OAK RIDGE NATIONAL LABORATORY

operated by
UNION CARBIDE CORPORATION • NUCLEAR DIVISION
for the
U.S. ATOMIC ENERGY COMMISSION

ORNL - TM - 2961

THIS DOCUMENT CONFIRMED AS
UNCLASSIFIED
DIVISION OF CLASSIFICATION
BY JH Kahn/amb
DATE 12/7/71

REFERENCE DESIGN OF A 3 kW(e) STEAM RANKINE CYCLE
SYSTEM FOR AN ISOTOPE POWER PLANT

A. P. Fraas

NOTICE This document contains information of a preliminary nature and was prepared primarily for internal use at the Oak Ridge National Laboratory. It is subject to revision or correction and therefore does not represent a final report.

R2341

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Contract No. W-7405-eng-26

Reactor Division

REFERENCE DESIGN OF A 3 kW(e) STEAM RANKINE CYCLE
SYSTEM FOR AN ISOTOPE POWER PLANT

A. P. Fraas

NOVEMBER 1971

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830
operated by
UNION CARBIDE CORPORATION
for the
U.S. ATOMIC ENERGY COMMISSION

Blank Page

CONTENTS

	<u>Page</u>
ABSTRACT	1
SUMMARY	1
HEAT BLOCK-SHIELD ASSEMBLY	2
STEAM TURBINE-GENERATOR UNIT	9
BOILER	14
Alternate Design	16
CONDENSER	18
DESIGN SUMMARY	18
Water Chemistry	22
Drainage	22
Control	23
DEVELOPMENT STATUS	30
REFERENCES	31

REFERENCE DESIGN OF A 3 KW(E) STEAM RANKINE CYCLE
SYSTEM FOR AN ISOTOPE POWER PLANT

A. P. Fraas

ABSTRACT

A reference design for a 3 kw(e) steam Rankine cycle system has been evolved for undersea applications. The system makes use of a heat block-shield assembly designed so that it is suitable for use with an organic Rankine cycle system or a thermoelectric system as well as the subject steam system. The turbine-generator used in the system is one developed by Swenson Research Laboratories, Inc. Unfortunately, only the initial development phase of this effort has been completed and funding is not available for further work. No other unit well suited to this application has been found.

The principal uncertainties and development problems associated with the proposed reference design are associated with the turbine-generator-pump unit. These include the problem of obtaining high reliability in the bearings, freedom from deterioration of the generator windings as a consequence of moisture permeation into the electrical insulation, and possible corrosion of the magnetic material exposed to steam in the generator. These problems appear tractable, but there is no funded effort directed toward their solution at the time of writing.

SUMMARY

The reference design presented in this report uses as its point of departure the conceptual design presented in the initial survey report prepared at the beginning of the Isotope Kilowatt Program.¹ The principal changes have stemmed from experimental work on the reentry tube boiler and revisions to the heat block shield design that have yielded a basic unit well suited to all three types of power plant chosen by the Navy for detailed study, i.e., the steam and organic Rankine cycle systems and the thermoelectric units. A review of turbine design considerations coupled with continued surveys of the development programs underway in the U.S. on small steam turbine-generator units indicates that the Swenson unit²

still provides as good a basis as any for a reference design. In general, the review has favored some small changes in the original conceptual design but no major changes.

The detailed analysis of the heat block-shield assembly is presented in the report covering the reference design for the organic Rankine cycle system³ and is not repeated here. Experimental tests on the reentry tube steam boiler are presented in another report.⁴ The problems of system control are so important from the reliability standpoint that a substantial section on system control is included in this report and another comprehensive topical report on system control is being prepared.

A review of the proposed reference design for the steam Rankine cycle system coupled to an isotope power source indicates that it has a number of important advantages as well as disadvantages. It appears that it will be possible to get a steam system in which radiation damage to the steam will have essentially no effect on the reliability or lifetime of the system, but this will require some new approaches in handling the water chemistry problem.⁵ If these new approaches do prove successful they would be applicable to much larger systems. This is important because in the larger sizes — above about 200 kw(e) — the steam systems will give higher thermal efficiencies than would be obtainable with either organic Rankine cycle or thermoelectric systems. Below 100 kw(e) the organic system is more efficient.

The principal barrier to the development of the steam Rankine cycle system is the fact that a suitable small turbine-generator unit has not been built and operated sufficiently to demonstrate the reliability of the design. However, there appears to be no basic reason why this could not be done at a reasonable cost, although similar projects have been costly.

HEAT BLOCK-SHIELD ASSEMBLY

A full account of the details of the design of the heat block-shield assembly is presented in the report covering the reference design for the organic Rankine cycle system.³ A further study comparing iron and nickel as materials for the heat block-shield assembly is presented in Ref. 6 and the design of the fuel capsule is presented in Ref. 7. Rather than repeat material from these presentations, suffice it to say here that the

basic design requirements for the heat block-shield unit for both the organic and steam Rankine cycle systems and for the thermoelectric system were set forth and a design was evolved to satisfy the requirements for all three systems. The compromises entailed in this work resulted in only minor increases in shield weight over shield designs tailored to each particular system.

The heat block-shield assembly chosen is shown in Fig. 1. The fuel is in the form of strontium titanate contained in 7 capsules made of Hastelloy as shown in Fig. 2. The shield is a large forging of low-carbon steel. The principal data on the dimensional and operating characteristics of the design are summarized in Table 1.

ORNL DWG 70-13981

Fig. 1. Sections through the heat block-shield assembly.

CAPSULE BODY

TOP CAP

BOTTOM CAP

NOTES:

1. ALL MAT'L SHALL BE HASTELLOY C-276
2. FINISH ALL OVER $\#8$
3. E.B. WELD BOTTOM CAP - .120 DIA. MIN
4. P. Rec. WELD TOP CAP - .055 DIA. MIN

NO REPRESENTATION OR WARRANTY, EXPRESSED OR IMPLIED, IS MADE THAT THE USE OR DISCLOSURE OF ANY INFORMATION, APPARATUS, METHOD OR PROCESS DISCLOSED IN THESE DRAWINGS MAY NOT INFRINGE PRIVATE RIGHTS OF OTHERS. NO LIABILITY IS ASSUMED WITH RESPECT TO THE USE OF, OR FOR DAMAGES RESULTING FROM THE USE OF, ANY INFORMATION, APPARATUS, METHOD OR PROCESS DISCLOSED IN THESE DRAWINGS. THE DRAWINGS ARE BEING MADE AVAILABLE FOR INFORMATION TO BIDDERS AND ARE NOT TO BE USED FOR OTHER PURPOSES, AND ARE TO BE RETURNED UPON REQUEST OF THE FORWARDING CONTRACTOR.

TOLERANCES UNLESS OTHERWISE SPECIFIED:

FRACTIONS $\pm \frac{1}{64}$ DECIMALS $\pm .005$ ANGLES $\pm$ SCALE: $\frac{3}{4}'' = 1''$

NO.		REVISIONS				DATE	APPD	APPD
DESIGNED	DATE	SUBMITTED	DATE	APPROVED	DATE	APPROVED	DATE	APPROVED
9-21-70								
DESIGNED	DATE	SUBMITTED	DATE	APPROVED	DATE	APPROVED	DATE	APPROVED
9-21-70								
CHECKED	DATE	APPROVED	DATE	APPROVED	DATE	APPROVED	DATE	APPROVED
9-21-70								

REFERENCE DRAWINGS	NUMBER
OAK RIDGE NATIONAL LABORATORY OPERATED BY UNION CARBIDE CORPORATION OAK RIDGE, TENNESSEE	
BLDG. NO.	
ISOTOPE KILOWATT PROGRAM FUEL CAPSULE	
SUBMITTED	APPROVED
DATE	DATE
CRD 2773	REV.

Fig. 2. Drawing showing the construction of the fuel capsule.

Table 1. Performance and dimensional data for the reference design for a 3 kW(e) isotope powered steam 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)	3.3
Net power output (end of life), kW(e)	2.9
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	1010 Steel
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
Heat Block-Shield and Fuel Temperatures	
Normal Operation	
Maximum fuel temperature, °F	1980
Maximum cladding temperature, °F	1440
Maximum block temperature, °F	1300
Block surface temperature, °F	1050
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	720
Pressure, psia	200
Total vapor flow rate, lb/hr	113
Boiler tube linear heat rating, Btu/hr·ft	1885-1640
Average heat flux, Btu/hr·ft ²	9730-8480
Number of tubes	12
Centerline diameter of tube array, in.	21
Outer tube OD, in.	1.050
Outer tube ID, in.	0.800
Inner tube OD, in.	0.500
Inner tube ID, in.	0.430
Turbine-Generator Unit	
Net power output, kW(e)	3.3
Output voltage, V	56
Output frequency cps	1000
Turbine rpm	60,000
Diameter, in.	6
Length, in.	10
Steam flow, lb/hr	80.5
Efficiency of turbine-generator without feed pump, %	38
Equivalent power loss in feed pump, kW(e)	0.45
Steam System	
Enthalpy of steam to turbine, Btu/lb	1384
Enthalpy of steam leaving nozzle, Btu/lb	966
Pressure of steam leaving nozzle	2.0
Moisture in steam leaving nozzle, %	13.4
Velocity of steam leaving nozzle, ft/sec	4580
Efficiency of turbine, %	50
Enthalpy of steam leaving turbine, Btu/lb	1175
Superheat of steam leaving turbine, °F	155
Saturation temperature at turbine exit, °F	100
Enthalpy of saturated steam at 100°F, Btu/lb	1105
Enthalpy of saturated water at 100°F, Btu/lb	68
Enthalpy of feedwater to boiler (300 psia), Btu/lb	384

Table 1 (continued)

Item	Data
Condenser	
Plate-coil outside diameter, in.	48
Plate-coil length, in.	60
Total steam side surface area, ft ²	30
Total external surface area, ft ²	62.8
Total inlet flow passage area, in. ²	4.1
Seawater temperature, °F	40
Condenser steam temperature, °F	100
Condenser steam pressure, psia	0.95
Pressure Vessel	
Material	Hy-100 Steel
Design depth, ft	6,000
OD, in.	49.75
ID, in.	48
Height, in.	88
Weight, lb	4350
Internal operating pressure, psia	0.1
Internal gas	N ₂
Total System Liquid Inventory, lb	23
Total Weight of Complete Power Unit, lb	22,000

STEAM TURBINE-GENERATOR UNIT

The design of all of the components in the steam system is dependent on the choice of steam conditions, and these in turn are determined by the choice of turbine-generator unit. In the course of the Phase I study only one small, hermetically sealed, steam turbine-generator unit with water lubricated bearings was found to be under development. The characteristics of this unit² (built by Swenson Research Laboratories) are summarized in Table 1. A total of around 1000 hr of testing was obtained with this unit, but further development work was stopped in mid-1969 because of lack of financial support. Swenson has done some further design work, and would probably use a homopolar generator rather than a permanent magnet rotor if they were to build another unit. This would avoid the demagnetization problem under load-shorting conditions, a problem that proved annoying both to Swenson and to TRW in the SNAP-2 program. They would also probably use a centrifugal pump on the turbine shaft for the system feed pump, and would use a jet pump to suppress cavitation in the centrifugal pump.

For lack of anything better it seemed reasonable to make use of the characteristics of the Swenson turbine-generator unit as a basis for developing the reference design for the steam Rankine system. However, it did seem desirable to analyze the turbine design problem sufficiently to draw some conclusions relative to the appropriateness of the Swenson design for Navy isotope power units. As a consequence, basic turbine-generator design considerations were reviewed and a series of analyses carried out in an effort to determine whether a considerably more attractive design might be evolved.

A review of bearing friction and rotor windage losses coupled with an examination of the rotor and bearing dynamics led to the conclusion that the Swenson selection of a single-stage impulse turbine was a sound choice. The single-stage Terry turbine also has the advantages of relatively low cost and simple construction. Further, once the choice of a single-stage turbine is made, the range of design conditions of interest can also be limited. In the first place, the moisture content of the steam leaving the nozzle ought not exceed 10 or 15% to avoid difficulties with turbine-bucket erosion. Since the system is to be designed for a 40°F

ambient sea water temperature, and since a 60°F allowance for the temperature drop through the surface films seems adequate, the turbine can be considered as discharging to a condenser steam temperature of approximately 100°F. If one fixes the nozzle exit quality and steam condenser pressure in this fashion, he fixes not only the enthalpy and entropy of the steam at the nozzle exit but also the entropy at the nozzle inlet because the expansion through the nozzle will be a very nearly perfect isentropic expansion. As a consequence, the steam inlet temperature is defined by the choice of steam inlet pressure, and thus it is possible to plot a series of curves such as that in Fig. 3 to show the effect of inlet steam pressure on the principal parameters affecting the design of the turbine. Note that two curves are given for the efficiency, one for an ideal generator with an efficiency of 100% and the second for a generator with an adiabatic efficiency of 85%. The latter value is consistent with experimental data obtained not only by Swenson but by other organizations that have tested small high speed, steam turbines. The better axial flow, partial admission wheels have given about the same turbine efficiency as the partial admission Terry turbine wheel in these small sizes.^{8,9} Of particular interest are the curves in Fig. 3 showing the steam velocity leaving the nozzle into the wheel, the specific volume of the steam leaving the nozzle, and the cross-sectional area of the jet entering the turbine wheel. The two former values define the area of the steam jet entering the turbine wheel and hence give an indication of the fraction of the turbine wheel perimeter utilized for admission, in this case for a turbine designed to deliver about 3 kw(e).

The curves of Fig. 3 indicate a number of important points. The first of these is that the turbine wheel tip speed must be quite high (it should be about 40% of the nozzle exit velocity) and hence the turbine bucket stresses induced by centrifugal force will be a controlling consideration. This in turn implies that a high strength-weight ratio material such as titanium is best suited for fabrication of the turbine wheel (this is the material that Swenson chose for their unit). The second important point is that the turbine nozzle will discharge steam into only a small portion of the perimeter of the turbine wheel, and hence it is important to keep

Fig. 3. Effects of Turbine Inlet Pressure on Rankine Cycle Efficiency for Nozzle Expansion to 10% Moisture and 0.9 psia (Entropy = 1.8, Enthalpy = 1000). The calculation of thermal efficiency did not include miscellaneous losses.

the turbine wheel diameter relatively small and allow the rpm to be quite high in order to have as large a fraction as possible of the turbine wheel perimeter engaged by the high velocity steam. This gives a higher efficiency in that it cuts the turbine rotor windage losses as a fraction of the power input.

An attempt was made to consider the relative merits of axial flow and Terry type-turbine wheels. Analyses were made both using the widely employed Balje' diagrams⁸ and by examining the individual losses with their various effects. Both studies indicated that there is little choice between an axial flow and a Terry type-turbine wheel from the standpoint of turbine efficiency. It appears that the Terry type wheel is more easily fabricated but that a somewhat lighter weight axial flow wheel can be obtained, and this would have the advantage of giving a somewhat more favorable situation relative to the dynamics of the rotor and bearings.

An important point shown by Fig. 3 is that the velocity of the steam leaving the nozzle going into the turbine wheel is very high — of the order of 4000 ft/sec. Inasmuch as the sonic velocity for steam at the turbine nozzle exit is around 1400 ft/sec, the Mach number of the jet leaving the nozzle is of the order of 3. For good efficiency in an impulse turbine, the turbine wheel tip speed should be about 40% of the nozzle exit velocity, and this in turn means that the relative Mach number between the jet and the buckets of the turbine wheel will be of the order of 1.3 to 1.5. This in turn means that substantial compressibility losses will be experienced in turning the flow through the turbine buckets. Thus, although increasing the turbine inlet pressure can be expected to increase the cycle efficiency for an ideal cycle, 10% or more of the increase will be lost to shock waves as the relative Mach number is increased beyond about 1.2.

In examining the bearing problem it seemed clear that ball bearings could not be expected to give a 5-yr life at the high speeds required with only water as the lubricant. In turning to hydrodynamic bearings, analysis of the various techniques that have been proposed for stabilizing lightly loaded bearings operating at high speed indicates that the only type that would be dynamically stable would be tilting pad bearings. Various groove and pad arrangements, including the herringbone type developed by MTI, will not give dynamic stability to a sufficiently high speed to be acceptable.

In searching for a way in which to avoid the mechanical complexities of the tilting pad bearings and the end seals that they require it appears that a good course may be to employ hydrostatic bearings. Analysis indicates that hydrostatic bearings might be employed because the rotor will be light and none of the operating loads will be large. The volume of liquid displaced by the shaft in moving from a central position radially outward to take up 90% of the clearance will be 12 times the rate of leakage out the ends of the bearing when the shaft is centered. Thus, by making use of three to five pockets, each supplied with lubricant through an orifice having a diameter of about 0.015 in., it appears possible to get a bearing and shaft system that should be stable throughout the desired speed range. If the bearing surfaces that can experience rubbing contact are coated with a very hard material such as plasma-sprayed titanium oxide, it appears that, under the relatively light loading conditions that could prevail with a light rotor (i.e., about 8 psi), there would not be difficulty with scuffing of the bearing when starting and stopping even if no pressure is supplied to the pads to float the shaft. This is a very important consideration because, in the isotope power unit, it is important that the system be both completely sealed and very simple. Thus it will not be practicable to provide bearing jacking pressure for startup.

As will be discussed in a later section, to simplify the control system it is desired that the turbine not be damaged by a runaway under a complete no-load condition. This in turn means that the turbine should be designed to operate not only at its normal design speed of, say, 60,000 rpm but also at its runaway speed which might be something like 95,000 rpm. Thus it is important to provide bearings that will function satisfactorily and not become unstable under the runaway speed conditions.

ORNL experience with small, high speed turbines employing a low viscosity working fluid such as water or potassium to lubricate the bearings indicates that a filter is required. As a consequence, the total flow of lubricant to the bearings should be directed from the pump through a micropore filter mounted on the side of the casing. This micropore filter can be attached to the casing with cap screws to maintain a sufficiently tight joint so that leakage will be unimportant because it will be caught in an enclosing can from which - along with leakage from the bearings - it will be "sniffled" to the condenser by turbine exhaust steam.

The feed supply to the bearings can be drawn from a port drilled into the region between the back of the pump impeller and the casing. The radial position can be chosen so that the bearing supply pressure will be lower than the pump discharge pressure to reduce the flow required for the bearings.

In view of the above considerations and analyses it was decided that the Swenson turbine shown in Fig. 4 is representative of what could be obtained from a well proportioned steam turbine-generator unit designed expressly for the isotope power application, and hence it was used as the basis for the subject reference design for a steam Rankine cycle system.

BOILER

The boiler design chosen for the reference design power plant is similar to that presented in the conceptual design of the Phase I survey report.¹ The reentry tube boiler was chosen to make it possible to couple the steam Rankine cycle system to the heat block-shield assembly simply by bayonetting the boiler tubes into the heat block-shield unit. One of the advantageous features of the reentry tube boiler is that this could be done with a cold boiler and a hot heat block-shield assembly without getting into difficulty with excessive thermal stresses or violent pressure fluctuations in the boiler as a consequence of alternating nucleate and vapor blanket type boiling heat transfer conditions in the boiler portion of the steam generator. Note, too, that the boiler tubes are inserted in holes close to the fueled portion of the heat block so that there is a relatively small temperature drop between the fueled region and the boiler tubes. If the boiler tubes were placed on the outside of the heat block a much larger temperature drop - of the order of 200°F - would have to be accepted.

A series of tests has been conducted on a single tube version of the reentry tube boiler with steam conditions and boiler tube proportions suited to the reference design. The short boiler tube length available made it advantageous to employ a finned surface on the inside of the outer tube, and some stainless steel tubing with internal spiral fins that was

Fig. 4. Cross Section Through the Swenson Turboalternator Rated at 3.5 kW(e). The length is 9.5 in. (Reproduced through courtesy of Swenson Research, Inc.)

available from a NASA-ORNL program was employed. The results of these tests are covered in a separate report.⁴ From these test data a set of proportions well suited to the reference design heat block-shield assembly was chosen to give the boiler defined by the data in Table 2. The key steps in the design calculations are shown in the table. Note that the length of the boiler tubes was determined by the proportions of the heat block-shield assembly, and that the steam output per tube was determined from the reentry tube boiler experiments. The data for the requirements for the turbine presented in Table 1 established the total steam flow rate required, and this divided by the steam output per boiler tube gave the number of tubes required in the boiler.

The major problem in the design of a reentry tube boiler intended to produce large amounts of superheat is to avoid loss of superheat in the exit steam as a consequence of exposure to the cold feedwater in the central tube. For this reason, the juncture between the central tube and the outer tube is designed to be immediately below the bottom of the heat block-shield assembly, and a 12-in.-long piece of 3/8-in. tubing was installed inside of the central tube to reduce the heat transfer between the high-temperature superheated steam and the relatively cold feedwater. A further step was taken to reduce this loss by heating the feedwater with prime steam bypassed from the boiler outlet. An orifice at the outlet of the steam side of the feedwater heater limits the flow to the condenser so that it would consist almost entirely of condensate. This arrangement has been designed to give a feedwater temperature within 8°F of the saturation temperature of the water in the boiler.

Alternate Design

The boiler design outlined above has the advantage that it is based on firm experimental data. However, it requires 18 tubes 0.75 in. OD whereas the reference design for the thermoelectric power unit employs 12 tubes 1.00 in. OD. In view of this a second boiler design was evolved for 12 tubes 1.00 in. OD. However, the complexity of the heat transfer relations involved on the inside of a finned tube is such that experiments would have to be run to validate the proportions used in preparing the alternate design of Table 2.

Table 2. Summary of Boiler Design Calculations

Initial Design	
Steam flow per tube, lb/hr	5.72
Enthalpy rise in steam, Btu/lb	1000
Heat load per tube, Btu/hr	5720
Tube OD, in.	0.75
Tube length, in.	55
Surface area per tube, ft ²	0.90
Tube wall temperature, °F	1000
Average surface heat flux, Btu/hr·ft ²	6360
Surface emissivity	0.60
Radiant heat flux, Btu/hr·ft ²	2900
Heat conducted thru air gap, Btu/hr·ft ²	3460
Average thickness of air gap, in.	0.005
Conductance of air gap, Btu/hr·ft ² ·°F	82
Temperature drop across air gap, °F	42
Total steam flow required, lb/hr	103
No. of boiler tubes required	18

Alternate Design for 1.0 in. Diameter Tubes

Steam flow per tube, lb/hr	8.59
Enthalpy rise in steam, Btu/lb	1000
Heat load per tube, Btu/hr	8590
Tube OD, in.	1.05
Tube length, in.	55
Surface area per tube, ft ²	1.20
Tube wall temperature, °F	1000
Average surface heat flux, Btu/hr·ft ²	7150
Surface emissivity	0.60
Radiant heat flux, Btu/hr·ft ²	2900
Heat conducted thru air gap, Btu/hr·ft ²	4250
Average thickness of air gap, in.	0.005
Conductance of air gap, Btu/hr·ft ² ·°F	82
Temperature drop across air gap, °F	52
Total steam flow required, lb/hr	103
No. of boiler tubes required	12

CONDENSER

The condenser consists of a plate-coil type of heat-exchanger surface rolled to form a cylinder that will fit just inside the pressure vessel.¹⁰ The plate-coil would not be a continuous cylinder but would have a slit down one side so that it could be expanded to fit snugly against the pressure vessel. A toggle linkage and spring arrangement would control the gap width, and would be operated to permit easy assembly of the pressure vessel over the condenser. After the condenser has been brought into position in the pressure vessel, this linkage would be operated to expand the condenser and force it into contact with the pressure vessel. In making the design calculations it was assumed that the plate-coil could be expanded so that the average distance between the outer surface of the plate-coil condenser and the bore of the pressure vessel would be 0.003 in. This gives an average temperature drop across the gap between the plate coil and the pressure vessel of 27°F. It seems likely that a somewhat smaller average gap could be obtained, but this can be determined only through actually building and testing a unit. The design calculations for the condenser and its principal proportions are summarized in Table 3.

The thermal insulation-thermal fuse would be installed on the inside surface of the condenser. This material would be the aluminum wire screen type of thermal insulation described in the report covering the reference design for the organic Rankine cycle system.³

DESIGN SUMMARY

The system that emerged from the design study is defined by the flow sheet of Fig. 5, the layout of Fig. 6, and the table of geometric and performance data presented in Table 1. Several points concerned with the system as a whole that emerged from these flow sheets and the layout include problems associated with liquid drainage under both normal operating and shutdown conditions, system startup, system control, and system water chemistry. Each of these deserves some discussion.

Fig. 5. Steam System Flow Sheet.

Fig. 6. Vertical section through isotope power unit coupled to a 5 kW(e) generator and contained in a pressure vessel designed for operation at 12,000 ft beneath the sea.

Table 3. Summary of Condenser Design Calculations

Steam flow to condenser, lb/hr	90
Enthalpy drop in condenser, Btu/lb	1040
Heat load on condenser, Btu/hr	93,600
Plate-coil OD, in.	48
Plate-coil height, in.	60
Plate-coil steam side surface area, ft ²	30
Plate-coil outer perimeter surface area, ft ²	62.8
Plate-coil outer perimeter heat flux, Btu/hr·ft ²	1490
Condensing heat-transfer coefficient, Btu/hr·ft ² ·°F	3000
Temperature drop from steam to condenser wall, °F	2
Temperature drop from plate-coil, °F	4
Gap between plate-coil OD and vessel, in.	0.003
Conductance of air gap, Btu/hr·ft ² ·°F	68
Temperature drop across air gap, °F	22
Temperature drop through vessel wall, °F	7
Grashof number for ambient sea water	3×10^9
Heat-transfer coefficient for sea water, Btu/hr·ft ² ·°F	48
Surface heat flux to sea water, Btu/hr·ft ²	1200
Temperature drop to sea water, °F	25
Total temperature drop from steam to sea water, °F	60

Water Chemistry

The extensive study of water chemistry under the TLPR Program in 1966 led to the conclusion that one might be able to build a hermetically sealed steam Rankine cycle system of stainless steel with no special provisions for water treatment.⁵ This is a very important consideration in the design of an unattended power plant that must operate for a 5-year period. Otherwise, a very complex set of water-treatment equipment together with its required controls would be necessary and the complexity of this would so degrade the reliability of the system as to make it completely unsuited to the application.¹¹ (As a matter of fact, as pointed out in the original survey report,¹ it is for this reason that consideration is being given only to completely hermetically sealed systems.) The original TLPR study of water chemistry also indicated that an all-titanium system should have a substantially higher probability of operating over periods of years without difficulties with water chemistry. As a consequence, the design presented here assumes that the entire system would be made of commercially pure titanium with the exception of the turbine wheel, which would be a high-strength titanium alloy, and the components in the generator such as the magnet iron, the electrical insulation, and the electrical conductors. As indicated in the section on the turbine, the turbine bearings and journals could be made of titanium with a plasma-sprayed coating of titanium oxide.

Drainage

The system has been designed so that the condensate will drain to the bottom of the condenser where it will be collected in a hot well. When the system is shut down the liquid level in the hot well will be relatively low because the lower portion of the system will be flooded. This includes the feed pump-turbine-generator unit and the lower portion of the boiler. (Slow leakage through the bearings will eventually flood the turbine-generator unit.) Thus the system has been designed so that it would be tilted on its side prior to an attempt to start so that nearly all the liquid will drain into the hot well, which serves as the expansion tank.

The bulk of the volume of this unit would be located at one side of the power plant. After the liquid had been drained into the hot well, the unit would be righted and electric power would be supplied to the generator to operate it as a motor to drive the feed pump and supply water to the boiler. When the boiler steaming rate became sufficiently high to drive the turbine, the electric power supply to the generator could be cut off and the generator could be connected into the line.

The system might be designed to operate with the heat block-shield assembly and boiler tubes in a horizontal position but it is believed that this would lead to serious difficulties with poor flow distribution between the various boiler tubes, and hence this would not be a desirable way in which to operate the boiler.

In order to assure that the unit would not tip over in the water as a consequence of landing on an uneven bottom, the system might be provided with floatation gear attached to the top of the pressure vessel so that the center of gravity of the unit would be well below the center of buoyancy. If the equipment were properly proportioned, the unit would tend to stand upright in the water even though its weight is greater than that of the water it displaces. This requirement would not be too difficult to meet for a 6000 ft depth, but it would be quite difficult indeed to meet for a 20,000 ft depth.

Control

The problems of system control are sufficiently difficult that, unless great care is taken to obtain a highly reliable control system, the reliability of the power plant will be determined by the reliability of the control system. A comprehensive investigation of the control of Rankine cycle power plants is underway at the time of writing, but it is not yet completed. The study is sufficiently far along, however, to lead the writer to favor the system presented in this section. The design considerations that have led to its selection are summarized in this section together with a description of the mode of operation envisioned.

Perhaps the most important consideration is that the Rankine cycle system must operate in conjunction with a set of storage batteries to

take care of short-term heavy loads and for emergency conditions if the Rankine cycle system should fail. In addition, for some applications it will be desirable to parallel two or more Rankine cycle systems, and in this event it will also be necessary for the systems to operate in conjunction with storage batteries for emergency conditions. Discussions with electrical systems experts indicate that there is such great difficulty in any attempt to parallel dc-ac inverters, either with each other or with ac generators, that paralleling simply isn't practical. Instead, it is necessary to switch from one power source to the other. If this is done, the switching perturbations lead to severe transient voltages and currents in equipment such as electronic gear that may be connected to the power supply. These voltage and current transients may be as much as ten times the normal steady-state values, and are likely to damage the solid-state equipment in the circuits. This, together with other complications, makes it highly desirable to have the turbine-generator operate in parallel with the battery bank to maintain continuity and avoid these transients. Parallel operation is possible if the output of the ac generator is rectified to yield dc current and a dc power distribution system is employed. A major advantage of a dc distribution system is that it does not require close control of the generator output frequency - it is only necessary to maintain a constant voltage. Studies of outage experience with conventional turbine-generators indicate that almost half the outages commonly stem from difficulties with either the throttle valve or the governor mechanism employed to actuate it.¹² Thus a marked improvement in system reliability can be obtained by eliminating these control components and allowing the turbine-generator speed to vary over a rather wide range. Extensive experience has been obtained in both automotive and aircraft electrical systems with this sort of an arrangement because in both cases the generators are mounted as auxiliary equipment on engines that must be permitted to operate over a wide range of speed. Excellent system reliability has been obtained with ac generators delivering their output through rectifiers to electrical systems that make use of storage batteries as a reserve supply. As a consequence, this type of system has been chosen for the reference design presented here.

As indicated above, 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.^{11,12} In view of the great 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. 7. 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

ORNL DWG 70-6660R

Fig. 7. Schematic diagram showing the control scheme proposed for the steam Rankine cycle system

to go from zero to full flow with a temperature variation of only 100 to 200°F in the heat source. 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.

The second major control element in the system depends on sensing the battery voltage (with the sensor $E_V E$ of Fig. 7) 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 component $E_V E$ 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 that is 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 45 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 11 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. 8. 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 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. 8.

Variations in the electrical load on the generator will lead to some changes in the heat load on the condenser. This will take the form of either a change in the amount of steam allowed to bypass the turbine or a change in the amount of superheat in the steam leaving the turbine. (A reduction in electrical load will increase the turbine speed, and this will increase the aerodynamic losses. These will reduce the useful work extracted from the steam, and hence increase its exit temperature.)

Increasing the superheat of the steam entering the condenser at low electrical loads will cause a small rise in the condenser temperature. This will be small because the overall cycle efficiency is low, and the maximum increase in heat load from the full electrical load condition to no electrical load would be only about 10%. For the design of Table 1, this would mean an increase in condenser temperature of only about 6°F.

Fig. 8. Effects of battery charge on major system parameters for typical electrical system load conditions

The homopolar generator with a conventional field winding yields a voltage that drops off rapidly as the current is increased, hence the field current must be regulated to obtain a constant voltage output. An ORNL study under the TLPR program¹³ indicates that the extra complication of this equipment can be avoided by employing a compound winding in the field. One of the two coils in the compound winding would be a conventional shunt winding coupled through a solid state rectifier to the generator terminals. A current transformer in series with the line would supply current through a rectifier to the "series" winding. Conversations with Westinghouse at Lima, Ohio, disclosed that they have built at least one model of alternator with a compound field winding to achieve this sort of characteristic.

DEVELOPMENT STATUS

The two principal uncertainties associated with the proposed reference design are those concerned with the reliability of the turbine-generator unit and the feasibility of operating a hermetically sealed steam system built of titanium with no provisions for feedwater treatment. The first of these would require a turbine-generator development program that would be likely to cost somewhere between 100,000 to \$500,000. There appears to be no basic reason why a suitable turbine-generator could not be developed, but the details of developing such a machine entail enough uncertainties so that it is hard to estimate what the development cost would be. On the other hand, the cost of investigating the compatibility of water with titanium at approximately the steam conditions anticipated would probably be much less - of the order of 100,000 to \$200,000. This problem could be investigated with a system similar to that projected for investigating the long-term stability of Dowtherm A under the combined stresses of temperature and nuclear radiation.

REFERENCES

1. R. A. Robinson, Isotopes Kilowatt Program Task 1 - Conceptual Design and Evaluation, USAEC Report ORNL-TM-2366, Oak Ridge National Laboratory, January 1970.
2. P. F. Swenson, Jr., Optimizing a 5 BHP Steam Turbine Design, 4th Intersociety Energy Conversion Engineering Conference, Washington, D. C., September 22-26, 1969, pp. 237-252.
3. G. Samuels and R. S. Holcomb, Reference Design for an Isotope Power Unit Employing an Organic Rankine Cycle, USAEC Report ORNL-TM-2960, Oak Ridge National Laboratory, January 1971.
4. R. S. Holcomb and M. E. Lackey, Performance Characteristics of a Short Reentry Tube Steam Generator at Low Steam Outputs, USAEC Report ORNL-TM-3236, Oak Ridge National Laboratory, June 1971.
5. G. H. Jenks and J. C. Griess, Water Chemistry in Pressurized and Boiling Water Power Reactors, USAEC Report ORNL-4173, Oak Ridge National Laboratory, November 1967.
6. G. Samuels, Comparison of Nickel and Iron Heat Block-Shields, USAEC Report ORNL-TM-3213, Oak Ridge National Laboratory, January 1971.
7. K. W. Haff, Isotopes Kilowatt Program, Fuel Capsule Design Evaluation and Safety Analysis, USAEC Report ORNL-TM-3230, Oak Ridge National Laboratory, May 1971.
8. O. E. Balje, A Study on Design Criteria and Matching of Turbomachines: Part A - Similarity Relations and Design Criteria of Turbines, J. Eng. Power A84, 83-102, January 1962.
9. O. E. Balje, Turbine Performance Prediction: Optimization Using Fluid Dynamic Criteria, R-6805 (AD-642 767), December 1, 1966.
10. A. P. Fraas, Heat Exchanger Design, John Wiley & Sons, 1965.
11. A. P. Fraas, Estimating the Reliability of Systems, USAEC Report ORNL-TM-2200, Oak Ridge National Laboratory, May 1968.
12. Reliability Study - Equipment and Components in Nuclear Power Plants, Gilbert Associates, Inc., Report No. 1512.
13. Terrestrial Low-Power Reactor Program Quarterly Progress Report for Period Ending September 30, 1967, USAEC Report ORNL-4216, pp. 29-35, March 1968.

Blank Page

Internal Distribution

1. S. E. Beall	33. A. M. Perry
2. T. A. Butler	34. R. A. Robinson
3. W. B. Cottrell	35. M. W. Rosenthal
4. W. C. Cox	36. A. F. Rupp
5. F. L. Culler	37-46. G. Samuels
6. R. G. Donnelly	47. Myrtle R. Sheldon
7-16. A. P. Fraas	48. A. C. Schaffhauser
17. J. H. Frye, Jr.	49. M. J. Skinner
18. J. H. Gillette	50. A. M. Smith
19. A. G. Grindell	51. I. Spiewak
20. K. W. Haff	52. D. A. Sundberg
21. H. W. Hoffman	53. D. B. Trauger
22. R. S. Holcomb	54. J. J. Tudor
23. P. R. Kasten	55. A. M. Weinberg
24. M. E. Lackey	56. G. D. Whitman
25. E. Lamb	57. J. V. Wilson
26. M. E. LaVerne	58. H. C. Young
27. D. B. Lloyd	59. Biology Library
28. M. I. Lundin	60-61. Central Research Library
29. R. N. Lyon	62. Y-12 Document Reference Section
30. R. E. MacPherson	63-87. Laboratory Records Department
31. H. C. McCurdy	88. Laboratory Records Department (RC)
32. A. J. Miller	

External Distribution

89. D. R. Bartz, Jet Propulsion Laboratory, NASA, 4800 Oak Grove Drive, Pasadena, California 91103
90. H. Bibb, Naval Facilities Engineering Command, Washington, D. C. 20390
91. Joe Clemente, Sundstrand Aviation, 1100 Connecticut Avenue, N.W., Washington, D. C. 20036
92. D. F. Cope, RDT Site Office, ORNL
93. W. M. Crim, Jr., Research and Technology Department, Building 322, Ft. Belvoir, Virginia 22060
94. R. R. Dahlen, Advanced Engineering, 3M Company, Building 551, 2501 Walnut Street, Roseville, Minnesota 55113
95. R. V. Degner, Rocketdyne, 6633 Canoga Avenue, Canoga Park, California 91303
96. R. E. English, NASA, Lewis Research Center, Cleveland, Ohio 44135
97. E. E. Fowler, Division of Isotopes Development, AEC, Washington, D. C. 20545
98. N. C. Gibbon, Cryogenic Products Department, Linde Division, P. O. Box 44, Tonawanda, New York 14150
99. J. C. Graf, General Electric, Valley Forge, Pennsylvania 19481

- 100-104. W. D. Holloman, Division of Reactor Development and Technology,
AEC, Washington, D. C. 20545
- 105. A. E. King, Westinghouse, Lima, Ohio 45801
- 106. M. Klein, Division of Reactor Development and Technology, AEC,
Washington, D. C. 20545
- 107. W. W. Logvin, Aerojet Liquid Rocket Company, Department 2400,
P. O. Box 13000, Sacramento, California 95813
- 108. J. J. Lynch, NASA Headquarters, Washington, D. C. 20546
- 109. S. V. Manson, NASA Headquarters, Washington, D. C. 20546
- 110. R. F. Mather, NASA, Lewis Research Center, Cleveland, Ohio 44135
- 111. W. Mathis, Westinghouse Astronuclear Laboratory, P. O. Box 10864,
Pittsburg, Pennsylvania 15236
- 112. T. A. McLaughlin, Isotopic Auxiliary Power Branch, AEC,
Washington, D. C. 20545
- 113. R. E. Niggeman, Sundstrand Aviation, 4747 Harrison Avenue,
Rockford, Illinois 61101
- 114-118. Nuclear Engineering Division, Naval Facilities Engineering
Command, Washington, D. C. 20390
- 119. J. Pidkowicz, RDT Site Office, ORNL
- 120. A. A. Pietsch, AiResearch Manufacturing Company, 402 S. 36th
Street, Phoenix, Arizona 85034
- 121. W. D. Pouchot, Westinghouse Astronuclear Laboratory, P. O. Box
10864, Pittsburgh, Pennsylvania 15236
- 122. C. R. Ross, Savannah River Laboratory, Aiken, S. C. 29801
- 123. G. W. Sherman, Aeronautical Space Power Division, Air Force
Aeronautical Propulsion Laboratory, Wright-Patterson Air Force
Base, Ohio 43433
- 124. Anthony Stathoplos, Nuclear Technology Corporation, 116 Main
Street, White Plains, New York 10600
- 125. B. Sternlicht, Mechanical Technology, Inc., 968 Albany-Shaker
Road, Latham, New York 12110
- 126. D. Stewart, Battelle Northwest Laboratory, Richland, Washington
99352
- 127. P. Swenson, Jr., Swenson Research Inc., 5135 Richmond Road,
Bedford Heights, Ohio 44146
- 128. J. E. Taylor, Thompson-Ramo-Wooldridge, 23555 Euclid Avenue,
Cleveland, Ohio 44117
- 129. W. B. Taylor, Technical Director, U.S. Army Mobility Equipment
Research and Development Center, Ft. Belvoir, Virginia 22060
- 130. T. F. Widmer, Thermo Electron Corporation, 85 First Avenue,
Waltham, Massachusetts 02173
- 131. E. S. Wilson, Space Nuclear Systems, AEC, Germantown, Maryland
20767
- 132-133. Division of Technical Information Extension (DTIE)
- 134. Laboratory and University Division, ORO
- 135-137. Director, Division of Reactor Licensing, AEC-Wash.
- 138-139. Director, Division of Reactor Standards, AEC-Wash.
- 140-144. Executive Secretary, Advisory Committee on Reactor Safeguards,
AEC-Wash.