	2.0
Number of fins	36
Fin height, in.	1
Fin thickness, in.	0.024
Effective length, in.	37
<u>Air Temperature, °F</u>	100

The waste-heat disposal system was designed to operate with either water or air cooling. The system has a cold frame that is cooled by boiling Freon for operating in air, and by direct conduction to the pressure vessel for operation in water. When the system is used in water, the Freon system provides cooling only during shipment to the site. After water cooling has been applied at the operating site, the Freon cooler can be removed.

The design shown in Fig. 10.1 is applicable to either air or water cooling. If the system were designed for only one application, the design could be simplified to meet the more specific requirements. For example, an air-cooled design would not require a pressure vessel. A water-cooled design would not require an air-cooling system.

Indirect-Conduction Thermoelectric System

The indirect-conduction system must meet the same 1800°F temperature limitation on the fuel cladding as the direct-conduction system, but, since the heat from the fuel is not conducted radially to the thermoelectric converter, the selection of the shield material is much easier than with the direct-conduction system.

A 30-kW(th) indirect-conduction thermoelectric power system is shown in Figs. 10.2 and 10.3. The system uses ^{90}SrO for fuel, and nickel for the shield material. The design data for the shield are given below:

Shield diameter, in.	37.1
Shield length, in.	42.9
Fuel length, in.	24.52
Number of fuel elements	12
Fuel spacing, in.	4.5
Total weight, lb	14,400
Maximum shield temperature, °F	1550
Maximum fuel cladding temperature, °F	1685

Boiling potassium was chosen as the working fluid for transporting heat from the shield block to the thermoelectric junctions because of its chemical stability, moderate vapor pressure, and high latent heat of vaporization at 1200°F. A potassium heat pipe was first considered as the means for removing the heat from the heat block, but because of its low heat-transfer capacity was abandoned in favor of a natural-circulation boiler. Analysis showed that capillary forces contributed only about 10% to the reflux flow rate in a vertical heat pipe. This small advantage was more than offset by the extra complications and design limitations of the heat pipe. For space power plant applications, capillarity is the only force available, and the capillary heat pipe is attractive. However, for terrestrial plants having the gravitational force component available, the surface tension contribution to the total driving force is small and the difficulties in obtaining it are large. Consequently, a natural-circulation boiler design was adopted. Over 100,000 hr of experience here was obtained during the space power plant program. This provided a good basis for the design work.

Fig. 10.2. Indirect-Conduction Thermoelectric Power System.

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Fig. 10.3. Cross Sections of Indirect-Conduction Thermoelectric Power System.

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The potassium boiler consists of 13 parallel tubes, 12 of which have an outside diameter of 1.5 in. and are located near the outer surface of the shield block and 1 which has an outside diameter of 3 in. and is located in the center of the shield block. These locations were chosen to decrease the temperature gradients in the shield block during normal operation. The vapor discharge ends of the boiler tubes are manifolded so that a single vapor line feeds the potassium condenser. The wall of the condenser supplies the heat for the hot junctions of the thermoelectrics. The liquid potassium from the condenser is collected at the bottom of the condenser and is returned by gravity through the liquid return manifold to the inlets of the boiler tubes.

Boiling Freon was chosen as the heat collector from the cold junctions of the thermoelectrics because it has a suitable vapor pressure and reasonable boiling and condensing heat-transfer coefficients. The Freon boiler consists of thirty 0.723-in. holes drilled into a 1-in.-thick cylindrical shell that receives the rejected heat from the thermoelectrics and transfers it to the Freon. The vapor discharge ends of the boiler tubes are manifolded into a ring header which serves as a moisture separator. The liquid Freon vapor is transferred to the condenser inlet header. The condenser consists of sixty 2-in.-dia copper tubes with thirty-six 1-in.-high copper fins and is cooled by natural circulation of the ambient atmosphere. The condensed Freon is collected at the bottom of the condenser and is returned by gravity through the liquid return manifold to the boiler inlet header.

The design shown in Figs. 10.2 and 10.3 requires a jacket of thermal insulation around the shield block to reduce heat losses to make it suitable for operation in air. An undersea unit would require the addition of a pressure vessel similar to that shown in Fig. 10.1 around the shield block and thermoelectric cold frame.

Freon Boiler-Condenser Heat Exchanger

A parametric study was made of the performance of a Freon boiler-condenser heat exchanger system to transport the waste heat from the thermoelectric converter to the atmosphere. The parameters used were the cold-frame temperature, the ambient air temperature, and the cold-frame heat flux. The geometry of the heat exchanger was held constant. The ratio of the area of the Freon condenser to the area of the Freon boiler was determined as a function of the cold-frame heat flux and the difference between the temperature of the cold frame and the ambient air with Freon-114 used as the heat exchanger fluid. The material of construction of both the boiler and the condenser was copper.

The cold-frame heat flux was varied from 5 to 40 W/in.², and the difference between the temperature of the cold frame and the ambient air was varied from 0 to 200°F. The values of parameters held constant in the study were as follows:

Tube outside diameter, in.	2.0
Tube inside diameter, in.	1.732
Fin height, in.	1.0
Fin thickness, in.	0.024
Number of longitudinal fins per tube	36

The Freon boiler consists of a 1-in.-thick copper ring in contact with the converter cold frame and with one 0.657-in. hole for every linear inch of cold-frame surface. The Freon boils in the 0.657-in. holes, condenses in the finned tube condenser, and returns to the boiler by gravity. Performance curves for this heat exchanger system are shown in Fig. 10.4. This type of heat-removal system was chosen for the reference design shown in Fig. 10.1.

Fig. 10.4. Freon Boiler-Condenser Performance Characteristics.

Finned Heat Exchanger Parametric Study

The use of a finned heat exchanger to transport the waste heat from the thermoelectric converter to the atmosphere was investigated. The parameters were the dimensions of the exchanger, with the electric output of the converter, the fin root spacing, and the fin heat flux held constant; the values of these constants are shown below:

Converter output, kW(e)	2.27
Fin root spacing, in.	0.1848
Fin thickness, in.	0.070
Fin heat flux, Btu hr ⁻¹ ft ⁻²	50

The fin length and weight were determined as a function of the fin root diameter and the fin tip diameter for two fin materials, copper and aluminum. The fin root diameter was varied from 8.5 to 42.5 in., and the fin tip diameter was varied from 30 to 60 in. The hot face of the thermoelectric converter was assumed to be located concentrically with the fin root at a radial distance of 2.75 in.

The heat-rejection capacity of the finned heat exchanger is given by

$$0.01389 \text{ NHL}\eta_f q = Q_R \quad , \quad (1)$$

where

- N = number of fins,
- H = fin height, in.,
- L = fin length, in.,
- η_f = fin efficiency,
- q = average fin heat flux, Btu hr⁻¹ ft⁻²,
- Q_R = heat rejected by exchanger, Btu/hr.

The heat supplied to the finned heat exchanger is given by

$$3413E \left(\frac{1}{\epsilon} - 1 \right) = Q_s \quad , \quad (2)$$

where E = thermoelectric converter output, kW; ϵ = thermoelectric converter efficiency; and Q_s = heat supplied to exchanger, Btu/hr.

Equating Q_R to Q_s and substituting the constant parameter values gives

$$8.9676 \times 10^{-5} \text{ NHL}\eta_f \epsilon + \epsilon = 1 \quad . \quad (3)$$

The number of fins, the fin height, and the fin efficiency are given by

$$N = \frac{\pi D_R}{0.1848} \quad , \quad (4)$$

$$H = \frac{D_T - D_R}{2} , \quad (5)$$

$$\eta_f = \frac{\tanh mh}{mh} , \quad (6)$$

where

D_R = fin root diameter, in.,

D_T = fin tip diameter, in.

$m = (2h/kt)^{1/2} \text{ (1/12) in.}^{-1}$,

k = thermal conductivity of the fin material, $\text{Btu hr}^{-1} \text{ ft}^{-2} (\text{°F/ft})^{-1}$,

h = unit thermal conductance of the coolant, $\text{Btu hr}^{-1} \text{ ft}^{-2} (\text{°F})^{-1}$,

t = fin thickness, ft.

Substituting Eqs. (4), (5), and (6) into Eq. (3) gives

$$1.525 \times 10^{-3} D_R \epsilon L \tanh \left(m \frac{D_T - D_R}{2} \right) + \epsilon = 1 . \quad (7)$$

Equation (7) was solved for L for the range of values of the parameters. The results are given in Figs. 10.5 and 10.6. The weight of the reference design Freon boiler-condenser heat exchanger system is ~1800 lb. The weights of the finned exchangers are given in Figs. 10.7 and 10.8. The

Fig. 10.5. Dimensions of a Finned Copper Heat Exchanger Operating with 75°F Cooling Air.

Fig. 10.6. Dimensions of a Finned Aluminum Heat Exchanger Operating with 75°F Cooling Air.

Fig. 10.7. Weight of a Finned Copper Heat Exchanger Operating with 75°F Cooling Air.

Fig. 10.8. Weight of a Finned Aluminum Heat Exchanger Operating with 75°F Cooling Air.

hot-face heat flux of the finned exchanger varies from ~ 37 W/in.² for a copper fin with an 8.5-in. root diameter and a 60-in. tip diameter to ~ 9 W/in.² for an aluminum fin with an 8.5-in. root diameter and a 30-in. tip diameter.

Shipment of Thermoelectric Systems

Some of the problems associated with the removal and disposal of the waste heat from the water-cooled direct-conduction thermoelectric converters during shipment have been studied. Some of the same problems would apply to shipping terrestrially operated plants, but they would be reduced because the use of the atmosphere as a heat dump is inherent in the design.

A general concept for a direct-conduction device consists of a fuel element embedded in a heat block which also serves as a shield. The thermoelectric converter hot frame is placed in direct contact with the heat block, and the entire assembly is enclosed in a containing vessel, the walls of which receive the waste heat from the converter cold frame. The waste heat is conducted from the vessel wall to cooling water by natural convection. The combination of a low water temperature and a high heat-transfer coefficient allows the vessel wall to have a temperature of approximately 100°F with a water temperature of 50°F.

The heat rejection capacity of the vessel, cooled by natural convection and radiation, during shipment on a 100°F day was compared to its capacity when cooled by 50°F water during operation. The comparison was made both with and without fins on the vessel surface, and the fins were assumed to have a fin efficiency of 100% and the surface to have an emissivity of 1 for the finned cases and either 1 or 0.6 for the unfinned case. The emissivities of 1 and 0.6 for the unfinned vessel give the equivalent of a finned case with ratios of total convective heat-transfer area to the vessel area of 1 and 0.6 respectively. The heat-rejecting capacity expressed as a performance ratio is given in Fig. 10.9 as a function of the area ratio and the vessel surface temperature. The vessel surface temperature required to give a performance ratio of 1 is shown in Fig. 10.10 as a function of the area ratio.

Fig. 10.9. Comparison of Heat-Rejection Capacity of a Finned Vessel Operating in 100°F Air with an Unfinned Vessel Operating in 50°F Water.

Fig. 10.10. Surface Temperature of a Finned Vessel Operating in 100°F Air with a Performance Ratio of 1.

860°F. With the same thermojunction design this would require an increase of approximately 28% in the number of couples just to pass the heat. A mechanical refrigeration system is required to operate during the entire shipment time, including the time from the completion of the assembly to its placement in the 50°F water operating environment. With this approach sufficient area would have to be provided to allow air cooling to prevent the fuel from being overheated in case the mechanical refrigeration failed during shipment.

An alternative to the mechanical refrigeration system is a natural-circulation Freon system to provide refrigeration. This method would allow the vessel surface temperature to be approximately 175°F during shipment. However, the operating temperature difference would have to be reduced to approximately 1025°F, which would require an increase of approximately 7% in the number of couples to pass the heat, compared with 28% for direct air cooling during shipment.

STEAM TURBINE POWER UNIT

A conceptual design for a 5-kW(e) steam turbine-generator unit coupled to a $^{90}\text{SrTiO}_3$ power source enclosed in a steel shield is presented in Figs. 10.11 and 10.12. A unit was designed so that the isotope heat source and shield would be shipped in a cage to permit removal of the decay heat by

Examination of Fig. 10.10 shows that a water-cooled converter designed to operate at a vessel wall temperature of 100°F and a hot-face temperature of 1200°F cannot be shipped with only air cooling without exceeding the 1200°F design temperature by at least 250°F. Two possible approaches can be used to solve this problem: the 1200°F design temperature can be used for the air-cooling requirement during shipment, which would result in a lower design temperature than 1200°F when the equipment is operated in water; or mechanical refrigeration could be provided to keep the vessel surface at 100°F during shipment. The latter approach would be consistent with the 1200°F design temperature of a heat block when the system is operated in water.

Both approaches have disadvantages. Direct air cooling requires that the operating temperature difference from the shield block to the vessel be reduced from 1100°F to approximately

Fig. 10.11. Vertical Section Through Isotope Power Unit Coupled to a 5-kW(e) Generator and Contained in a Pressure Vessel Designed for Operation at 12,000 ft Beneath the Sea.

Fig. 10.12. Horizontal Cross Section Through Isotope Power Unit of Fig. 10.11.

thermal convection of the surrounding air, and the steam plant would be shipped inside the outer pressure vessel. The principal performance and dimensional data for the power unit are given in Table 10.2.

Table 10.2. Dimensional and Performance Data for
a Steam Turbine-Generator Isotope Power Unit

Power Output

Net electrical output, kW	5
Isotope heat output, kW(th)	50
Heat available for crew quarters, kW(th)	40

Isotope Capsules

Material	$^{90}\text{SrTiO}_3$
Capsule material	Hastelloy C-276
Capsule diameter, in.	2.94
Capsule length, in.	11.15
Number of capsules in-line	4
Number of capsules in plane of array	19
Total number of capsules	76
Heat output per capsule, W	658
Total heat output, kW(th)	50

Heat Block-Shield

Shield material	Steel
Capsule hole diameter, in.	3.0
Capsule hole center-line spacing, in.	4.0
Capsule hole pattern envelope diameter, in.	19
Fueled length, in.	44.6
Boiler tube pattern mean diameter, in.	21
Number of holes for boiler tubes	20
Boiler tube hole diameter, in.	0.625
Shield outside diameter (over fin tips), in.	41.1
Axial fin height, in.	1.0
Shield overall height, in.	69
Assembly weight, lb	21,100
Peak capsule temperature, °F	1625
Peak temperature in heat block, °F	1525
Temperature in heat block at rim of capsule array, °F	1000

Boiler

Steam outlet pressure, psia	200
Steam outlet temperature, °F	800
Outer tube inside diameter, in.	0.60
Outer tube outside diameter, in.	0.50
Inner tube outside diameter, in.	0.25
Inner tube inside diameter, in.	0.21
Number of outer tubes	20
Length of outer tubes, in.	69

Table 10.2 (Continued)

Boiler (Continued)

Surface area of outer tubes, ft ²	18.1
Surface heat flux for outer tube, Btu hr ⁻¹ ft ⁻²	8500
Temperature drop across air gap, °F	200
Radiant heat flux across air gap, Btu hr ⁻¹ ft ⁻²	2800
Thermal conduction through air gap, Btu hr ⁻¹ ft ⁻²	5700

Turbine-Generator Unit

Power output, kW(e)	5
Output voltage, V	28
Output frequency, cps	1000
Turbine rpm	60,000
Diameter, in.	6
Length, in.	10
Steam flow, lb/hr	105

Condenser

Number of tubes	120
Tube outside diameter, in.	0.25
Tube inside diameter, in.	0.20
Tube length, in.	60
Tube total internal surface area, ft ²	30
Tube total inlet flow passage area, in. ²	4.1
Seawater temperature, °F	40
Condenser temperature, °F	50
Steam inlet specific volume, ft ³ /lb	860
Steam inlet temperature, °F	70
Steam inlet velocity, fps	750

Outer Pressure Vessel

Outside diameter, in.	46
Wall thickness, in.	1.3
Overall height, in.	8
Weight, lb	5800
Design depth, ft	12,000

Shield Temperatures, °F, If Shipped in Air at 100°F

Shield surface temperature	600
Heat block perimeter temperature	825
Heat block maximum temperature	1350
Peak capsule surface temperature	1450

Isotope Heat Source and Shield

The proportions of the heat source and shield assembly were chosen from the parametric data presented in Chapter 4. To minimize the cost, a steel heat block and shield was employed with $^{90}\text{SrTiO}_3$ in Hastelloy C-276 capsules. The outer perimeter of the shield is fitted with fins to provide extra heat-transfer surface so that the heat can be removed at a moderate shield temperature by thermal convection of atmospheric air under shipping conditions. If the unit were shipped by plane in a cargo compartment with a reduced ambient air density, an auxiliary blower would probably have to be provided to maintain cooling during the flight.

Turbine-Generator

The turbine-generator of Fig. 10.11 is under development by the Swenson Research Laboratory. A cross section through this unit is shown in Fig. 8.4, a photograph of the unit installed in a housing (which also encloses the reciprocating feed pump) is shown in Fig. 10.13, and a performance curve is shown in Fig. 10.14. Steam enters the single-stage Terry turbine at 800°F and 200 psia, and is expanded to condenser pressure in a single nozzle with a wheel efficiency of 45%. The overall thermal efficiency of the steam power plant with no regenerative feed heating has been determined by Swenson as 9.5%. With a regenerative feed heater and several other refinements, Swenson estimates that the overall thermal efficiency could be increased to 12%, but no credit for this potential improvement was included in the data given in Table 10.2.

Fig. 10.13. Swenson Turbine-Generator and Feed Pump Assembly Enclosed in Its Housing. (Reproduced through courtesy of Swenson Research, Inc.)

The Swenson unit currently employs a reciprocating feed pump. It seems likely that a greater reliability could be obtained by mounting a centrifugal-pump impeller on the outboard end of the generator shaft if cavitation in the pump inlet can be suppressed satisfactorily. The titanium rotor runs on water-lubricated tilting-pad bearings at 60,000 rpm. The generator stator is potted in epoxy resin and delivers three-phase 1000-cps ac current at 60 V. The windings can be modified readily to produce 28 V for rectification to charge the storage batteries of a 24-V dc system.

Fig. 10.14. Indicated, Brake, and Drag Power vs Shaft Speed. (Reproduced through courtesy of Swenson Research, Inc.)

Boiler

The re-entry-tube boiler shown in Fig. 10.11 has been employed in place of a conventional simple recirculating boiler partly because it is particularly well suited for the rather highly superheated steam required in the turbine and partly to avoid possible difficulty with thermal shock effects when the cold boiler is installed in the hot shield-heat block assembly. The re-entry-tube boiler is a new concept that has been given only a few tests, but a fairly complete analysis is presented elsewhere.¹ The preliminary tests indicate excellent stability and control characteristics and performance consistent with Table 10.2.

The major advantage of this type of boiler stems from the buffering effect of the superheated steam layer in the annulus between the outer tube and the central tube. As shown in the analysis in Ref. 1, the buffer layer makes it possible to operate over a wide range of conditions and still ensure the production of superheated steam. Placing the boiler tubes in holes

close to the array of fuel capsules rather than at the outer perimeter of the shield makes it possible to produce high-temperature steam and yet keep the maximum heat-block temperature to less than 1525°F.

Condenser

The turbine performance is improved by minimizing the condenser temperature; consequently, the condenser tubes will be brazed to the outer pressure vessel. In this design the exhaust steam flows upward through two 1-in.-dia pipes to a ring manifold at the top and then downward from the steam manifold through the condenser tubes to the condensate manifold at the bottom. The lower 1-ft length of the 0.25-in.-OD condenser tubes is partly flattened to taper the flow passage toward the outlet to assist in the entrainment of bubbles of noncondensibles produced by radiolytic decomposition of the water into hydrogen and oxygen. The entrained bubbles of noncondensibles are redissolved in the liquid after the pressure is increased in the feed pump, and the dissolved hydrogen and oxygen recombine when the water becomes ionized by entering a radiation field.²

Capacity for Supplying Heat

The basic power unit of Fig. 10.11 can be modified to provide heat for an undersea manned installation such as SeaLab. This can be done by wrapping two 1-in. spiral pipes in parallel around the upper portion of the pressure vessel as shown in Fig. 10.15a and brazing them to the pressure vessel. The entire unit can then be surrounded with a jacket of thermal insulation, the inlet and outlet pipes connected to heating coils in SeaLab as indicated in Fig. 10.15b, and the water circulated by either thermal or forced convection. Thermal insulation should be provided around the connecting lines to reduce heat losses. The layer of water that flows between the pressure vessel and the outer jacket of thermal insulation will be heated to the same temperature as the water circulating in the heating system. If titanium were employed, there would be no difficulty with saltwater corrosion of the heating circuit. Also, the seawater pressure will not be high for an installation such as SeaLab, and therefore the walls of the jacket surrounding the thermal insulation need not be particularly heavy. Since water has a rather poor thermal conductivity, a multiplicity of layers of foil similar to those used for reflective insulation can be employed if it is desired to inhibit thermal convection currents and thus cause the water to serve as the thermal insulation. This arrangement would make the unit relatively light since it would not have to withstand any pressure differential between a gas region in the thermal insulation and the sea.

A heating system could be designed that used low-pressure (~10-psia) steam rather than hot water, thereby improving the rate of thermal convection. However, leaktight joints would have to be used in the system rather than the simple hose connections that can be employed with circulating hot seawater. The hose must, of course, be suitable for use at ~160°F.

Fig. 10.15. 5-kW(e) Isotope Power Unit with Water Coils for Removing 40 kW of Heat. (a) Sectioned outer jacket of thermal insulation showing coils on pressure vessel; (b) inlet and outlet pipes connected to heating coils in SeaLab.

Weight Reduction by Thinning the Shield

The weight of the shield-heat block assembly could be greatly reduced after the unit is immersed in water by letting the surrounding water serve as a shield. A cage should be mounted around the unit to prevent anyone under water from approaching it nearer than ~5 ft.

The principal problem associated with thinning the shield for use under water is associated with provisions for both heat transfer between the central and outer regions of the steel shield during shipment when a heavy shield is required and also easy assembly and disassembly. One way of doing this would be to make use of greatly extended surfaces by providing 2-in.-high axial fins on a 1/2-in. center along the parting surface between the inner and outer portions of the steel shield as shown in Fig. 10.16. With this arrangement the local heat flux is reduced to the point that just the thermal radiation across the air gap

Fig. 10.16 Horizontal Section Through Split Steel Shield Showing Finned Region Between Inner Portion Containing the Isotope and Outer Portion Used To Meet ICC Shipping Requirements.

will be sufficient to transmit the heat from the inner portion to the outer portion of the shield, with the inner portion running at 1000°F. Table 10.3 summarizes the principal data for this modification of the layout of Fig. 10.11 and Table 10.2. Figure 10.17 shows one type of thermal insulation jacket that could be used around the heat block in the central shield assembly after the boiler and steam system is installed. Before assembly, the inner and outer portions of the steel shield will have to be cooled to close to room temperature while the complete shield assembly is still in air so that no serious thermal shock problems will occur when the unit is immersed in water. A special water-cooling system that fits into the holes for the re-entry-tube boiler would serve this purpose. After the shield is cooled and the complete shield assembly immersed, the outer portion of the shield can be removed, the re-entry-tube boiler assembly installed in the heat block, and the outer thermal insulation jacket installed over the heat block. With the unit submerged only 5 or 10 ft, the water enclosed by the thermal insulation jacket can be displaced by pumping air into the unit and

Table 10.3. Size and Weight of Steel Shields and Steel Pressure Vessels with Insulating Liners for Design Radiation Dose Levels at Shield Surface

	At Radiation Dose at Shield Surface of				
	200 mR/hr	2 R/hr	20 R/hr	200 R/hr	2000 R/hr
<u>Shield</u>					
Thickness, in.	10.25	8.0	5.75	3.5	1.75
Diameter (fin root), in.	39.1	34.6	30.1	25.6	21.1
Diameter (fin tip), in.	41.1	38.6	34.1	29.6	25.1
Height, in.	69	66.8	64.5	62.3	60
Assembly weight, lb	21,100	16,100	12,100	8,800	6,100
<u>Pressure Vessel at 1000 ft</u>					
Thickness, in.	0.25	0.25	0.25	0.25	0.25
Outer diameter, in.	42.6	41.1	36.6	32.2	27.7
Height, in.	84.5	82.3	80.0	77.8	75.5
Weight, lb	1800	1700	1470	1250	1050
<u>Pressure Vessel at 12,000 ft</u>					
Thickness, in.	1.30	1.22	1.08	0.95	0.81
Outer diameter, in.	45.7	43.0	38.1	33.4	28.7
Height, in.	86.6	84.2	81.9	79.1	76.9
Weight, lb	5760	5000	4000	3160	2340
<u>Pressure Vessel at 24,000 ft</u>					
Thickness, in.	2.60	2.44	2.16	1.90	1.62
Outer diameter, in.	48.3	45.5	40.2	35.3	30.3
Height, in.	89.2	86.7	84.2	81.0	78.8
Weight, lb	11,360	9,830	7,600	5,700	4,160

Fig. 10.17. Outer Pressure Vessel and Thermal Blanket for Isotope Heat Source with a Thin Shield.

below their normal operating level. Also, the thermal insulation jacket is designed so that if the steam system were to stop removing heat from the heat block the inner layer of aluminum in the thermal insulation jacket would melt, the vacuum would be destroyed, heat would be rejected from the outer portion of the jacket, and the maximum temperature of the heat block assembly would not exceed 1500°F during transition.

Emergency Heat Removal

If there should be a failure in the water supply to the boiler after the unit is assembled, the heat block and shield assembly will heat up, from an initial temperature of about 1000°F, at the rate of about 100°F per hour. The aluminum blocks between the fins of the shield will be of an alloy that will melt at approximately 1150°F, and in any overheated region molten aluminum will flow through the thermal insulation to the pressure vessel, where it will be frozen to establish a fairly good thermal bond between the steel shield and the pressure vessel. The layer of thermal insulation is divided into small regions by a network of ceramic blocks arranged to fill the gap between the shield and the pressure vessel. Thus the surface of the shield is segmented into regions so that the molten aluminum is confined to one region, and no substantial area of the steel shield will be without an adequate aluminum thermal bond through the thermal insulation to the pressure vessel. Inasmuch as the heat loss through ceramic blocks is substantially greater than that through low-density thermal insulation, the area between the lines of closely fitted ceramic blocks consists of layers of reflective aluminum insulation, alternate layers of aluminum and ceramic fabric insulation, or rock-wool type of thermal insulation. Any of these materials

venting the water from the bottom. A separate condenser mounted to one side of the power unit would eliminate plumbing from having to be connected after the boiler and turbine assembly is installed in the heat block.

The weight of the complete unit would depend a great deal on the underwater depth for which it was designed, because much of the weight is in the thermal insulation jacket and pressure vessel surrounding the heat block assembly. Typical weights for representative depths are included in Table 10.3. If the unit were accidentally dropped into much deeper water than the depth for which it was designed, the outer jacket would simply crush inward on the steel heat block and make the unit inoperative, but the isotope capsules would remain intact and their temperature would drop

will be penetrated by molten aluminum if the aluminum bars between the fins become hot enough to melt. A heat flow path having a cross-sectional area of only 50 in.² of aluminum traversing the gap between the shield and the pressure vessel will hold the shield assembly at its normal operating temperature if the area in the thermal bonds is distributed fairly uniformly over the surface of the shield. Fortunately, the situation is stable in that any uncooled region will get hot and cause local melting of the aluminum to produce a local thermal bond, and this should occur irrespective of the attitude of the unit.

Assembly Procedure for Marine Applications

Assembly of the unit, which would take place at the dock, is briefly outlined below:

1. removal of steam power unit from pressure vessel and of heat source and shield assembly from shipping cage,
2. installation of steam power unit, including boiler, turbine-generator, and connecting piping, in a special fixture,
3. lowering of heat source and shield assembly into place on the steam power unit to position the re-entry tubes of the boiler in the heat source,
4. installation of array of aluminum blocks (shown between the fins in Fig. 10.12) in the heat source and shield assembly,
5. installation of thermal insulation blanket by means of a set of hoops to hold the various elements in place,
6. lowering of the upper portion of the pressure vessel into position around the heat source,
7. making connections, with Swagelok fittings, between condenser (the tubes of which are brazed to the upper portion of the pressure vessel) and the housing surrounding the turbine-generator feed pump unit,
8. raising the water level in the re-entry-tube boiler to an elevation sufficient for boiling to be initiated.

Once the boiler begins to deliver steam, the feed pump starts to operate and the system pressures and temperatures build up to their normal full-load values. Then the assembly proceeds as follows:

9. coupling of the generator to an electrical load (a partially charged bank of storage batteries would be sufficient),
10. installation of a water jacket temporarily around the upper portion of the pressure vessel to remove heat rejected from the condenser,

11. after an hour or so of performance testing to assure that unit is functioning properly, raising the lower portion of the pressure vessel into place and bolting the flange shut.

The temporary water jacket surrounding the upper portion of the pressure vessel will have to be supplied with cooling water until just before the unit is lowered over the side into the sea. Fortunately, the thermal capacity of the steel shield is so great that the rate of temperature rise would be less than 100°F per hour with no cooling whatsoever; hence there would be plenty of time to disconnect the cooling jacket and remove it before the assembly is lowered into the sea.

ORGANIC RANKINE-CYCLE SYSTEMS

Since there is a wide range of possible applications and environmental conditions for isotope power systems, organic Rankine-cycle systems include several working fluids and different equipment arrangements. Final selection of the working fluid will probably be dictated by the availability of power conversion equipment; therefore the fluids discussed are restricted to those that have been used in power cycles or are under development for this purpose.

Fluids other than water are considered because there is need for a good turbine efficiency combined with a low power output. The turbine design is greatly simplified for a fluid having a small enthalpy change upon expansion and having a large volume flow. Several organics that meet these requirements and have good thermal stability are being used in applications in which the temperature and power ranges are similar to those of this study. Dowtherm A is being used by both Aerojet-General Corporation³ and Sundstrand Corporation⁴⁻⁷ and monochlorobenzene and o-dichlorobenzene (Dowtherm E) are being used by Ormat Turbines, Ltd.,^{8,9} of Yavne, Israel.

A schematic of a system using one of these fluids is shown in Fig. 10.18. The basic difference between this system and the conventional steam cycle is the use of a regenerator in the turbine exhaust. Because of their properties, the fluids expand into the superheat region; hence a regenerator is added to improve the cycle efficiency. There are several possible variations of the arrangement shown in Fig. 10.18. Depending upon the pump suction requirement, the condensate may have to go to the pump suction before it is passed through the regenerator. Also, as is discussed later, by special equipment arrangements the pump can be eliminated and natural circulation can be used for the power cycle.

The vapor pressures of these organics, along with that of water, are shown in Fig. 10.19. The boiler pressures will be much less than that of water in all cases, and for the Dowtherms may be sufficiently low to eliminate the need for a boiler feed pump. For Dowtherm A the pressure at 475°F is about 11.5 psia, or about 30 ft of liquid. Thus, if the condenser can be located 30 ft above the boiler, the system can be operated by natural convection rather than with a feed pump. For Dowtherm E the boiler temperature would have to be reduced to about 350°F to operate by natural convection with a 30-ft liquid head.

Fig. 10.18. Schematic of Organic Rankine-Cycle System.

The cycle efficiency depends on the boiler and condenser temperatures, the turbine-generator efficiency, and the amount of heat recovered in the regenerators. An evaluation of the current development work with these fluids indicates that the actual cycle efficiency should be 50 to 60% of the Carnot-cycle efficiency (see Fig. 10.20). The low vapor pressures of Dowtherm A and Dowtherm E can, for some applications, be a detriment to the cycle efficiency. At the low-temperature end of the cycle the low pressure and thus very large vapor volume

may be the limitation rather than the heat dump temperature. An example of this is given in Table 10.4, where the condenser temperature is assumed to be either 100°F or the temperature corresponding to a vapor pressure of 0.1 psia. Thus the effective condenser temperature for Dowtherm A is 220°F and for Dowtherm E is 115°F. Dowtherm A also has the disadvantage of a high (53°F) freezing point, which may limit its use for some applications.

The conceptual design of four variations of a 5-kW(e) organic Rankine-cycle system is shown in Figs. 10.21-10.24. The designs in Figs. 10.21 and 10.22 are for underwater applications, and those in Fig. 10.23 and

Fig. 10.19. Vapor Pressure of Several Organic Fluids and Water.

Fig. 10.20. Carnot-Cycle Efficiency.

Table 10.4. Carnot-Cycle Efficiency^a and Freezing Point of Several Fluids

Fluid	Carnot Efficiency (%)			Condenser Temperature (°F)	Freezing Point (°F)
	for Boiler Temperature of 550°F	450°F	350°F		
Dowtherm A	32.7	25.3	16.0	220	+ 53
Dowtherm E	43.1	36.8	29.0	115	- 7
Monochlorobenzene	44.6	38.5	30.9	100	- 49
Water	44.6	38.5	30.9	100	+ 32

^aFor condenser temperature of 100°F or the temperature corresponding to 0.1 psia.

Fig. 10.21. Forced-Convection Organic-Cycle Power System for Marine Applications.

Fig. 10.22. Natural-Convection Organic-Cycle Power System for Marine Applications.

0 1 2 3 4 5
 FT SCALE FT

DWG D 115-27

Fig. 10.23. Forced-Convection Organic-Cycle Power System for Land-Based Applications.

1

2

3

BLANK

4

5

6

7

8

9

10

Fig. 10.24. Natural-Circulation Organic-Cycle Power System for Land-Based Applications.

BLANK

Figs. 10.24 are for terrestrial use. All four variations are based on the concept that the system will be shipped in two parts: (1) the entire working-fluid system, including boiler, turbine-generator, and condenser, and (2) the heat source. It is assumed that the first part will be fabricated, filled with the working fluid, sealed, and tested at some fabrication facility before being shipped and that the system need not be opened in the field during installation. The heat source would be shipped in a cage that permits free circulation of air for removal of the heat but prevents accidental contact with the hot surface. The final assembly of the two parts would be done at the site, or perhaps some dockside facility, and would entail removing the cage and placing the heat source into or around the boiler tubes. This assembly operation would not require that the power conversion system be opened.

The arrangement shown in Fig. 10.21 would require special or temporary cooling if it were kept out of water for an extended time period after being assembled. The rate of rise of the temperature of the steel or nickel heat sources is between 50 and 100°F per hour, and for uranium or tungsten is about 400°F per hour. The bare heat source before being assembled will have a surface temperature of 400 to 550°F for steel or nickel, and will not require cooling for about 4 hr. For the uranium or tungsten heat source the surface temperature will be 650 to 750°F and cooling would be required within 30 min of the time that the source is placed inside the insulation. The temporary cooling system could be either a water spray over the condenser portion of the vessel or a jacket which provides a pool of water over the condenser.

To prevent an excessive fuel cladding temperature in the event that the temporary cooling is lost, an aluminum reflective type of insulation would be used around the side of the heat source. By choosing the proper alloy, the melting point of the aluminum sheets may be from 900°F to over 1200°F. Choosing a material that will melt in the range of 1000 to 1100°F will protect the fuel elements in systems using steel or nickel shields. With uranium or tungsten shields the smaller surface area and thus high heat flux may lead to excessive fuel element temperatures unless the fuel and heat source configuration is originally selected to meet this accident condition. This would require that some of the weight advantage of the uranium or tungsten be sacrificed.

The systems shown in Figs. 10.22-10.24 would not require special cooling after assembly. The system shown in Figs. 10.23 and 10.24 are designed for air operation and have several safety devices. After assembly, the system should be in its normal operating condition. If the working fluid fails to remove sufficient heat to keep the air temperature in the boiler at the desired point, two dampers, controlled by the air temperature, are available for circulation of air through the boiler to reduce the temperature and dump the excess heat to the atmosphere. If the normal heat dump and both dampers fail, the temperature in the boiler would rise until it reached 800 to 900°F, at which point the large flange above the heat source would drop off. Upon release of the flange (which is held in place by a low-melting alloy), air would circulate through the boiler to remove the heat

by natural circulation. All these emergency heat dump methods depend on natural circulation of air and therefore require to some extent that the unit be upright. To provide for an accident in which the units are not upright, the side insulation would again be an aluminum reflective type in which the melting of the insulation would protect the fuel elements.

The system shown in Fig. 10.22 is similar to those in Figs. 10.23 and 10.24 except that the air recirculates by natural convection within the vessel after assembly. Although the vessel has sufficient surface area to reject the heat to the atmosphere by radiation and natural convection without excessive temperatures within the system, it would be advisable to spray water over the vessel after assembly.

The very large and bulky units shown in Figs. 10.22 and 10.24 are the results of going to a natural-circulation power cycle in which the boiler pressure is obtained with the liquid head from the condenser. The reason for emphasizing the natural-circulation system is not so much the elimination of the boiler feed pump as the possibility of using a very simple control scheme. One such control concept would be to use a turbine-dc generator in which the tip speed would run about one-half the nozzle velocity (approximately the speed for peak efficiency for an impulse turbine) at full load. If the load were reduced, the turbine speed would increase to a maximum of less than twice its full-load speed, at which point the torque and therefore the power produced would become zero; that is, the system energy would be passed directly to the condenser. This concept allows the turbine efficiency to vary from its maximum to zero in such a manner that it will extract from the system whatever power is demanded by the load (up to its maximum design output). The concept does require that a voltage regulator be used and that the turbine generator be designed to permit operation at its "no-load" speed.

In comparing the merits of the different organic fluids for use in under-sea or terrestrial applications, it becomes apparent that Dowtherm A is not as good a working fluid as either Dowtherm E or monochlorobenzene because of its very low vapor pressure and thus high specific volume below 225°F and also its rather high freezing point (53°F). The reason for including Dowtherm A is that it has received the most attention in this country. However, the developmental effort on this fluid was directed toward space applications in which the heat dump requirements lead to high condenser or radiator temperatures. For space applications the superior thermal stability of Dowtherm A would make it better than either Dowtherm E or monochlorobenzene. In the present study Dowtherm A would be restricted to applications in which the condenser is cooled by very high temperature ambient air (design temperature > 125°F) and in which there is no danger of exposure to sufficiently low temperatures to cause freezing of the system.

The choice between Dowtherm E and monochlorobenzene will depend on the application. In general, monochlorobenzene appears to be the better all-purpose working fluid for power levels in the range of 1 to 2 to 10 kW(e) because of its higher vapor pressure and also its lower freezing point. Dowtherm E has the advantage of being applicable for natural-circulation

systems in which the boiler feed pump is eliminated or for lower power output cases. Natural-circulation systems would not be practical with monochlorobenzene.

The layouts shown in Figs. 10.21-10.24 show the general arrangement and approximate size of equipment items. No special effort was made to minimize the size or weight of the units. Table 10.5 gives specific dimensional and performance data for a unit similar to that shown in Fig. 10.21 with monochlorobenzene used for operation in water at 40°F.

Table 10.5. Dimensional and Performance Data for an Isotope-Fueled Monochlorobenzene Rankine-Cycle Power Unit

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Power Output

Net electric, kW	5
Isotope heat, kW(th)	27

Fuel Capsules

Fuel material	⁹⁰ SrO
Fuel cladding material	Hastelloy C-276
Fuel outside diameter, in.	2.50
Cladding outside diameter, in.	2.94
Capsule length, in.	10.14
Number of capsules in-line	2
Number of capsules in plane of array	13
Total number of capsules	26
Heat output per capsule, kW	1.04
Maximum fuel temperature, °F	2050
Maximum cladding temperature, °F	1475

Heat Block-Shield Assembly

Shield material	Nickel
Capsule hole diameter, in.	3.0
Capsule hole center-line spacing, in.	3.5
Capsule hole pattern envelope diameter, in.	15.12
Fueled length, in.	20.28
Boiler tube pattern mean diameter, in.	27
Boiler tube hole diameter, in.	1.56
Number of holes for boiler tubes	24
Shield outside diameter (over fin tips), in.	38
Fin height, in.	2.5
Shield overall height, in.	38.6
Assembly weight, lb	11,000
Maximum heat block temperature, °F	1350
Temperature of heat block at outer surface of capsule array, °F	800

Boiler

Fluid outlet temperature, °F	375
Fluid outlet pressure, psia	57
Outer tube outside diameter, in.	1.5
Outer tube inside diameter, in.	1.4

Table 10.5 (Continued)

Boiler (Continued)

Inner tube outside diameter, in.	0.5
Inner tube inside diameter, in.	0.4
Number of tubes	24
Effective length of tubes in shield, in.	33
Surface area of outer tubes, ft ²	25.9
Total heat to boiler, kW(th)	25
Total heat to boiler, Btu/hr	85,300
Surface heat flux, Btu hr ⁻¹ ft ⁻²	3300
Mean temperature of heat block holes, °F	625
Temperature drop from block to fluid, °F	250
Radiant heat transfer, Btu hr ⁻¹ ft ⁻²	900
Thermal conduction, Btu hr ⁻¹ ft ⁻²	2400
Fluid flow rate, lb/hr	430

Regenerator

Effectiveness, %	90
Heat load, Btu/hr	9500
Vapor temperature in, °F	193
Vapor temperature out, °F	93
Liquid temperature in, °F	80
Liquid temperature out, °F	155
Weight flow, lb/hr	430
Number of rows	13
Number of tubes per row	6
Number of liquid passes per row	3
Tube outside diameter, in.	0.375
Tube inside diameter, in.	0.325
Face dimensions, in.	12 x 12
Length, in.	30
Vapor side surface area, ft ²	416
Liquid side surface area, ft ²	6.7
Type of fin	Plate
Fin width, in.	2
Fin length, in.	12
Number of fins per inch	8
Minimum vapor flow area, ft ²	0.65

Condenser

Heat load, Btu/hr	68,000
Pressure, psia	0.25
Temperature in, °F	93
Condensing temperature, °F	80
Flow rate, lb/hr	430
Water temperature, °F	40
Vessel surface area for condensing, ft ²	36

Table 10.5. (Continued)

<u>Outer Pressure Vessel</u>	
Vessel design depth, ft	12,000
Lower vessel inside diameter, in.	50
Lower vessel wall thickness, in.	1.5
Lower vessel length, in.	108
Upper vessel inside diameter, in.	24
Upper vessel wall thickness, in.	0.75
Upper vessel length, in.	48
Total length, in.	156
Weight, lb	9200
<u>Heat Source Temperature, °F, If Shipped in Air at 100°F</u>	
Shield surface temperature	500
Heat block perimeter temperature	650
Heat block maximum temperature	1200
Peak cladding temperature	1350
Peak fuel temperature	1950

BRAYTON-CYCLE SYSTEM

A substantial analytical and experimental program has been under way at both NASA Lewis Research Center¹⁰⁻¹² and AiResearch^{13,14} since 1962 on Brayton-cycle systems for nuclear space power plants. Some indication of the scale of the effort is indicated by the funding: Lewis Research Center spent around \$20 million from 1962 to 1968. The major objective of the NASA program has been to refine the compressor and turbine to obtain exceptionally high component efficiencies, a difficult objective in small units in which the Reynolds numbers are inherently low. In a small machine of a given output it is helpful to make use of a high-molecular-weight inert gas such as xenon, but this leads to very poor heat-transfer coefficients, which in turn entail serious pressure loss penalties in the heater, regenerator, and cooler. The best compromise has been found to be a mixture of helium and xenon having a mean molecular weight of 83, and the bulk of the AiResearch and NASA work since 1964 has been carried out on such a mixture. The program has progressed to the point that the first unit of a new model turbine-generator unit has been operated at an output of about 5 kW(e) for 500 hr at Lewis Research Center. Of this, 375 hr of operation was with argon and 125 hr was with krypton (which has a molecular weight of 83). By varying the system pressure the power output can be varied from 2 to 10 kW(e). This unit represents the most advanced small Brayton-cycle machine available and was therefore used as the basis for the conceptual design presented in this section.

The conceptual design for a Brayton-cycle isotope power unit for undersea applications is presented in Fig. 10.25. The layout is based on the use of ⁹⁰SrTiO₃ fuel encapsulated in Hastelloy C-276. The capsules are installed in a steel heat block and shield assembly. To prevent excessive gamma-ray leakage, the gas ducts are designed with wavy walls in the form

Fig. 10.25. Brayton-Cycle Power Unit for Marine Applications.

of surfaces of revolution. The outer duct supplies gas from the regenerator to the shield region immediately outside the envelope of the fueled region. The gas emerging from the lower end of this set of cooling passages flows radially inward through a plenum and then vertically upward through cooling passages around the isotope capsules (see Fig. 10.26). The hot gas flow is collected in a plenum at the top end of the heat block region, from which it flows upward through the central annular passage to the turbine inlet.

The proportions of the turbine-generator unit and regenerator were taken from drawing SK51365 of Ref. 14. The layout presented here differs somewhat from the AiResearch layout in that the regenerator is placed at the top rather than at the side of the turbine-generator unit, and the cooler used by AiResearch has been eliminated. The air leaving the regenerator flows to a volute distributor, from which it passes up through a set of finned passages between a baffle and the inside of the pressure vessel to the compressor inlet. This arrangement eliminates the need for a secondary coolant system to transport the heat from the cooler to a separate radiator, as planned for the space power plant.

A cutaway view of the turbine-generator is shown in Fig. 8.5. Since the rotor runs on gas bearings, bearing frictional losses are small. To obtain high efficiencies, the contours of the gas passages, particularly in the rotors, are held to very close tolerances and the running clearances are small.

An extensive set of performance analyses for both Brayton-cycle systems in general and the particular system for which the unit of Fig. 8.5 was designed has been carried out at the NASA Lewis Research Center and at AiResearch, and over 30 reports have been prepared to summarize the results. In reviewing this work, it was found that a fairly straightforward perturbation could be made on the design and performance data presented by AiResearch.¹⁴ Because of temperature limitations on the isotope capsules and allowances for emergency heat removal, it was necessary to reduce the turbine inlet temperature from 1600°F to 1480°F. According to Fig. 5 of Ref. 14 this would reduce the net electrical output from 6 kW to 5 kW, a value consistent with other phases of this study. The estimated performance presented in Table 1 of Ref. 14 was modified accordingly to yield Table 10.6 of this report. The estimated heat loss through the thermal insulation surrounding the heat block-shield assembly is included (at the bottom of Table 10.6) and with this allowance the overall thermal efficiency is 20%. Whether or not this relatively high thermal efficiency can be achieved depends on the component efficiencies. The test program underway at the Lewis Research Center should yield firm values for the component efficiencies by the latter part of 1969.

The temperature distribution in the system is indicated in Fig. 10.27, and the estimated temperature distribution at the top end of the heat block is summarized in Table 10.7, together with data on the shield, pressure vessel, and overall assembly.

Fig. 10.26. Horizontal Cross Section Through Isotope Power Unit of Fig. 10.25.

Table 10.6. Brayton-Cycle Performance

Net generator output, kW	5.0
Gross generator output, kW (1200 cps)	5.187
Working fluid, Xe-He, mol wt	83.8
Shaft speed, rpm	36,000
Recuperator effectiveness, %	90
$\beta \sim 1 - (\Delta P/P)_{\text{total system}}$	0.92
Compressor mass flow rate, lb/sec	0.796
Compressor bleed flow, %	2
Compressor inlet temperature, °F	80
Compressor inlet pressure, psia	14.2
Compressor diameter, in.	4.25
Compressor pressure ratio	1.9
Compressor efficiency, %	80
Compressor specific speed	0.11
Turbine inlet temperature, °F	1460
Turbine inlet pressure, psia	25.8
Turbine diameter, in.	4.97
Turbine pressure ratio	1.75
Turbine efficiency, %	87
Turbine specific speed	0.092
<u>Generator</u>	
Generator diameter, in.	3.3
Generator efficiency, %	92
Generator windage loss, kW	0.468
Brayton rotating unit thermal losses, kW	0.236
Bearing friction loss, kW	0.271
<u>Electrical Losses</u>	
Field excitation, kW	0.037
Speed control, kW	0.100
Voltage regulator equipment, kW	0.050
<u>Heat Loss</u>	
Shield surface, kW	5
Cycle efficiency, $\eta_{\text{cy}} = \text{net output}/\text{total heat in}$	0.20
* Pressure loss factor.	

In estimating the temperature distribution in the heat block-shield assembly, it was found that the film temperature drop between the metal wall and the bulk free gas stream could be kept to less than 100°F. In view of the temperature rise in the gas stream being a little less than 400°F, the local metal temperature will be determined mainly by the local gas temperature. Large-diameter holes were used in the first pass so that the temperature rise in that region would be only about 100°F, and much smaller passages were employed in the second pass, in which the gas flows directly over the fuel capsules themselves. Consequently, the bulk of the temperature rise will occur in the second pass. The metal temperature in some portions of the heat block immediately outside the fuel capsules will actually be a bit below the local gas temperature; some of the heat transferred from the fuel capsules to the gas will flow through the gas to the heat block walls and hence out to the first-pass cooling passages. This will help to alleviate the severe thermal stresses induced by the large radial temperature gradient in the heat block. One means of further reducing the stresses associated with this large thermal gradient is to cut radial slits between the holes in the heat block in regions in which there would be no transverse heat flow if the slit were not there. A set of such slits is shown in Fig. 10.26. The extent to which these slits will alleviate thermal stresses in the heat block and shield is difficult to estimate and probably will require an experiment.

Table 10.7. Dimensional and Performance Data for a Brayton-Cycle Turbine-Generator Isotope Power Unit

<u>Power Output</u>		
Net electrical output, kW	5	
Isotope heat output, kW(th)	25	
Heat available for crew quarters, kW(th)	15	
<u>Isotope Capsules</u>		
Isotope material	$^{90}\text{SrTiO}_3$	
Capsule material	Hastelloy C-276	
Capsule diameter, in.	2.94	
Capsule length, in.	11.15	
Number of capsules in-line	2	
Number of capsules in plane of array	19	
Total number of capsules	38	
Heat output per capsule, W(th)	658	
Total heat output, kW(th)	25	
<u>Heat Block-Shield Assembly</u>		
Shield material	Steel	
Capsule hole diameter, in.	3.0	
Capsule hole center-line spacing, in.	4.0	
Capsule hole pattern envelope diameter, in.	19	
Fueled length, in.	22.3	
First-pass coolant passage pattern mean diameter, in.	21	
Number of holes for first-pass coolant passages	60	
First-pass coolant passage diameter, in.	0.50	
Shield outside diameter (over fin tips), in.	41.1	
External axial fin height, in.	1.0	
Shield overall height, in.	59	
Assembly weight, lb	18,100	
Peak capsule temperature (normal), °F	1740	
Peak temperature in heat block (normal), °F	1525	
Temperature in top of heat block at rim of capsule array, °F	1160	
<u>Brayton-Cycle Heater</u>		
Coolant pass	1st	2nd
Coolant density (mean), lb/ft ³	0.127	0.114
Coolant passage equivalent diameter, in.	0.50	0.25
Coolant passage flow area, ft ²	0.0833	0.145
Coolant passage effective surface area, ft ²	15.5	16
Coolant flow rate, lb/sec	0.796	0.796
Coolant mass velocity, lb sec ⁻¹ ft ⁻²	9.55	5.5
Coolant Reynolds number	11,000	3,000
Coolant dynamic head, psi	0.08	0.028
Coolant friction factor	0.42	0.52
Coolant passage length, in.	24	22.3
Coolant pressure drop, psi	0.24	0.16
Coolant heat-transfer coefficient, Btu hr ⁻¹ ft ⁻² (°F) ⁻¹	17.2	12.8
Coolant temperature rise, °F	83	300
Coolant specific heat, Btu/lb	0.0625	0.0625
Heat added to coolant, Btu/sec	4.12	14.9
Film temperature drop, °F	56	262
<u>Cooler^a</u>		
Fin height, in.	0.40	
Fin center-line spacing, in.	0.30	
Fin thickness, in.	0.10	
Fin length, in.	64	
Total gas side surface area, ft ²	200	
Gas side total flow passage area, ft ²	145	
Gas side total flow rate, lb sec ⁻¹ ft ⁻²	5.5	
Seawater temperature, °F	40	
Gas exit temperature, °F	80	
Gas heat-transfer coefficient, Btu hr ⁻¹ ft ⁻² (°F) ⁻¹	10	
Gas film temperature drop, °F	30	
Water film temperature drop, °F	5	
Temperature drop through pressure vessel, °F	5	
Gas mean velocity, lb sec ⁻¹ ft ⁻²	5.5	
Gas mean dynamic head, psi	0.011	
Gas friction factor	0.05	
Gas pressure drop, psi	0.14	

Table 10.7. (continued)

<u>Outer Pressure Vessel</u>	
Outside diameter, in.	46
Wall thickness, in.	1.3
Overall height, in.	141
Weight, lb	9500
Design depth, ft	12,000
<u>Shield Temperatures, °F, If Shipped in Air at 100°F</u>	
Shield surface temperature	600
Heat block perimeter temperature	825
Heat block maximum temperature	1350
Peak capsule surface temperature	1650

^aUpper portion of pressure vessel inner surface.

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Fig. 10.27. Flowsheet of Operating Conditions for Brayton Cycle.

A major consideration in the design of the heat block and shield region was provision for heat removal under emergency conditions when the Brayton-cycle machinery would not be circulating gas through the heat block assembly. A fusible alloy of aluminum mounted between the fins around the outer perimeter of the shield will melt if the shield temperature exceeds $\sim 1200^\circ\text{F}$, thus providing a thermal bond between the shield and the surrounding pressure vessel. To make this mechanism as effective

as possible, a thermal insulation system similar to that described in "Steam Rankine-Cycle Systems" would be employed. Unfortunately, fluting the bores of the holes in the heat block in which the fuel capsules are mounted reduces the conductivity of the gap between the fuel capsule and the heat block and hence substantially increases the temperature difference between the two under emergency conditions. This makes it necessary to reduce the normal operating temperature of the capsule wall to well below the 1800°F nominal peak allowable temperature.

A heat exchanger using removable outer insulation as an emergency safeguard was also considered. This heat exchanger is suitable for operation with the Brayton-cycle components described previously.

A conceptual design (Figs. 10.28 and 10.29) was made of a parallel-flow Brayton-cycle heat exchanger fueled with 30 kW of $^{90}\text{SrTiO}_3$ and utilizing exterior fins for emergency heat removal. This heat exchanger weighs ~22,000 lb, and contains 19 fuel tubes with an edge-to-edge distance of 2.0 in., with a length of 37.4 in., and with an inside diameter of 2.0 in. The fuel tube cladding is Hastelloy C-276. The fuel is shielded by 10.5 in. of iron, which is sufficient to reduce the surface dose rate to 200 mrad/hr. The heat exchanger is designed to raise the temperature of a He-Xe mixture (mol wt, 83.8) from 1039°F to 1500°F at a flow rate of 3456 lb/hr with less than 3% pressure drop. The other design characteristics for this unit are contained in Table 10.8.

As part of the overall safety analysis, a study was made to determine what effect a loss of gas flow would have on the fuel capsule cladding temperature. One way to ensure that the maximum cladding temperature does not exceed 2000°F is to add fins to the outer surface of the heat exchanger. The added surface area allows the heat to be transferred to the environment with reasonable internal temperatures. An important consideration in this type of emergency cooling is the rapid removal of the outer insulation, thus exposing the outer fins to the surroundings. Several means of removing the insulation were considered, such as explosive bolts, metal foil insulation which would melt at a fairly low temperature, or spring tension in conjunction with a pressurized piston. For a conceptual design, spring tension was used to remove the insulation quickly as the inner cladding temperatures begin to rise.

The emergency heat dump system for the heat exchanger requires the removal of the outer insulation so that the exterior fins can "see" the environment. The outer insulation for the heat exchanger consists of six segments, which are removed during emergency conditions by release of spring tension after a pressure-held piston is released. The springs are held in tension during normal operating conditions, and hence each insulation segment is held in place by one of six detent-pin systems. This system consists of a detent pin that is extended into a notch in the insulation stripping assembly rod by compression from a bellows in the detent cylinder. The pressure to this bellows is supplied by pressurized inert gas at 100 psig. Under emergency operating conditions a fusible plug located on the surface of a fuel capsule will melt at 1950°F and release the pressure on the detent bellows manifold to the atmosphere.

Table 10.8. Design Characteristics of $^{90}\text{SrTiO}_3$ -Fueled Parallel-Flow Heat Exchanger

<u>Fuel Tube Properties</u>	
Fuel form	$^{90}\text{SrTiO}_3$
Power density, W/in.	13.3
Thermal conductivity at 1472°F, $\text{Btu hr}^{-1} \text{ft}^{-2} (^\circ\text{F/ft})^{-1}$	1.93
Number of tubes	19
Length, in.	37.4
Outside diameter, in.	2.46
Inside diameter, in.	2.0
Edge-to-edge distance between tubes (triangular pitch), in.	2.0
Cladding (Hastelloy C-276) thickness, in.	0.23
Maximum allowable cladding temperature, °F	2000
Bundle outside diameter, in.	20.3
<u>Gas Properties</u>	
Working fluid (mol wt, 83.8)	He-Xe
Inlet temperature, °F	1089
Exit temperature, °F	1500
Inlet pressure, psia	26.3
Maximum allowable pressure drop, %	3
Density, lb/ft^3	0.133
Viscosity, $\text{lb hr}^{-1} \text{ft}^{-1}$	0.153
Thermal conductivity, $\text{Btu hr}^{-1} \text{ft}^{-2} (^\circ\text{F/ft})^{-1}$	0.0338
Flow rate, lb/hr	3456
<u>Shielding Requirements</u>	
Solid fuel block (Fe), in.	9.3
Open fuel block (Fe), in.	10.5

The plug is contained in a seal tube that is open to the atmosphere. An insulated flange is provided at the shield insulation junction to reduce heat losses. When the plug melts, the pressurized gas escapes through the exhaust ports to the seal tube and thus to the atmosphere. Although the seal tube is shown as straight in Fig. 10.29, an offset would be provided in an actual model to prevent radiation beaming. Two of these fusible-plug sensing devices are provided to assure that the system is fail-safe. An additional safeguard is provided by a fragile section (Fig. 10.29) in the gas pressure line which is designed to rupture and release the gas pressure in the event of severe shock.

Table 10.9. Summary of Dimensional and Performance Data for the Isotope Power Units Presented as Conceptual Designs

Power Conversion Unit	Thermoelectric	Steam Turbine	He-Xe Turbine	Organic Turbine
<u>Power Output</u>				
Net electrical output, kW	2.27	5	5	5
Isotope heat output, kW(th)	30	50	25	27
Heat available for crew quarters, kW	20	40	15	17
<u>Isotope Capsules</u>				
Isotope material	^{90}SrO	$^{90}\text{SrTiO}_3$	$^{90}\text{SrTiO}_3$	^{90}SrO
Isotope capsule material	Hastelloy C	Hastelloy C	Hastelloy C	Hastelloy C
Isotope capsule diameter, in.	2.94	2.94	2.94	2.94
Isotope capsule length, in.	11.27	11.15	11.15	10.14
Number of capsules in-line	2	4	2	2
Number of capsules in plane of array	13	19	19	13
Total number of capsules	26	76	38	26
Heat output per capsule, W	1154	658	658	1040
<u>Heat Block-Shield</u>				
Shield material	Tungsten	Steel	Steel	Nickel
Capsule hole diameter, in.	3.0	3.0	3.0	3.0
Capsule hole center-line spacing, in.	4.5	4.0	4.0	3.5
Capsule hole pattern envelope diameter, in.	18.59	19	19	15.12
Fueled length, in.	22.53	44.6	22.3	20.28
Shield outside diameter, in. (over fin tips)	27	41.1	41.1	38
Axial fin height, in.		1.0	1.0	2.5
Shield overall height, in.	30.8	69	59	38.6
Assembly weight, lb	10,600	21,100	18,100	11,000
Peak capsule temperature, °F	1640	1625	1740	1475
Peak temperature in heat block, °F	1490	1525	1525	1350
Temperature in heat block at rim of capsule array, °F	1270	1000	1160	800
<u>Electric Generator Unit</u>				
Output voltage, V	28	28	208/120	120
Output frequency, cps	dc	1000	1200	1000
Turbine rpm		60,000	36,000	20,000
Length, in.		10	30	18
Diameter, in.		6	21	13
Fluid flow, lb/hr		105	2500	430
<u>Outer Pressure Vessel</u>				
Vessel outside diameter, in.	41	46	46	53
Vessel wall thickness, in.	1.3	1.3	1.3	1.5
Vessel overall height, in.	78	84	141	156
Vessel weight, lb	4800	5800	9500	9200
Vessel design depth, ft	12,000	12,000	12,000	12,000
<u>Shield Temperature If Shipped in Air at 100°F</u>				
Shield surface temperature, °F	700	600	600	500
Heat block perimeter temperature, °F		825	825	650
Heat block maximum temperature, °F		1350	1350	1200
Peak capsule surface temperature, °F		1450	1650	1350
Total System Weight, lb	18,400	28,000	29,000	22,000
Height of Complete System, in.	140	84	141	156
Diameter of Complete System, in.	62	46	46	53

Fig. 10.28. Schematic of Conceptual Parallel-Flow Heat Exchanger with Structural Supports.

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Fig. 10.29. Enlarged View of Insulation Removal System of Heat Exchanger Shown in Fig. 10.28.

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Appendix A

FUEL PROPERTIES FOR $^{90}\text{SrTiO}_3$, ^{90}SrO , $^{144}\text{Ce}_2\text{O}_3$, and ^{60}Co

(Reproduced from *ORNL-4358 and **ORNL-4185)

B. STRONTIUM TITANATE (SrTiO_3)1. Composition

a. Radionuclidic abundance

The radionuclidic abundances are given in Section I.A.1.

b. Radiochemical purity

The radiochemical purity is given in Section I.A.1.

c. Chemical composition*

(1) Range of composition

<u>Cation</u>	<u>Maximum, wt %</u>	<u>Minimum, wt %</u>
Sr	97	92
Ca	5	2
Ba	2	0.5
Mg	2	0

One gram of ^{90}Sr will produce an average of 0.0232 g of ^{90}Zr after one year's decay.

(2) Average chemical composition

<u>Metal oxide</u>	<u>Content, wt %</u>	<u>Titanate compound</u>	<u>Content, wt %</u>
SrO	52.5	SrTiO_3	92.5
CaO	2.3	CaTiO_3	5.6
BaO	0.5	BaTiO_3	0.8
MgO	0.4	MgTiO_3	1.1
TiO_2	44.3		

The strontium metal content of pure fission-product SrTiO_3 is 48.16%.

The strontium metal content of average fission-product SrTiO_3 is 44.55%.

The as-processed SrTiO_3 will contain 24.5% ^{90}Sr . The ^{144}Ce constitutes 0.03% of the radioactivity at a maximum. The ^{137}Cs has been found to be as high as 0.003% of the total activity. The ^{106}Ru activity has not been found. These activities can be neglected compared to the bremsstrahlung radiation.

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*Results of analyses on strontium titanate feed samples of the Fission Products Development Laboratory at Oak Ridge National Laboratory.

STRONTIUM TITANATE (SrTiO₃)REFERENCE
COLUMN2. Specific Power

2, 4, 5

- a. 0.247 w/g of pure SrTiO₃ (26.5% ⁹⁰Sr metal)
 0.235 w/g of average SrTiO₃ (24.5% ⁹⁰Sr metal)

It is assumed that there are 139 Ci/g of ⁹⁰Sr and
 148.9Ci of ⁹⁰Sr per thermal watt.

- b. 36.8 Ci of ⁹⁰Sr per gram of pure SrTiO₃ (26.5% ⁹⁰Sr metal)
 34.1 Ci of ⁹⁰Sr per gram of average SrTiO₃ (24.5% ⁹⁰Sr metal)

3. Radiation

The radiation is the same as given under Section I.A.3.

4. Critical Mass

Strontium-90 and ⁹⁰Y are not fissionable.

5. Compatibility with Materials of Containment

The compatibilities of various inactive strontium com-
 pounds with a number of container materials were determined
 and have been summarized in the following table.

Source material and exposure	Container material								
	Superalloys				Refractory metals				
	Hastelloy X	Haynes 25	Ni-o-nel	N-155	Mo	Nb	Ta	W	TZM
Strontium metal (Sr)									
1000°C, 1000 hr	SR	NR	SR	MR	NR		SR	NR	NR
1000°C, 5000 hr		MR		MR			SR	NR	
1000°C, 10,000 hr		MR	MR	MR	NR		SR	SR	
Strontium oxide (SrO)									
1000°C, 1000 hr	SR		SR	SR	NR	NR	SR	SR	
1850°C, 168 hr					SR	MR	SR	SR	
Strontium-9% beryllium oxide (SrO-9% BeO)									
1000°C, 1000 hr		MR	MR	SR	NR			NR	NR
Strontium fluoride (SrF ₂)									
1000°C, 1000 hr	SR	SR	SR	SR	NR		SR	SR	
Strontium disilicide (SrSi ₂)									
1000°C, 168 hr	DR	DR	DR	DR	DR		DR	DR	DR
Strontium titanate (SrTiO ₃)									
1000°C, 1000 hr	SR	SR	SR	SR	NR		NR	NR	NR
1850°C, 1000 hr					NR	SR	SR	NR	MR

Legend: NR - Negligible or no reaction (compatible)
 DR - Destructive reaction (incompatible)

SR - Slight reaction
 MR - Major reaction

A number of reactions were dependent on time and temperature; for example, Haynes 25 with strontium metal showed no attack at 1000°C up to 1000 hr but fell into the major reaction classification after 5000 and 10,000 hr at 1000°C. This indicates that Haynes 25 should probably not be considered for use with strontium metal at temperatures >1000°C and times >5000 hr.

STRONTIUM TITANATE (SrTiO_3)

For periods up to 1000 hr at 1000°C, strontium metal showed little or no reaction with any metal except N-155, in which areas of severe subsurface void formation were found up to 22 mils.

At 1000°C SrO showed only slight surface oxidation reaction with most of the metals.

Strontium fluoride (SrF_2) showed very little attack with the metals tested.

At 1000°C SrSi_2 showed extensive reaction with Hastelloy X, Haynes 25, Ni-o-nel, and N-155. Molybdenum, tantalum, and TZM were resistant to penetration by silicon metal which formed from the dissociation of SrSi_2 at 1000°C.

Strontium titanate was compatible with all the container materials exposed at 1000°C. Molybdenum and tungsten were compatible at 1850°C, while niobium and tantalum had a slight reaction at this temperature. TZM reacted to a depth of 5 mils for a major reaction classification.

More extensive compatibility tests of Haynes 25, Hastelloy C, and type 316 stainless steel with inactive SrTiO_3 and Sr_2TiO_4 have not indicated any serious interactions after 1000 and 5000 hr at 1100°C. However, these materials were attacked to a maximum depth of 0.007 in. when exposed to SrO under the same conditions. It is believed that these latter results are due to reaction of the container materials with water vapor or excess oxygen dissolved in the SrO. Mechanical properties tests of specimens exposed to all three strontium compounds did not reveal any deterioration when compared with control specimens heat treated in argon under similar conditions of time and temperature.

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6. Thermophysical Properties

a. Density

5.11 g/cm³ for pure SrTiO_3
5.03 g/cm³ for average SrTiO_3

25

b. Coefficient of thermal expansion

$\alpha = 11.2 \times 10^{-6}/^\circ\text{C}$ (100-700°C)

25

The coefficient of expansion of SrTiO_3 has been determined up to 1400°C with a Brinkman Model 5440 dilatometer. To simulate a typical radioactive

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STRONTIUM TITANATE (SrTiO₃)REFERENCE
COLUMN

strontium fuel, samples were prepared with a composition of 92 wt % Sr, 3.7 wt % Ca, 3.7 wt % Ba, and 0.3 wt % Mg. The hot-pressed SrTiO₃ sample was 1 in. in length and contained 5% excess TiO₂ over the stoichiometric ratio of SrO·TiO₂.

The following table gives the values obtained for the mean coefficient of expansion for SrTiO₃ for the temperature intervals shown. The mean coefficient of expansion, A_m , is defined as

$$A_m(t_o, t) = \Delta l / l \Delta t ,$$

where

$A_m(t_o, t)$ = mean coefficient of expansion in
the temperature interval, Δt ,
from t_o to $t^\circ\text{C}$,

Δt = temperature interval,

Δl = change in sample length over Δt ,

l = sample length at temperature $t_o^\circ\text{C}$.

Temperature interval, $^\circ\text{C}$	Mean linear coefficient of expansion for SrTiO ₃
0-100	$10.8 \times 10^{-6}/^\circ\text{C}$
0-200	$10.7 \times 10^{-6}/^\circ\text{C}$
0-300	$10.7 \times 10^{-6}/^\circ\text{C}$
0-400	$10.8 \times 10^{-6}/^\circ\text{C}$
0-500	$10.9 \times 10^{-6}/^\circ\text{C}$
0-600	$11.0 \times 10^{-6}/^\circ\text{C}$
0-700	$11.1 \times 10^{-6}/^\circ\text{C}$
0-800	$11.2 \times 10^{-6}/^\circ\text{C}$
0-900	$11.3 \times 10^{-6}/^\circ\text{C}$
0-1000	$11.4 \times 10^{-6}/^\circ\text{C}$
0-1100	$11.6 \times 10^{-6}/^\circ\text{C}$
0-1200	$11.7 \times 10^{-6}/^\circ\text{C}$
0-1300	$12.0 \times 10^{-6}/^\circ\text{C}$
0-1400	$12.1 \times 10^{-6}/^\circ\text{C}$

c. Specific heat and enthalpy

(1) Specific heat in $\text{cal g}^{-1} ^\circ\text{C}^{-1}$

$$0.154 + 0.96 \times 10^{-5}T - 2.54 \times 10^{-3}T^2 \quad (^\circ\text{K}) \quad 27$$

(2) Enthalpy in cal/mole

$$H_T - H_{298} = 28.23T + 0.88 \times 10^{-3}T^2 + 4.66 \times 10^5 T^{-1} - 10,058 \quad (^\circ\text{K}) \quad 28$$

STRONTIUM TITANATE (SrTiO₃)REFERENCE
COLUMN

d. Temperatures of phase transformations

(1) Melting point

2040°C

29

(2) Boiling point

2500-3000°C

10

(3) Phase changes at 1440 and 1640°C

29

e. Latent heats of phase transformations

ΔH vaporization = 71 kcal/mole (calculated from
Trouton's rule)

12

f. Vapor pressure

g. Thermal conductivity

(1) SrO:TiO₂ = 1.00

30

The following equation was derived from data
by a method of least mean squares:

$$1/k = 55.76 + 0.0673T \text{ (}^\circ\text{K)}$$

where k is the thermal conductivity in
cal cm⁻¹ sec⁻¹ °C⁻¹. The data covered
the range 350 to 900°C. The useful range
of equation is 200 to 1400°C.

Thermal conductivity, ^a cal cm ⁻¹ sec ⁻¹ °C ⁻¹	Temperature, °C
0.0114 ± 0.0015	200
0.0099 ± 0.0003	400
0.0088 ± 0.0001	600
0.0078 ± 0.0001	800
0.0071 ± 0.0002	1000
0.0065 ± 0.0003	1200
0.0060 ± 0.0003	1400

^aDetermined on theoretically dense pellets
of stoichiometric SrTiO₃ with a composi-
tion of 51.03 wt % SrO, 1.92 wt % BaO,
2.42 wt % CaO, 0.28 wt % MgO, and 44.35
wt % TiO₂ simulating a freshly prepared
Hanford product.

(2) Thermal conductivity of various SrO-TiO₂
compositions

30, 31

The curves in Fig. 1 were obtained by
interpolating the data for the stoichiometric
SrO, Sr₂TiO₄, SrTiO₃, and TiO₂.

Fig. 1. Thermal Conductivity of Various SrO-TiO₂ Compositions.

- (3) Thermal conductivity of radioactive strontium titanate

32, 33

Data were obtained on three hot-pressed pellets having 92, 92, and 98% theoretical densities with TiO₂/alkaline-earth oxide mole ratios of 0.86, 0.95, and 1.25. The ages of the SrTiO₃ in the pellets were 0.5, 3.5, and 4.25 years. It was found that the age of SrTiO₃ and the variation in pellet composition and theoretical density did not have an effect on the results within the limit of experimental error. A single equation for the thermal conductivity as a function of temperature was derived from the data by the method of least mean squares:

$$1/k = 82.65 + 0.112T \text{ (}^\circ\text{K)}$$

where k is the thermal conductivity in $\text{cal cm}^{-1} \text{ sec}^{-1} \text{ }^\circ\text{C}^{-1}$. The data were measured

STRONTIUM TITANATE (SrTiO₃)REFERENCE
COLUMN

over a range of 350 to 850°C and are considered usable from 200 to 1400°C. Values of k as a function of temperature are given below at the 95% confidence level.

Thermal conductivity, cal cm ⁻¹ sec ⁻¹ °C ⁻¹	Temperature, °C
0.00737 ± 0.00090	200
0.00632 ± 0.00025	400
0.00554 ± 0.00015	600
0.00493 ± 0.00016	800
0.00444 ± 0.00023	1000
0.00403 ± 0.00028	1200
0.00370 ± 0.00031	1400

h. Thermal diffusivity

(1) 100% dense inactive SrTiO₃ pellet

Thermal diffusivity, ^a cm ² /sec	Temperature, °C
0.0151	200
0.0124	400
0.0107	600
0.0093	800
0.0083	1000
0.0075	1200
0.0068	1400

^aThese values were calculated by dividing the thermal conductivity by the product of the specific heat and the density.

(2) Thermal diffusivity of radioactive strontium titanate (94.7% of theoretical density)

Thermal diffusivity, ^a cm ² /sec	Temperature, °C
0.01068	200
0.00869	400
0.00739	600
0.00645	800
0.00574	1000
0.00511	1200
0.00461	1400

^aThese values were calculated by dividing the thermal conductivity by the product of the specific heat and the density.

STRONTIUM TITANATE (SrTiO_3)REFERENCE
COLUMN

- i. Viscosity
- j. Surface tension
- k. Total hemispherical emittance
- l. Spectral emissivity
- m. Crystallography
- Cubic: $a = 3.899 \text{ kX}$

20

- n. Solubilities

Leach rates were determined on five $^{90}\text{SrTiO}_3$ fuels 33, 34 having densities ranging from 62 to 92% of theoretical. The low density samples were recovered from SNAP-7A fuel capsules after seven years of operation. The high density samples were prepared by hot pressing. The samples were intact after an immersion of 76 days in distilled water. The data were fitted to a single equation by a method of least squares and are presented in Fig. 2 with the 95% confidence limits. A single equation to represent the data is as follows:

$$\ln R = -0.5235 \ln T - 2.5987$$

where

R = leach rate, $\text{mg cm}^{-2} \text{ day}^{-1}$
 T = time, days.

Fig. 2. Leach Rate of $^{90}\text{SrTiO}_3$ as a Function of Time.

STRONTIUM TITANATE (SrTiO₃)

o. Diffusion rates

7. Mechanical Properties

a. Hardness

b. Crush strength

19,100 lb/in.²35 (CaTiO₃)

c. Bend strength

43 lb/in.²

36

8. Chemical Properties

a. Heat and free energy of formation, entropy

(1) Heat of formation

 $\Delta H_f^\circ = -399$ kcal/mole

37, 38

(calculated from available data)

(2) Free energy of formation

 $\Delta F_f^\circ = -379$ kcal/mole

37, 38

(calculated from available data)

(3) Entropy

 $S_{298}^\circ = 26.0$ eu

39

b. Chemical reactions and reaction rates

40

(1) Oxygen — no reaction

(2) Nitrogen — no reaction

(3) Water — decomposes

(4) Inorganic acids — soluble

9. Biological Tolerances

The ⁹⁰Sr tolerances are given under Strontium-90 Metal Fuel Form, Section I.A.9.

10. Shielding Data

Bremsstrahlung dose rates from ⁹⁰Sr power sources of 100, 200, 500, 1000, 2000, 5000, 10,000, and 20,000 w with iron, lead, and uranium shielding are given in Figs. 3-6 for a SrTiO₃ matrix. These sources require significant shielding because of the high energy of some of the photons from the bremsstrahlung radiation.

4

Fig. 3. Bremsstrahlung Dose Rates from Unshielded Isotopic Power Sources of Strontium-90 (Strontium Titanate) as a Function of Distance from Center of Source.

Fig. 4. Bremsstrahlung Dose Rates from Iron-Shielded Isotopic Power Sources of Strontium-90 (Strontium Titanate). Center of source to dose point separation distance = 100 cm.

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Fig. 5. Bremsstrahlung Dose Rates from Lead-Shielded Isotopic Power Sources of Strontium-90 (Strontium Titanate). Center of source to dose point separation distance = 100 cm.

Fig. 6. Bremsstrahlung Dose Rates from Uranium-Shielded Isotopic Power Sources of Strontium-90 (Strontium Titanate). Center of source to dose point of separation of distance = 100 cm.

C. STRONTIUM OXIDE (SrO)1. Composition

a. Radionuclidic abundance

The radionuclidic abundances are given in Section I.A.1.

b. Radiochemical purity

The radiochemical purity is given in Section I.A.1.

c. Chemical composition*

(1) Range of composition

<u>Cation</u>	<u>Maximum, wt %</u>	<u>Minimum, wt %</u>
Sr	97	92
Ca	5	2
Ba	2	0.5
Mg	2	0

One gram of ^{90}Sr will produce an average of 0.0232 g of ^{90}Zr after one year's decay.

(2) Average chemical composition

<u>Cation</u>	<u>Metal, wt %</u>	<u>Metal oxide, wt %</u>
Sr	95	94.3
Ca	3.5	4.1
Ba	1.0	0.9
Mg	0.5	0.7

The strontium metal content of pure SrO is 84.77%.

The strontium metal content of average SrO is 79.9%.

The as-processed SrO will contain 44.0% ^{90}Sr . The ^{144}Ce constitutes 0.03% of the radioactivity at a maximum. The ^{137}Cs has been found to be as high as 0.003% of the radioactivity at a maximum. The ^{106}Ru activity has not been detected. These activities can be neglected compared to the total radiation.

2. Specific Power

2, 4, 5

- a. 0.435 w/g of pure SrO (46.6% ^{90}Sr metal)
0.411 w/g of average SrO (44.0% ^{90}Sr metal)

It is assumed that there are 139 Ci/g of ^{90}Sr and 148.9 Ci of ^{90}Sr per thermal watt.

- b. 64.8 Ci of ^{90}Sr per gram of pure SrO (46.6% ^{90}Sr metal)
61.2 Ci of ^{90}Sr per gram of average SrO (44.0% ^{90}Sr metal)

*Results of analyses on strontium titanate feed samples of the Fission Products Development Laboratory at Oak Ridge National Laboratory.

3. Radiation

The radiation will be the same as given under Section I.A.3.

4. Critical Mass

Strontium-90 and ^{90}Y are not fissionable.

5. Compatibility with Materials of Containment

See Section I.B.5.

6. Thermophysical Properties

a. Density

$$\rho = 4.975 \text{ g/cm}^3$$

41

b. Coefficient of thermal expansion

The coefficient of expansion of SrO has been determined up to 1400°C with a Brinkman Model 5440 dilatometer. To simulate a typical radioactive strontium fuel, samples were prepared with a composition of 92 wt % Sr, 3.7 wt % Ca, 3.7 wt % Ba, and 0.3 wt % Mg. The SrO sample was 1/2 in. in length and was prepared by hot pressing.

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The following table gives the values obtained for the mean coefficient of expansion for SrO for the temperature intervals shown. The mean coefficient of expansion, A_m , is defined as

$$A_m(t_0, t) = \Delta l / l \Delta t ,$$

where

$A_m(t_0, t)$ = mean coefficient of expansion in the temperature interval, Δt , from t_0 to $t^\circ\text{C}$,

Δt = temperature interval,

Δl = change in sample length over Δt ,

l = sample length at temperature $t_0^\circ\text{C}$.

Temperature interval, °C	Mean linear coefficient of expansion for SrO
0-100	$11.0 \times 10^{-6}/^\circ\text{C}$
0-200	$11.0 \times 10^{-6}/^\circ\text{C}$
0-300	$11.1 \times 10^{-6}/^\circ\text{C}$
0-400	$11.1 \times 10^{-6}/^\circ\text{C}$
0-500	$11.1 \times 10^{-6}/^\circ\text{C}$
0-600	$11.1 \times 10^{-6}/^\circ\text{C}$
0-700	$11.1 \times 10^{-6}/^\circ\text{C}$

STRONTIUM OXIDE (SrO)REFERENCE
COLUMN

Table continued

Temperature interval, °C	Mean linear coefficient of expansion for SrO
0-800	$11.2 \times 10^{-6}/^{\circ}\text{C}$
0-900	$11.2 \times 10^{-6}/^{\circ}\text{C}$
0-1000	$11.2 \times 10^{-6}/^{\circ}\text{C}$
0-1100	$11.2 \times 10^{-6}/^{\circ}\text{C}$
0-1200	$11.3 \times 10^{-6}/^{\circ}\text{C}$
0-1300	$11.3 \times 10^{-6}/^{\circ}\text{C}$
0-1400	$11.3 \times 10^{-6}/^{\circ}\text{C}$

c. Specific heat and enthalpy

(1) Specific heat in $\text{cal g}^{-1} ^{\circ}\text{C}^{-1}$

$$0.117 + 1.22 \times 10^{-5}T - 1.50 \times 10^{-8}T^2 \quad 14$$

(T is in °K)

(2) Enthalpy in cal/mole

$$H_T - H_{298} = 12.13T + 0.63 \times 10^{-3}T^2 \quad 14$$

$$+ 1.55 \times 10^{-5}T^3 - 4192$$

(T is in °K)

d. Temperatures of phase transformations

(1) Melting point

$$2457^{\circ}\text{C} \quad 42$$

(2) Boiling point

$$3227^{\circ}\text{C} \quad 42$$

e. Latent heats of phase transformations

$$\Delta H \text{ fusion} \quad 16.7 \text{ kcal/mole} \quad 43$$

 ΔH vaporization (not available)

f. Vapor pressure

$$\log P = 3.07 \times 10^4 T^{-1} + 13.12 \quad 44$$

(T is in °K and P is in atm)

g. Thermal conductivity

(1) SrO = 100% 30

The following equation was derived from data by a method of least mean squares:

$$1/k = 109.51 + 0.0916T (^{\circ}\text{K})$$

where k is the thermal conductivity in $\text{cal cm}^{-1} \text{ sec}^{-1} ^{\circ}\text{C}^{-1}$. The data covered the range 350 to 900°C. Useful range of the equation is 200 to 1400°C.

STRONTIUM OXIDE (SrO)REFERENCE
COLUMN

Thermal conductivity, ^a cal cm ⁻¹ sec ⁻¹ °C ⁻¹	Temperature, °C
0.0066 ± 0.0006	200
0.0058 ± 0.0003	400
0.0053 ± 0.0001	600
0.0048 ± 0.0002	800
0.0044 ± 0.0003	1000
0.0041 ± 0.0004	1200
0.0038 ± 0.0004	1400

^aDetermined on theoretically dense pellets of SrO with a composition of 91.69 wt % SrO, 3.46 wt % BaO, 4.34 wt % CaO, and 0.51 wt % MgO simulating a freshly prepared Hanford product.

- (2) Thermal conductivity of various SrO-TiO₂ compositions

30, 31

These values are given in Section I.B.6.g.(2).

h. Thermal diffusivity

Thermal diffusivity, ^a cm ² /sec	Temperature, °C
0.0122	200
0.0103	400
0.0091	600
0.0080	800
0.0072	1000
0.0066	1200
0.0060	1400

^aThese values were calculated by dividing the thermal conductivity by the product of the specific heat and the density.

i. Viscosity

j. Surface tension

1200 dyn/cm

45 (MgO)

k. Total hemispherical emittance

STRONTIUM OXIDE (SrO)REFERENCE
COLUMN

1. Spectral emissivity

Total emissivity	Temperature, °C	44 (BeO)
0.36	1000	
0.46	1400	
0.50	1800	

Small amounts of impurities such as carbon or radiation darkening can increase these values significantly to ~0.9.

m. Crystallography

Cubic, face centered: $a = 5.1396 \text{ kX}$ 20

n. Solubilities

Reacts with water or HCl and goes into solution. 20

The dissolution rate studies were performed using stable SrO pellets prepared by hot pressing at 1300°C and 4000 psi for 1 hr. In one test, a specimen weighing 43.962 g with an apparent density of 4.54 g/cm³ was suspended from an analytical balance beam in a container through which a constant linear flow rate of 4.4 cm/min of distilled water was maintained. Periodic weighings of the specimen were recorded during a 7-day span. The dissolution rates for SrO calculated from the data in the experiment are shown in the following table. The values were calculated on the basis of the initial surface area of the pellet. These solubilities are a factor of 10⁴ to 10⁵ greater than those of strontium titanate. 46

Rate, g/cm ² .day	Time interval, days
1.485	0-1
1.22	1-2
1.22	2-3
1.24	3-4
0.96	4-5
1.12	5-6
0.76	6-7

o. Diffusion rates

STRONTIUM OXIDE (SrO)7. Mechanical Properties

a. Hardness

3.5 mohs

47

b. Crush strength

8. Chemical Properties

a. Heat and free energy of formation, entropy

(1) Heat of formation

$$\Delta H_f^\circ = -141.0 \text{ kcal/mole} \quad 48$$

$$\Delta H_f^\circ = -144.2 \pm 1.5 \text{ kcal/mole} \quad 27$$

$$\Delta H_f^\circ = -140.50 \pm 0.20 \text{ kcal/mole} \quad 49$$

(2) Free energy of formation

$$\Delta F_f^\circ = -133.8 \text{ kcal/mole} \quad 48$$

(3) Entropy

$$S_{298}^\circ = 13.0 \text{ eu} \quad 27, 48$$

b. Chemical reactions and reaction rates

50

(1) Oxygen - no reaction

(2) Nitrogen - no reaction

(3) Water - forms $\text{Sr}(\text{OH})_2$

(4) Inorganic acids - soluble

(5) CO_2 - reacts to form SrCO_3 9. Biological Tolerances

The ^{90}Sr tolerances are given under Strontium-90 Metal Fuel Form, Section I.A.9.

10. Shielding Data

Bremsstrahlung dose rates from ^{90}Sr power sources of 100, 200, 500, 1000, 2000, 5000, 10,000, and 20,000 w with iron, lead, and uranium shielding are given in Figs. 7-10 for a SrO matrix. These sources require significant shielding because of the high energy of some of the photons from the bremsstrahlung radiation. 4

Fig. 7. Bremsstrahlung Dose Rates from Unshielded Isotopic Power Sources of Strontium-90 (Strontium Oxide) as a Function of Distance from Center of Source.

ORNL DWG 63-8459

Fig. 8. Bremsstrahlung Dose Rates from Iron-Shielded Isotopic Power Sources of Strontium-90 (Strontium Oxide). Center of source to dose point separation distance = 100 cm.

Fig. 9. Bremsstrahlung Dose Rates from Lead-Shielded Isotopic Power Sources of Strontium-90 (Strontium Oxide). Center of source to dose point separation distance = 100 cm.

Fig. 10. Bremsstrahlung Dose Rates from Uranium-Shielded Isotopic Power Source of Strontium-90 (Strontium Oxide). Center of source to dose point separation distance = 100 cm.

D. STRONTIUM FLUORIDE (SrF₂)1. Composition

a. Radionuclidic abundance

The radionuclidic abundances are given in Section I.A.1.

b. Radiochemical purity

The radiochemical purity is given in Section I.A.1.

c. Chemical composition*

(1) Range of composition

<u>Cation</u>	<u>Maximum, wt %</u>	<u>Minimum, wt %</u>
Sr	97	92
Ca	5	2
Ba	2	0.5
Mg	2	0

One gram of ⁹⁰Sr will produce an average of 0.0232 g of ⁹⁰Zr after one year's decay.

(2) Average chemical composition

<u>Cation</u>	<u>Metal, wt %</u>	<u>Metal fluoride (MF₂), wt %</u>
Sr	95	93.5
Ca	3.5	4.7
Ba	1.0	0.9
Mg	0.5	0.9

The strontium metal content of pure SrF₂ is 70.10%.

The strontium metal content of average SrF₂ is 65.5%.

The as-processed SrF₂ will contain 36.0% ⁹⁰Sr. The ¹⁴⁴Ce constitutes 0.03% of the radioactivity at a maximum. The ¹³⁷Cs has been found to be as high as 0.003% of the total activity. The ¹⁰⁶Ru activity has not been found. These activities can be neglected compared to the bremsstrahlung radiation.

2. Specific Power

2, 4, 5

- a. 0.360 w/g of pure SrF₂ (38.6% ⁹⁰Sr metal)
0.336 w/g of average SrF₂ (36.0% ⁹⁰Sr metal)

It is assumed that there are 139 Ci/g of ⁹⁰Sr and 148.9 Ci of ⁹⁰Sr per thermal watt.

- b. 53.7 Ci of ⁹⁰Sr per gram of pure SrF₂ (38.6% ⁹⁰Sr metal)
50.0 Ci of ⁹⁰Sr per gram of average SrF₂ (36.0% ⁹⁰Sr metal)

*Results of analyses on strontium titanate feed samples of the Fission Products Development Laboratory at Oak Ridge National Laboratory.

STRONTIUM FLUORIDE (SrF₂)REFERENCE
COLUMN3. Radiation

The radiation will be the same as given under Section I.A.3.

4. Critical Mass

Strontium-90 and ⁹⁰Y are not fissionable.

5. Compatibility with Materials of Containment

Metallographic examination of inactive SrF₂ test specimens exposed for 19 months at 925°C gave the following results.

36

Encapsulating material	Penetration, in.	
	Maximum	Average
<u>Simulated SrF₂ fuel^a</u>		
Haynes 25	0.0006	nil
Ta liner in Haynes 25	Complete penetration of Ta liner	
Hastelloy C	0.001	0.0004
Hastelloy N	0.002	0.001
Hastelloy X	0.0075	0.0014
<u>Simulated aged feed material^b</u>		
Haynes 25	0.0015	0.0008
Ta liner in Haynes 25	Complete penetration of Ta liner	
Hastelloy C	0.0007	0.0003
Hastelloy N	0.0028	0.0011
Hastelloy X	0.0055	0.0011

^a1 wt % Ca, 0.5 wt % Fe.

^bSrF₂ plus 2-1/2 wt % Zr as ZrO₂.

Also see Section I.B.5.

6. Thermophysical Properties

a. Density

51

4.29 g/cm³ for pure SrF₂

4.20 g/cm³ for average SrF₂

b. Coefficient of thermal expansion

$\alpha/^{\circ}\text{C} = 15.72 \times 10^{-6} - 0.63 \times 10^{-8}t + 1.17 \times 10^{-10}t^2$

52

(t is in °C; temperature range is 28-284°C)

STRONTIUM FLUORIDE (SrF₂)REFERENCE
COLUMN

c. Specific heat and enthalpy

(1) Specific heat in cal g⁻¹ °C⁻¹

$$0.125 + 5.57 \times 10^{-5}T$$

43

(T is in °K)

(2) Enthalpy in cal/mole

$$H_T - H_{298} = 15.7T + 3.5 \times 10^{-3}T^2 - 4980$$

43

(T is in °K)

d. Temperatures of phase transformations

(1) Melting point

1463°C

53

(2) Boiling point

2477°C

43

e. Latent heats of phase transformations

 ΔH fusion 4.3 kcal/mole

43

 ΔH vaporization 71 kcal/mole

43

f. Vapor pressure

Vapor pressure, torr	Temperature, °C
8.5	1822
15.0	1880
25.7	1927
31.0	1959

20

g. Thermal conductivity

Thermal conductivity, cal cm ⁻¹ sec ⁻¹ °C ⁻¹	Temperature, °C
0.0120	20
0.0055	300
0.0030	725

54

h. Thermal diffusivity

Thermal diffusivity, cm ² /sec	Temperature, °C
0.0219	20
0.0091	300
0.0043	725

These values were calculated by dividing the thermal conductivity by the product of the specific heat and a density of 3.86 g/cm³.

STRONTIUM FLUORIDE (SrF₂)REFERENCE
COLUMN

i. Viscosity	
Log $\eta = 0.781 + 436/T$	55
(η is in centipoises and T is temperature in °K)	
j. Surface tension	
184 dyn/cm	56
k. Total hemispherical emittance	
l. Spectral emissivity	
A value of 0.9 can be assumed.	18
m. Crystallography	
Cubic: $a = 5.781 \text{ \AA}$	51
n. Solubilities	
0.0009 mole/liter in water 18°C	20
o. Diffusion rates	
7. <u>Mechanical Properties</u>	
a. Hardness, moh	
$H_M = 3.5$	20
b. Crush strength	
8. <u>Chemical Properties</u>	
a. Heat and free energy of formation, entropy	
(1) Heat of formation	
$\Delta H_f^\circ = -290.3 \text{ kcal/mole}$	48
(2) Free energy of formation	
$\Delta F_f^\circ = -277.2 \text{ kcal/mole}$	48
(3) Entropy	
$S_{298}^\circ = 20.5 \text{ eu}$	43
b. Chemical reactions and reaction rates	50
(1) Nitrogen — no reaction	
(2) Water — slightly soluble	
(3) Inorganic acids — insoluble	

STRONTIUM FLUORIDE (SrF_2)9. Biological Tolerances

The ^{90}Sr tolerances are given under Strontium-90 Metal Fuel Form, Section I.A.9.

10. Shielding Data

See the sections under Strontium Titanate, I.B.10, and Strontium Oxide, I.C.10, for shielding data.

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CERIUM-144 DATA SHEETS

FUEL FORM (as processed)

REFERENCE COLUMNCEROUS OXIDE (Ce₂O₃)

HALF-LIFE: 285 d

1

Composition

Radionuclidic abundance

With one year of cooling time after reactor discharge, the isotopic composition will be: 2, 3

<u>Isotope</u>	<u>% Abundance</u>
¹⁴⁴ Ce	13.3
¹⁴² Ce	42.7
¹⁴⁰ Ce	44.0

At this isotopic composition, the specific activity of the cerium isotopes is 430 curies of ¹⁴⁴Ce per gram of cerium isotope or $7.91 \times 0.43 = 3.40$ watts/g of cerium isotope. The specific activity of the cerium isotopes decreases with increasing cooling time after reactor discharge as shown in the following table.

Specific Activity of ¹⁴⁴Ce per Gram of Cerium Isotopes
as a Function of Cooling Time After Reactor Discharge
With an Irradiation Time of 100 Days

<u>Cooling time, days</u>	<u>Specific activity, curies of ¹⁴⁴Ce per g of cerium isotope</u>
50	812
100	760
200	594
300	500
350	456
400	408
450	362
500	322

Radiochemical purity

The principal radiochemical impurity present in the ¹⁴⁴Ce product is ¹⁴¹Ce ($T_{1/2} = 32.5$ d), whose content depends on the length of time since reactor discharge as given in the following table.

REFERENCE COLUMN

Calculated Ratio of $^{141}\text{Ce}/^{144}\text{Ce}$ Activities
as a Function of Cooling Time Since Reactor
Discharge With an Irradiation Time of 200 Days

Cooling time, days	Calculated ratio of $^{141}\text{Ce}/^{144}\text{Ce}$ activities
50	0.98
100	0.38
150	0.15
200	0.06

The ^{141}Ce content is almost negligible after 200 days of cooling time.

The ^{147}Pm , ^{90}Sr , and ^{137}Cs contaminants are negligible compared with the ^{144}Ce - ^{144}Pr radiation.

Cation composition

After purification, the cerium product contains >99% cerium and <1% neodymium. Neodymium also grows in from the decay of ^{144}Ce as shown in the following table. Hence, it is assumed that an average cerium product contains 95% cerium and 5% neodymium.

Time, days	^{144}Ce , %	^{142}Ce , %	^{140}Ce , %	^{144}Nd , %
0	13.3	42.7	44.0	0
57	11.6	42.7	44.0	1.7
114	10.1	42.7	44.0	3.2
171	8.7	42.7	44.0	4.6
228	7.6	42.7	44.0	5.7
285	6.65	42.7	44.0	6.65

Specific Power (based on fabrication 1 yr after reactor discharge)

2.91 watts/g of pure Ce_2O_3 (85.4% Ce metal) 2
2.76 watts/g of 95% Ce_2O_3 -5% Nd_2O_3

367 curies ^{144}Ce per gram of pure Ce_2O_3 (85.4% Ce metal)
349 curies ^{144}Ce per gram of 95% Ce_2O_3 -5% Nd_2O_3

REFERENCE COLUMNRadiation

Alpha particles - none

Beta particles

4

Nuclide	Max E, Mev	Avg E, Mev	Abundance, %	w/kilocurie	Particles w ⁻¹ sec ⁻¹
¹⁴⁴ Ce	0.31	0.090	65	0.44	3.03 x 10 ¹²
	0.22	0.060	5		0.233 x 10 ¹²
	0.17	0.045	30		1.40 x 10 ¹²
¹⁴⁴ Pr	2.98	1.21	97.7	7.08	4.55 x 10 ¹²
	2.3	0.89	1.3		0.061 x 10 ¹²
	0.8	0.28	1.0		0.047 x 10 ¹²
Total beta power				7.52	

Gamma

4

Nuclide	Energy, Mev	Abundance, %	w/kilocurie	Particles w ⁻¹ sec ⁻¹
¹⁴⁴ Ce	0.134	17	0.20	0.792 x 10 ¹²
	0.10	2		0.093 x 10 ¹²
	0.08	2		0.093 x 10 ¹²
	0.04	17		0.792 x 10 ¹²
¹⁴⁴ Pr	2.2	0.8	0.19	0.037 x 10 ¹²
	1.5	0.25		0.0117 x 10 ¹²
	0.7	1.6		0.075 x 10 ¹²
Total gamma power			0.39	
Total beta power			7.52	
Total gamma and beta power			7.91	

Bremsstrahlung

4

The production of bremsstrahlung photons from the 3.0-Mev ¹⁴⁴Pr beta in a cerium oxide matrix is given in the table on the following page.

Neutrons - none

Production of Bremsstrahlung Photons From
Praseodymium-144 Beta in Cerium Oxide Matrix

Maximum beta-particle energy   3.00 Mev
Average beta-particle energy   1.23 Mev

Bremsstrahlung energy group, Mev	Within ΔE energy group	
	Photons/beta particle	Photons $w^{-1} \text{ sec}^{-1}$
0.100 $\pm$ 0.05	9.016×10^{-2}	4.10×10^{11}
0.200 $\pm$ 0.05	3.703×10^{-2}	1.68×10^{11}
0.300 $\pm$ 0.05	2.054×10^{-2}	9.35×10^{10}
0.400 $\pm$ 0.05	1.289×10^{-2}	5.86×10^{10}
0.500 $\pm$ 0.05	8.620×10^{-3}	3.92×10^{10}
0.600 $\pm$ 0.05	6.001×10^{-3}	2.73×10^{10}
0.700 $\pm$ 0.05	4.284×10^{-3}	1.95×10^{10}
0.800 $\pm$ 0.05	3.107×10^{-3}	1.41×10^{10}
0.900 $\pm$ 0.05	2.277×10^{-3}	1.04×10^{10}
1.000 $\pm$ 0.05	1.677×10^{-3}	7.63×10^9
1.100 $\pm$ 0.05	1.238×10^{-3}	5.63×10^9
1.200 $\pm$ 0.05	9.123×10^{-4}	4.15×10^9
1.300 $\pm$ 0.05	6.698×10^{-4}	3.05×10^9
1.400 $\pm$ 0.05	4.883×10^{-4}	2.22×10^9
1.500 $\pm$ 0.05	3.527×10^{-4}	1.60×10^9
1.600 $\pm$ 0.05	2.516×10^{-4}	1.14×10^9
1.700 $\pm$ 0.05	1.766×10^{-4}	8.04×10^8
1.800 $\pm$ 0.05	1.216×10^{-4}	5.53×10^8
1.900 $\pm$ 0.05	8.176×10^{-5}	3.72×10^8
2.000 $\pm$ 0.05	5.334×10^{-5}	2.43×10^8
2.100 $\pm$ 0.05	3.353×10^{-5}	1.53×10^8
2.200 $\pm$ 0.05	2.011×10^{-5}	9.15×10^7
2.300 $\pm$ 0.05	1.135×10^{-5}	5.16×10^7
2.400 $\pm$ 0.05	5.915×10^{-6}	2.69×10^7
2.500 $\pm$ 0.05	2.760×10^{-6}	1.26×10^7
2.600 $\pm$ 0.05	1.095×10^{-6}	4.98×10^6
2.700 $\pm$ 0.05	3.363×10^{-7}	1.53×10^6
2.800 $\pm$ 0.05	6.451×10^{-8}	2.94×10^5
2.900 $\pm$ 0.05	3.898×10^{-9}	1.77×10^4
3.000 $\pm$ 0.05	0.000	
Total bremsstrahlung energy, Mev/beta particle		5.075×10^{-2}

REFERENCE COLUMNCritical Mass

Cerium-144 and ^{144}Pr are not fissionable.

Compatibility With Materials of ContainmentThermophysical Properties

Density

6.867 grams/cm³ - density of pure Ce₂O₃ 5

6.89 grams/cm³ - density of 95% Ce₂O₃-5% Nd₂O₃ 5

Coefficient of thermal expansion

Linear coefficient
of expansion, °C⁻¹

Temperature, °C

8.90 x 10 ⁻⁶	300	6 (Nd ₂ O ₃)
10.60 x 10 ⁻⁶	500	
11.35 x 10 ⁻⁶	800	
11.41 x 10 ⁻⁶	1050	
11.0 x 10 ⁻⁶	268	7 (CeO ₂)
11.1 x 10 ⁻⁶	415	
11.3 x 10 ⁻⁶	503	
11.5 x 10 ⁻⁶	616	
11.7 x 10 ⁻⁶	720	
12.0 x 10 ⁻⁶	845	
12.3 x 10 ⁻⁶	975	
12.7 x 10 ⁻⁶	1131	
12.9 x 10 ⁻⁶	1310	

Specific heat and enthalpy

Specific heat in cal g⁻¹ °C⁻¹

$$9.24 \times 10^{-2} + (2.35 \times 10^{-5} T) - (6.76 \times 10^{-2} T^{-2}) \quad 8$$

(T is in °K)

Enthalpy in cal/mole for β-form Ce₂O₃

$$H_T - H_{298} = 30.30 T + (3.86 \times 10^{-3} T^2) \quad 8$$

$$+ (2.22 \times 10^5 T^{-1}) - 10,122$$

(T is in °K)

Temperatures of phase transformations

Melting point - 2190°C 9
2142 (in H₂) 10

Boiling point - 3230°C 11

REFERENCE COLUMN

Latent heats of phase transformations

 ΔH (fusion) 20 kcal/mole 11 ΔH (vaporization) 80 kcal/mole 11

Vapor pressure

 $\log_{10} P = -24,000/T + 5.56$ 12 (Nd_2O_3)

(T in °K and P in atm)

Thermal conductivity

<u>cal cm⁻¹ sec⁻¹ °C⁻¹</u>	<u>Temperature, °C</u>	
0.0245	100	13 (ThO_2)
0.0204	200	
0.0143	400	
0.0104	600	
0.0081	800	
0.0073	1000	
0.0060	1200	
0.0059	1400	

Thermal diffusivity

<u>cm²/sec</u>	<u>Temperature, °C</u>
0.0443	100
0.0235	400
0.0118	800
0.0074	1400

Calculated by dividing the product of the specific heat and density into the thermal conductivity.

Viscosity

Surface tension

Total hemispherical emittance

Spectral emissivity

0.20 to 0.57 14 (ThO_2)

The exact value of the emissivity will depend on the roughness of the surface of the material, the presence of impurities, and the effects of radiation.

REFERENCE COLUMN

Crystallography

hexagonal

5

a = 3.888 Å   c = 6.062 Å

Solubilities

Water — insoluble

14

Sulfuric acid — soluble

Diffusion rates

Mechanical Properties

Hardness

5 Mohs

15

Crush strength

1900 kg/cm²15 (CeO₂)Chemical Properties

Heat and free energy of formation, entropy

Heat of formation

 $\Delta H^\circ_f = -435$ kcal/mole

7

Free energy of formation

 $\Delta F^\circ_f = -415$ kcal/mole

7

Entropy

 $S^\circ_{298} = 33.5$ eu

7

Chemical reactions and reaction rates

16

(oxygen, nitrogen, water, steam, hydrogen,
liquid metals, other)

Oxygen at room temperature — will react

Oxygen at elevated temperature — pyrophoric

Nitrogen — no reaction

Water — reacts

Inorganic acids — soluble

Biological Tolerances

17

Maximum permissible body burdens and maximum
permissible concentration of ¹⁴⁴Ce-¹⁴⁴Pr in air
and water were taken from Ref. 17 and are given in
the table on the following page.

Maximum Permissible Body Burdens and Maximum Permissible Concentrations
for Radionuclides in Air and in Water for Occupational Exposure¹⁷

Radionuclide and type of decay	Organ of reference (critical organ underscored)	Max. permissible burden in total body, q(μ c)	Maximum permissible concentrations, μ c/cm ³			
			For 40-hr week		For 168-hr week	
			Water	Air	Water	Air
⁵⁸ Ce ¹⁴⁴ (α, β^-, γ)	(Sol) {	<u>GI (LLI)*</u>	<u>3×10^{-4}</u>	8×10^{-8}	<u>10^{-4}</u>	3×10^{-8}
		<u>Bone</u>	0.2	<u>10^{-8}</u>	0.08	<u>3×10^{-9}</u>
		<u>Liver</u>	0.3	<u>10^{-8}</u>	0.1	4×10^{-9}
		Kidney	10	2×10^{-8}	0.2	7×10^{-9}
		Total Body	20	3×10^{-8}	0.3	10^{-8}
	(Insol) {	<u>Lung</u>		<u>6×10^{-9}</u>		<u>2×10^{-9}</u>
		<u>GI (LLI)*</u>	<u>3×10^{-4}</u>	6×10^{-8}	<u>10^{-4}</u>	2×10^{-8}

*The abbreviations GI and LLI refer to the gastrointestinal tract and lower large intestine, respectively.

Shielding Data

The combined gamma and bremsstrahlung dose rates from ^{144}Ce - ^{144}Pr power sources of 100, 200, 500, 1000, 2000, 5000, 10,000, and 20,000 watts with iron, lead, and uranium shielding are given in Figs. 1 to 4.

4

Fig. 1. Gamma Plus Bremsstrahlung Dose Rates From Unshielded Isotopic Power Sources of Cerium-144 as a Function of Distance From Center of Source.

ORNL DWG 63-8474

Fig. 2. Gamma Plus Bremsstrahlung Dose Rates from Iron-Shielded Isotopic Power Sources of Cerium-144. Center of source to dose point separation distance = 100 cm.

Fig. 3. Gamma Plus Bremsstrahlung Dose Rates from Lead-Shielded Isotopic Power Sources of Cerium-144. Center of source to dose point separation distance = 100 cm.

ORNL DWG 63-8476

Fig. 4. Gamma Plus Bremsstrahlung Dose Rates from Uranium-Shielded Isotopic Power Sources of Cerium-144. Center of source to dose point separation distance = 100 cm.

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COBALT-60

The physical properties for ^{60}Co , shown below, were measured for the non-radioactive isotope ^{59}Co . The effects of irradiation and decay on physical properties with ingrowth of up to 20% nickel have been shown to be minor.

Radiation Properties of Cobalt-60 for Heat Sources^a

Property	Value
Half-life	5.14 yr
Beta energy	
Maximum	0.312 Mev
Average	0.095 Mev
Gamma energies	1.172 Mev 1.333 Mev
Total recoverable energy	2.60 Mev
Curies per recoverable watt	64.2 curies

^aW. C. Windley, Jr., Properties of ^{60}Co and Cobalt Metal Fuel Forms, DP-1051 (Rev. 1) (1966).

Properties of Cobalt Metal Fuel Forms^a

Density	8.8 g/cm ³ (solid forms)	
Phase transformations	417 ± 7°C - transformation from α to ϵ phase is sluggish and typically occurs at 390°C on cooling or 430°C on heating; volume increase of 0.30 ± 0.06% occurs on heating from α to ϵ phase. 1495°C - melting point (>1480°C as nickel grows in) 3100°C - boiling point	
Specific heat	Temperature, °C	Specific heat, cal/g·°C
	0	0.099
	100	0.106
	500	0.126
	1100	0.204
	1400	0.168
	Molten	0.141

Properties of Cobalt Metal Fuel Forms^a (Continued)

Thermal conductivity	<u>Temperature, °C</u>	<u>Conductivity, cal/sec·cm·°C</u>
	50	0.22
	100	0.20
	150	0.19 _b
	500	0.13 _b
	1000	0.09 _b
Coefficient of thermal expansion	14.2 μin./in. ·°C between 20 and 300°C	
	15.7 μin./in. ·°C at 400°C	
	16.8 μin./in. ·°C at 750°C	
Oxidation rate in air	<u>Temperature, °C</u>	<u>Parabolic scaling constant, (mg/cm²)²/hr</u>
	400	0.0004
	600	0.1
	800	2.1
	1000	70.0
	1200	400.0

^aW. C. Windley, Jr., Properties of ⁶⁰Co and Cobalt Metal Fuel Forms,

^bCalculated from electrical resistivity.

Appendix B

METHOD OF CALCULATING FUEL, CLADDING, AND
HEAT BLOCK TEMPERATURES

The temperature distribution in the fuel, cladding, and inner portions of the heat block was calculated by means of Hexpintemp-2, a computer code written for this project. This code handles only two dimensions, using $R\theta$ geometry, and implicitly assumes that axial heat flow is small compared with radial heat flow and may be neglected. The code calculates the temperature distribution, taking into account heat conduction, radiation, heat generation, and the prescribed outer boundary condition. The thermal conductivity of any material may be temperature dependent; it is represented by a polynomial of the 4th degree or less in the temperature. Radiation between fuel and cladding and between cladding and the heat block is calculated on the assumption that the radiating surfaces are concentric circular cylinders; no account was taken of the possibility of the fuel pin being off-center. The heat generation rate may be space-dependent, and in particular the heat generated by absorption of gamma radiation is calculated for every location in the block by means of the buildup factor method.¹ The outer boundary condition may be either a prescribed temperature or a prescribed temperature gradient; the first was used in all cases except when there were coolant channels, in which case a zero gradient at the outer boundary was assumed. No account was taken of thermal expansion of fuel, cladding, or heat block.

Hexpintemp-2 is a development from HETRARTH (Heat Transfer in R-Theta Geometry), which was developed from code HETRARZ.² The method by which a code basically using $R\theta$ geometry was adapted for calculations involving circular pins on a hexagonal lattice is described in some detail below.

For the purpose of these calculations the following model was used. Each fuel pin is circular, with a central circular fuel region, surrounded successively by a gas gap, and the cladding. The pin is inserted into a hole in the heat block, with another gas gap existing between the cladding and the heat block. Radiation and conduction both occur across the gas gaps. Each pin is at the center of one hexagon of a regular hexagonal array. A circle is inscribed in the hexagon; within each circle, temperature is calculated by using difference equations proper for $R\theta$ geometry. (These circles are, of course, tangent.) The problem in adapting an $R\theta$ code to a hexagonal lattice is to handle the temperatures in the regions between circles; this is done by developing difference equations for the temperatures in such regions — and it will be demonstrated that these involve no greater order of error than the difference equations for $R\theta$ geometry. For simplicity, the heat block outside the region containing the fuel pins is modeled as a continuation of the array of hexagonals. This represents the outer circular boundary by a succession of straight-line segments which are the outward sides of the outer hexagons and which form a figure that is nearly circular. A coolant channel is modeled similarly to a fuel pin, with the coolant and its surrounding pipe replacing the fuel, gas gap, and cladding; the coolant temperature is held constant during the calculation.

Calculation of Heat Generation by Gamma-Radiation
Absorption in the Heat Block

When ^{60}Co fuel pins are used, a large fraction of the heat generated is absorbed outside the fuel pin in the heat block, and this heat generation by gamma-ray absorption must be calculated; such a calculation is not required for SrO or SrTiO_3 since essentially all of the heat is generated within the fuel. For this calculation, each fuel pin is represented by a line source of strength S_L , $\text{Btu hr}^{-1} \text{ft}^{-1}$ at its center. Then the heat generation rate $H(\vec{r})$ at any point $\vec{r}$ in the heat block is

$$H(\vec{r}) = \frac{S_L}{2\pi} \sum_{\substack{\text{all fuel} \\ \text{pins}}}^{}_{i=1} B(\vec{r} - \vec{r}_i) \frac{e^{-\mu|\vec{r}-\vec{r}_i|}}{|\vec{r} - \vec{r}_i|} \mu_a ,$$

where

$\vec{r}_i$ = the center of the i th fuel pin,

$B(\vec{r} - \vec{r}_i)$ = the buildup factor,

μ = the total gamma attenuation coefficient of the heat block,

μ_a = the gamma-ray-energy attenuation coefficient of the heat block.

A linear buildup factor was used such that in an infinite heat block all energy would be absorbed:

$$B = k_1 \left(\frac{\mu}{\mu_a} - 1 \right) ,$$

with k_1 chosen to make $\int_{\infty} H(\vec{r}) d\vec{r} = S_L$ when there is only one fuel pin.

Physically $k_1 S_L / 2\pi$ is the source strength at the outer surface of the fuel pin which must be assumed to cause all energy leaving the pin to be absorbed in the heat block when this form of approximation to the buildup factor is used.

In the case of ^{60}Co pins on 2-1/2-in. centers in a tungsten block, this choice results in a calculated deposition of about 1/4 of the heat in the ^{60}Co pin and most of the remainder immediately outside the pin within the hexagon of the basic lattice within which the pin is centered. For ^{60}Co pins on 2-1/2-in. centers in a steel or nickel block, about 27% of the heat emitted is absorbed in the pin itself, 33% in the hexagon surrounding that pin, and 40% outside.

Calculation of Heat Generation
Inside the Fuel Pins

In the case of the beta-ray emissions of $^{90}\text{SrO}_2$, $^{90}\text{SrTiO}_3$, and ^{60}Co it was assumed that the heat generation was uniform throughout the area of the pin and was zero elsewhere.

In the case of the gamma-ray emitting ^{60}Co pins, a significant amount of the heat deposited in any pin results from absorption of gamma radiation arising in other pins. The simplest approximation would be to concentrate the gamma source which is uniformly spread throughout the pin into a line source of strength $S_L \text{ Btu hr}^{-1} \text{ ft}^{-1}$ at the center of each pin; then the heat deposition rate in a pin of radius R is

$$H = \int_0^R \frac{S_L \mu_a}{2\pi} \frac{e^{-\mu \rho}}{\rho} 2\pi \rho \, d\rho + \text{contribution from other pins}$$

or

$$H = \frac{S_L \mu_a}{\mu} (1 - e^{-\mu R}) + \text{contribution from other pins.}$$

To reverse the concentrating process above, this heat is now assumed to be uniformly spread out over the pin area, and the heat generation rate in a pin from its own gamma emissions is thus

$$\frac{S_L \mu_a}{\pi \mu R^2} (1 - e^{-\mu R}) \approx \frac{S_L \mu_a}{\pi R} \text{ Btu ft}^{-2} \text{ hr}^{-1}.$$

As it turns out, this approximation would be adequate for the purpose of our present calculation. However, a more elaborate development, taking into account the distribution of the source and the buildup factor of the form $(1 + k\mu r)$, leads to a similar heat generation rate at radius r of

$$\frac{S_L \mu_a}{\pi R} \left\{ \frac{2}{\pi} E \frac{r}{R}, \frac{\pi}{2} + \frac{1}{2} (k - 1) \mu R \right\} \text{ Btu ft}^{-2} \text{ hr}^{-1},$$

where E denotes the complete elliptic integral. This latter expression is the one used in the Hexpintemp-2 code; the contribution to the heat deposition due to other pins is calculated for each point in a fuel pin using the same equations as are used for the heat block.

Material Properties

In these calculations the following values were used for the attenuation coefficients for gamma rays from ^{60}Co :

	$\mu \text{ (ft}^{-1}\text{)}$	$\mu_a \text{ (ft}^{-1}\text{)}$
Nickel	14.24	6.71
Steel	14.24	6.71
Tungsten	31.34	17.40

thermal conductivity, $k [\text{Btu hr}^{-1} \text{ ft}^{-2} (\text{°F/ft})^{-1}]$,

$$k = C_0 + C_1T + C_2T^2 + C_3T^3 + C_4T^4 ,$$

where T is in °F.

	<u>C₀</u>	<u>C₁</u>	<u>C₂</u>	<u>C₃</u>	<u>C₄</u>
Steel	4.87879	9.00864E-2*	-1.22513E-4	5.90812E-8	-9.4697E-12
Nickel	-38.7836	1.50034E-1	-1.14896E-4	3.98359E-8	-5.2083E-12
Uranium	10				
Tungsten	50				
⁹⁰ SrO	1.809	-5.672E-4	8.573E-8		
⁹⁰ SrTiO ₃	3.15	-1.11E-3	1.75E-7		
Hastelloy	4.69	4.63E-3			
Air	1.89E-2	1.44E-5			

*Read: 9.00864 × 10⁻², etc.

For the cobalt in the fuel pins the conductivity of steel was used.

Method Used for Calculating the Temperature

Code Hexpintemp-2 calculates the steady-state temperatures within each hexagon of its lattice by solving iteratively a system of linear equations relating the temperatures at nodal points, or nodes. The nodes are at intersections of coordinate lines of constant radius r or constant angle θ , with their origin at the center of the hexagon within which the node lies. Associated with each node is an area defined by the bisector of the interval between that node and each of its radial or azimuthal neighbors. Thus, the node at (r_i, θ_j) has associated with it the area bounded by radii $0.5(r_i + r_{i-1})$ and $0.5(r_i + r_{i+1})$ and by azimuths $0.5(\theta_j + \theta_{j-1})$ and $0.5(\theta_j + \theta_{j+1})$. The temperature at these nodes is calculated by using the relation

$$\frac{\text{heat generated within}}{\text{associated area}} = \frac{\text{heat loss by}}{\text{conduction}} + \frac{\text{heat loss by}}{\text{radiation}} .$$

But each quadrant at a node may have a different material with different properties, so we have

$$\left\{ \begin{array}{l} \text{sum of heat} \\ \text{generated in} \\ \text{each of four} \\ \text{quadrants} \end{array} \right\} = \left\{ \begin{array}{l} \text{sum of heat lost} \\ \text{by conduction at} \\ \text{eight half-} \\ \text{faces} \end{array} \right\} + \left\{ \begin{array}{l} \text{sum of heat} \\ \text{radiated from} \\ \text{four (or fewer)} \\ \text{surface areas} \end{array} \right\} .$$

The four quadrants q , radiating surfaces s , and conducting faces f associated with point (r_i, θ_j) are shown above. At each conducting face

heat lost by
conduction = conductivity $\times$ area $\times$ temperature gradient,

and at each radiating surface of temperature T_i facing a surface of temperature T_j

$$\text{heat lost by radiation} = \sigma \frac{1}{\left(\frac{1}{\epsilon_i} + \frac{1}{\epsilon_j} - 1\right)} (T_i^4 - T_j^4) \times \text{area}.$$

The temperature gradients in the radial and azimuthal directions are calculated simply as

$$\frac{\text{temperature difference between neighbors}}{\text{distance along coordinate line between neighbors}}$$

and the surface areas are calculated by the usual geometric relations. Thus the temperature at any node inside a hexagon is

$$T_{i,j} = \frac{\sum_q Hq + \sum_f \frac{k_f A_f T_f}{d_f} + \sum_s \sigma \epsilon_m A_s (T_s + T_{i,j})(T_s^2 + T_{i,j}^2) T_s}{\sum_f \frac{k_f A_f T_f}{d_f} + \sum_r \sigma \epsilon_m A_s (T_s^2 + T_{i,j}^2)(T_s + T_{i,j})},$$

where H , k , A , d , ϵ_m , and σ denote heat generated, thermal conductivity, area, separation distance, mutual emissivity, and the Stefan-Boltzmann constant respectively.

On the perimeter of the hexagons the basic differential equation for the temperature,

$$k\nabla^2 T + H = 0, \quad (1)$$

must be solved in a different manner. When — as in Hexpintemp-2 — 30° sectors are used, the points on the bounding hexagon are of two types — points of tangency, as at A or V , and corners, as at C . The points S , U , W , X , and Y shown in the sketch are also mesh points. For the present problem the material outside the circle through W , X , and Y is the block material, and is assumed to have uniform conductivity and heat generation. The temperatures at point T are expressed in terms of those at S , W , and U and the corresponding points S' , W' , and U' of the hexagon below, and the temperatures at point C are expressed in terms of those at U , U' , and U'' .

If the temperature is expanded in a series

$$T(x,y) = T(0,0) + ax + bx^2 + cy + dy^2 + exy$$

and substituted in Eq. (1) for each point, sets of simultaneous equations will be produced. These lead to the relations

$$T(C) = \frac{1}{3} \{T(U) + T(U') + T(U'')\} + \frac{H}{4k} h^2,$$

in which h is the distance $CU = CU' = CU''$ and

$$T(A) = \frac{a^2 \{T(U') + T(S)\} + (c^2 - b^2) \{T(W) + T(W')\} + c^2 a^2 (H/k)}{2(a^2 + c^2 - b^2)},$$

where $a = \text{distance } TW = TW$, $c = 0.5 \times \text{radius } OA$, and $b = 0.13396 \times \text{radius } OA$.

For points on the hexagons, relations like that given above for $T(C)$ and $T(A)$ are used in the iterative procedure of calculating the temperature. Finally, the temperature at point U is determined from that at points A, C, X, and V by the method used for other points inside the hexagon. Since both the difference equations used in the $R\theta$ geometry portion of the calculation and those used at the corners are based upon a second-degree approximation to the temperature, they are approximations of comparable accuracy.

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Appendix C

ISOTOPE KILOWATT STUDYSCOPE OF WORKTASK I - PARAMETRIC AND ENGINEERING ANALYSIS

- A. Fuel Capsule and Fuel Block Analysis: Prepare parametric engineering analysis and conceptual design layouts for individual ^{90}SrO , $^{90}\text{SrTiO}_3$, and ^{60}Co fuel capsules and fuel blocks. These conceptual designs shall incorporate preliminary safety provisions and alternate thermal insulation concepts as required for coupling of the fuel block with various energy conversion systems described in Item B, below. Analysis shall be provided concerning these conceptual designs over the range of 5 to 50 kW(th) and for operating lifetimes from 6 months to 5 years.
- B. Primary Heat Exchanger Evaluation: On the basis of the above fuel capsule/fuel block conceptual designs, develop engineering data including operating temperatures and heat fluxes regarding primary heat exchanger requirements for the following energy conversion systems:
1. Thermoelectric; segmented lead telluride or advanced thermoelectric materials.
  2. Helium-xenon Brayton cycle.
  3. Rankine cycle; liquid metal, water and organic.
  4. Sterling engine.
- C. Energy Conversion Study: On the basis of the above heat exchange requirements, expected performance characteristics for each specific energy conversion concept shall be determined for heat sink conditions of -0°F (air), 40°F (air and water), and 130°F (air). Data provided will include conversion efficiency, weight rate of fluid flow, secondary heat exchanger requirements; temperature distribution, pressure drop, efficiency vs primary heat exchanger temperature changes and mass energy balance flowsheets.
- In addition to the preceding analytical effort, a comprehensive evaluation will be conducted on each candidate energy conversion system in order to realistically assess its current state of technology. This evaluation shall consist of literature survey and technical briefings with representative equipment designers and manufacturers.
- D. Shielding Study: Perform fundamental shielding studies for each fuel block/heat exchanger subsystem described in Items A and B, above. These studies will be used as a basis for establishing size and weight characteristics and should include, as a minimum, the following:

1. shield thickness vs dose rate (power level parameters),
2. shield thickness vs dose rate (distance parameters),
3. shield weight vs shielded volume (power level parameters),
4. minimum weight of shield, capsules, fuel vs power level, and
5. minimum volume of shield, capsules, fuel vs power level.

Tungsten and depleted uranium shielding materials will be evaluated in this study. Consideration shall be given to performance advantages resulting from the internal shielding concepts as opposed to shielding external to the fuel block/heat exchanger subsystem.

- E. Conceptual Power System Design: Prepare conceptual system layout drawings at a nominal 5-kW(e) rating for specified fuel block-heat exchanger-energy conversion combinations. A total of four combinations will be specified by AEC. Safety criteria will be established and techniques for temperature control and emergency cooling (active and passive) provided in these specified power system combinations. Preliminary estimates of performance for each power system combination will be determined. When direct scale up (or scale down) is practical, performance data will be parametrically generated at output ratings of 2 kW(e) and 10 kW(e).
- F. Power Systems Conceptual Evaluation Report: Prepare for AEC approval a Power Systems Conceptual Evaluation Report describing the features and principal characteristics of specified fuel block-heat exchanger-energy conversion combinations. This evaluation should consider the current status of technology as well as the utility or adaptability of this technology to the marine or terrestrial environment. Recommendations shall be provided with respect to the two most promising concepts for subsequent detailed engineering study and evaluation in Task II. These recommendations shall consider performance capability, maintainability and operational capability, status of technological development, safety considerations, and cost effectiveness analysis.

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