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MASTER

REFERENCE DESIGN FOR AN ISOTOPE POWER UNIT EMPLOYING AN ORGANIC RANKINE CYCLE

G. Samuels
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G. Samuels
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JANUARY 1971

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REFERENCE DESIGN FOR AN ISOTOPE POWER UNIT
EMPLOYING AN ORGANIC RANKINE CYCLE

G. Samuels and R. S. Holcomb

ABSTRACT

A reference design for a 5 kw(e) Rankine cycle power unit using Dowtherm A has been evolved to serve as the basis for evaluation tests in Phase II of the Isotope Kilowatt Program. The heat source consists of seven SrTiO_3 fuel elements 3.40 in. in diameter and 38.4-in. in length encapsulated in Hastelloy C-276. The unit is designed for undersea operation at a depth of 6000 ft and 5 yr life. The required beginning-of-life thermal output is 34 kw.

The fuel elements are located on a triangular pitch in the center of a low-carbon steel heat block-shield which serves as both the shield and as a heat conductor to transfer the heat from the fuel to reentry boiler tubes located in the outer part of the shield. Saturated vapor at 600°F from the 12 reentry boiler tubes passes through a combined separator boiler drum, to the turbine and then through the regenerator to the upper head of the pressure vessel which serves as the condenser. The liquid from the condenser flows to the feed pump, passes around and cools the generator, then through the regenerator and back to the boiler drum.

INTRODUCTION

The objective of the organic Rankine cycle power conversion system study is to evolve by the end of 1972 a Conceptual System Design Description, a detailed Technical Development Plan which highlights the status of existing technology and describes areas requiring technological advancements, and recommends, in detail, experimental and developmental activities necessary to achieve this advancement. The system is to be designed for a lifetime of 5 years of unattended operation and have an output of 5 kw(e) at the end of 5 years. The system design is to be based on the use of SrTiO_3 fuel with sufficient flexibility that SrF_2 may be substituted at a later date without extensive redesign or re-evaluation. Other groundrules for the study are:

1. The fuel block-shield will be designed to be safely transported and stored in a 120°F still air environment utilizing passive heat dissipation techniques only. Both AEC and DOT regulations governing the transport of radioactive materials will be met.

2. The system will be designed to operate in seawater at a depth of 6000 ft and at a temperature of 40°F. Design flexibility will be provided to permit eventual operation at a depth of 20,000 ft.

3. The system will be designed to permit field replacement or maintenance of the power conversion equipment and regulating equipment without the need for hot cell facilities.

The present design work on the organic Rankine cycle is restricted to the use of Dowtherm A as the working fluid. During the Phase I part of the program,¹ which was complete 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 the data that is available concerning their properties, such as thermal stability, and experience with rotating equipment is limited. 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.

The system 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 low temperature (100 to 200°F). Although the design effort is directed toward undersea use, the basic concepts must be applicable to land-based use. In addition to these design objects, 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 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 waste 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 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 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.

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 unattended for long periods of time. A second closely

related factor which 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 unit which will be sufficiently flexible to serve a variety of possible applications, be able to accommodate one of the CRU's now under development and still meet the design precepts listed above. This design will then be the basis for relatively simple tests to demonstrate the adequacy of the components selected. Following the demonstration tests, a conceptual system design description 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 feed pump on a single shaft. Unlike steam, saturated Dowtherm A vapor, when expanded through the turbine, becomes superheated rather than wet. 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%.

The design of the system is shown in Fig. 2. The vapor flow, starting at the boiler, is as follows: from the boiler to a combined separator-vapor drum, to the turbine and then through the regenerator to the condenser. The liquid from the condenser flows to the feed pump, passes around and cools the generator, then through the regenerator and back to the vapor drum. A summary of the performance and dimensional data is given in Table 1.

Table 1. Performance and Dimensional Data for the Reference Design
5 kw(e) Isotope Powered Organic Rankine Cycle
Energy Conversion System

Item	Data
Plant Performance	
Isotope heat output (beginning of life), kw(t)	34
Isotope heat output (end of life), kw(t)	30
Net power output (beginning of life), kw(e)	5.7
Net power output (end of life), kw(e)	5.0
Design life, yr	5
Fuel Capsules	
Fuel material	SrTiO ₃
Number of fuel capsules	7
Fuel capsule OD, in.	4.10
Fuel pellet OD, in.	3.40
Fuel capsule length, in.	40.4
Fuel length, in.	38.4
Fuel capsule centerline spacing, in.	5.65
Heat Block-Shield	
Material	Iron
Diameter over fin tips, in.	40.5
Height, in.	59.2
Fin height, in.	2.50
Fin thickness, in.	0.50
Fin centerline spacing, in.	2.50
Number of fins	45
Radiation dose at surface, mr/hr	200
Heat block-shield assembly weight, lb	17,200
<u>Heat Block-Shield and Fuel Temperatures</u>	
Normal Operation	
Maximum fuel temperature, °F	1900
Maximum cladding temperature, °F	1360
Maximum block temperature, °F	1210
Block surface temperature, °F	820
Shipping	
Maximum fuel temperature, °F	1580
Maximum cladding temperature, °F	1020
Maximum block temperature, °F	870
Block surface temperature, °F	480

Table 1.(continued)

Item	Data
Accident Conditions (loss-of-coolant)	
Maximum fuel temperature, °F	2400
Maximum cladding temperature, °F	1890
Maximum block temperature, °F	1810
Block surface temperature, °F	1360
Heat loss during normal operation, kw	3
Boiler	
Heat load, kw	31-27
Temperature, °F	613-600
Pressure, psia	51.6-45.6
Total vapor flow rate, lb/hr	500-436
Boiler tube linear heat rating, Btu/hrft	1885-1640
Average heat flux, Btu/hrft ²	9730-8480
Number of tubes	12
Centerline diameter of tubes, in.	31.0
Outer tube OD, in.	1.050
Outer tube ID, in.	0.742
Inner tube OD, in.	0.540
Inner tube ID, in.	0.302
Vapor riser OD, in.	0.840
Vapor riser ID, in.	0.546
Boiler drum OD, in.	8.625
Boiler drum ID, in.	7.981
Maximum radiation dose rate in boiler tubes, r/hr	3.5
Maximum radiation dose rate in boiler drum, r/hr	0.15
Regenerator	
Effectiveness, %	90
Liquid temperature in, °F	223
Liquid temperature out, °F	420
Vapor temperature in, °F	485
Vapor temperature out, °F	250
Log mean temperature difference, °F	43
Liquid flow rate, lb/hr	436
Vapor flow rate, lb/hr	436
Liquid volumetric flow rate, ft ³ /sec	0.00205
Vapor volumetric flow rate, ft ³ /sec	51.8
Heat load, Btu/hr	40,000
Required (U x A), Btu/hr.°F	930
Vapor pressure loss, psia	0.05
Vapor-side face area, in. ²	162
Length of matrix, in.	12

Table 1. (continued)

Item	Data
Condenser	
Temperature, °F	223
Pressure, psia	0.10
Vessel surface area, ft ²	8.3
Volume, ft ³	10
Pressure Vessel	
Material	Hy-100 Series Steel
Design depth, ft	6,000
OD, in.	49.75
ID, in.	48
Height, in.	156
Weight, lb	7,000
Internal operating pressure, psia	0.1
Internal gas	air
Total system liquid inventory, lb	23
Total weight of complete power unit, lb	25,000

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

Fig. 2. Vertical Section Through the Reference Design for a 5 kw(e) Organic Rankine Cycle System.

ORGANIC RANKINE CYCLE EFFICIENCY

A parametric study was made of the efficiency of the Rankine cycle employing Dowtherm A as the working fluid. The parameters varied were the turbine inlet temperature, condenser pressure, and turbine-generator efficiency. The range of the parameters was as follows: turbine inlet temperature - 500 to 700°F, condenser pressure - 0.1 to 0.5 psia and turbine-generator efficiency - 61 to 66%.

The thermodynamic properties of the superheated vapor at pressures as low as 1 psia were obtained from Ref. 2. The properties of the superheated vapor at pressures from 1 to 0.1 psia were calculated from the published values at 1 psia by assuming that the properties of the vapor were related according to the perfect gas law. The entropy of the vapor was calculated from the thermodynamic relationship

$$\left(\frac{\partial S}{\partial P}\right)_T = - \left(\frac{\partial V}{\partial T}\right)_P$$

which becomes for a perfect gas

$$\left(\frac{\partial S}{\partial P}\right)_T = - \frac{R}{P} .$$

From the published and calculated values of the vapor properties a Mollier chart (enthalpy-entropy diagram) was constructed and is presented in Fig. 3.

It may be observed from the Mollier chart that an isentropic or real expansion process beginning with saturated or superheated vapor proceeds into the superheat region so that the vapor would leave the turbine exhaust with a considerable amount of superheat. While this is desirable as a means for preventing erosion or moisture losses in the turbine, it results in a rather low thermal efficiency unless the superheat is conserved in the cycle by regenerative heating of the condensate with the turbine exhaust vapor. The cycle efficiency is increased by about 50% with a regenerator of high effectiveness over the efficiency

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Fig. 3. Mollier Diagram of Superheated Vapor Region for Dowtherm A.

of the cycle without a regenerator. Because of the incentive to employ as effective a regenerator as is practical, a regenerator effectiveness of 90% was assumed for the cycle study. A slight penalty resulting from the use of a regenerator is the pressure loss of the vapor flowing through the vapor side of the regenerator from the turbine to the condenser. A constant pressure loss of 0.05 psi was assumed for all cases and, as a consequence, the turbine exhaust pressure was taken as being 0.05 psi higher than the condenser pressure for each case.

Single-stage turbine efficiencies of about 70% and generator efficiencies of about 92% have been reported.³ The bearing losses and other mechanical losses can be accounted for by a mechanical efficiency of about 94%. The efficiency of the feed pump was assumed to be 40%, which is about the value that can be expected from a pump mounted on the turbine shaft and hence operating at the high speed of the turbine. The work done by the turbine wheel was calculated for a given turbine efficiency, the pump work was calculated and subtracted, the difference was multiplied by the mechanical efficiency of 94%, and this quantity was multiplied by the generator efficiency of 92% to obtain the net power output. Two values of turbine efficiency were assumed; 70.6 and 76.1%, which resulted in an overall effective turbine-generator efficiency of 61 and 66% when multiplied by the mechanical and generator efficiencies.

The relations between the cycle efficiency, turbine inlet temperature, condenser pressure and turbine-generator efficiency are shown in Fig. 4. For a turbine-generator efficiency of 61%, the cycle efficiency varied from 15.8 to 22.1% as the turbine inlet temperature was changed from 500 to 700°F at a condenser pressure of 0.1 psia. It is interesting to note that decreasing the condenser pressure from 0.25 to 0.10 psia for a fixed value of turbine inlet temperature gives a slightly larger increase in cycle efficiency than increasing the turbine inlet temperature by 50°F for a constant value of 0.25 psia for the condenser pressure. This is an important factor when selecting the optimum conditions for the cycle when the effect of the turbine inlet temperature on the fluid decomposition is taken into account.

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Fig. 4. 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.

RADIOLYTIC AND PYROLYTIC DECOMPOSITION

One of the most important factors influencing not only the cycle efficiency but also the design of the system components is the decomposition of Dowtherm A and the effects of the products of decomposition on the system operation. As will be shown later, pyrolytic rather than radiolytic decomposition is expected to be the primary cause of decomposition.

Data for decomposition and hydrogen formation rates are shown in Figs. 5 thru 9. The hydrogen formation rate for Dowtherm A is assumed to be the same as the biphenyl. These data are the same as reported in Ref. 1 where they are discussed in more detail.

Figures 10 and 11 give the total decomposition and volume of hydrogen as a function of cycle efficiency for various boiler temperatures and turbine exhaust pressures after 50,000 hr of operation. The efficiencies shown here are different than those given in the cycle efficiency section because of different assumptions concerning component efficiencies. However, the values shown in Figs. 10 and 11 are internally consistent and illustrate the effect of cycle efficiency on decomposition and the gas volume that must be accommodated in the system.

The total decomposition of the liquid that can be tolerated in the system is not well defined. For the system presently under study the decomposition of the liquid is not expected to be a major problem. The products of decomposition⁴ for long periods of operation are primarily hydrogen and higher boiling point liquids which are approximately equal quantities of ortho-, meta- and para-terphenyl. The effect of these terphenyls can be illustrated by the following example which is a gross simplification of the actual processes. If one assumes a total decomposition of 30% of the Dowtherm, the mole fraction of the components of the liquid will be approximately 79% Dowtherm and 7% of each of the terphenyls. Assuming that Raoult's law is valid for the solution, the total vapor pressure can be determined. The vapor pressures at 600°F for the pure components are 45.6, 8.3, 3.8 and 2.7 psia for Dowtherm, ortho-, meta- and para-terphenyl respectively. Using Raoult's law the vapor pressure of the solution is 37.0 psia, a reduction of 8.6 psia from that for pure Dowtherm.

Considering that the vapor is mostly Dowtherm and that the heat of vaporization is not too different for the components, the total weight of liquid evaporated for a given heat input is the same as for pure Dowtherm. The turbine will have supersonic nozzles for which the flow rate is directly proportional to the upstream pressure. For the turbine output to remain constant, the pressure must be increased to its original design point for pure Dowtherm or 45.6 psia. This in turn means that the boiler temperature for the solution must be increased to about 625°F. There will be a small increase in the molecular weight of the vapor which will effect the flow through the turbine nozzles, however, the amount of terphenyls in the vapor phase will be less than 5%.

The above discussion was for a constant heat input in the system. For an isotope powered unit, the heat input decreases with time and this decrease will partly offset the effect of the high boilers on the boiling temperature. For example, with Sr fuel and a five year lifetime the decrease in heat output is about 11.3% which would offset about one-half the temperature increase caused by the terphenyls.

Thus, the effects of the terphenyls are minor for a system with a recirculating type of boiler. The effect of the terphenyls on a once-through type of boiler are different. In this case, the boiler temperature must be increased to evaporate the terphenyls or else there will be some liquid carry over to the turbine. Boiler temperatures were not analyzed for this case.

For systems in which the Dowtherm is exposed to low radiation doses and temperatures in the range of 600 to 650°F, the formation of solids or tars is not expected to be significant. The hydrogen generated will affect the design of the condenser and this will be discussed in the condenser section of this report.

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Fig. 5. 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).]

Fig. 6. Volume of Hydrogen Produced from Pyrolysis of Biphenyl.

Fig. 7. Volume of Hydrogen Produced from Addition of Pyrolytic Effects to Radiolytic Effects of Gamma Rays on Biphenyl for 5×10^4 hr.

ORNL DWG. 69-14115

Fig. 8. First-Order Rate Constant for Pyrolysis of Dowtherm A.
 [From: Aerojet-General Corporation, Organic Rankine-Cycle Power System Demonstration Programs, AGC-3486 (December 1967), p. 24.]

Fig. 9. Pyrolytic Degradation of Dowtherm A.

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

Fig. 11. Volume of Hydrogen Evolved from Biphenyl per Pound of Boiler Inventory as a Function of Cycle Efficiency After 5×10^4 hr Exposure.

HEAT BLOCK-SHIELD

In developing the design of the heat block-shield, a set of shipping and design precepts, listed below, was established:

1. The fuel block-shield will be designed to be safely transported and stored in a 120°F still air environment utilizing passive heat dissipation techniques only. Both AEC and DOT regulations governing the transportation of radioactive materials will be met.

2. The shielded isotope heat source will 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 surface of the shield.

3. The shield and heat block will be designed so that under normal conditions the temperature of the Hastelloy-276 cans encasing the isotope will not exceed 1700°F, and in no foreseeable set of accident conditions would the temperature of the Hastelloy cans exceed 2000°F.

4. 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 will be at least 0.020 in.

5. 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.)

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 iron or steel units 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 can be used with either a steam or an organic Rankine cycle system, and 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 presently projected AEC waste management program at Hanford. Starting this year, the AEC is expected to be routinely removing 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 (Fig. 12). The advantage of this arrangement is that the power output with all seven positions fueled with SrTiO_3 (power density = 0.85 w/cc) is within one percent of that with SrF_2 (power density = 1.0 w/cc) using only the outer six fuel positions.

Fuel Capsules

Two fuel diameters were considered for the seven element design. The first size considered (4.15 in.) is the same as presently used in the SNAP 23 units. As will be shown later in the thermal analysis section, the maximum cladding temperature for this size fuel pellet during accident conditions was about 2000°F and for normal operation with the steam Rankine cycle system the maximum cladding temperature was 1700 to 1800°F. 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 of most cases the dimensions will be governed by the accident conditions. For example, with the organic Rankine cycle system operating with a boiler temperature of

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Fig. 12. Heat Block-Shield.

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 shield block design using a 3.4-in. diameter fuel body was 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 reentry boiler tube which greatly improves the performance of this type of boiler. It was also found that the smaller fuel diameter leads to a heat block-shield that can be used with the thermoelectric power conversion system. This then will give a single heat block-shield design that can be used with all of the systems presently being studied.

The fuel and cladding dimensions and heat rates for the two fuel diameters are shown in Table 2. The total heat output required is 34 kw. Although the thermal requirements for 5 kw(e) is only 30 kw the design lifetime of 5 years and the 28 year half-life of Sr lead to a beginning-of-life requirement of 34 kw. The dimensions shown in Table 2 allow a diametrical clearance between the fuel and the cladding of 0.050 in. The capsule length includes a 0.25 in. allowance for the 4.15 in. diameter pellets and 0.20 in. for the 3.4 in. diameter pellets. The length of the holes in the heat block-shield was assumed to be 0.20 in. longer than the capsules for both cases and the diameter of the holes was assumed to be 0.050 in. larger than the fuel capsule diameter.

Table 2. Fuel and Capsule Parameters for 34 kw(t)
and Seven Element Design with SrTiO_3

Fuel Pellet			Fuel Capsule		
Diameter (in.)	Linear Rating (kw/in.)	Element Length (in.)	Diameter (in.)	Element Length (in.)	Heat Flux ^a (Btu/hr·ft ²)
4.15	0.186	26.1	5.00	27.9	5830
3.40	0.126	38.4	4.10	40.4	4750

^aBased on fuel length.

Shielding Analysis

For the shielding calculations, the power density in the SrTiO_3 was assumed to be 0.85 w/cc and the gamma emission rate (photons/w-sec) were taken directly from Ref. 5 and are shown in Table 3.

Table 3. Gamma Emission Rates
for SrTiO_3

Energy (mev)	Gamma Emission Rate (photons/w-sec)
0.25	1.705×10^{11}
0.50	6.612×10^{10}
0.80	1.958×10^{10}
1.10	6.103×10^9
1.40	1.650×10^9
1.70	3.027×10^8
2.00	2.082×10^7

The method used to calculate the surface dose rate was to first determine the dose rate for a single element and then add the contributions of the individual elements to obtain the total dose rate. The calculation procedure used to determine the dose rate for a single element are those described in Chapter 11 of Ref. 6. For the case of the end shields the upper limit value was used. The shield block thickness calculated for the end shield differed by less than 0.1 in. from the side thickness and the side thickness was used for the weight, size, and heat flux data which follows.

The results of the calculations for the 4.15 in. diameter SrTiO_3 fuel pellets are shown in Fig. 13. The upper solid lines are for the total amount of shielding required. The lower dashed lines are for the thickness required for the iron shield block. The difference represents the thickness of the cladding and end caps used to encapsulate the fuel. A comparison of the total side shield thickness required for various fuel pellet diameters is shown in Fig. 14.

ORNL DWG. 71-374

Fig. 13. Surface Dose Rate vs. Shield Thickness for 4.15 in. Diameter SrTiO_2 Fuel and an Iron Shield.

ORNL DWG. 71-375

Fig. 14. Dose Rate vs. Side Shield Thickness for SrTiO_3 Fuel and an Fe Shield for Three Representative Fuel Capsule Diameters.

The heat block-shield dimensions and weight are shown in Figs. 15 and 16 as a function of the surface dose rate and for two ligament thickness. The ligament thickness is the distance between the fuel holes in the block. The weight includes the fuel and cladding material. It should be noted that Figs. 15 and 16 do not include the axial fins which will be located on the outer diameter of the block. These fins will be 2-1/2 in. high and will add 5 in. to the diameter.

All of these results were for SrTiO_3 . For identical designs, the shielding requirements for SrF_2 will be greater than that for SrTiO_3 because of its greater power density and also because of its larger gamma emission rate. Although no data were found for the production of bremsstrahlung photons for SrF_2 the rates should be between those reported for SrO and SrTiO_3 . The combination of these two factors will give a gamma source strength for SrF_2 about 1.4 times that for SrTiO_3 . This increased source strength will require an additional 0.3 in. of side shield thickness to keep the surface dose rate the same as for the SrTiO_3 . This additional shielding can be obtained by reducing the ligament thickness between the elements. As will be seen in the thermal analysis section, the spacing for the six element SrF_2 design can be decreased by 1/2 in. (thus increasing the shielding by 1/2 in.) and still maintain the capsule cladding temperature slightly below that for the seven element SrTiO_3 design. No additional end shield thickness will be required for the SrF_2 . The elimination of the central fuel position more than compensates for the increased source strength.

Thermal Analysis

The thermal analysis of the heat block-shield was separated into three parts; radiation and conduction in the inner fueled region, conduction through the shield and radiation, and convection from the outer shield surface.

Inner Fueled Region

The temperature distribution within the inner fueled region is quite complex and depends to a large extent on the manner in which

Fig. 15. Heat Block-Shield Dimension and Weight vs. Surface Dose Rate and Ligament Thickness for 34 kv and 4.15 in. Diameter Fuel.

Fig. 16. Iron Heat Block-Shield Dimensions and Weight vs Surface Dose Rate and Ligament Thickness for 34 kw and 3.40 in. Diameter Fuel.

the heat is transferred from the fuel to the cladding and from the cladding to the shield block. These calculations were made using a two-dimensional, (radial and angular), computer code. The method of calculation is described in Appendix B of Ref. 1. The code calculates the steady-state temperature within the fuel, cladding, and shield block, taking into consideration the internal heat generation in the fuel, the radiation and conduction of heat from the fuel to the cladding, the 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 temperature-dependent variables. The transfer of heat from the fuel to the cladding is assumed to be by conduction through a 0.025 in. gap filled with argon and by radiation, 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 conduction through a 0.025 in. gap filled with air and by radiation with the emissivity of each surface again equal to 0.6.

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 Btu/hr·ft·°F, are given below, with the temperature, t , in °F:

SrF₂

$$k = 0.65$$

SrTiO₃

$$k = 2.10 - 9.29 \times 10^{-4}t + 2.56 \times 10^{-7}t^2 - 2.98 \times 10^{-11}t^3$$

Hastelloy C-276

$$k = 5.32 + 5.59 \times 10^{-3}t$$

Iron

$$k = 4.19 - 1.56 \times 10^{-2}t - 8.17 \times 10^{-6}t^2 + 5.09 \times 10^{-9}t^3$$

Argon

$$k = 1.00 \times 10^{-2} + 1.18 \times 10^{-5}t$$

Air

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

A 30° section of a 7-element fuel block is shown in Fig. 17, and a typical temperature distribution through the region is given in Table 4. In this case the fuel is SrTiO_3 , the fuel diameter is 3.40 in. and the ligament thickness is .1 in. The temperatures shown in Table 4 are based on an assumed temperature of 1000°F at the outer diameter of the fueled region which is defined as the distance across the three holes in the heat block-shield plus the ligament thickness.

Table 4. Temperature Distribution^a for
3.4-in. Diameter Fuel Pellet and
a 1.0-in. Ligament Thickness

Line No. ^b	Temperature Distribution (°F) for Radial Positions a-e ^c				
	a	b	c	d	e
1	1971	1593	1433	1296	1289
2	1971	1593	1432	1286	1275
3	1837	1531	1376	1289	1289
4	1837	1521	1360	1261	1260
5	1837	1495	1320	1180	1176
6	1837	1464	1270	1064	1050
7	1837	1442	1237	1015	1007
8	1837	1431	1222	1015	1002
9	1837	1427	1217	1011	1001

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

^bCorresponds to numbers shown on Fig. 17.

^ca = center of fuel element; b = OD of fuel; c = ID of cladding; d = surface of shield block; e = 1/2 in. into shield block.

The use of a liquid metal bond to increase the heat transfer coefficient between the fuel capsule and the shield block was considered during the Phase I study. This 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

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Fig. 17. Typical Section of a Seven Element Heat Block.

cause material compatibility problems; also, if the liquid metal were lost, excessive fuel capsule temperatures might result.

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. Helium, because of its high thermal conductivity, would be preferred. 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 using the thermal conductivity of air 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 very low value of thermal conductivity. Thermal conductivity values of several gases and saturated vapors at 1400°F are given below:

	Thermal Conductivity (Btu/hr.ft.°F)
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.

The results of the calculations for the fueled region of the shield block are shown in Tables 5 and 6. Because of the variation in both thermal conductivity and radiation with temperature, the calculations were made for temperatures of 1000 and 1400°F at the OD of the fueled region. Although the temperature differences between the various components and the reference temperature are valid only for the reference temperature shown, they may be used for other reference temperatures varying by 100 to 200°F with very little error.

Table 5. Maximum Fuel, Cladding and Block Temperature for 7-Element Design with SrTiO_3 Fuel

Reference Temperature (°F)	Fuel Diameter (in.)	Ligament Spacing (in.)	Temperature (°F)		
			Fuel	Cladding	Block
1000	4.15	0.50	2578	1799	1680
1000	4.15	0.75	2493	1712	1576
1000	4.15	1.00	2440	1656	1509
1000	4.15	1.50	2371	1583	1420
1000	3.40	1.00	1971	1433	1296
1000	3.40	1.50	1932	1392	1246
1400	3.40	1.00	2343	1820	1727
1400	3.40	1.50	2304	1781	1683

Table 6. Maximum Fuel, Cladding and Block Temperature for 6-Element Design with SrF_2 Fuel

Reference Temperature (°F)	Fuel Diameter (in.)	Ligament Spacing (in.)	Temperature (°F)		
			Fuel	Cladding	Block
1000	3.40	1.00	2262	1382	1248
1000	3.40	1.50	2253	1352	1203
1400	3.40	1.00	2549	1746	1661
1400	3.40	1.50	2547	1730	1638

Table 5 is for SrTiO_3 with seven fueled positions in the central region of the block. The 1000°F reference temperature is in the range expected at this block position during normal operation. For the organic system this temperature will be about 1000°F while for the steam or thermoelectric systems the temperature will be about 1100°F . The 1400°F reference temperature cases are expected to be typical of those existing at this position during accident conditions.

Table 6 shows similar data for SrF_2 fuel pellets. As the power density in SrF_2 is about 18% greater than for SrTiO_3 , six rather than seven fuel elements are required. The arrangement of the SrF_2 elements will be the same as for the SrTiO_3 unit except that the central element will be omitted. One important point that should be noted about the results shown in Tables 5 and 6 is that the ligament spacing for the SrF_2 can be made by 1/2 in. less than that for the SrTiO_3 without increasing the capsule cladding temperature above that for the SrTiO_3 . Thus, one can obtain additional shielding for the SrF_2 design without the cladding temperature exceeding that for the SrTiO_3 unit.

Shield Region

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.ft. $^\circ\text{F}$

L = fueled length, ft

D_2 = outside diameter of the shield

D_1 = outside diameter of the fueled region.

The temperature difference across the side shield using only the fueled length is shown in Fig. 18 for both the 3.40 and 4.15-in. diameter fuel pellets and for 1 and 1.5-in. ligament thicknesses. The thermal conductivity for all cases was assumed to be 24 Btu/hr.ft.°F. These temperature differences will always be too large for the applications and types of operation expected in the present design. For all cases in which the heat is conducted through the shield, the length over which the heat is removed from the outer part of the shield will be greater than the fuel length. For example, with the 3.40-in. fuel pellets, a surface dose rate of 200 mr/hr and for accident conditions in which the heat is being transferred from a 4-ft axial length of the shield surface, the temperature drop across the shield will be about 10% less than that shown in Fig. 18. For the case in which the entire shield surface is losing heat, as during shipping, the effective temperature drop across the shield will be about 25% rather than 10% less than that shown in Fig. 18.

Outer Shield Surface

The temperature of the outer surface of the shield is dependent on the manner in which the heat is being removed and thus on the specific type of operation. During normal operations the heat is removed from within the shield and the outer surface temperature is of interest only to determine the heat lost through the insulation. However, during shipping and during accident conditions all of the heat generated within the fuel must be transferred from the shield surface.

The surface heat flux from the various sizes considered are shown in Table 7 assuming that different parts of the surface are transferring heat. The surface areas considered are: the shield OD over a length equal to the fueled length, the total length, and a 4-ft length (except for the 4.15-in. fuel size for which the total length is less than 4 ft) and for the total surface area. The reason for including the 4-ft length

Fig. 18. Temperature Difference Across Side Shield vs. Surface Dose Rate for 1.0 and 1.5 in. Ligament Thickness and 3.40 and 4.15 in. Fuel Pellets.

Table 7. Shield Surface Heat Flux

Fuel Pellet Diameter (in.)	Surface Dose Rate (mr/hr)	Ligament Spacing (in.)	Surface Heat Flux (Btu/hr.ft ²)			
			Fueled Length	Total Length	4 ft Length	Total Surface
3.40	100	1.0	3870	2460	3100	2140
		1.5	3770	2390	3010	2080
	200	1.0	4020	2620	3220	2280
		1.5	3910	2530	3130	2200
	500	1.0	4220	2820	3380	2470
		1.5	4100	2740	3280	2390
	1000	1.0	4400	3010	3530	2640
		1.5	4270	2920	3410	2550
	1500	1.0	4520	3130	3620	2750
		1.5	4380	3030	3500	2650
	2000	1.0	4590	3220	3670	2820
		1.5	4450	3110	3560	2720
4.15	100	1.0	5280	2970	-	2460
		1.5	5160	2890	-	2380
	200	1.0	5450	3150	-	2610
		1.5	5320	3060	-	2530
	500	1.0	5720	3420	-	2840
		1.5	5550	3330	-	2750
	1000	1.0	5920	3660	-	3040
		1.5	5770	3560	-	2950
	1500	1.0	6080	3810	-	3160
		1.5	5890	3700	-	3050
	2000	1.0	6170	3920	-	3260
		1.5	5980	3810	-	3150

case is that the side insulation that serves as a safety device which will melt during accident conditions is expected to be about 4-ft long.

The heat is transferred from the surface of the shield by thermal radiation and convection. The shield surface will have axial fins along the full length which are 2-1/2-in. high by 1/2 in. thick located on 2-1/2-in. centers. Figure 19 shows the heat flux from the side surface of the shield as a function of surface temperature for conditions typical of shipping and handling. This curve is based on the assumption that the unit is exposed to air and surrounding at 120°F. The emissivity of the surface is assumed to be 1.0. By treating the surface an emissivity of about 0.9 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. Figure 20 shows the heat transferred from the shield surface to a second surface having a variable emissivity and a temperature of 1200°F. These curves are applicable to the accident condition and will be discussed later. For both of these curves the area used for radiation was based on the OD of the shield rather than the OD of the fins. This difference amounts to 10 to 15% and more than compensates for the effect of fin efficiency which was taken as 1.0.

Heat Block-Shield Reference Design

The design selected for the heat block-shield uses the 3.40-in. diameter fuel pellets with the shield thickness necessary to reduce the surface dose rate to 200 mr/hr. The primary reason for selecting the 3.40 in. rather than the 4.15-in. diameter fuel pellet size is to obtain a single unit that can serve all three of the systems being studied and also to keep down the capsule cladding temperature during accident conditions. The shield thickness was chosen to met DOT shipping regulations without the need for auxiliary shielding or packaging and also to keep dose levels sufficiently low to minimize problems in the final assembly of the unit in the field.

The dimensional and weight data for this unit using SrTiO_3 is:

Fuel pellet diameter, in.	3.40
Cladding OD, in.	4.10

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Fig. 19. Heat Flux by Thermal Radiation and Natural Convection to 120°F Ambient.

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Fig. 20. Heat Flux by Thermal Radiation to Surface at 1200°F with Variable Emissivity.

Ligament spacing, in.	1.50
Spacing between elements, in.	5.65
Heat block-shield length, in.	59.2
Heat block-shield diameter, in.	35.5
Fin tip diameter, in.	40.5
Total weight, lb	17,200

If SrF_2 fuel is used in place of SrTiO_3 all of the above data would remain the same except that the ligament spacing would be reduced to 1.2 in. and the spacing between the elements would then be 5.35 in.

Shipping Regulations

Shielding Requirements

The Department of Transportation shielding requirements for shipping casks depend on whether the vehicle is used exclusively for shipment of the unit. The regulations governing the transport of shipping casks are reproduced below.

"All radioactive materials, liquid, solid and gaseous must be packaged in suitable containers (shielded, if necessary) so that any time during transportation the radiation dose rate does not exceed any of the limits specified in the following subparagraphs.

.....

1. 200 millirem per hour at any point on the external surface of the package.
2. 10 millirem per hour at three feet from any accessible external surface of the package.

"Packages for which the radiation dose rate exceeds the limits specified above but does not exceed at any time during transportation any of the limits specified in subparagraphs (1) through (4) below, may be transported in a vehicle (except aircraft) assigned for the sole use of that consignor, and unloaded by the consignee from the transport vehicle in which originally loaded.

1. 1000 millirem per hour at three feet from the external surface of the package (closed transport vehicle only);
2. 200 millirem per hour at any point on the external surface of the car or vehicle (closed transport vehicle only);
3. 10 millirem per hour at six feet from the external surface of the car or vehicle; and

4. 2 millirem per hour or equivalent in any normally occupied position in the car or vehicle except this does not apply to private motor carriers."

Assuming exclusive use of the vehicle, the limiting dose rate will be the 10 mr/hr at 6 ft from the external surface of the car or vehicle. Using the data reported in Ref. 5 to determine the dose rate at a distance from the shield surface and assuming that a point 6 ft from the external surface of the vehicle is 9 ft from the surface of the shield, the dose rate is 10 mr/hr.

Structural Standards for Packaging

There are two standards⁷ which apply to the packaging of large quantities of radioactive material for shipment. The first set is for normal conditions of transport and the second set is for hypothetical accident conditions.

Normal Conditions of Transport

A package used for the shipment of a large quantity of radioactive material shall be so designed and constructed and its contents so limited that under the normal conditions of transport:

- a. There will be no release of radioactive material from the containment vessel;
- b. The effectiveness of the packaging will not be substantially reduced;
- c. There will be no mixture of gases or vapors in the package which could, through any credible increase of pressure or an explosion, significantly reduce the effectiveness of the package;
- d. Radioactive contamination of the liquid or gases primary coolant will not exceed 10^{-7} curies of activity of Group I radionuclides per milliliter, 5×10^{-6} curies of activity of Group II radionuclides per milliliter, 3×10^{-4} curies of activity of Group III and Group IV radionuclides per milliliter; and
- e. There will be no loss of coolant or loss of operation of any mechanical cooling device.

The normal conditions of transport are:

- a. Heat — Direct sunlight at an ambient temperature of 130°F in still air.

- b. Cold - An ambient temperature of -40°F in still air and shade.
- c. Pressure - Atmospheric pressure of 0.5 times standard atmospheric pressure.
- d. Vibration - Vibration normally incident to transport.
- e. Water Spray - A water spray sufficiently heavy to keep the entire exposed surface of the package, except the bottom, continuously wet during a period of 30 min.
- f. Free Drop - Within 2.5 hr after conclusion of the water spray, a free drop through the distance specified below onto a flat essentially unyielding horizontal surface. Impact with the surface occurs in a position for which maximum damage is expected.

Free Fall Distance

Package weight (lb)	Distance (ft)
<10,000	4
10,000 to 20,000	3
20,000 to 30,000	2
>30,000	1

- g. Corner Drop - A free drop onto each corner of the package in succession, or, in the case of a cylindrical package, onto each quarter of each rim, from a height of 1 ft. This test does not apply to packages that are not constructed primarily of wood or fiberboard, to packages exceeding 10,000 lb in weight.
- h. Penetration - Impact of the flat circular end of a vertical steel cylinder, 1.25-in. in diameter and weighing 13 lb, dropped from a height of 4 ft impacting normally onto the exposed surface (of the package) that is expected to be the most vulnerable to puncture.
- i. Compression - For packages not exceeding 10,000 lb in weight, a compressive load equal to either five times the weight of the package or 2 lb/in.^2 multiplied by the maximum horizontal cross section of the package, whichever is greater. The load shall be applied, for a period of 24 hr, uniformly against the top and the bottom of the package. The package should be positioned as in normal transport.

None of the above conditions are expected to present any problems.

Figure 19 shows the side heat flux as a function of the surface temperature for radiation and convection to 120°F surroundings. For the reference design heat block-shield and these conditions the maximum surface temperature of the block will be about 480°F and the temperature drop through the

shield at the midplane will be about 140°F giving a maximum temperature of 620°F at the OD of the fueled region. Referring to the data in Table 5, it can be seen that the maximum temperature of the block will be an additional 250°F and for the fuel element cladding an additional 400°F. Thus the maximum internal block temperature will be about 870°F and the maximum cladding temperatures about 1020°F.

The effect of increasing the ambient temperature to 130°F and adding the solar heat load has little effect on these temperatures. The maximum solar heat load is about 300 Btu/hr·ft². Assuming the 300 Btu/hr·ft² incident on all surfaces of the shield will increase the temperatures by only 30°F. Reducing the ambient temperature to -40°F will reduce the temperature by about 100°F.

It is difficult to visualize the other conditions, with the possible exception of item 6, causing serious damage to the shield. The effect of a free drop will be discussed under the hypothetical accident conditions.

Hypothetical Accident Conditions

A package used for the shipment of a large quantity of radioactive material shall be so designed and constructed and its contents so limited that if subjected to the hypothetical accident conditions

- a. The reduction of shielding would not be sufficient to increase the external radiation dose rate to more than 1000 milliroentgens per hour or equivalent at one meter from the external surface of the package.
- b. No radioactive material would be released from the package except for gases and contaminated coolant containing total radioactivity exceeding neither:
 1. 0.1% of the total radioactivity of the package contents; nor
 2. 0.01 curie of Group I radionuclides, 0.5 curie of Group II radionuclides, and 10 curies of Group III and Group IV radionuclides, except that for inert gases, the limit is 1000 curies.

The hypothetical accident conditions are:

- a. Free Drop - "A free drop through a distance of 30 ft onto a flat, essentially unyielding, horizontal surface, striking the surface in a position for which maximum damage is expected.

- b. Puncture - "A free drop through a distance of 40 in. striking in a position maximum damage is expected, the top end of a vertical cylindrical mild steel bar mounted on an essentially unyielding horizontal surface. The bar shall be 6 in. in diameter, with the top horizontal and its edge rounded to a radius of not more than 1/4 in., and of such a length as to cause maximum damage to the package, but not less than 8 in. long.
- c. Thermal - "Exposure for 30 min within a source of radiant heat having a temperature of 1475°F and an emissivity coefficient of 0.9, or equivalent. For calculational purposes, it shall be assumed that the package has an absorption coefficient of 0.8. The package shall not be cooled artificially until after the 30-min test period has expired and the temperature at the center of the package has begun to fall.
- d. Water Immersion - "Immersion in water for 24 hr to a depth of at least 3 ft."

Because of the uncertainties and inherent approximations in engineering design, compliance with these regulations must sometimes be demonstrated by tests rather than analytical treatment. Tests⁵ have been made on a spent fuel shipping cask similar to the reference heat block-shield. These tests were made using a Nodular Iron (Spheriodal-Graphite Cast iron). The test model was 42 in. in diameter and 43 in. long with a 7-in. hole in the center. The unit weighed 15,600 lb. Drop tests were made from a height of 9 meters onto a concrete pad covered with a steel plate. The unit was dropped on each end with the axis vertical, with the axis horizontal and on one corner. After all drops the only deformation of the cast body was a very small bevel around the outer edge. No cracks could be found. The unit penetrated the steel plate on the corner drop.

Extensive model tests are currently being conducted at ORNL for the Liquid Metal Fast Breeder Program spent fuel shipping casks. These casks are expected to weigh up to 250,000 lb and be made of steel. SAE 1020 steel is presently being used in the model tests.

The only concern with damage to the heat block-shield when subjected to these conditions is damage to the plugs that are welded to the top of the block. As these plugs serve as the end shield for the fuel holes, any loss of a plug would result in radiation streaming from the top of the shield. The possibility of this type of damage can be minimized in

several ways. One, the plugs can be threaded into the block and then welded, or the plug can be made in two parts with the inner part threaded and the outer part threaded and welded. An alternate method is to add a shock absorbing material or device to protect this area.

The thermal and water immersion conditions will not present any problems. The easiest manner to show that the thermal condition is no problem is to assume the 1475°F flame as a heat sink rather than a heat source. Using a 1475°F heat sink the surface temperature of the shield will be 1520°F, the maximum iron temperature will be 1900°F and the maximum cladding temperature 2000°F. Actually, the heat capacity is so large that the temperature increase for the various parts of heat block-shield will be only about 20% of that given by the above assumption.

The immersion of the heat block-shield in water will lead to a high thermal stress in the shield. However, the worst that one would expect would be minor cracking of the outer surface.

BOILER

The boiler chosen for the reference design is a recirculating type unit, which uses a reentry tube in the boiling section (see Fig. 21). The selection of a recirculating boiler was based on the need for a simple, stable unit and the fact that with Dowtherm A one does not want to get superheated vapor. The reentry tube feature of the design was chosen to meet the design precepts that the arrangement of 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 the piping or equipment containing the Dowtherm A.

The alternative to the recirculating boiler is a once-through unit. The once-through boiler has a possible advantage in that the amount of Dowtherm at the boiling temperature may be less than in a recirculating boiler, which could result in a smaller amount of hydrogen being formed. However, the hydrogen generation rate is a strong function of the temperature and at temperatures in the range of 600 to 650°F a reduction of 25°F in the boiler temperature would compensate for the increased inventory in

Fig. 21. Reentry Tube Boiler for 5 kw(e) Organic Rankine Cycle.

the recirculating type unit. It can be seen from the cycle efficiencies shown in Fig. 4 that 25°F reduction in the boiler temperature (i.e., turbine inlet temperature) results in a less than 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 longer to dry the vapor completely. This in turn makes it very difficult to devise a design to meet the precept of coupling the heat source to the boiler at dockside without locating the boiler tubes outside the finned heat block-shield. This would increase the diameter of the pressure vessel and also increase the operating temperature of the heat block-shield.
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 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. Also the effects of high boiling point components such as the terphenyls could require considerably higher wall temperatures in order to obtain a dry vapor from the boiler.
4. The feed flow to the boiler must be very tightly coupled to the thermal output of the heat source. 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 unit operates in a vertical or near vertical position. Another difference between space and terrestrial applications is that the weight and size of the systems, especially the space radiators is heavily dependent on the cycle efficiency and thus on the operating temperature, whereas this is not the case for an undersea power plant.

The boiler design consists of 12 reentry tubes connected to a boiler drum where the vapor and liquid are separated. The tubes have an active boiling length of about 56 in. and the height of the liquid leg is 68 in. The total heat load at the beginning-of-life is 31 kw and the vapor flow rate per tube is 41.7 lb/hr.

A hydrodynamic analysis of the boiler was made to determine the boiler tube dimensions required to get a boiler exit quality of about 30%. The two-phase flow analysis was based on the dimensionless parameters reported for water-steam mixtures⁸ at a boiler pressure (350 psia) where the ratio of the liquid-to-vapor density is the same as for Dowtherm A at 600°F saturated conditions.

A summary of the boiler and heat block dimensions and operating conditions are given in Table 8. The boiler tube holes in the shield block are the holes into which the reentry tubes are inserted. The surface temperature of these holes is 900°F. This high temperature is one reason why the recirculating rather than the once-through boiler was selected and also why one must limit the exit quality from the boiler. For high exit qualities the heat transfer coefficient decreases to a very low value and the boiler tube approaches the 820°F block temperature. The decomposition rate at this temperature is orders of magnitude greater than at 600°F.

The total liquid inventory in the reentry tubes and the vapor risers is 138 in.³ or 3.95 lb. The inventory in the boiler drum is about 50 in.³ or 1.43 lb for each in. of liquid level in the drum. Assuming that the operating level in the drum is 6 in. and that all of the liquid in the drum, tubes and risers is at 600°F, the total amount of hydrogen formed by thermal decomposition is about 1.9×10^{-3} ft³ at standard conditions of temperature and pressure of a volume of 0.37 ft³ at the condenser pressure (0.1 psia) for 50,000 hr of operation. The amount of liquid at 600°F that is decomposed is about 18% (see Fig. 10). Combining the liquid at 600°F with the remaining systems inventory gives the total decomposition of 10% for the total inventory of the system.

The actual amount of decomposition can be reduced by baffling the boiler drum in such a manner that part of its inventory is at a temperature equal to the mixed mean temperature of the return liquid from the regenerator (420°F) and the recirculating liquid in the boiler (600°F). For the 30% exit quality from the boiler tubes the mixed mean temperature in the boiler drum, one must prevent or minimize the heat transfer (primarily condensation) between the vapor in the drum and the colder

Table 8. Boiler and Heat Block Dimensions
and Operating Conditions

Item	Data
Boiler	
Temperature, °F	613-600
Pressure, psia	51.6-45.6
Heat load, kw	31-27
Total vapor flow rate, lb/hr	500-436
Boiler tube linear heat rating, Btu/hr.ft	1885-1640
Average heat flux, Btu/hr.ft ²	9730-8480
Number of tubes	12
Outer tube OD, in.	1.050
Outer tube ID, in.	0.742
Inner tube OD, in.	0.540
Inner tube ID, in.	0.302
Vapor riser OD, in.	0.840
Vapor riser ID, in.	0.546
Boiler drum OD, in.	8.625
Boiler drum ID, in.	7.981
Maximum radiation dose rate in boiler tubes, r/hr	3.5
Maximum radiation dose rate in boiler drum, r/hr	0.15
Liquid Inventory	
Inner tube, in. ³	62.4
Annuli, in. ³	62.4
Vapor riser, in. ³	13.4
Boiler drum, in. ³ /in.	50.0
Heat Block-Shield	
Boiler tube hole diameter, in.	1.070
Hole circle diameter, in.	31
Boiler tube hole surface temperature, °F	820
Maximum heat block temperature, °F	1210
Maximum fuel cladding temperature, °F	1360
Maximum fuel temperature, °F	1900

liquid. There are several ways in which this can be done. One very simple way is to insert a perforated plate across the drum at a level of about 4 in. and bring the liquid from the regenerator into the drum just below the plate. This will leave 2 in. of hot liquid above the plate to prevent condensation and serve as an insulating layer between the vapor and colder liquid below the plate. As the flow of hot liquid down through the plate is $2\frac{1}{3}$ times the flow rate from the regenerator, the plate will prevent the agitated cooled fluid from being mixed with the hot insulating layer. This will reduce the amount of hydrogen formed by about 50% and reduce the volume of hydrogen at condenser operating conditions to about 0.2 ft^3 . The decomposition of liquid based on total inventory will be about 6.5%.

The maximum radiation dose rate expected in the tubes is 3.5 r/hr and in the boiler drum is less than 0.15 r/hr. The decomposition due to those dose rates are negligible in comparison to the thermal decomposition rates.

REGENERATOR

The heat transfer and pressure drop characteristics were calculated for a series of regenerators for the Dowtherm A Rankine cycle. A set of design conditions which was considered typical for the cycle were selected as a basis to determine approximately the regenerator size required for the system. The design conditions selected were as follows: turbine inlet temperature - 600°F , condenser pressure - 0.1 psia, turbine efficiency - 70%, and regenerator effectiveness - 90%. The net output of the generator was 5 kw which required an input of 25 kw thermal in the boiler for the cycle efficiency calculated for these conditions. The corresponding flow and thermal characteristics which are required of the regenerator to meet these cycle conditions are given in Table 9.

A finned-tube type of construction was selected on the basis of providing a design which could provide a favorable ratio of vapor-to-liquid velocity and be readily fabricated. Since the vapor side pressure drop is a primary consideration, one pass was selected for the vapor side and multiple passes for the liquid side. Since the liquid volumetric

flow rate is quite low, it was initially apparent that the liquid flow area should be kept small in order to achieve a reasonable liquid-side heat transfer coefficient. Preliminary calculations indicated that two liquid-side configurations would give satisfactory results: (1) liquid flow through a single 1/2-in. OD 0.035-in. wall tube with multiple passes in a bank and multiple banks, (2) liquid flow through two tubes 3/8-in. OD 0.035-in. wall with multiple passes in a bank and multiple banks.

Table 9. Flow and Thermal Characteristics of Regenerator

Item	Data
Liquid temperature in, °F	223
Liquid temperature out, °F	420
Vapor temperature in, °F	485
Vapor temperature out, °F	250
Log mean temperature difference, °F	43
Liquid flow rate, lb/hr	403
Vapor flow rate, lb/hr	403
Liquid volumetric flow rate, ft ³ /sec	0.0019
Vapor volumetric flow rate, ft ³ /sec	47.9
Heat load, Btu/hr	36,980
Required (U × A), Btu/hr.°F	860
Liquid inventory, lb	
Single tube	8
Double tube	4

Since the Dowtherm A vapor has a low thermal conductivity and the vapor side pressure drop must be kept low, it is necessary to use extended surfaces on the vapor side to provide sufficient heat transfer area to give satisfactory overall performance. Plate fins made of nickel with a thickness of 0.010 in. and spaced at 16 fins per inch were chosen as the extended surface for the vapor side. A tube pitch of 1 in. in both the transverse and longitudinal direction was chosen with the location of the tubes staggered from one bank to the next in the direction of the vapor flow.

The calculation procedure was as follows: vapor side face dimensions with an aspect ratio of 1 were chosen, the overall heat transfer coefficient was determined, the number of banks required was found, and the

vapor-side and liquid-side pressure drops were calculated. The liquid-side heat transfer coefficients were calculated to be about 190 Btu/hr.ft².°F and the vapor-side heat transfer coefficients were calculated to be in the range of 5 to 7 Btu/hr.ft².°F. The vapor side face area was varied from 144 to 196 in.² for both the single 1/2-in. tube configuration and the two 3/8-in. tube design.

The resulting regenerator length and vapor-side pressure drop for these cases are shown in Fig. 22. For a vapor-side pressure drop of 0.05 psi the volume required for the regenerator matrix is about 1 ft³ for both the 1/2-in. tube and the two 3/8-in. tube designs.

The reference design regenerator employs the two 3/8-in. tube design. It was sized for the end-of-life power level of 27 kw instead of the 25 kw used in the design study, consequently the vapor-side face area required is slightly greater than that obtained for a pressure drop of 0.05 psi from the curve in Fig. 22. The characteristics of the reference design regenerator are given in Table 10.

Table 10. Characteristics of Reference Design Regenerator

Item	Data
Effectiveness, %	90
Liquid temperature in, °F	223
Liquid temperature out, °F	420
Vapor temperature in, °F	485
Vapor temperature out, °F	250
Log mean temperature difference, °F	43
Liquid flow rate, lb/hr	436
Vapor flow rate, lb/hr	436
Liquid volumetric flow rate, ft ³ /sec	0.00205
Vapor volumetric flow rate, ft ³ /sec	51.8
Heat load, Btu/hr	40,000
Required (U × A), Btu/hr.°F	930
Vapor pressure loss, psi	0.05
Vapor-side face area, in. ²	162
Length of matrix, in.	12
Liquid inventory, lb	4.3

Fig. 22. Length and Vapor-Side Pressure Drop as a Function of Vapor-Side Face Area of a Finned-Tube Regenerator.

CONDENSER

There are two features of the organic Rankine cycle system that require special attention in the design of the condenser. The first is that the system is hermetically sealed and has a design lifetime of 5 years which requires the condenser to be able to accommodate the non-condensables formed by the decomposition of the Dowtherm without seriously effecting the condensing heat transfer coefficient. The second factor is the rather high condensing temperature (225°F) of the Dowtherm. This high condensing temperature coupled with the low temperature (40°F) of the seawater can lead to excessive subcooling of the condensate which, in turn, has the potential of reducing the cycle efficiency by about $1/4$ or 4 to 5 points.

As can be seen in Fig. 23, the top head of the pressure vessel provides the condenser surface for rejecting heat from the cycle. The heat transfer from the vessel to the water is by natural convection and the heat transfer coefficient will be in the range of 100 to $125 \text{ Btu/hr.ft}^2.^{\circ}\text{F}$ for a water temperature of 40°F .

The heat transfer coefficient of the condensing Dowtherm will depend on the amount of non-condensables, the configuration of the condensing surface and the condensing temperature. For a maximum system temperature of 600°F and the boiler design presently contemplated, the amount of hydrogen that will be accumulated in the condenser will be about 2% by volume. The effect of the hydrogen on the condensing coefficient of Dowtherm is not known. Also, with the ribbed type condensing surface, the coefficient of pure vapor is not known very accurately. Thus one has the problem of providing sufficient surface area to compensate for the lack of good data for the condensing coefficient and at the same time prevent excessive subcooling of the condensate.

One method of accomplishing this is to design the condenser in such a manner that it will act as a direct contact regenerator so that any subcooled liquid must fall through the incoming vapor and be reheated. Figure 23 shows a design which uses ribs on the vessel head to both increase the condensing surface and also to collect the condensate from the upper head and prevent it from draining across the head. The condensate from the ribs then drops through the vapor spaces. To further increase

Fig. 23. Condenser for 5 kw(e) Organic Rankine Cycle.

the heat transfer between the condensate and the incoming vapor, a screen is located below the ribs to increase the heat transfer area by breaking up the droplets or any stream of condensate that may form.

The use of the ribs on the condensing surface has additional advantages for the present application. The ribs are "extended surfaces" or fins and by properly sizing their thickness one can make their effectiveness a function of the condensing coefficient. If the heat transfer coefficient is high, then the efficiency of the rib surface area is lower than for the case of a low heat transfer coefficient. Thus, as noncondensables build up in the system and cause a decrease in the heat transfer coefficient, the effectiveness of the ribs will increase to partially compensate for this. Another advantage of the ribs is that by increasing the product of the heat transfer coefficient and area one can increase the operating temperature of the vessel wall. This reduces the potential for subcooling the condensate.

In determining the size of the condenser, the following heat transfer coefficients ($\text{Btu/hr.ft}^2.\text{°F}$) were used: 110 for the water side, 275 for the vessel head based on a 6,000 ft design depth and 100 for condensing. The condenser heat load was assumed to be 22 kw. A summary of the condenser operating condition, volume and surface areas are given in Table 11. It should be noted that these conditions are for the end-of-life. At the beginning of life the condenser heat load will be about 25 to 26 kw but the condensing coefficient across the upper head of the vessel will be in the range of 200 to 400. The required coefficient at the beginning-of-life is less than 200 and the condenser pressure will be less than 0.1 psia. The condensing coefficient will decrease as noncondensables are accumulated in the condenser and a coefficient of 100 $\text{Btu/hr.ft}^2.\text{°F}$ was used for the end-of-life conditions.

An alternate design which is used with the organic space system is a jet condenser. Again the major advantages 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 jet condenser reentrains the noncondensables and recycles

them through the system. 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 shut down operation. The second advantage is the pressure recovery characteristics of the unit which give a high pressure at the pump suction.

Table 11. Condenser Operating Conditions and Dimensions

Item	Data
Heat load, kw	22
Design temperature, °F	223
Design pressure, psia	0.1
Cone base diameter, in.	39
Centerline diameter of ribs, in.	6,12,18,24,30,36
Height of ribs, in.	2
Thickness of ribs, in.	1/4
Water side area, ft ²	8.3
Condensing side base area, ft ²	7.6
Condensing side rib area, ft ²	11.5
Outer vessel wall temperature, °F	122
Inner vessel wall temperature, °F	155
Fin efficiency of ribs, %	30
Condenser volume, ft ³	10
Estimated liquid inventory, lb	3

There are several disadvantages in using a jet condenser. In addition to the jet condenser there must also be a heat dump to reject heat from the cycle, thus, this type of system 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 an excess 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.

COMBINED ROTATING UNIT AND SYSTEM CONTROL

There is little that can be discussed, with any assurance, about the combined rotating unit at this time because there are no units available that have demonstrated the reliability required for the present application. A project report discussing possible control schemes will be issued at a later date. This section of the report will be limited to a description of a type of pump which will enable that organic system to operate in a prescribed manner independent of the turbine-generator speeds or type of power output. This type of pump is one of several control schemes presently under consideration.

As pointed out in the introduction, the power plant is to be designed as a general purpose unit and as such it is expected that the plant must have load following capability. For a system with a constant heat source to have load following capability either the turbine shaft power must be kept constant and the energy dissipated by some parasitic device or the turbine speed allowed to increase with the excess energy dumped to the condenser. Figure 24 shows a pump which can decouple the design of the organic system and components from the power conversion system and also has characteristics such that the conventional level controls used in Rankine cycle systems are unnecessary regardless of turbine-generator speed.

The principle of operation is the same as that of a conventional pump with the exception that the inlet or eye is undefined and can vary to meet the system demands. The following analysis is simplified to illustrate the manner in which the pump operates. For solid body rotation of a fluid the head rise across the fluid is:

$$H = \frac{V_1^2}{2g} - \frac{V_2^2}{2g} = \frac{(\pi N)^2}{2g} (d_1^2 - d_2^2) = \frac{(\pi N d_1)^2}{2g} \left(1 - \frac{d_2^2}{d_1^2}\right) \quad (1)$$

where

H = head rise, ft·lb/lb

V = fluid velocity, ft/sec

Fig. 24. Constant Head Pump.

N = rotational speed, rps

g = gravitational acceleration, ft/sec²

d = diameter, ft

and the subscripts 1 and 2 refer to the outer and inner fluid diameter respectively. Thus, the head delivered is a function of the pump speed and the inner fluid diameter.

The flow to the pump is by gravity from the condenser and is

$$Q = C h^{0.5} \quad (2)$$

where h is the elevation in ft of the liquid surface in the condenser outlet above the center line of the pump. For the present design the system flow rate will decrease by about 11% in 5 yr due to decay of the Sr heat source. This means that the liquid level in the condenser outlet will be reduced by 21% of the distance between the initial level and the pump center line. Assuming this initial difference to be 1 ft, the change in liquid level will be between 2 and 3 in. and result in an insignificant transfer of fluid from the condenser to the boiler drum. It should be noted that the flow rates and liquid levels are independent of pump speed so long as the pump has sufficient head to deliver the incoming flow.

The head delivered by the pump is given by Eq. (1) and is shown in Fig. 25 where the head is shown as a function of the speed in rpm and the ratio of the inner to outer fluid diameter, for an outer fluid diameter of 1 in. For a boiler temperature of 600°F the required pump head is about 105 ft. By drawing a horizontal line on Fig. 25 for a head of 105 ft, one can determine the inner fluid diameter as a function of the pump speed.

For an isotope powered system in which the heat input to the boiler is slowly varying with time, the amount of vapor generated will also vary with time. For the present system in which the flow rate is determined by the size of the turbine nozzles (assuming critical flow through nozzles) the boiler pressure is directly proportional to the flow rate and thus the heat input. For a 5 yr operation life with Sr fuel the initial flow rate is about 11% greater than the end-of-life rate and the boiler temperature will start out at about 613°F ($p = 51.6$ psia). For these conditions

Fig. 25. Pump Head as a Function of Speed and the Ratio of Inner-to-Outer Fluid Diameter for a 1 in. Outer Fluid Diameter.

the required pump head is 120 ft and again Fig. 25 can be used to determine the pump conditions.

To illustrate the use of Fig. 25 let us assume that the design speed of the pump is 24,000 rpm. At the beginning-of-life the required head is 120 ft and the inner fluid diameter, or interface between the liquid and the vapor, will be 0.55 in. If the pump speed increases to 32,000 rpm the inner fluid diameter will increase to about 0.77 in. and continue to deliver the same flow and head. As the power decreases toward the end-of-life the required head drops to 105 ft and at 24,000 rpm the inner fluid diameter becomes 0.61 in. In all cases the flow through the pump is governed by the liquid level in the line from the condenser, which in turn depends on the flow rate or isotope heat output and not on the pump speed. Any tendency of varying pump speed to change the output head and flow will change the inner fluid diameter so that the head matches the boiler pressure. Thus the type of pump shown in Fig. 24 will maintain stable system operation with both varying heat input to the system and widely varying pump speed.

SAFETY PROVISIONS

During shipping and storage the heat block-shield will be exposed to air and surroundings at a maximum temperature of 130°F. As discussed previously the temperature of the shield, cladding and fuel for this condition is about 500°F less than for normal operating conditions with heat loss only by natural circulation and thermal radiation. A light weight cage structure would be placed around the unit to prevent accidental contact with the thermally hot surface.

The most serious incident in regard to the heat block-shield temperatures is a loss of normal cooling after the heat block-shield has been coupled to the power conversion system and enclosed in the pressure vessel. For this condition it is assumed that no heat can be given to the working fluid; thus there is no means of heat transport to the condenser.

In order to cope with this type of incident and also meet the design precepts that the cladding not exceed 2000°F and that the emergency heat-removal system must function satisfactorily with the unit in any attitude,

including on its side or inverted, a 4-ft long section of an aluminum screen type insulation is used between the heat block-shield and the pressure vessel. This safety concept takes advantage of the relatively low (1200°F) melting point of aluminum and the high (~ 0.9) thermal radiation absorptivity of a porous material. 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.

Figure 26 shows the heat block-shield surface temperature for the loss-of-coolant accident as a function of the shield and screen emissivity. As all heat transfer is by radiation the effective surface diameter of the block was taken as the diameter of the fin tips and the effective length was 4 ft. The heat flux, $2740 \text{ Btu/hr}\cdot\text{ft}^2$, was based on the beginning-of-life heat rate of 34 kw. The effective emissivity of the block surface, considering the effects of fin efficiency, will be about 0.9. The emissivity, or absorptivity, of the screen, before melting occurs, will be about 0.9. Thus, the surface temperature of the heat block-shield will be about 1300°F when the screen starts to melt.

The largest uncertainty with this safety provision is the behavior of the molten aluminum during melting. The screen, before melting occurs, has a void volume of about 85%. As melting proceeds it is probable that some of the porosity effects will be lost and that the emissivity of the melting surface will decrease. If one assumes that the emissivity drops to a rather pessimistic value of 0.25, the surface temperature of the block would be about 1480°F . Table 12 lists temperatures throughout the heat block-shield for several values of emissivity of the screen.

A test program is presently underway to determine not only the safety aspects of the screen but also its thermal conductivity. The thickness of the screen will be sized to limit the heat loss through the screen to about 5% of the block thermal output.

Fig. 26. Heat Block-Shield Surface Temperature During Loss-of-Coolant Accident as a Function of Shield and Screen Emissivity

Table 12. Heat Block-Shield Temperatures (°F)
for Several Values of Insulating
Screen Emissivity

Screen Emissivity	Block Surface	Maximum Block	Maximum Cladding	Maximum Fuel
0.9	1296	1745	1840	2360
0.7	1320	1770	1860	2380
0.5	1357	1810	1890	2400
0.25	1478	1940	2000	2480

PERMEATION OF HYDROGEN THROUGH METALS

All of the previous sections concerning hydrogen generation rates and accumulation in the system have ignored any effects of hydrogen permeation into or from the system. For a unit enclosed in a pressure vessel and operating in an air atmosphere at 0.1 psia, the effects of hydrogen permeation are small and for long periods of operation are beneficial. The initial reason for reducing the pressure in the vessel was to minimize gas inleakage into the system. As will be shown later, if the vessel is filled with air, the reduced pressure will also reduce hydrogen permeation into the system.

The equation giving the quantity of gas permeating a metal wall is

$$\phi = \frac{CA}{X} (P_1^{0.5} - P_2^{0.5}) \exp \left(\frac{-Q}{RT} \right)$$

where

ϕ = rate of gas permeating the wall, cc(STP)/hr

C = permeation constant, cc(STP)mm/hr·cm²·atm^{0.5}

A = area, cm²

P₁ = upstream pressure, atm

P₂ = downstream pressure, atm

X = wall thickness, mm

Q = activation energy, cal/mol

R = gas constant, cal/mol. $^{\circ}$ K

T = absolute temperature, $^{\circ}$ K.

The value of the permeation constant⁹ as a function of temperature for a number of materials is shown in Fig. 27. It should be noted that the above question assumes that Sievert's law ($\phi = KP^{0.5}$) is valid. This is not true for very low gas pressures where the active surface is not completely covered by the gas. In this case the permeation rate tends to vary linearly rather than with the square root of the pressure. However, for the rather low temperature (600 to 650 $^{\circ}$ F) being considered here, the diffusion rates are low and the effect of the varying pressure dependence should be relatively small. The following discussion neglects this deviation from Sievert's law.

For the present design there are a total of 13,420 cm² of surface area with a wall thickness of 3.91 mm and 3,230 cm² with a wall thickness of 8.18 mm which is operating at 600 to 625 $^{\circ}$ F and exposed to the Dowtherm. The amount of hydrogen in dry atmospheric air is 5×10^{-6} mole % which for a total pressure of 0.1 psia has a partial pressure of 3.4×10^{-9} atm. The only solubility data found for hydrogen is for solubility in biphenyl¹⁰ at 125 $^{\circ}$ C. At this temperature the solubility, S in cc(STP)/g, is given by

$$S = 0.0035p$$

where p is psia. Because of the lack of data relating solubility with temperature, the above equation was used for the condenser condition and first the equilibrium conditions in the loop were determined.

The total equivalent system inventory at 600 $^{\circ}$ F is about 8 lb and the generation rate of hydrogen is about 2.4×10^{-8} ft³(STP)/hr or 6.8×10^{-4} cc(STP)/hr. Using the permeation equation and the permeation constant for stainless steel at 600 $^{\circ}$ F one can calculate the pressure of hydrogen required in the condenser so that the permeation rate in the high temperature part of the loop is equal to the generation rate. Using the data listed above the required hydrogen pressure in the boiler is 1.9×10^{-6} psia and in the condenser is 1.5×10^{-3} psia. For the condenser operating at 0.1 psia the hydrogen percentage is 1.5 and for a total condenser volume of 10 ft³ the hydrogen volume at STP is 7.4×10^{-4} ft³.

Fig. 27. Effects of Temperature on Permeation Constants for Typical Metals and Alloys. USE THE RIGHT SCALE FOR NIOBIUM AND PALLADIUM AND THE LEFT SCALE FOR ALL OTHERS. (Taken from Ref. 9.)

Calculations were also made to determine the hydrogen build-up rate as a function of time. In making these calculations, it was first assumed that all of the hydrogen was in the condenser in order to determine the partial pressures throughout the system. Once the partial pressure is known then the rate of change of pressure with time and thus the pressure can be determined. The error involved in determining the system pressures in this manner is only a few percent. The results of these calculations are shown in Fig. 28. The hydrogen pressure in the condenser after 5 yr is about 1.0×10^{-3} psia or 1.0% of the total pressure. This is 40 to 50% of the amount calculated neglecting the effects of permeation. Thus the inflow of atmospheric hydrogen during the initial operation will be more than offset by the later outflow.

It is interesting to note that if the system were operating in a pressure of 1 atm the external hydrogen pressure would be 5×10^{-7} atm. For this condition the hydrogen pressure in the condenser would have to be 1.16×10^{-2} psia or 11.6% by volume for the system to be at equilibrium. The pressure as a function of time is shown in Fig. 28. Although this case does not apply to the undersea units, it does point out the need for using care in interpreting results from loops or capsule tests using metal walls exposed to atmospheric conditions.

There is one additional factor that should be considered in the design of systems in which very small amounts of hydrogen may be troublesome. The amount of hydrogen in the form of water vapor is several orders of magnitude greater than the amount of free hydrogen in air. This hydrogen may become available to diffusion through the wall by either corrosion or a surface dissociation mechanism. If further analysis shows this to be a problem for the present design, then either very dry air or an alternate gas low in both hydrogen and water vapor can be used in the pressure vessel.

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Fig. 28. Hydrogen Pressure in Condenser vs. Time.

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