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**ISOTOPE KILOWATT PROGRAM
QUARTERLY PROGRESS REPORT**

**For Period Ending
March 31, 1970**

A. P. Fraas, Program Director

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REACTOR DIVISION

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JULY 1970

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee
operated by
UNION CARBIDE CORPORATION
FOR THE
U. S. ATOMIC ENERGY COMMISSION

FOREWORD

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 under-sea 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 first phase of the program was a parametric and engineering analysis comparing the principal isotope fuels and the principal types of energy conversion system that have been proposed for applications of this sort, and the preparation of a set of conceptual designs for the more attractive systems. That work was completed in the summer of 1969 and was reported in Ref. 1. In October of 1969 ORNL was asked to proceed with a detailed engineering study of the three most promising systems selected from those covered in Phase I of the program. These three systems now under study are a 2-kw(e) thermoelectric system, a 3-kw(e) steam Rankine cycle system, and a 5-kw(e) organic Rankine cycle system.

The first step in the effort was to evolve a program plan for a three-year effort to be carried out in calendar years 1970, 1971 and 1972. The conceptual designs presented in Task I were reexamined and possible improvements were considered with particular attention to the difficult development problems. A variety of engineering tests was considered as a means of evaluating the technology, solving the principal technical problems, and investigating engineering uncertainties that should be resolved before settling on the design of a prototype power plant. In view of the limited funds and the desirability of narrowing the field to a single Rankine cycle system, particular attention was given to the relative merits of the steam and organic Rankine cycle systems.

The first quarter of 1970 was devoted to firming up reference designs for the three types of system, selecting the most crucial experiments required to evaluate the technology, and settling on a program plan for the three-year effort. With this first quarter's work as a foundation, the second quarter will be devoted to firming up the details of the reference designs, firming up details of the experiments to be conducted, and

preparation of topical reports covering the three reference designs with their associated experiments. Subsequent quarters will be devoted to a series of tests designed to investigate vital questions in the technology. As the results of these tests become available, the reference designs will be revised accordingly. The objective is to evolve by December 1972 a set of designs for two prototype power plants with a clear delineation of the development program required in each case including the cost and time for the various programmatic steps.

This is the first in a series of quarterly progress reports that will be issued during the course of the projected three-year program. Topical reports will be prepared to present the work carried out in particular areas when key tasks are completed. For example, four reports are now being prepared; three presenting the reference designs for the three systems under study and one on heat pipes. Others will follow when the work in an area has progressed to the point that a report is in order.

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1.0 SUMMARY

The first step in attempting to evolve a reference design for an organic Rankine cycle system was to review the information available on the various organic working fluids and the experience at various operating temperatures. This review led to the conclusion that only Dowtherm A has been subjected to sufficient operating experience in the U.S. to warrant its selection without extensive endurance testing. Although the bulk of the work on Dowtherm A Rankine cycle systems has been directed toward the space power plant for which a temperature of at least 650 and preferably 700°F is desired, there seems to be little incentive to use so high a temperature for a naval power unit. As a consequence, the design study has been directed toward the use of Dowtherm A at a temperature of about 600°F where thermal degradation should not be serious, even for periods of as much as five years.

In reviewing the various organic Rankine cycle power plant configurations considered in the Phase I study, it appeared that a still better configuration could be obtained to give important advantages with respect to control of the liquid inventory while at the same time reducing the overall size of the power unit. Thus a new system configuration has been evolved and chosen as the reference design for the Phase II portion of the program.

It appears that both the cost of fabricating the isotope capsules and the size and weight of the heat block-shield assembly can be reduced by increasing the diameter of the fuel capsules and increasing their length. This has led to the development of a new heat block-shield design with a cluster of 7 instead of 19 fuel capsules. Materials considerations, particularly the 10 meter drop test, have led to the choice of low carbon iron as the heat block-shield material. A heat block-shield assembly has been evolved that will satisfy not only the requirements of the organic Rankine cycle system, but also those of the steam Rankine cycle and thermoelectric systems.

An initial investigation of the thermal insulation-thermal fuse problem led to the hope that a ceramic foam type of thermal insulation could be chosen to give good thermal insulation at normal operating temperatures

and yet melt out if the temperature increased by 200°F above the design temperature. Tests of this material showed that this approach was not satisfactory and a different concept, the use of a large number of layers of aluminum wire screen, has been investigated. Analyses indicate that it should satisfy both the thermal insulation and thermal fuse requirements, and a test is planned.

The conceptual design for the steam Rankine cycle system presented in Ref. 1 (which covers the Phase I study) appears to be the best configuration available, hence it has been chosen as the reference design for the Phase II effort. The principal problems other than the turbine generator associated with this system appear to be those of system control and the detailed performance characteristics of the re-entry tube boiler. Preliminary studies indicate that there is a good possibility that an exceptionally simple but unorthodox control system can be employed to yield a very high degree of reliability. This concept will be evaluated by making a detailed comparison with other control system concepts. Tests have been initiated on a re-entry boiler tube of the type and proportions desired. These tests indicate that a suitable re-entry tube boiler can be developed.

All of the thermoelectric power unit configurations evolved in the course of the Phase I study have serious shortcomings; either they are too large and bulky or they present serious assembly problems. Recent tests of a new type of wick structure for heat pipes have been developed at Los Alamos, and this has led to the development of a new modular design for a thermoelectric power unit that appears attractive from the standpoint of the principal criteria that have been evolved for naval isotope power units. A new reference design for a complete power unit employing a set of thermoelectric modules and heat pipes has been worked out and is now under critical review. It is expected that this design will serve as the reference design for the Phase II effort.

2.0 ORGANIC RANKINE CYCLE SYSTEM DESIGN

G. Samuels R. S. Holcomb

Dowtherm A has been chosen as the working fluid for the organic Rankine cycle for the second phase of the program. During the Phase I effort completed in the summer of 1969, monochlorobenzene and dichlorobenzene were also considered. Although these latter two fluids offer possible advantages for some applications, they have not received much attention in this country and only limited data are available concerning their operating characteristics including thermal stability and experience with rotating equipment. In contrast, thermal stability investigations of Dowtherm A have been performed at the National Bureau of Standards, Aerojet-General Corporation, and Sundstrand Corporation. Both Aerojet-General and Sundstrand have operated Dowtherm A turbine-generator units. Also, for applications in which the heating requirements are relatively large in comparison to the electrical requirements, such as an undersea habitat, the high condenser temperature (225°F) of the Dowtherm A system becomes an advantage. Heat can be supplied from the condenser at a temperature of 100 to 200°F without changing the turbine back pressure or affecting the turbine-generator operating conditions.

Design Objectives and Precepts

The system is not, and is not expected to be, directed toward one specific application; i.e., it will be designed as a general purpose unit to supply up to 5 kw of electric power and be capable of supplying heat at a moderate temperature (100 to 200°F). Although the design effort is directed toward undersea use, the basic characteristics of the organic system make it easy to require of the design that it be applicable to land-based use. In addition to these design objectives, a set of design precepts, listed below, was established:

1. The arrangement of the components should be such that the power conversion part of the system can be tested without the heat source and then joined with the heat source at dockside without opening of the piping or equipment containing the Dowtherm A. Furthermore, the

installation should be designed to facilitate startup and thorough checkout of the unit before the outer pressure vessel is sealed.

2. In transferring the heat from the heat source to the working fluid, not more than 10% of the heat should be lost through the thermal insulation and safety devices under normal full-power conditions.
3. Provisions 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. Under these emergency conditions the fuel cladding should not exceed 2000°F.
4. The emergency heat-removal system should be able to function satisfactorily with the unit in any attitude, including on its side or inverted.
5. 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 power conversion system.
6. 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.
7. The output of the generator should be either 56-v dc current or be converted to 56-v dc current so that it can be used in conjunction with a 48-v set of storage batteries that will handle heavy power demands for short periods, as well as provide emergency power.

There are several factors that restrict the degree to which a number of design features can be fixed at this time. The major factor is the lack of a CRU (turbine-generator and pump unit) that has demonstrated the ability to operate for long periods of time. A second closely related factor that will probably depend on the CRU is the need for a highly reliable control system. The present program at ORNL does not include work on a CRU, and the design of this component will depend on the results of work being done elsewhere.

The objective of the present design work on the organic system is to devise a system that will be sufficiently flexible to serve a variety of possible applications, be able to accommodate any of the CRU's now under

development, and meet the design precepts listed above. This system design will then be the basis for relatively simple tests to demonstrate the adequacy of the components selected. Following the demonstration tests, a final detailed design will be prepared.

System Description

A flow diagram for the system is shown in Fig. 1. With the exception of the regenerator, this is a conventional Rankine cycle with the turbine, generator, and boiler feedpump on a single shaft. Unlike steam, saturated Dowtherm A vapor becomes superheated rather than wet when expanded through the turbine. The regenerator recovers most of the enthalpy difference between the superheated vapor and saturated vapor at the turbine back pressure. For the cycle conditions of interest a regenerator with an effectiveness of 90% increases the cycle efficiency by about 50%.

A preliminary design of the organic system is shown in Fig. 2. The main features of the system shown are the re-entry boiler tube in a recirculation boiler and the condenser with a central region to trap noncondensable gases. The vapor flow, starting at the boiler, is as follows: the vapor-liquid mixture flows from the boiler to a combined vapor separator-header drum, the separated vapor flows through the turbine, and then through the regenerator to the condenser. The liquid from the condenser flows to the feedpump, passes through and cools the generator, through the regenerator, and back to the boiler header drum.

The boiler tube and heat block arrangement shown in Fig. 2 was selected to meet the design precept that the organic system can be fabricated and tested separately from the heat block, and that the dockside coupling or assembly of the two should be simple and not require opening the sealed organic system. The arrangement also allows the shipment of the heat block so that it will be cooled by ambient air until it is slipped into an insulated pressure vessel. After coupling the heat block, organic Rankine cycle system, and pressure vessel, the system can be checked out prior to sealing the pressure vessel.

In order to provide the protection listed in precepts 3 and 4, it is presently intended to use the aluminum screen type of insulation about

Fig. 1. Flow Diagram of the Organic Rankine Cycle System

ORNL DWG. 70-7686

Fig. 2. Preliminary Design of Organic Rankine Cycle System

4 ft long between the heat source and the pressure vessel. This safety concept takes advantage of the relatively low (1200°F) melting point of aluminum and also the high (~ 0.9 or greater) thermal radiation absorptivity of a porous material. The melting point of aluminum is high enough to be used not only with the organic systems, but with the thermoelectric and steam systems. In the event of an accident or system failure that prevents the normal removal of heat from the heat block, the temperature of the block will increase until its surface temperature is high enough to reject the heat to the aluminum insulation. When the aluminum reaches 1200°F it will start melting and provide a constant temperature heat sink. The melting will continue until enough insulation has been removed so that the heat loss through the insulation to the pressure vessel is equal to the heat generated in the block. Preliminary calculations indicate that, with a block temperature of about 1300°F and a maximum fuel capsule temperature of 1800 to 1850°F, all of the heat generated can be transferred to the aluminum by thermal radiation alone. Tests to verify this mechanism and also to determine the thermal conductivity of the aluminum screen are discussed in a later section.

The selection of a recirculating boiler was based on the need for a simple, stable unit and the fact that with the Dowtherm A one does not want to get superheated vapor. The alternative to the recirculating boiler is a once-through unit. The once-through boiler has a major advantage in that the amount of Dowtherm A at the boiler temperature is much less than in a recirculating boiler and hence the resulting thermal decomposition will be less. However, the decomposition rate is a very strong function of the temperature, and at temperatures in the range of 650°F, a recirculating boiler operating at a temperature 50°F less than a once-through unit would have decomposition rates similar to the once-through unit. It can be seen from the cycle efficiencies shown in Fig. 3 that a 50°F reduction in the boiler temperature (i.e., turbine inlet temperature) results in a loss of only one point in cycle efficiency.

The disadvantages of the once-through boiler are:

1. It does not have the inherent stability of the recirculating type.
2. The tubes must be much longer to dry the vapor completely.

Fig. 3. Relations Between Dowtherm A Rankine Cycle Efficiency, Turbine Inlet Temperature, Condenser Pressure, and Turbine-Generator Efficiency; for a Regenerator Effectiveness of 0.90, Pump Efficiency of 40%, and a Pressure Drop of 0.05 psi from Turbine Exhaust to Condenser.

3. The internal heat transfer coefficient in the tubes will be much lower in the high quality portion of the tube than in the boiling region, and probably lower in the liquid preheating portion. This, coupled with the fact that the heat block is operating at very close to a constant temperature along its length, will lead to varying tube wall temperatures, with high wall temperatures in the areas of low heat transfer coefficient. This in turn will give a higher boundary layer temperature which could offset any thermal decomposition advantages.
4. The feed flow to the boiler must be very tightly coupled to the thermal output of the heat block. This places a restriction on the types of control system that can be used.

The organic systems designed for space applications have used once-through boilers primarily because the once-through unit has the advantage that it eliminates a free liquid surface which in zero-g is very difficult to control. In the designs considered here, it is assumed that the power units will operate in the vertical or near-vertical position. Another difference between space and terrestrial applications is that the weight and size of space radiators is heavily dependent on the operating temperature, whereas this is not the case for an undersea power plant.

As can be seen in Fig. 2 the top head of the pressure vessel provides the condenser surface for rejecting heat from the cycle. A cylindrical baffle is located in the center of the condenser to trap noncondensables. Most of the surface area (sufficient to carry the complete load) is located outside of the baffled region and most of the condensation will occur in this outer region. The 10 to 15% of the condenser surface inside the baffle will draw vapor and noncondensable into the center region where they tend to remain at the center of the vortex formed by the high velocity of the incoming vapor.

An alternate design which is used with the organic space system is a jet condenser. Again the major advantages of the jet condenser are related to zero-g conditions in that this type of unit eliminates a free liquid surface and also supplies a high net positive suction head to the pump. This type of condenser could have two advantages for the present application. First, the pressure recovery characteristics of the unit

give a high pressure for cavitation suppression at the pump suction. Second, the jet condenser reentrains the noncondensables and recycles them through the system. However, there is a limit to the amount of noncondensables that can be recycled. Eventually they may reach a concentration sufficient to interfere with the nozzles and liquid jets and thus interfere with operation.

There are several other disadvantages to using a jet condenser. In addition to the jet condenser there must also be a heat dump to reject heat from the cycle so that this introduces an additional component. In order to operate the high velocity jet, the head required of the feed pump is much higher than required for the more conventional surface condenser. Also, the jet condenser requires subcooled liquid which, in turn, requires the heat dump to reject a larger amount of heat from the system. This will reduce the cycle efficiency by several points. The jet condenser system also rejects heat at a lower temperature than a system with a surface condenser and the heat available for auxiliary purposes will be at a lower temperature.

The cycle efficiencies for an organic Rankine cycle with a regenerator and a conventional surface condenser are shown in Fig. 3. For these cases the regenerator effectiveness was 90%, the feed pump efficiency was 40%, and the pressure drop across the regenerator was 0.05 psia. Curves are shown for turbine-generator efficiencies of 61 and 66%. These turbine-generator efficiencies result from combining a 94% mechanical efficiency with turbine-generator efficiencies (without mechanical losses) of 65 and 70%.

The layout shown in Fig. 2 uses the earlier heat source design with the 4.1 in. diam fuel bodies. The later source design will reduce the diameter of the system by about 3 in. and increase the length by about 12 in. In either case the weight of the heat block-shield unit is well over half the total weight and any weight savings by compacting the power conversion system will be relatively small. Also, for most applications of interest, small savings in weight are not important. For this reason, the primary considerations in the system design are reliability, simplicity, and safety rather than weight or size.

Heat Block-Shield Design

In developing the design of the heat block-shield unit, a set of shipping and design precepts was established. These precepts are as follows:

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 will permit free circulation of air for removal of the decay heat, but will 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 1700°F, and under all foreseeable conditions the temperature of the Hastelloy cans would not 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. 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 not only a conduction problem but also severe corrosion.)

Based on the results of the parametric studies¹ of fuels and heat block-shield materials made during the first phase of the program, it was decided to concentrate future design and evaluation tests on an iron or steel unit fueled with SrTiO_3 . Restrictions on the design of the heat block-shield unit, in addition to the design precepts listed above, are that the unit be suitable for use with either a steam or an organic Rankine cycle system or with a thermoelectric power unit; and also that the

unit be adaptable to the use of SrF_2 in place of SrTiO_3 . The SrF_2 is of interest because of the present AEC waste management program at Hanford. Starting this year, the AEC is expected to be removing routinely all of the ^{90}Sr and ^{137}Cs from the fuel reprocessing wastes at Hanford prior to solidification of the remaining waste for long-term storage. This is being done to minimize waste mobility, and, depending on accounting procedures, the separated ^{90}Sr could be considered as paid for at that point. As a further extension of this Waste Management Program, it is planned, by about 1975, to convert routinely the separated Sr to solid form, probably SrF_2 , and encapsulate the solid in Hastelloy C-276 cans to reduce further waste-handling costs. Thus, the SrF_2 may become available at costs considerably less than those of SrTiO_3 .

The possibility of replacing the SrTiO_3 with SrF_2 led to a reconsideration of the designs evaluated during the Phase I study, and was a factor in the selection of a fueled region consisting of seven elements located on a triangular pitch (see Fig. 4). The advantage of this arrangement is that the power output with all seven positions fueled with SrTiO_3 (power density = 0.85 w/cm^3) is within 1% of that with SrF_2 (power density = 1.0 w/cm^3) using only the outer six fuel positions.

Two fuel diameters have been considered for the seven element design. The first size considered (4.15 in.) is the same as presently used in the SNAP 23 units. Both heat block-shield and fuel weights and temperature profiles through the shield and fuel were determined for ligament thicknesses (distance between holes in the heat block-shield) of 1/2 in. to 1-1/2 in. Figure 5 shows the dimensions and weight as functions of the surface dose rate. The thermal power output of the unit is 34 kw, which, for a 5-yr life gives an end-of-life output of 30 kw.

The maximum temperature at which the Hastelloy C-276 cladding can operate is not very well defined. During the Phase I study, temperatures as high as 1800°F for continuous operation and 2000°F for accident conditions were considered. At this stage of the design work a maximum continuous operating temperature of 1600 to 1650°F appears more prudent. Reducing the normal operating temperature will probably have little effect on the final design of the unit because in most cases the dimensions will

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Fig. 4. Heat Block-Shield Arrangement

Fig. 5. Heat Block-Shield Dimension and Weight vs. Surface Dose Rate and Ligament Spacing for 34 kw and 4.15 in. Diameter Fuel.

be governed by the accident condition. For example, with the organic Rankine cycle operating with a boiler temperature of 600°F, the maximum cladding temperature will be about 1500°F with the above design and a 1-1/2 in. ligament spacing. However, during accident conditions in which the boiler is inoperative, the temperature will approach 2000°F with the safety devices now under study.

Because of this narrow safety margin and other reasons discussed below, a second heat block-shield design using a 3.40-in. diameter fuel body is now being analyzed. This unit will have a smaller diameter and be about 12 in. longer. The longer shield is a decided advantage to the steam Rankine cycle system because it allows a longer re-entry boiler tube which greatly improves the performance of this type of boiler. Preliminary analysis indicates that the longer shield block can also be used for the thermoelectric system. This then will give a single heat block-shield design for all of the systems presently being studied.

The dose rate as a function of shield thickness for the 4.15 and 3.40-in. fuel bodies is shown in Fig. 6. Data for a 2.50-in. diameter fuel body as determined during the Phase I study is also shown.

Armco iron or an equivalent very low carbon iron has been tentatively selected for the heat block-shield material. The emphasis here is to obtain a low carbon content in order to increase the transformation temperature at which the material undergoes a crystallographic change from body-centered cubic to a face-centered cubic lattice, or at least to minimize the percent of the material that may undergo this change. This transformation temperature for pure iron is about 1650°F, and decreases as the carbon content increases. At a carbon content of 0.025 weight percent, some of the material (the amount increases with carbon content) will undergo this transformation at about 1350°F. The problem with operating above this temperature is that some part of the block will be at the transformation temperature and thermal cycling can lead to dimensional changes. Although severe thermal cycling is not expected with the systems under study, the use of a very low carbon material will eliminate the problem.

The disadvantages of the very low carbon material are: 1) The material is not as easy to machine as the conventional low carbon steels and 2) procurement of high quality ingots may be more difficult or more expensive.

ORNL DWG. 70-7676

Fig. 6. Dose Rate vs. Side Shield Thickness for SrTiO₃ Fuel and an Fe Shield.

With the design using a 4.1-in. fuel body, the central fueled region of the block will be above the 1350°F transformation temperature for all systems. With the design using a 3.4-in. fuel body, the central block temperature with the organic system will probably be below 1350°F. However, with the steam or thermoelectric units the central temperature will be very close to 1350°F. The final choice of material will depend on the detailed thermal analysis of this latter block design.

Thermal Fuse and Insulation

The purpose of the Thermal Fuse and Insulation Task is to develop a means to prevent overheating and melting of the isotope fuel cladding in the event of an accidental loss of coolant flow to the heat block. The approach to the solution of this problem was to find a type of insulation for the heat block which would limit the heat losses to about 5 to 10% during normal operation, but melt and flow from around the heat block so that the total heat generated in the fuel could be removed by thermal radiation from the shield to the pressure vessel without melting of the fuel cladding after loss of coolant flow.

The insulation material should have characteristics such that it would suffer no significant loss of insulating value at a hot face temperature of 800°F for a period of 5 years, but melt at a temperature of about 1200°F. Glass and aluminum were the two materials which appeared to most nearly possess the desired properties.

After a literature review of glass insulation, a cellular glass insulation product of the Pittsburgh-Corning Company marketed under the trade name of Foamglas was selected for preliminary evaluation. Samples were placed in a small furnace for a period of one hr at 1200°F and then removed for examination. The outer surface showed a change in color and a slight amount of sagging and fusion of the edges of the glass cells. These effects did not appear significant, however, and the material seemed to have remained in a condition to be quite effective as an insulator. The samples were again put into the furnace for a period of one hour at 1300°F. Examination revealed some additional fusion on the edges of the glass cells at the surface, but no gross melting or powdering had occurred.

A photograph of one of the test samples after being exposed to 1200° and 1300°F for one hour at each temperature is shown beside a piece of the unheated material in Fig. 7. It was concluded that the glass had neither a low enough range of softening temperature nor was brittle enough to fracture and powder at 1200 to 1300°F, and therefore would not fulfill the requirements of the heat block insulation.

Aluminum wire screen was selected as the next material for testing. An insulation matrix composed of a large number of layers of aluminum screen appears to offer the possibility of keeping the operating heat loss within acceptable limits with a thickness of 2 to 4 in. The thermal conductivity was estimated from experimental results obtained on stainless steel screen matrices for gas-cooled reactors in France.² The thermal conductivity is estimated to be about 0.042 Btu/hr-ft-°F at a mean insulation temperature of 400°F for a screen composed of 0.010-in. diameter wire on a mesh of 16 x 18 wires per inch.

It is anticipated that the voids in the screen will yield an effective emissivity of from 0.9 to 1 so that the temperature difference required to radiate the heat from the heat block will be minimized. Under accident conditions, the temperature of the block and screen would increase until the screen reached its melting point. From that point, the temperatures should remain constant as the screen continued to melt. Eventually, the screen thickness would be decreased to the point where the heat losses through the screen would equal the heat generated in the block and no further melting would occur.

Preliminary tests are planned to measure the thermal conductivity of an aluminum screen test specimen and to determine the temperature history of the surface of an electric heater rod as the aluminum screen is melted. The thermal conductivity will be measured at 100°F intervals for a hot face temperature ranging from 600°F up to the melting point of aluminum. After the thermal conductivity tests have been completed, the power input to the heater will be increased until melting of the screen occurs, and the heater surface temperature will be recorded.

The test specimen is composed of a roll of screen 32-in. long with an ID of 1-1/2 in. and an OD of 3-1/2 in. The screen consists of 0.010-in. diameter wire on a mesh of 16 x 18 wires per inch.

PHOTO 77840

Fig. 7. Photograph of Foamglas Insulation Test Specimens. The Specimen on the Left Has Been Exposed for One Hour at 1200°F and One Hour at 1300°F. The Specimen on the Right Is in the As-received Condition

The heater is an electric cartridge heater with a power rating of 750 w. The heater has a stainless steel sheath with an OD of $3/4$ in., an overall length of 40 in., and a heated length of 12 in. The heater will be placed inside the test specimen and centered with a webbed spacer at the bottom and a solid spacer at the top end.

A longitudinal section through the test assembly is shown in Fig. 8. Thermocouples will be located at axial positions of 4, 6, and 8 in. along the heated length on the heater surface, and on the inner and outer surface of the screen specimen. Fibrous ceramic insulation will be used at the top and bottom to minimize end heat losses and to prevent air streaming along the heater and out the top by natural convection. The assembly will be placed in a stainless steel can which will contain the molten aluminum which flows from the screen.

The test specimen has been fabricated, and testing is planned upon completion of installation of the thermocouples and electrical power supply.

Provided the preliminary tests give satisfactory results, it is planned to proceed to test this insulation concept on a $1/4$ scale heat block. The $1/4$ scale test would serve to verify the thermal conductivity and melting characteristics on a larger diameter test specimen. In addition, it should disclose possible design and fabrication problems on the larger insulation sections should such problems develop.

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Fig. 8. Longitudinal Section Through Heat Block Insulation Test Assembly

3.0 STEAM RANKINE CYCLE SYSTEM DESIGN

A. P. Fraas R. S. Holcomb

The conceptual design for a steam Rankine cycle power unit delineated in the Phase I study¹, remains the most promising steam power unit configuration evolved in the course of the program to date, and has been chosen as the basis for the reference design. With this configuration as the point of departure, some modifications are being effected as a consequence of the reduction in design power output from 5 kw(e) to 3 kw(e). The proportions of the heat block-shield assembly are also being modified so that it will meet the requirements of all three types of system now under consideration, i.e., the thermoelectric, organic Rankine cycle, and steam cycle systems as discussed above. The design of this unit was discussed in the previous section on the organic Rankine cycle system. Similarly, the design of the thermal insulation is essentially the same as for the organic system.

System Control

Two major problem areas peculiar to the steam system that have been given attention this quarter have been a preliminary analysis of the problems of control and an experimental investigation of the effects of geometry on the re-entry tube boiler contemplated for the design.

The principal uncertainties remaining in establishing the reference design are those concerned with the system control.

Studies of forced outage experience with full-scale steam power plants indicate that the control system is likely to be the principal cause of a forced outage.^{3,4} In view of the high importance of achieving an exceptionally high degree of reliability in isotope power units, it appears essential that the control system be very simple, in fact much simpler than any steam Rankine cycle system that has been used to date. This in turn implies that one should start from first principles and examine critically the really essential requirements of the instrumentation and control system. These appear to be four in number, that is,

1. Avoid excessive temperatures in the isotope heat source
2. Avoid excessive turbine-generator rpm

3. Avoid excessive battery charge
4. Maintain the battery charge.

These requirements are simple and straightforward, but there are many complications. For example, the first requirement poses problems because, as the isotope decays, the heat input to the boiler will drop off. If one wishes to maintain a given set of steam conditions in terms of temperature and pressure, this in turn means that the feed flow rate of water to the boiler should be varied to suit the output of the isotope. Unless some step of this sort is taken the amount of superheat in the steam will drop rapidly as the isotope decays and turbine bucket erosion may become a problem.

A schematic diagram showing the layout of one of the systems contemplated is presented in Fig. 9. To meet the first of the requirements itemized above, a thermostatically controlled valve has been provided in the line between the feedwater pump and the once-through boiler. An outer thimble of titanium would extend up into the heat block-shield assembly. Attached to its upper end would be a stainless steel rod which would extend down through the thimble to the feedwater control valve. A variation of 100°F in the temperature of the thimble would lead to a differential thermal expansion of about 0.030 in. between the thimble and the stainless steel rod. Since the feedwater flow rate at full power for this unit will only run around 100 lb/hr, the port in the valve will have an equivalent orifice diameter of only about 1/8 in., hence the valve could be designed to go from zero to full flow with a temperature variation of only 100 to 200°F in the heat source. Thus as the isotope decays and the heat generation rate falls off, the valve will tend to close, reduce the feedwater flow rate, and the heat source temperature will drop less than 100°F for a change in the heat source power output of a factor of 2. Thus this control device should maintain a nearly constant heat source temperature irrespective of the heat generation rate.

The steam will be delivered to what will in effect constitute a pair of critical pressure drop orifices in parallel. As a consequence, the boiler discharge pressure will be essentially directly proportional to the steam generation rate, hence no control action will be required to regulate the pressure of the steam delivered by the boiler.

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Fig. 9. Schematic Diagram of a Small Steam Power Plant Making Use of an Isotope Heat Source, a Once-through Boiler, and a Very Simple Control System

The second major control element in the system depends on sensing the battery voltage (with the sensor E_vE of Fig. 9) and using that to control a valve venting steam directly to the condenser from the line between the boiler and the turbine. The valve would be fitted with two poppets on a common stem so that as one opened the other would close. The valve seat diameters would be proportioned so that, with a given supply pressure, the total flow rate would be constant irrespective of valve position and the consequent ratio of turbine flow to bypass flow. The E_vE can be a simple silicon controlled rectifier and zener diode combination that actuates a control relay. The on-off condition is realized by the selection of a zener diode that will sense voltage changes within the limits required. The zener controls the gate power of the SCR which in turn actuates the relay. This relay switches power on or off to the bypass control valve. If a simple on-off control scheme does not prove satisfactory, the elements of this control device could be modified to give proportional control. Under any circumstances, a stop should be provided so that with the bypass valve fully open, the turbine should still receive sufficient steam to overcome frictional losses and keep the generator speed above the lower limit of the design value range with no electrical load.

The control scheme proposed depends on the increase in battery voltage with charge characteristic of lead-acid storage batteries. If the fully-charged no-load battery voltage were 56 volts and the no-load voltage with its lowest acceptable charge were 24 volts, the generator could be designed to deliver its full output to the battery with a net difference between its output and the back emf of the battery of only, say 4 volts. The voltage output of the generator would be directly proportional to the speed, and this could be allowed to float. It would increase with the battery charge as indicated in Fig. 10. The power output would be at the full capacity of the turbine until the generator voltage output reached the voltage at which the battery would be fully charged. The bypass valve described in the previous paragraph would then begin to open and limit the speed and voltage.

The speed and voltage output of the generator will depend in part on the external electrical load imposed on the generator-battery system. If

Fig. 10. Effects of Battery Charge and Electrical System Load on Major Performance Parameters for a Typical Steam Turbine-Generator-Lead Acid Storage Battery System Having a Generator Rated at 5 kw with Electrical System Loads up to 10 kw

this load represents half the capacity of the generator, half the generator output will go to the load and half to charging the battery. The reduced I^2R losses in the battery at the lower charging rate will cause the generator speed and voltage to stabilize at a lower level than for the zero external load condition. Similarly, if the external electrical load exceeds the capacity of the generator, the voltage across the battery terminals will drop still further, and the load will drag down the turbine-generator speed to a still lower level. These effects are shown in Fig. 10.

Efforts to find a still simpler control system are underway. One variation of the above system that is under investigation allows the turbine-generator speed to float over a wide range and regulates the strength of the field to maintain a constant output voltage. The principal disadvantage of this approach is that, as the load on the generator varies from the full design load to zero-load, the turbine-generator speed will increase from the design speed to almost double the design speed. This approach appears to be satisfactory with an organic Rankine cycle turbine, but it appears likely to give difficulty with the steam turbine because of the much higher turbine tip speed inherent in the use of steam. The higher turbine tip speed not only gives the threat of difficulty with excessive stresses in the turbine wheel, but it also raises problems relative to the dynamic stability of the rotor in its bearings. In addition, if the feed pump is mounted on the turbine-generator shaft, the cavitation problem at the inlet to the feed pump becomes very much more difficult at the runaway speed. One approach that is under consideration is to inject liquid into the generator rotor cavity when the turbine-generator speed exceeds, say, 110% of normal speed. The churning of the liquid in the generator cavity would introduce large losses and these in turn would slow down the generator and absorb a substantial portion of the load. The liquid flow for this purpose could be obtained from a small zero-flow pump mounted on the generator shaft and designed for a low head. When the generator speed exceeded a specified level, the output of the pump could be sufficient to lift liquid to the top of a standpipe so that it would overflow into the generator rotor cavity.

The detailed design of a series of typical control systems will be examined from the reliability standpoint in an effort to appraise the relative reliability of each of the systems that appears to be of interest.

Re-entry Tube Boiler Investigation

The principal effort on the steam system during this quarter has been directed toward the development of the boiler. A once-through boiler of the re-entry type was selected previously¹ as the one which would best meet the assembly and operating requirements of a radioisotope electric power plant. In this type of boiler the feedwater flows upward inside a central tube where the water is vaporized and the steam may become superheated. The steam then flows downward in an annulus formed by an outer tube concentric with the central tube where the steam receives additional superheat. The heat is supplied to the outside of the outer tube. The steam conditions to be achieved were selected as 800°F at 200 psia.

An existing steam system test facility was utilized to conduct experiments on one full-scale tube of the re-entry boiler design. The test facility consists of a pressurized water tank, the boiler, preheaters, condenser, power supplies, and instrumentation.

The boiler consists of two concentric tubes connected to a pipe tee which provides an outlet connection for the steam flow. Analyses indicated that the outer tube should be fitted with internal fins. Fortunately, such a tube was available as a spare from another program. While the properties are not ideal, the tube employed is much better than a plain tube. It is a 3/4-in. OD by 50-in. long stainless steel tube with 8 spiral fins machined on the inside of the tube to provide increased heat transfer surface area. The fins are 0.050-in. high by 0.050-in. wide arranged on a pitch of 0.3 in. The bore of the tube at the fin tips is 0.530 in. The inner tube is a 0.50-in. OD stainless steel tube with a wall thickness of 0.035 in. At the lower end the diameter was reduced and fitted with a concentric sleeve. The sleeve is open at one end to the steam in the annulus, allowing the space inside the sleeve to be filled with a stagnant layer of steam that provides thermal insulation between the central tube and the superheated steam in the annulus. The smaller tube has an ID of 0.245 in. At the

upper end of the central tube, a loosely fitted plug is located inside the inner tube to form a flow passage with a reduced width and consequently increase the heat transfer coefficient in the region where the steam is of high quality or superheated. A cross-section of the boiler assembly is shown in Fig. 11.

Five models of the central tube were tested. The length of the insulated section was varied from 0 to 24 in. in three of the models. The length of the plug was varied from 0 to 24 in. in three different models. The characteristics of the five models are given in Table 1.

Table 1

Characteristics of Re-entry Tube Boiler Test Models

Model No.	Central Tube Plug Length (in.)	Insulating Sleeve Length (in.)
1	3	0
2	3	12
3	0	12
4	24	12
5	3	24

The outer tube wall, water inlet, return steam, and boiler exit steam temperatures were measured with thermocouples. The boiler was heated by four individually-controlled clamshell electric heaters which were set to give a uniform outer tube wall temperature of approximately 1000°F. The water was a few degrees below the saturation temperature at the boiler pressure.

A series of tests were run in which the water flow rate and the boiler pressure were the independent variables. For each value of flow and pressure, the heat input and the temperatures were measured and

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Fig. 11. Diagram of a Re-entry Boiler Tube

recorded. The flow rate was varied from 1 to 7 lb/hr and the pressure was set in most cases at either atmospheric pressure or about 200 psia.

The results obtained on Model 1 are presented in Fig. 12. The central tube exit steam temperature is the temperature at the point where the steam leaves the central tube and enters the annulus for its return pass downward. The steam temperature leaving the superheater is measured at a point in the steam outlet line which is 1 in. from the boiler outer tube. At atmospheric pressure, the central tube exit temperature is very high at a low flow rate, but decreases rapidly to the saturation temperature at a flow of about 3 lb/hr, and then remains at the saturation temperature. The resistance of the exit steam line causes a slight increase in boiler pressure with flow rate and consequently a small increase in the saturation temperature. The superheater exit steam temperature exhibits a pronounced minimum value at a flow rate of 2.75 lb/hr and has a maximum value of 480°F at about 4 lb/hr. At 200 psia, the central tube exit steam temperature decreases gradually with flow rate, tends to level off, and then decreases rapidly toward the saturation temperature. The superheater exit steam temperature gradually increases with flow rate and appears to level off at 7 lb/hr. The superheater exit steam temperatures achieved with Model 1 were considerably lower than the desired goal of 800°F. The heat flux which can be radiated from the heat block at the design temperature for the radioisotope system would limit the allowable flow per tube to about 5 lb/hr. The superheater exit steam temperature measured at 200 psia and 5.2 lb/hr was only 450°F.

Since the superheater exit steam temperature was lower than the central tube exit steam temperature for flows up to 6 lb/hr, it appeared that the steam in the lower portion of the superheater annulus was losing too much of its heat to the boiling water in the central tube. This led to the selection of a design utilizing a central tube fitted with a loose sleeve to create a steam gap and thus provide thermal insulation between the annulus and the central tube. The length of this sleeve was chosen as 12 in. for Model 2.

The steam temperatures found with Model 2 are presented in Fig. 13. The central tube exit steam temperature at atmospheric pressure dropped

Fig. 12. Model 1 Re-Entry Boiler Central Tube Exit and Superheater Exit Steam Temperature vs. Flow Rate at Atmospheric Pressure and 200 psia.

Fig. 13. Model 2 Re-Entry Boiler Central Tube Exit and Superheater Exit Steam Temperature vs. Flow Rate at Atmospheric Pressure and 200 psia.

more quickly with an increase in flow rate, but had the same general trend as for Model 1. The superheater exit steam temperature increases rapidly with flow rate, with the rate of increase diminishing up to 6 lb/hr. No minimum similar to that for Model 1 was observed at atmospheric pressure.

The results at 200 psia showed a trend in the central tube exit steam temperature similar to that seen with Model 1, except that the temperature decreased more quickly, had a broader level region, and reached the saturation temperature at 7 lb/hr. The superheater exit steam temperature gave much higher values than those for Model 1. The temperature measured at 5.2 lb/hr flow was 720°F, which would be quite acceptable for a steam Rankine cycle since it would give a cycle efficiency of only slightly less than the original goal of 800°F.

In Models 3 and 4, the length of the plug in the top of the central tube was varied to determine its effect on the performance. With Model 3, where the plug was removed, the central tube exit steam temperature was reduced, but the superheater exit steam temperature was nearly the same as for Model 2. In the case of Model 4, where the plug was made 24-in. long, the central tube exit steam temperature was increased, but again the superheater exit steam temperature was practically unchanged from Model 2. Since the effects on the superheater exit steam temperature were so small, no curves are presented here for Models 3 and 4.

In an attempt to see if an even higher superheater exit steam temperature could be achieved, Model 5 was made with an insulated section 24-in. long. These results are given in Fig. 14. At atmospheric pressure, the central tube exit temperature was slightly lower than for Model 2. The superheater exit steam temperature gave somewhat surprising results since it was slightly lower than for Model 2. The central tube exit steam temperature at 200 psia fell more quickly to the saturation temperature than for Model 2. Again, the superheater exit steam temperature was lower than that for Model 2, i.e., the temperature was 655°F at a flow of 5.2 lb/hr as compared to 720°F for Model 2.

The reduction in superheater exit temperature experienced when the insulating sleeve length was increased from 12 to 24 in. confirms the analysis indicating that there is an optimum length of the insulating

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Fig. 14. Model 5 Re-Entry Boiler Central Tube Exit and Superheater Exit Steam Temperature vs. Flow Rate at Atmospheric Pressure and 200 psia.

sleeve. The superheater exit steam temperature at 200 psia and a flow rate of 5.2 lb/hr is plotted vs insulated length in Fig. 15. The data indicate that an exit steam temperature of about 740°F might be achieved for a sleeve length of 16 in. A boiler with this sleeve length will be tested to determine the actual exit temperature which it would produce.

A topical report is in preparation to present more completely the analyses that have been made, the results obtained from the tests, and the conclusions that can be drawn from the analyses and tests.

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Fig. 15. Superheater Exit Steam Temperature vs. Insulating Sleeve Length at 5.2 lb/hr Flow and 200 psia.

4.0 THERMOELECTRIC SYSTEM DESIGN

A. P. Fraas

A number of difficult problems have been encountered in attempts to develop detail designs for thermoelectric power conversion units using isotope heat sources. If the thermoelectric junctions are placed between the heat source and the surrounding shield, or between the outer perimeter of a hot shield and the cooling medium, the temperature drop through the shield greatly reduces the temperature drop available through the thermoelectric element. The extent of this reduction depends on the shielding material. If iron is employed, the temperature drop is likely to run a third to half of the total temperature drop available. This loss can be cut about in half through the use of nickel in place of iron at a cost of something like \$30,000 for nickel as compared to \$3,000 for iron for a 1 to 2 kw(e) output unit. A further reduction in the temperature drop through the shield can be effected through the use of tungsten. Tungsten has the disadvantage that its cost would be something like \$100,000, and the tungsten itself is brittle and would not stand the required drop test.

Two other systems that have been considered entail the use of a boiling liquid metal. In one, the heat is transported from the interior of the heat block-shield assembly out to the thermoelectric junctions by a set of heat pipes. In the other, a somewhat similar arrangement can be employed using a natural circulation loop of much simpler construction because no capillary passages would be required. This would make use of boiling to carry heat out of the heat block to the thermoelectric elements. The latter approach was favored in the initial ORNL studies in this area.¹ However, subsequent work on heat pipes in another ORNL program on space power plant applications has led to the development of designs for high performance heat pipes of reasonable geometry from the fabrication standpoint,⁵ and the soundness of the detail design has been demonstrated by the recent examination of five heat pipes at ORNL after long endurance tests, one lasting 20,000 hr.⁶ Further, it appears that these heat pipes can be designed with a performance margin sufficient to allow normal functioning of the power unit even if it were turned upside down. As a

result, an attractive modular design employing heat pipes has been evolved for naval isotope power units. This new concept is presented below. As in the Phase I studies, the reference design conditions have been taken as an electrical output of 2 kw and a life of 5 yr.

Choice of Thermoelectric Converter Type and Geometry

In an effort to develop very compact thermoelectric modules that will fit a small gallery in a space reactor shield, both Westinghouse and RCA have recently developed compact converter modules that entail the use of a heat source along the central axis of a cylinder and a large number of thermoelectric junctions placed radially between the central heat source and the cooled cylindrical perimeter. These arrangements make use of brazing and swaging techniques to effect a good thermal bond between the elements of the thermoelectric units and both the inner heat source and the cooled outer tube. Unfortunately, it seems likely that if these units are built in moderately large diameters, e.g., 3 in., and in substantial lengths, e.g., 20 in. or more, the differential thermal expansion between the hot inner tube and the cold outer tube will lead to fractures in either the thermoelectric units themselves or in the various bonds in the electrical and thermal insulation required in the elements. Further, these units yield a relatively poor efficiency, this type of structure appears to be very expensive, and the fabrication procedures and processes are not yet well developed.

In view of the very limited funds available for the Navy Isotope Power Plant program and the current status of the development of thermoelectrics, it seems best to predicate the design of a thermoelectric power unit on a set of modules employing well-developed thermoelectric junctions. Further, these modules ought to be designed so that they will not present any difficult problems from the standpoint of thermal stresses, differential thermal expansion, or materials compatibility. The concept proposed meets these requirements and provides wide latitude in the choice of materials, i.e., the coefficients of thermal expansion of the various materials involved need not be matched nearly so precisely as is required for the compact converter type. While the proposed thermoelectric module

is heavy and bulky when compared with the compact converter designs, the increased size and weight are trivial considerations for the naval applications of this program, and there is no evident incentive to employ a more fancy and expensive design of the sort being developed for the space power program.

Basic Module

The module envisioned employs a thermoelectric element 1 cm in diam and 3 cm long. These modules would be assembled in a radial array as indicated in Fig. 16. The cold outer junction would be forced against the outer cooled wall by a hermetically sealed, flattened tube containing inert gas under pressure. The hot junction would float freely in a closely fitted recess between nickel blocks around the central heat pipe. The close clearance between the sides of the hot shoes and the nickel blocks would lead to a temperature drop of only about 30°F even if all of the heat were transmitted by conduction through the gas gap. However, at the high temperatures prevailing at the hot junction, i.e., around 1100°F, an appreciable fraction of the heat would also be transmitted by thermal radiation, hence the temperature drop would be reduced by perhaps 10%.

Each module formed in this way would consist of a set of thermoelectric junctions having a capacity of about 190 w(e). The complete thermoelectric module and heat pipe assembly would be prepared in the factory and shipped as a unit such as the module shown in Fig. 17. It would be installed in the isotope power unit at dockside when that unit was prepared for operation.

Complete Power Unit

A drawing showing the general configuration of a complete assembly of the power unit is presented in Fig. 18, and the principal dimensional and performance data are summarized in Table 2. Note that the thermoelectric units are inside the tubes that project from the top of the pressure vessel containing the heat block and shield assembly. With this arrangement, caps at the top of the tubes containing the thermoelectric units can be removed, the seal welds cut, and an individual module can be

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Fig. 16. Horizontal Cross Section Through a Thermoelectric Module

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Fig. 17. Vertical Cross Section Through a Thermoelectric Module

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Fig. 18. Complete Thermoelectric Power Unit

Table 2

Dimensional and Performance Data for the Reference
Design 2 kw(e) Isotope Power Unit with
Thermoelectric Converters

Power Plant Performance

Net power output (beginning of life), kw(e)	2.26
Net power output (after 5 yr), kw(e)	2.0
Isotope heat output (beginning of life), kw(t)	34
Isotope heat output (after 5 yr), kw(t)	30
Voltage output, v	48

Thermoelectric Modules

Thermoelectric junction diameter, in.	0.40
Thermoelectric junction length, in.	1.20
Thermoelectric junction hot shoe temperature, °F	1000
Thermoelectric junction cold shoe temperature, °F	100
Number of thermoelectric junctions per module	192
Output per module, w	192
Output per module, v	12
Number of modules	12
Total number of junctions	2304
Heat pipe OD, in.	1.0
Heat pipe ID, in.	0.9
Heat pipe length, in.	79
Heat pipe working fluid	Potassium
Heat pipe operating temperature (beginning of life), °F	1064
Module weight, lb	70

Heat Block-Shield

Material	Armco Iron
Diameter over fin tips, in.	40.5
Height, in.	59.2
Radiation dose at surface, mr/hr	200
Cooling fin height, in.	2.50
Cooling fin thickness, in.	0.50
Cooling fin centerline spacing, in.	2.50
Number of cooling fins	45
Heat block-shield assembly weight, lb	16,800

Table 2
(continued)

Fuel Capsules	
Fuel material	SrTiO ₃
Number of fuel capsules	7
Fuel capsule OD, in.	4.15
Fuel slug OD, in.	3.4
Fuel capsule length, in.	40
Fuel capsule centerline spacing, in.	5.70
Temperatures	
Fuel capsule maximum surface temperature during normal operation, °F	1520
Fuel capsule maximum surface temperature during shipping in 100°F air, °F	1300
Shield surface temperature during shipping in 100°F air, °F	450
Heat block temperature at heat pipes during normal operation, °F	1087
Heat pipe vapor temperature, °F	1064
Pressure Vessel	
Material	HY-100 Series Steel
Design depth, ft	12,000
OD, in.	51
ID, in.	48
Height, in.	72
Weight, lb	6400
ID of tubes for thermoelectric modules, in.	3.75
OD of tubes for thermoelectric modules, in.	4.25
Length of tubes for thermoelectric modules, in.	20
Overall height of complete power unit, in.	90
Total weight of complete power unit, lb	24,000

removed and replaced as desired. Note that the threaded caps have been provided to protect the seal welds from accidental damage.

The heat block-shield assembly is essentially the same as that designed for the organic Rankine cycle. It makes use of 7 isotope fuel capsules arranged on 5.70-in. centerlines at the center of an iron shield. Axial fins on the sides serve for heat removal by thermal convection in the surrounding air during shipping or storage.

The power unit would be assembled by first attaching the support skirt at the bottom of the shield assembly to the flange provided in the lower part of the pressure vessel (see Fig. 18). The thermal insulation under the shield would be installed prior to this operation, of course, and that over the sides and top would be installed after attaching the support skirt. Note, too, that the support skirt joint employs a set of radial keys of thermal insulation to reduce heat losses and to accommodate differential thermal expansion between the hot shield and the cold pressure vessel.

The thermoelectric modules would be installed in the upper portion of the pressure vessel, and this assembly would then be lowered into place with the heat pipes bayonetting into the heat block-shield assembly. The flange in the pressure vessel would be seal-welded and then secured with a hoop clamp.

The thermal insulation surrounding the shield will be of the same design as that chosen for the organic system. The design selected will give good thermal insulation at normal operating temperatures and yet will melt out to give a high heat loss in the event that the shield overheats as a consequence of some emergency.

A rough draft of a topical report presenting details of the design analysis has been prepared and is being reviewed critically. When this review is completed, a test program to investigate a promising heat transport module geometry will be evolved.

5.0 FUEL CAPSULE DESIGN

R. A. Robinson

Strontium Fuel Capsule Design

A 1000-W SrTiO_3 capsule has been designed to withstand an external pressure of 10,000 psi at a temperature of 400°F. This capsule differs from earlier designs in size (larger diameter) and end closure.

The larger diameter capsule presented here should reduce the overall weight of the fuel block as well as lower the capsule surface operating temperature.

The top cap of the capsule is threaded to provide strength without the necessity of deep penetration welding with electron beam techniques. The electron beam equipment requires several days to install in a shielded cell which is costly. If a quality assurance method such as ultrasonic inspection of the weld is required, several additional days are required to install and calibrate the ultrasonic equipment. By providing a threaded end cap, the capsule can be sealed with a GTA weld. The GTA welding equipment can be installed and the capsule welded in the same day. A helium leak check can be used for assurance of the weld integrity. As in the case of the GTA welding equipment, the helium leak detection equipment can be quickly installed in a shielded cell.

The bottom cap of the capsule is designed for a deep penetration (100 mil) electron beam weld. Since this closure is made prior to installation in the shielded cell, none of the above mentioned problems of equipment installation and calibration in a shielded cell are confronted.

The same stress analysis used in the Isotope Kilowatt Study Phase I was used to determine the wall and end cap thicknesses of the capsule. Figure 19 is a preliminary drawing of the capsule. The length dimension can vary up to 24 in.

An empty capsule of essentially the same design as shown in Fig. 19 was hydrostatically tested at ORNL recently in connection with another program and it successfully withstood 10,000 psig with only minor distortion of the end caps. The end cap failed at 13,200 psig. In an actual

Fig. 19. Modified SNAP 23 Type Capsule.

use situation the end caps will obtain additional support from the fuel pellet stack. The effect of long periods at the proposed operating temperature on the strength of Hastelloy C-276 are not known. It is known that there is a substantial loss of ductility after a relatively short time (<100 hr) at temperatures in the range of 1500°F; however, if proper clearances are maintained between the capsule and fuel pellets, support offered by the fuel should compensate for the ductility loss.

Fuels

Hastelloy C-276 has been chosen as the material of construction for the fuel capsules because of its excellent compatibility with seawater and its demonstrated safe use with $^{90}\text{SrTiO}_3$ fuel in small marine and terrestrial thermoelectric generators. As noted in the Phase I report, there may be some advantage to considering the use of $^{90}\text{SrF}_2$ as the fuel form in large future systems because of its higher power density and its future availability in large quantities from the Hanford Waste Management program.

A preliminary thermodynamic analysis has been made of potential chemical reactions between SrF_2 and each of the alloying constituents in Hastelloy C-276. The total free energy changes (ΔF_{react}) in all of the reactions were positive indicating that there should be no simple incompatibilities. However, it is emphasized that this type of analysis is only useful in identifying obvious incompatibilities. It does not take into account the possibility of complex compound formation or solubility between constituents. In addition, it is necessary to consider the compatibility of the radioactive decay produce (^{90}Zr) in the fuel and capsule system.

A brief survey has been conducted to investigate current activities in the area of SrF_2 compatibility studies at different organizations and to assess the results of their studies. Only a limited amount of data generated at ORNL, the Martin Company, and BNWL was available. An attempt was made to compare the available data from different sources by consolidating the key information in Table 3.

It should be noted in Table 3 that all the simulated fuel materials used in the studies were nonradioactive with the exception of Series No. 10.

Table 3. Summary of SrF_2 Compatibility Studies Conducted or in Progress

Series No. ^a	Organization	Simulated Fuel Material			Encapsulation		Test Conditions		Depth of Penetration in Mils for Test Materials ^b												
		Composition	Method of Compaction	Bulk Density (% theo.)	No. of Capsules	Material	Temp. (°C)	Time (hr)	Hastelloy			Haynes 25		Ni-O-Ni	N-155	Inconel	Mo	Nb	Ta	TZM	W
									C	N	X	No Liner	Ta Liner								
1	ORNL	SrF ₂	Pressed (40,000 psi) and sintered pellet (6 mm diam x 5 mm high)	60-100	2	Outer: Inconel X Inner: Test Matl.	1000	1000	(c)	(c)	3 (V.F.)	4.5 (Rxn) 3 (V.F.)	(c)	10 (Rxn) 4 (V.F.)	2 (V.F.)	(c)	No Rxn deposit of small particles at interface	(c)	2 (Rxn)	(c)	Slight Rxn at interface
2	Martin	SrF ₂ (1 wt % Ca, 0.5 wt % Fe)	Step-pressed into test capsule at 50 psi	85-95	2	Outer: SS 321 Inner: Test Matl.	925	726 - 13,870 (19 mo)	0.8	0.5	1.6	Nil	Complete penetration of liner	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)
3	Martin	SrF ₂ + 2.5 wt % Zr as ZrO ₂	Step-pressed into test capsule at 50 psi	85-95	2	Outer: SS 321 Inner: Test Matl.	925	726 - 13,870 (19 mo)	0.3	1.1	1.1	0.8	Complete penetration of liner	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)
4	Martin	SrF ₂ + 5 wt % Zr as ZrO ₂	Step-pressed into test capsule at 50 psi	85-95	2	Outer: SS 321 Inner: Test Matl.	925	726 - 13,140 (18 mo)	0.8	0.8	2	0.6	Complete penetration of liner	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)
5	BNWL	SrF ₂ (~96.6 wt %; <1% Na, ~2% Ca, bal. Fe, Al, Mg)	Step-pressed into test capsule	80-90	2	Outer: Glass (Pyrex or Vycor) Inner: Test Matl.	400 800	3-36 mo	(d)	(c)	(d)	(d)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)
6	BNWL	SrF ₂ + 6 mol % Zr + 6 mol % ZrF ₄ (comp. of SrF ₂ same as above)	Step-pressed into test capsule	80-90	2	Outer: Glass (Pyrex or Vycor) Inner: Test Matl.	400 800	3-36 mo	(d)	(c)	(d)	(d)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)
7	BNWL	SrF ₂ + 13 mol % Zr + 13 mol % ZrF ₄ (comp. of SrF ₂ same as 5)	Step-pressed into test capsule	80-90	2	Outer: Glass (Pyrex or Vycor) Inner: Test Matl.	400 800	3-36 mo	(d)	(c)	(d)	(d)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)
8	BNWL	SrF ₂ (Same composition as 5)	Step-pressed into test capsule	80-90	2	Outer: Glass (Vycor) Inner: SS 304L	800	3 mo (2160 hr)	(d)	(d)	(d)	(d)	Uclimet	Rene 41	Waspalloy	Inconel	Incoloy 804	Multimet SS 304L	SS 316L	(d)	(d)
9	BNWL	SrF ₂ + 6 mol % Zr + 6 mol % ZrF ₄ (comp. of SrF ₂ same as 5)	Step-pressed into test capsule	80-90	2	Outer: Glass (Vycor) Inner: SS 304L	800	3 mo (2160 hr)	(d)	(d)	(d)	(d)	(d)	(d)	(d)	(d)	(d)	(d)	(d)	(d)	(d)
10	BNWL	⁹⁰ SrF ₂ (>95%)	Step-pressed into test capsule	80-90	1	SS 304L	400	3 mo (2160 hr)	(d)	(c)	(d)	(d)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)
		⁹⁰ SrF ₂ + 13 mol % ⁹⁰ Zr + 13 mol % ⁹⁰ ZrF ₄	Step-pressed into test capsule	80-90	1	SS 304L	400	3 mo (2160 hr)	(d)	(c)	(d)	(d)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)	(c)

^aExcept for Series No. 10 all the simulated fuel materials are non-radioactive.^bRxn: Depth of reaction, V.F.: Void formation; Depth of penetration is the average figure for the maximum exposure time.^cMaterial not tested in this series.^dThe runs either have been completed or are in progress, but no results are available as yet.

A comparison between Nos. 1 (1000°C, 1000 hr) and 2 (925°C, 13,870 hr) for Hastelloy X and Haynes No. 25 indicates that the extent of attack (in terms of the depth of penetration) on No. 1 appears to be more severe than that on No. 2. This result seems contrary to what can be expected from the difference in the test conditions between the two; that is, while the temperatures for the two were nearly the same (1000°C vs 925°C), the duration of exposure for No. 2 was approximately 14 times longer than that for No. 1.

The apparently contradictory result may be partially explained by the difference in the method of compaction of the simulated fuel material. The step-pressing method used by the Martin Company may tend to produce a compact that disintegrates readily, as evidenced by Fig. 1 of their monthly progress report No. 56, Strontium-90 Fuel Program, MND-2848-46, April 1966. This situation would result in a poor contact between the fuel material and the container material, which may account for the smaller depth of penetration for No. 2 than for No. 1.

As can be seen by comparing Nos. 2, 3, and 4, there does not seem to be any definite relation between the composition of the strontium material (ZrO_2 content) and the depth of penetration of the container material.

Also, the result for the tantalum liner (No. 2) cannot be compared directly with that for the tantalum container used in No. 1 because the thickness and other details on the liner were not given. According to the result of No. 1, molybdenum and tungsten are quite compatible with SrF_2 under the test conditions. In general, the available data appear to indicate only a slight degree of attack on various superalloys, while the attack on refractory metals was almost negligible.

The technical data on the compatibility of SrF_2 with various potential container materials are somewhat fragmentary and inconclusive at present. For example, the exposure time in Series No. 1 is not long enough to permit prediction of the long-term performance of the container materials. Nevertheless, the result is helpful in selecting suitable materials for long-term studies.

The results from Series Nos. 2 through 4 are satisfactory from the standpoint of exposure temperature and time. However, the validity of their results is rather questionable, especially in view of the pressing

technique that has been used in preparation of the simulated fuel material. The step-pressing of the fuel material into a container not only does not represent the actual encapsulation method of the radioisotope fuel, but also would tend to introduce stresses into the container. In addition, their photographs reveal that the pressed material has a tendency to disintegrate.

Practically all studies have been based on nonradioactive materials. The preliminary evidence with $^{90}\text{SrF}_2$ and $^{90}\text{SrF}_2 \cdot \text{Zr} \cdot \text{ZrF}_4$ from BNWL (Series No. 10, official result not yet available) indicates $\sim 50\%$ more attack by the radioactive $^{90}\text{SrF}_2$ than by the nonradioactive material. However, no conclusive result can be expected from their studies since both the temperature (400°C) and time (3 months) are far from the conditions to be encountered in the actual applications ($\sim 1000^\circ\text{C}$, ~ 10 years). Furthermore, the studies entailed so-called "scouting tests" with all the test capsule materials in a single container. This also introduces another uncertainty because of the possible interactions between the dissimilar materials.

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