ANL-6857

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VAPOR CYCLE COOLANT REQUIREMENTS

FOR NUCLEAR SPACE POWER PLANTS

by

Marion J. Janicke

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ANL-6857
Propulsion Systems and
Energy Conversion
(TID-4500, 34th Ed.)
AEC Research and
Development Report

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9700 South Cass Avenue
Argonne, Illinois 60440

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Reactor Engineering Division

March 1964

Operated by The University of Chicago
under
Contract W-31-109-eng-38
with the
U. S. Atomic Energy Commission

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VAPOR CYCLE COOLANT REQUIREMENTS FOR NUCLEAR SPACE POWER PLANTS

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ABSTRACT

Interplanetary space travel will require electric power in excess of 1-10 MW. Present designs show that for these power demands, a coolant, rather than a radiation or conduction, heat-transmission scheme will be required for the economic use of power-plant weight. Of the various electrical conversion systems, the turbo-electric system appears to be the most promising method for supply of this energy. The choice of the coolant is the most important decision in the planning of any space power plant.

Liquid metals are especially suited for reactors with high thermal fluxes and high operating temperatures, because of their high thermal conductivities, low vapor pressures, and relatively high volumetric heat capacities. They are stable at high temperatures and in intense radiation fields. The main disadvantage of liquid metals is that they may be corrosive at high temperatures. However, the temperature at which this condition occurs is that at which most other liquids are no longer usable as coolants. Mercury, rubidium, potassium, and sodium are coolants that vaporize at a temperature within the present state-of-the-art metallurgical limits.

Chapter 1

INTRODUCTION

Power sources for space application are categorized as nuclear, solar, and chemical. Corliss,^(1.1) Saldin and Schuh,^(1.2) and Szego and Cohn^(1.3) have indicated how power may represent the life of the mission. As a crude indication, at 10^4 hr, radioisotopes may have a useful power range from 0.1 to 1.0 kW(e); solar devices could be used from 0.1 to 50 kW(e), and nuclear fission devices from 50 to 10^4 kW(e). Chemical power may be between 0.1 and 10^4 kW(e) for approximately 10 hr.

The usual objectives for a space power plant are that the system be highly reliable, unattended, lightweight, simple, efficient, and made to operate for at least one year. A successful space power plant will have to meet these and other requirements. The biggest problems today appear to be hardware problems concerning corrosion, seals, bearings, etc. Many of the so-called direct-conversion devices (fuel cells and thermionic, thermoelectric, and MHD devices) require pumping devices to circulate high-temperature fluids through a circuit; therefore, the problems of rotating machinery remain, and turboelectric conversion systems continue to be desirable for large space-power-plant systems.

Most of the successful high-temperature reactors have been of the heterogeneous type in which the reactor core is built of solid rods shaped so as to permit the coolant to pass parallel to the rods. The minimum weight of the reactor is determined by nuclear criticality requirements, and the maximum size is determined by heat-transfer and reactor-control consideration.^(1.4)

Anderson^(1.5) compared nuclear reactor-system weights for the SNAP-2 and SNAP-8 turboelectric systems, and the SNAP-10A thermoelectric coolant-cooled system. (SNAP is an acronym for Systems for Nuclear Auxiliary Power.) The most extensive vapor-cycle turboelectric study was by English,^(1.6) who studied a sodium-vapor cycle for a nuclear space power plant. The specific weight for this 20-MW(e) plant was approximately 6 lb/kW(e), as compared with 330 lb/kW(e) for a 3-kW(e) system like SNAP-2, -8, or -10A. Sodium vapor entered the turbine at 2500°R . Temperatures mentioned by English may appear to be high, but an experimental sodium-vapor loop reported by Congelliere^(2.7) shows operating temperatures of 2330°R . The reason for this temperature limitation is that the oxidation resistance of the Haynes Stellite-25 construction material is sharply reduced at 2400°R . A sodium-vapor space system, however should probably be considered an advanced state-of-the-art concept.

The nuclear reactor and the Brayton cycle have been studied in combination to eliminate the alkali vapor-cycle problems of corrosion and

boiling-condensing in a zero-gravity field.^(1.8) The major disadvantage of the Brayton cycle for space duty is that the radiator areas can be ten times as great as that for the isothermal vapor cycle. The Stirling cycle^(1.9) is another possible single-phase cycle system that could use a reactor heat source but is currently receiving attention in conjunction with the solar energy source.

Some of the so-called direct-conversion devices with reactor heat sources are worth mentioning. The single-phase magnetohydrodynamic (MHD) systems^(1.10) are slightly more popular and well-reported than the two-phase MHD devices. Thermionic systems have been proposed both for in-pile and out-of-pile nuclear-heat-source systems.^(1.11) Clark^(1.12) has studied the feasibility of a 3-MW(e), nuclear, thermoelectric, power plant. In these direct-conversion devices, a coolant scheme was used rather than a conduction or radiation heat-transport method.

For systems of capacity smaller than one kW(e), Corliss^(1.1) indicates that radioisotope energy may be competitive for energy conversion. A system involving such a power source does not require a critical mass, and the source can be made of materials that do not require heavy shielding. A disadvantage is that the output is proportional to the quantity of isotope present and therefore is continually depleting, with a consequential loss in power with mission time. A problem would also arise if a rocket abort melted the protective shell housing the radioisotope. However, radioisotope energy will probably continue to receive favorable attention from designers.^(1.13, 1.14)

Mackay and Leventhal^(1.14) point out that nuclear power plants have two principal disadvantages. First, because of its radiation environment, the power plant must be adequately shielded, and/or special equipment must be used for performing maintenance functions. Secondly, in the event of a malfunction during the rocket launch, the resultant radioactive debris would require an extensive cleanup procedure. A solar-energy source does not have the radiation problems of a nuclear system, but a large solar collector must be oriented toward the sun most of the time. A current estimate of the maximum system output for a solar-energy source is approximately 50 kW(e). The maximum output is based largely on the fabrication of the solar collector. Heath and Maxwell^(1.16) reported on the various collector configurations and the status of their development. Fischer and Menetrey^(1.17) have described a turboelectric solar-energized plant for supplying auxiliary power requirements. Zwick and Zimmerman^(1.18) have written about the feasibility of solar energy for fuel cell and thermoelectric power. The Stirling engine has received most attention in connection with the solar-collector energy source. Welsh and Monson^(1.19) claim high reliability and long, unattended life as the chief advantages of a 3-kW(e) earth-orbiting Stirling-cycle space power plant.

For very short-duration requirements, Wayman and Mills^(1.20) have reported a chemically fueled power supply for space vehicles up to 50 hp.

Electric power in excess of 1 MW will be required for interplanetary space travel. A coolant circuit scheme will be required for the economic use of power plant weight. Of the various electrical conversion systems, the turboelectric system appears to be the most promising method for supplying this energy.

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Chapter 2

COMPACT POWER SOURCES

Previous Research

The Kearny Station of New Jersey, the Shiller Stations of New Hampshire, and the South Meadow Generating Station of Hartford, Connecticut, are binary-system power plants that have operated successfully with a mercury-vapor cycle.^(2.1, 2.2, 2.3) Mercury was used to add heat to the cycle at a temperature higher than that of plants operating on a water cycle. The thermal efficiency of the Kearny system is about 37%; the system would be further pursued today if it were not for the high price of the coolant inventory. As an example, the Kearny Station produces 45 MW(e), and the mercury weight requirement for this system is approximately 400,000 lb.^(2.4) At a mercury market value of \$2.82 per pound,^(2.5) the mercury inventory represents an investment of \$1,128,000. Plans for future 1000-MW(e) stations make the inventory costs of such a coolant system prohibitively high. The experience from these conventional power plants, however, has been useful in designing the SNAP-2 space power plant.

SNAP Technology

The SNAP-2 is a liquid-metal-cooled reactor coupled to a mercury-vapor cycle producing 3 kW(e). The reactor is uranium-235 fueled, zirconium hydride-moderated, and beryllium-reflected and -controlled. Power is converted basically by a rotating single shaft that combines the turbine, pumps, and alternator rotor. The turbomachinery units are contained within one hermetically sealed housing. Heat carried away from the reactor by the NaK coolant is transferred to the mercury in a boiler superheater. After the mercury has expanded through the turbine, it is condensed in a radiator which forms the outer skin of the power plant. For energy conversion, the SNAP-2 has an integrated pump, turbine, and alternator supported by two journals and a thrust bearing. NaK reactor coolant and mercury-working fluid are separated from each other. A magnetic-type pump mounted at the end of the shaft accelerates the NaK by means of magnetic drag forces. The unit is hermetic and separately seals both of the liquid-metal loops. Recent tests of a Combined Rotating Unit, CRU-IV-M, have shown improved performance. The efficiency of the mercury pump increased from 30 to 50%. NaK pump efficiency increased from 1.9 to 2.3%. A change in the alternator design from single- to two-phase output increased efficiency from 80 to 90%. A change from full to balanced partial admission increased the turbine efficiency from 47 to 55%.^(2.6)

Other even-numbered SNAP projects incorporate a reactor as a heat source but are not reported as well in the unclassified literature. (See Table 2.1 for reactor SNAP systems and Table 2.2 for isotope-powered systems.) The SNAP-8 is an effort to develop a 30-kW(e) nuclear power

supply system. The design will essentially be a magnification of SNAP-2. Previous plans for SNAP-8 were to use two individual units to supply 60 kW(e). A recent evaluation of reliability, nuclear shielding, and zero-gravity condensing resulted in one turbomachinery package instead of two, a sensible heat radiator instead of a latent heat radiator, and a package weighing more than 100 lb per kW(e).^(2.7, 2.8)

Table 2.1

CATEGORIZATION OF REACTOR HEAT SOURCES FOR SNAP(S)

System	Electrical Power	Fuel	Energy Converter	Mission Life	Purpose
SNAP-2	3 kW(e)	Fully enriched uranium, ZrH ₂ moderated, NaK-cooled reactor	Mercury vapor turboelectric		Satellite power source
SNAP-4	0.3-3 MW(e)	SNAP-2 hydride fuel technology	Boiling water, single-loop turboelectric	1-5 yr with 12-month maintenance intervals	Unattended underwater or remote land location
SNAP-6					
SNAP-8	30 kW(e)	SNAP-2 fuel technology	Potassium vapor turboelectric	90 days (1964) 10,000 hr (1965)	Electric propulsion source
SNAP-10	300 W		Thermoelectric (conduction-cooled)		Air Force satellite program
SNAP-10A	500 W	SNAP-2 fuel technology	Thermoelectric (convection-cooled)	90 days	Air Force satellite program
SNAP-12					
SNAP-14					
SNAP-50	300 kW(e) 1970 1 MW(e) eventually	UC UO ₂ , BeO-UO ₂ lithium-cooled reactor	Boiling potassium turboelectric	10,000 hr	AEC-NASA-AF venture with advanced turboelectric, MHD and thermionic are still being considered

Table 2.2

CATEGORIZATION OF ISOTOPE HEAT SOURCES FOR SNAP(S)

System	Electrical Power (W)	Fuel	Energy Converter	Mission Life	Purpose
SNAP-1	500	Cerium-144	Mercury turboelectric	60 days	
SNAP-3	2.7	Polonium-210	Thermoelectric	5 yr	Transit 4A and 4B satellites
SNAP-5					
SNAP-7A	5.0	Strontium-90	Thermoelectric		Experimental Coast Guard buoy in Chesapeake Bay Md
SNAP-7B	30.0	Strontium-90	Thermoelectric		Coast Guard land-based navigation light
SNAP-7C	5.0	Strontium-90	Thermoelectric		Navy automatic weather station in Antarctic
SNAP-7D	30.0	Strontium-90	Thermoelectric		Navy unmanned floating weather station
SNAP-7E	5.0	Strontium-90	Thermoelectric		Navy experimental navigational beacon
SNAP-9A	25	Plutonium-238	Thermoelectric	5-10 yr	Navy's operational prototype for TRANSIT 5 navigational satellites
SNAP-11	18.6	Curium-242	Thermoelectric	90 days	NASA's unmanned soft lunar exploration program PROJECT SURVEYOR
SNAP-13	12.5	Curium-242, thermionic	Cesium-vapor	120 days	(Alternate for SNAP-11)

SNAP-10 will use a SNAP-2 type reactor as a heat source for a thermoelectric conversion system. Electrical output of this system will be approximately 0.5 kW(e). A principal feature of this system will be the elimination of the rotating equipment. For more SNAP literature see Nuclear Science Abstracts.^(2.9)

SNAP has been primarily a joint effort of the Atomic Energy Commission (AEC) and the National Aeronautics Space Administration (NASA). However, the AEC, NASA, and United States Air Force are conducting a joint research and development program to develop a space power plant called SPUR or SNAP-50.^(2.10) The equipment will be essentially an extension of SNAP-2 except for a lithium- or sodium-cooled reactor and a potassium-vapor turbine. One year's service should be a minimum operating requirement. More advanced space power plants will be made repairable so that electrical service may be extended from 2 to 5 yr.

A significant difference in space and earth power plants is that the rocket transportation vehicle imposes a weight limitation on the space power plant. For vapor cycles with isothermal radiators, maximum work output per unit of radiator area is obtained when the radiator temperature is approximately three-fourths of the turbine inlet temperature. Increasing the system temperature decreases the radiator weight. This objective is important for large systems where the radiator is a major portion of the system weight. However, a high peak temperature of the system imposes the problem of optimizing the high-temperature capabilities of the turbine.

The plant equipment is determined by the geometric arrangement of the neutron-activated components. A nuclear space power plant essentially consists of a nuclear-fission heat source, heat exchangers, turbines, pumps, generators, and electrical conversion equipment (see Figures 2.1 and 2.2 for equipment arrangements presently preferred). An indirect cycle using two coolant circuits will provide sufficient flexibility so that up-dated technology will cause minimum design changes. The primary coolant will be lithium or sodium, and the secondary coolant will be potassium or mercury. The nuclear reactor is as far as possible from other components aboard the spacecraft. To protect the other components from the neutron and gamma radiations arising from the reactor, a shield is constructed from a judicious selection of geometrically placed materials. The electrical equipment is subject to both neutrons and gamma rays and must be protected so that tolerable limits are not exceeded. This restriction will put the transformer, motors and generators under the radiation shield and as far from the radiation source as possible. Because the generator and turbine are considered an integral unit, the turbine will also be included in this shield. The radiator also requires protection from the neutron irradiation to prevent the radiator

from becoming a significant contributor of gamma rays. The primary coolant flowing through the reactor absorbs neutrons and becomes a source of gamma rays. The coolant flows around the neutron shield and into the heat exchanger to vaporize the secondary circuit fluid. To prevent radiation damage to the electrical equipment from the activated primary coolant, the secondary coolant within the heat exchanger will provide some mass for gamma shielding. Additional shielding for the primary coolant can be obtained by placing a dense material between the exchanger and the electrical equipment, and placing the electrical equipment as far as possible from the gamma source through and in the vaporizer.

Fig. 2.1
Cylindrical Radiator
Concept of Turbo-
space Power Plant

Fig. 2.2. Flat Radiator Concept of Turbo-space Power Plant

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Chapter 3

COOLANT CONSIDERATIONS

Introduction

The use of a vapor cycle is favored for the generation of electrical power greater than one kW in space vehicles. This approach achieves the lowest power plant weight per electrical kilowatt output. The isothermal heating and cooling regions of the saturated Rankine cycle use the available temperature ratios between heat addition and heat rejection better than do the Stirling or Brayton cycles. Of the various cycles, the saturated Rankine cycle approaches the Carnot cycle the closest. It is therefore proposed that the most profitable area of investigation, based on a minimum weight requirement, would be that of a vapor cycle.

Vapor-cycle Components

Basically the use of the vapor cycle requires four major pieces of apparatus: a heat source vaporizes the coolant, the turbine expands the working fluid, the radiator condenses the exhausted vapor, and the pump raises the pressure of the coolant for entrance to the vaporizer to resume the cycle.

In a direct cycle, the liquid-metal coolant would boil within the reactor. However, because of the zero-gravity environment and the general lack of understanding of the dynamic nuclear behavior of a boiling liquid-metal reactor, the indirect cycle has been favored for space power plant designs. In either case, a vapor separator is inserted between the vaporizer and the turbine to ensure high-quality vapor; the liquid that is separated is combined with the condensate from the radiator and returned to the reactor or intermediate heat exchanger. The intermediate heat exchanger isolates the highly radioactive reactor coolant from the nonradioactive system. This could be especially important if the primary coolant was to convey heat away from a component that was susceptible to radiation damage. Figures 3.1 and 3.2 show coolant arrangements for typical direct and indirect vapor cycles for space power plants.

Fig. 3.1
Flow Circuit for In-
direct Vapor Cycle

operating pressures, and high-pressure coolants mean excessive radiator systems weight due to stress considerations. Conversely, high-temperature, low-pressure coolants result in excessive weight due to high specific-volume requirements. Figure 3.3 shows the variation of condensing vapor pressure of the more popular liquid metals with temperature. A boxed-in

region has been shown by some authors, between 600 and 1400°F and between 2 and 20 psia, to illustrate approximately what pressure and temperature may be used in future power-plant designs.^(3.3, 3.4) A space power plant with water as coolant would violate excessive requirements, but is included in Figure 3.3 because of its earthly popularity.^(3.5) Lithium, a liquid-metal coolant with a large specific volume due to its low pressures, has not been entered on the graph because of its waning popularity. Mercury, rubidium, potassium, and sodium are the most attractive high-temperature, two-phase working fluids available.^(3.6-3.10)

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Fig. 3.3 Vapor-cycle Condensing Conditions

values from this reference line, Table 3.1 evaluates these vapor-cycle systems. It was assumed that the turbine inlet temperature was four-thirds of the respective condensing temperature and the power plants were of equal mechanical output. The mercury system appears to be slightly

Figure 3.3 indicates that condensing conditions for the "existing" power plant designs approximate a straight line. By use of operating

Table 3.1

EVALUATION OF "EXISTING" VAPOR-CYCLE SYSTEMS

Characteristics	H ₂ O	Hg	K	Na
Volumetric Heat Capacity (per*)	1.00	0.438	0.134	0.247
Pumping-power Index (per*)	1.00	113	213	43.8
Isothermal Heat Addition (%)	0	26.8	23.9	25.3
Rankine-cycle Efficiency (%)	0	8.85	0.45	7.08
Mass Flow Rate (per*)	1.00	6.03	0.98	0.47
Turbine (Inlet) Vol Flow Rate (per*)	1.00	3.95	10.0	16.8

*Compared with water.

superior to the other three coolants considered. Mercury was chosen over water because the water system has problems in radiolytic disassociation and low radiator temperature. Mercury was chosen over potassium- and the sodium-vapor systems because of its higher volumetric heat capacity and efficiency and lower volumetric flow rate.

Corrosion

As in most high-temperature system designs, the liquid-metal coolant system is limited by the properties of the structural materials. The materials containing the liquid-metal coolant are particularly susceptible to various mechanisms of corrosion:

1. Solution attack: The coolant uniformly attacks its container surface.
2. Direct alloying: Surface films or diffusion layers formed by the interaction of the coolant and the solid surface can result in a loosely adherent scale or can slow down additional diffusion.
3. Selective reaction: Liquid-metal coolants may react with constituents of the surface and result in intergranular penetration, or may deplete minor components of the solid. This type of attack is usually very concentrated, can seriously affect the physical properties of a material, and is thought to be accelerated by a stressed condition.
4. Corrosion by contaminants: Contaminants such as oxygen, nitrogen, or carbon may react with the surface and can inhibit further attack or become nonadherent and cause a weight loss.

In a liquid-metal system, a temperature differential may produce mass transport as a result of a variable coefficient of solubility. The solid component may dissolve from the zone of higher solubility and precipitate in a zone of lower solubility. This precipitated material may eventually accumulate in certain parts of the system and reduce the flow of the coolant. Impurities such as oxygen may accelerate the thermal-gradient transfer effect; in some cases, inhibitors would tie up the oxygen as an insoluble oxide.

An isothermal system composed of several metals may experience the mass transfer of materials from one part of the system to another. A metal can be dissolved upstream and can alloy and precipitate with a different material further downstream.

Miller^(3.11) states: "Any actual design of liquid-metal heat-transfer systems, particularly at the higher temperatures, should be based on configuration of the data by direct experimental evidence, duplicating as nearly as possible the contemplated conditions of operation."

Corrosion plays an important part of the coolant choice and will affect the weight of the system. The literature to date, however, has been scarce on the relationship between corrosion and system weight. This relationship appears to be an area for a comprehensive investigation.

Coolant Temperatures

The maximum and minimum allowable system temperatures are involved in coolant choice. The maximum temperature the system may be permitted to attain will be limited by metallurgical changes of the fuel, by corrosion susceptibility of the cladding, by boiling of the reactor coolant, by pump cavitation, or by structural strength. The nuclear heat source can develop a cycle of very high efficiency because it can add heat to the cycle at a very high temperature, but it is limited by the metallurgical composition of the system.

A coolant temperature can also be too low. The most restrictive low temperature is the temperature at which the coolant solidifies. A low temperature may cause excessive participation and restrictive flow of the coolant. Because of heat rejection at a low temperature, a large radiator area may result. In addition, if a temperature is low with respect to another temperature in the system, thermal gradients may produce stresses above tolerable limits.

Vapor-cycle Efficiency

For maximum work per unit radiator area, the radiator temperature should be approximately three-fourths of the turbine inlet temperature.^(3.12) Therefore, increasing the inlet temperature to the turbine will decrease the radiator weight. The vapor temperature to the turbine can be increased by using higher pressure with saturated vapor, or by using superheated vapor at the same pressure.

For a mercury vapor cycle, a change of initial state 1 to 1' on the temperature-entropy (T-S) diagram of Figure 3.4 shows a decrease in the

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Fig. 3.4. Increased Temperature and Pressure with Saturated Mercury Vapor

heat rejected, that is, the area under 2' to 3 is less than the area under 2 to 3. The change in the heat supplied is not as evident because the area representing heat added is decreased by the amount of the horizontal lined area, and increased by the amount of the vertical lined area. The Rankine-cycle efficiency is also plotted as a function of saturated inlet turbine pressure in this figure. The use of saturated vapor at high pressures results in a decrease in both the heat rejected and the heat supplied. Thermal efficiency increases with increasing pressure until a maximum value is obtained when the decreasing rate of heat added exceeds the decreasing rate of heat rejected. Mercury-cycle conditions were evaluated as far as data would permit.^(3.7) The maximum efficiency shown here is a conjecture using similar results based on water data.

Another effect is that of increasing the turbine inlet temperature at constant pressure, thereby increasing thermal efficiency. The heat added increases faster than the heat rejected, and the efficiency increases but does not experience a maximum value. A disadvantage of this scheme to raise both the turbine inlet temperature and efficiency is the inefficient use of heat exchanger weight due to poor heat transfer of superheat vapor. Slight superheating may be justified if moisture-bearing vapor causes excessive wear of the turbine components. Increases in Rankine-cycle thermal efficiency may also be obtained by decreasing the temperature of heat rejection.

The mercury T-S diagram shown in Figure 3.5 indicates that for a fixed initial state at p_2 reduced to p'_2 , the heat supplied is increased by the area under 3' to 3. The heat rejected is increased by the area shown by double hatching, and decreased by the area shown by single hatching. Therefore, decrease of the temperature of heat rejection also increases the efficiency of the Rankine cycle. High thermal efficiency, however, is not the prime quest for space power plant systems. The low temperatures that promote cycle efficiency also promote large radiator area.

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Fig. 3.5. Decreased Exhaust Pressure for Mercury Vapor Cycle

The best combination of upper and lower Rankine-cycle conditions should be a balance of the high-temperature metallurgical limitation of the structure, the allowable moisture in the vapor expanding through the turbine, and the heat-rejection capabilities due to thermal radiation. To promote cycle efficiency, a low coolant temperature should be sought for heat rejection. However, the radiator size increases in inverse proportion to the fourth power of the temperature. Therefore, to keep the radiator from adding too much weight to the system, a paradoxically high coolant temperature is sought for heat rejection. The ratio of radiator temperature to turbine inlet temperature mentioned in the first sentence of this section was based on the Carnot cycle; Rankine cycles with turbine inefficiency show this ratio to be closer to 0.8.^(3.13) The value of this ratio helps to establish a starting place for any design work. The best combination of conditions for a minimum-weight system will require balancing trade-off conditions of the power plant, which in turn will require hard work and astute judgment.

Reactor Heat Removal

The reactor is a heat source that is limited in thermal output by the heat-removal capabilities of the coolant. In other words, the rate of heat removal will determine the power output of the reactor core. A widely used performance index for comparing the heat-removal capabilities for the various coolants is the heat-transfer coefficient (usually denoted by h or U). For a given heat rate, a higher value of h will permit a smaller reactor-core heat-transfer area. Also, a reactor design can use a coolant that has an inherently high h to reduce the flow requirements and thus the pumping power requirements.

Ordinarily, reducing the coolant flow rate decreases h , so that extra reactor surface area accompanies a required decrease in pumping power. However, because of the especially high h of the liquid metals, a flow rate decrease does not significantly decrease h . To illustrate this point, the h of a nonboiling sodium reactor core, with a coolant equivalent diameter of 0.387 cm and with coolant properties taken at 700°C, was reduced only 3.9% for a flow reduction from 1000 to 900 cm/sec. A comprehensive review of the status of various liquid-metal heat-transfer investigations is contained in a report by Lubarsky and Kaufman.^(3.14)

Reflector Cooling

Reactor control can be accomplished by inserting a neutron-absorbing material into the core, or by varying the neutron leakage rate out of the core. The absorbing material is usually a control rod that passes into the core. Fabrication of this control mechanism is complex and usually requires penetrations into the reactor vessel. A reflector control that would vary the neutron leakage rate offers a design that is simple and does not penetrate the core; this control also offers a minimum number of perturbations

of the neutron flux within the core so that the power generated in the core may be more uniform. Designs for nuclear reactor space power plants have used this reflector method of control.

The reflector is placed so as to capture neutrons and gamma rays emitted from the core. As a result, heat will be produced in the reflector which will amount to approximately 5% of the total heat produced by the reactor core.^(3.15) This requires some means of cooling the reflector. The calculation of the heat generated involves a detailed analysis of the high-energy particles that must be accurate or at least conservative. The methods for cooling the reflectors are also applicable to cooling the generator and will be discussed in Chapter 5.

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Chapter 4

RADIATOR

Overall Radiator Design

The radiator area is a function of the design of the working fluid passages, their extended surfaces, the general overall radiator arrangement, meteor consideration, and transportation vehicle geometry. A principal objective in designing the coolant passage is maximum meteor protection at minimum weight. The only means of providing protection against meteor penetration (and consequent working-fluid loss) is to increase the thickness of the material over what would be ordinarily required from pressure, temperature, and corrosion considerations. For high meteor flux, extensive finning and tube thicknesses are required; but for low meteor fluxes, less finning and tube thicknesses may be possible. Figure 4.1 shows typical coolant passages that provide meteor protection. Figure 4.2 indicates the amount of heat transferred out of a radiator having two, three, or four sections ($\phi = 180^\circ$, 120° , or 90° , respectively).

112-2139

Fig. 4.1. Typical Radiator Coolant Passages

112-2933

Fig. 4.2

Configuration Factor vs
Radiator Dimensions

A further refinement for the radiator analysis will include the effect of meteors on radiator size as a function of site location and radiator material. Data concerning meteoroid frequency and their penetrating power can then be correlated with radiator thicknesses, material density, and vulnerable area to determine the probability of radiator puncture.^(4.1)

Pressure Loss

The flow of the vapor is especially important after expanding through the turbine. A pressure loss of the exit turbine vapor traveling to the radiator causes a reduction in the condensing temperature; reducing the vapor pressure reduces the condensing temperature. Radiator area (thus radiator weight) is inversely proportional to the fourth power of the condensing temperature and indicates the high price paid for pressure losses. To reduce pressure losses, large piping and low vapor velocities should be used whenever possible. The vapor, after exiting from the turbine, will contain enough moisture so that relationships like those reported by Lockhart and Martinelli,^(4.2) or modifications thereof^(4.3) for a two-phase flow, should be used for evaluating the pressure losses.

The pressure loss in the condensing section can be approximated by taking half the conventional pressure drop computed entirely on gas-inlet conditions; Kern^(4.4) notes that such a calculation will give conservative results.

Coolant Inventory

Coolant inventory (or holdup) in the condensing section will vary from almost all gas at the inlet to all liquid at the outlet. The conservative approximation as to the inventory in the condensing channels is to assume a straight-line variation between all gas and all liquid. A more analytic way of calculating the inventory would be to use the Martinelli relationships and calculate for small increments along the condensation process. However, because of the complexity of the real situation, only an actual design will be the decisive method for determining the amount of coolant within the condensing section.

Noncondensable Gases

If a direct cycle is used for the system, the coolant will probably have radioactive gases trapped in the coolant system. These gases can produce adverse effects in the radiator where the vapor is condensing. Because maintenance must be minimal, this problem can be serious once the power plant is in operation.

At the inlet of the radiator tubes, a composition of coolant vapor and noncondensable gases may be substantially all vapor, with the film

coefficient nearly that of the condensing coefficient for the vapor alone. However, as the vapor is condensed, the radiator outlet may be a mixture consisting primarily of a noncondensable gas having a low film coefficient.

If the noncondensable gases are not removed before the coolant is carried to the pump, the gas volume increases enough to unprime the pump. A current philosophy of expelling this gas is that a porous-walled tube may pass the liquid coolant and squeeze out the gas to the vacuum atmosphere.^(4.5)

Condensing Heat Transfer

The basic equation, developed by Nusselt, for calculating a coefficient of heat transfer is an expression of the heat transfer across the film of liquid that is continually condensing on the surface and continually draining away. Film condensation is one of two types of condensation. When condensation is drop-wise, rather than film-wise, as is characteristic of metallic heat-transfer media, the condensation coefficient varies from Nusselt's predictions. In film condensation, the condensed liquid covers the entire condensing surface; in drop-wise condensation, the condensed liquid gathers in globules, which leaves some of the condensing surface in direct contact with the vapor. Bonilla^(4.6) and Kutateladze^(4.7) agree that for drop-wise condensation of mercury vapor, the heat flux is essentially independent of the temperature difference between the condenser surface and the mercury vapor, that is, the coefficient of condensation heat transfer is inversely proportional to the temperature difference. Condensing heat transfer will also be affected by a zero-gravity environment.

A reduced or zero-gravity condition affects the way the condensation breaks away from the surface on which it was formed. Experimental investigations are being performed to determine the mechanism of the removal of this drop-wise condensation.^(4.8) This type condensation will affect the two-phase pressure loss; the condensate drops may be pushed along the surface by the vapor or will be intimately combined with the vapor to produce fog flow. Experimental data in the literature on liquid metals condensing in a zero-gravity environment have been mostly for mercury. The other liquid metals have properties that will require additional experimental data before an analytic correlation can be made to unify a generalized theory to explain these mechanisms.^(4.9)

Considerations of Temperature Loss

For preliminary calculations, the radiator temperature is assumed to be equal to the temperature of the vapor exiting the turbine. However, in the real situation the two temperatures are different.

The mechanisms whereby temperature is reduced may be summarized as follows:

1. A temperature loss occurs when a net pressure loss is caused by flow of the coolant from the turbine to the region of condensation.
2. A temperature drop is experienced when the heat is transferred across the condensate. This drop may be increased if the gas is noncondensable.
3. A temperature drop is caused by the thermal resistance of the structural material that confines the coolant. This temperature drop depends upon the thickness of the material which, in turn, is a function of pressure, temperature, corrosion, and meteor penetration.
4. A temperature drop is caused by the high-emissivity material that is placed on the structural material to promote good radiation heat transfer. Most of the high-emissivity materials have low thermal conductivity so that even a small thickness will cause an appreciable temperature drop.
5. A temperature drop is caused by the bond resistance between the structural and high-emissivity materials. The temperature structure for this bond can be used to find the heat transfer for radiation after appropriate geometric and emissivity view-factors are applied.

Radiator Leak Detection

Although the probability of complete outage decreases with an increasing number of units, a large number of units would not be concomitant with repair equipment capabilities. Aspects of isolation, such as instrumentation to locate and detect both large and small leaks, and the ability to repair a damaged radiator quickly and remotely, will influence the degree with which radiator sectionalization can be achieved. Diemer and Stocking^(4.10) have analyzed numerous systems for detecting and correcting the leakage resulting from meteor penetration in space-vehicle radiators; Reference 4.10 discusses in detail a promising mass-flow leak-detection system.

Coolant Loss

A radiator cannot be 100% protected against meteor penetration. Time elapsed between penetrations will be a function of the radiator area and the thickness of radiator material to be penetrated. There is no certainty concerning when a meteor puncture will occur. A radiator system that would provide for repairing penetrations would have a greater likelihood of accomplishing its mission than a system that is not expected to undergo meteor penetration.

Once the working fluid loss has been sensed and located, cutoff valves at the inlet and outlet header of the radiator would temporarily remove the malfunctioning radiator section from the main flow until the component can be made operational again. An ideal radiator surface would be constructed of a material (or materials) that would react with the meteor, the coolant, or the radiator materials themselves to prevent leakage due to meteor penetration.

The size and frequency of the meteor penetrations can determine the amount of fluid expected to be lost during the life of the power plant. If it is assumed that all coolant leaks are detected and stopped in the same manner, the total weight of the working fluid to be transported with the power plant will influence the selection of the fluid.

Another consideration of a liquid-metal coolant loss is that some of the radiator surfaces might be coated with the coolant. This covering will reduce the emissivity and the heat-rejection capabilities of the radiator. As a consequence, even though the puncture can be repaired, the coolant coating must be removed to maintain the original heat-rejection capacity.

Lunar and Other Planetary Environments

On the moon, the heat-sink conditions are similar to those in outer space. The atmosphere on the moon^(4.11) has a pressure approximately 2×10^{-13} of the earth's atmosphere. This near vacuum makes thermal radiation the only practical method of transferring heat out of the thermodynamic cycle. Even the heaviest gases (for example, the noble gases - krypton and xenon) cannot be retained on the moon. The density of the lunar atmosphere is greatly reduced by collisions with protons of the solar wind. Singer^(4.12) calculated the average permanent gas density in the vicinity of the lunar surface to be about 80 hydrogen atoms per cubic centimeter. The temperature of the lunar surface varies from +100 to -150°C.^(4.13) The sink temperature of a power plant on the moon will be greatly affected by these surface-temperature changes. The temperature changes occur in a 29.5-earth-day cycle, with approximately 15 days of darkness (because the moon is fixed with respect to the earth). The sun's influence on the effective sink temperature will depend on the power plant's latitudinal position on the moon. The amount of solar energy received by the radiator will depend on the angle between the sun's rays and the normal to the radiator surfaces.

Nuclear-powered stations on planets that can contain their atmosphere will probably incorporate gas-heat exchangers to remove the heat emitted in the Rankine cycle. Equipment using this method of removing heat should weigh less than radiator-type equipment. On planets where rivers or lakes are available, the weight of rejection equipment will be reduced still further. A power plant using gas as a coolant can use some of

the planet's atmosphere as a coolant in the event of a coolant loss. Ibele and Irvine^(4.14) have investigated some of the thermodynamic properties of the atmospheres of Venus, Mars, and Jupiter, and their investigations may be the basis for preliminary power plant designs.

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Chapter 5

ENERGY-CONVERSION EQUIPMENT

Turbine

The ability of the turbine to operate at a high temperature affects the size of the dominant system weight, the radiator. Two circumstances make this fact apparent. Radiator area is inversely proportional to the fourth power of the radiating temperature, and the radiator temperature for maximum turbine work per unit radiator area is approximately three-fourths of the turbine inlet temperature. The problem of decreasing the radiator weight becomes a problem of increasing the high-temperature capabilities of the turbine.

Vapor that is not superheated from the reactor or the intermediate heat exchanger should be passed through a vapor separator to remove as much moisture as possible before entering the turbine. This separator must be light and compact because of the transportation vehicle limitations. A disadvantage of the separator is that vapor pressure must be reduced considerably for adequate results, and the pressure loss will mean an increase in entropy and a decrease in the availability of energy per pound of vapor flow.

Liquid-metal vapor turbine experience is limited to mercury, (5.1) and no unclassified literature on rubidium, potassium, and sodium vapor turbines exists. Turbine blading materials, although structurally capable of withstanding the strength requirements, may not be able to withstand the corrosive nature of the liquid-metal vapor. Tungsten carbide, titanium carbide, and chromium carbide, bound with either nickel or cobalt, have been used for liquid-sodium bearings and journals, and have shown excellent corrosion resistance up to 1000°F. The above carbide materials and Stellite therefore appear to offer the best possibilities for turbine blading.

In addition to the corrosiveness of the liquid-metal coolant vapor, moisture in the vapor, upon expanding through the turbine, can cause serious erosion. Moisture content (and thus erosion) increases as the vapor passes through the turbine. Erosion may wear away some of the turbine bucket contours and cause a permanent decrease in efficiency. Moisture can also affect the immediate turbine efficiency by lagging behind the flow of vapor and interfering with the rotation of the buckets. Salisbury(5.2) and Lee(5.3) relate that for moisture in steam turbines, power loss is proportional to the percentage of moisture in the steam.

As the vapor expands through the turbine, the blade height increases to provide for the required allowable annular area. To avoid large frictional losses, the turbine blade annular area must be sufficient

to accommodate the total fluid flow at a velocity below Mach 0.95. Therefore, blade height is a function of the total annular area required to pass the fluid. A desirable blade height will range from 5% to 20% of the mean blade-ring diameter. In extreme cases, 30% can be allowed if special tapered and twisted blades are used. The minimum is imposed because of energy losses due to fluid path distortion of short blades; a maximum is dictated solely by stress considerations. A minimum height of $\frac{3}{4}$ in. has been recommended, blades as short as $\frac{3}{8}$ in. being allowed.

Turbine blading is the main consideration in turbine design. There are no definite rules as to best blade design. However, much of turbine blade design is learned from past experience. Special attention is usually focused on the allowable stresses a blade may be subjected to in its temperature environment. After the blade height has been determined through a stress consideration, attention is directed to developing the correct blade shape. Any deviation from the optimum shape will result in corresponding losses in efficiency. Only centrifugal stresses will be discussed here. However, through this evaluation the spindle, disc, and blade dimensions can be approximately determined.

Experience shows that approximately 15% of the annular area is taken up by the blades. Therefore, when the gross annular area is used to calculate the blade height, the net flow area is multiplied by 1.15. Net flow area depends on the exhaust-volume rate of flow and the last-stage exhaust-axial velocity. Leaving-loss velocities used in practice may range from 300 fps in steam turbines to 900 fps in jet engines. Another parameter closely associated with turbine dimensioning is the mean blade speed. Steam turbines generally use speeds of 800 to 1000 fps, 1200 fps being exceeded by jet engines.

As the operation temperature increases, the blading is particularly susceptible to stress limitations imposed by centrifugal forces. For a blade of constant cross section, the force exerted by the total centrifugal force at the root is found by means of the equation

$$F = \int_{r(\text{root})}^{r(\text{tip})} \frac{\gamma}{g} A \omega^2 r dr,$$

where

- $r(\text{tip})$ = radius at tip of blade (in.)
- $r(\text{root})$ = radius at root of blade (in.)
- γ = density of blade (lb/in.³)
- g = conversion factor [(lb_{mass}-ft)/(lb_{force}-sec²)]
- A = cross-sectional area of blade (in.²)
- ω = angular velocity (rad/sec)
- r = generalized coordinate for radius (in.).

The stress at the blade root is this force divided by the constant cross-sectional area. For added convenience, the stress at the blade root is given by

$$\sigma = 2.84 \times 10^{-5} \Omega^2 \gamma r_m h_b,$$

where

Ω = rotational speed (rpm)

γ = density of blade (lb/in.³)

r_m = mean blade radius (in.)

h_b = blade height (in.).

A curve of blade centrifugal stress versus rotor speed (see Figure 5.1) for a one-MW sodium turbine indicates that high stresses are reached at comparatively slow speeds. The stress equation shows that for

Fig. 5.1. Turbine Blade Centrifugal Stress vs Rotor Speed for a One-MW Sodium Turbine

a given volumetric flow rate and last-stage exiting velocity, the quantity $\gamma r_m h_b$ remains a constant. Stresses have been evaluated with values of $\gamma = 0.3$ and $\gamma = 0.7$ lb/in.³. Refractory materials have a value for γ of approximately 0.7 lb/in.³, stainless steel of about 0.3 lb/in.³. Refractories have a higher strength at elevated temperatures than iron alloys, but because refractories also have a density twice that of steel, the imposed stress is correspondingly increased.

Even though the actual strength advantage of refractory materials for this purpose is small, they have more corrosion resistance than iron alloys. For this reason, a refractory material may be a better choice should the stress increase of refractories outweigh the strength increase. Figure 5.2 shows how the blade, wheel, and spindle may vary in size with rotational speed for a one-MW sodium turbine.

The turbine is probably the most important component in the system for reducing the overall weight. The system weight depends upon the

radiator weight, and the radiator weight depends on the high-temperature capabilities of the turbine. A space power plant with one-MW(e) output and a mercury coolant would have a turbine weight one-tenth of the overall system weight, and a radiator weight one-half of the overall system weight. (5.4)

Turbine weight will depend on the kind and number of turbine stages, which in turn will depend on the available energy in the expanding vapor.

Generator

An electromagnetic generator is believed to be applicable for a combined pump, turbine, and generator design. Much of the information for such a generator design appears readily available from experience with previous liquid-metal canned-rotor pumping devices, that is, the generator is envisioned to be of the canned-rotor type. In this type of generator, as in the canned-rotor pump, a permanent-magnet rotor would be separated from its stator by a thin, metallic liner. For canned-rotor pumps, the working fluid is permitted to fill the air gap; a generator may also have to use this provision. Because the pump, turbine, and generator are hermetically sealed, the pump and turbine compartments will probably leak into the generator compartment.

Fig. 5.2. Turbine Size vs Rotor Speed for a One-MW Sodium Turbine

Generator efficiency losses will be a heat-producing source and may require a coolant to carry the heat away. If a typical generator has an efficiency of 80% and is to have an output of one MW(e), the heat to be removed from the generator will be 250 kW. This heat is appreciable, and equipment to carry it out of the cycle will present a weight penalty to the transportation vehicle. The extent of the penalty will depend on the method of removing the heat.

Therefore, in addition to the required heat rejection for the thermodynamic cycle, the generator heat must be considered. Four methods are available for removing this heat.

1. A refrigeration system would permit the generator to operate at a temperature below that of the radiator-condenser. The unfavorable characteristics of such a system include a high specific weight and the deterioration of the refrigerant in a nuclear radiation environment. (5.5)

2. The heat from the generator can be removed by a gas and transferred to a radiator similar to that of the main, liquid-metal, coolant-rejection system. A gas-to-liquid-metal heat exchanger^(5.6) would have a comparatively poor heat-transfer coefficient and would have high pumping-power requirements. Unless high gas temperatures can be obtained (which means a high generator temperature), a gas heat exchanger or radiator appears to be unfavorably heavy.

3. A high-temperature (approximately 500°C) generator would be ducted or ported and could use the returning liquid-metal condensate to carry away the heat. This method of removing heat seems plausible, but the electrical characteristic changes of such a system have not been fully investigated.

4. A system could operate at a sufficiently high temperature to radiate its heat to outer space with a minimum of conveyor mechanisms. Temperatures of approximately 700°C would be reasonable for these self-radiators and probably are future state-of-the-art concepts.

In summation, generator heat rejection means additional radiator area and extra weight charged to the system. The amount of weight will depend on whether the generator uses a single-phase or two-phase coolant. Single-phase coolants require much more radiator area than two-phase coolants, and a second two-phase coolant system may be justified.

Pump

At a given temperature, the coolant has a definite vapor pressure. Therefore, vapor bubbles may form if the liquid coolant pressure drops below the vapor pressure, resulting in cavitation. As the vapor mixture flows into a region of higher pressure, the bubbles collapse. Action of this kind in a pump, if it occurs with high frequency, can cause serious erosion and damage the pump. To prevent this condition, the liquid is required to have a certain NPSH (net positive suction head) at the inlet to the pump. The NPSH is the energy that must exist in the system inlet above the vapor pressure of the liquid. An experimental parameter called the Thoma Coefficient is useful in describing the condition when cavitation begins (called the critical value). This relationship is given by ^(5.7)

$$\text{Thoma Coefficient} = \frac{\text{Net Positive Suction Head}}{\text{Pump Pressure Difference}}$$

and the conditions of the system should be such that the Thoma Coefficient is higher than the critical value.

Special attention is required to supply a sufficient NPSH for the pump to prevent cavitation, thus promoting longevity. Ordinarily, the inlet conditions for the pump in a conventional power plant can be made conservative at the expense of extra heat-exchanger area. Were this to be used for the space power plant, very costly additional weight for the transportation vehicle would be required. Thus a pump must be designed with inlet conditions that will promote a minimum-weight subcooling radiator. The pump will probably be required to operate close to the critical value of the Thoma Coefficient. A report by Hammitt^(5.8) on the cavitation performance of a centrifugal pump shows that the Thoma Coefficient for a given Reynolds number is approximately twice as large for mercury as for water. The other liquid metals (rubidium, potassium, and sodium) will require additional investigation.

Another paper by Hammitt^(5.9) is devoted to the problems and the desired research for investigating liquid-metal cavitation. Such research will involve considerable time to develop the experience necessary to predict the damage caused by cavitation. Hobbs^(5.10) has reviewed the problems of predicting cavitation erosion from accelerated tests. He compared the various techniques that are useful in acceleration tests and he concluded that the absolute determination of the damage rate is not possible.

Bearings

Liquid metals are not good lubricants because they are corrosive and have a very low viscosity at high temperatures.^(5.11) Nonetheless, a liquid metal as a lubricant has the advantage that it does not deteriorate when subjected to nuclear radiation. With the coolant as the lubricant, the system can be hermetically sealed, and thus any leakage or possibility of a conventional lubricant contaminating the coolant system is eliminated.

In one of the first SNAP systems,^(5.12) mercury was supplied as a lubricant to two hydrosphere bearings. One bearing was located between the pump and the alternator, the other at the far end of the shaft just beyond the turbine. The pressure supplied to the two bearings compensated for any axial load variations and prevented the collapse of the bearing. The hydrosphere bearing consisted of a spherical journal rotating in a non-grooved hemispherical socket of slightly larger diameter.

Experience has shown some difficulties with the use of the SNAP bearings of the hermetically sealed mercury turboalternator rotor. "Some failures attributed to bearings had been caused by distortion in bearing support structures and new manufacturing control, which resulted in incorrect bearing clearances. During early developments, distortions caused by the press fit of bearings in their housing were experienced. Also, alignment difficulties of bearings were experienced because of the reference surfaces used in assembly."^(5.13) There have been some problems with the cavity-type bearing, and some movement toward using journal bearings is being made.

However, a journal bearing in zero gravity will be unloaded and therefore will be subject to undesirable whirl. Radzimovsky^(5.14) has indicated that a bearing for reducing such film whirl is the so-called NUTCRACKER type.

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Chapter 6

CONCLUSIONS

The requirements of a space power plant are extraordinary as compared with those of conventional steam power plants. For example, the usual objectives for a space power plant require that the system be highly reliable, unattended, lightweight, simple, efficient, and able to operate for at least one year. Conversely, steam power plants operate with regular maintenance as an accepted practice. Also, there are no unattended power stations in operation. In a space power plant, failure of any one component that interrupts the generation of electrical energy is interpreted as plant failure. A minimum number of components simplifies plant operation and promotes a high system reliability. For this reason, the simple, direct or indirect, vapor cycle is chosen

Some of the general problems that will be of practical significance for space power plants will include

1. design for optimum weight,
2. creep and evaporation of the structural materials at high temperatures,
3. loss of coolant through diffusion and leakage,
4. compatibility between the coolant and construction materials at high temperatures,
5. probability of damage by meteorite incident, and
6. fabrication problems associated with large, lightweight, hermetically sealed structures which must be packageable, assembled in space, and able to withstand the necessary launch accelerations.

Liquid metals are especially suited for reactors with high thermal fluxes and high operating temperatures. In general, they have a high thermal conductivity, low vapor pressure, and relatively high volumetric heat capacity. They are stable at high temperatures and in intense radiation fields. The main disadvantage (at high temperatures) is the corrosive nature of these coolants. However, at such temperatures most other liquids are no longer usable as coolants. Mercury, rubidium, potassium, and sodium are coolants that vaporize at a temperature that appears to be within the present state-of-the-art metallurgical limits.

The maximum temperature of the coolant can be limited by the

1. metallurgical composition of the fuel,
2. low efficiency of the generator,

3. cavitation in the pump,
4. corrosion of the structural materials, and
5. strength of the structural materials in general.

The minimum operating temperature, if there is one, can be limited by one of these considerations:

1. solidification of the coolant,
2. large radiator area,
3. precipitation of impurities in a cold portion of the system, and
4. thermal stress considerations.

A weight optimization of the system will require compromises between the various desirable characteristics of the system components.

1. As the turbine back pressure is decreased,
 - a. cycle efficiency increases,
 - b. radiator temperature decreases, and
 - c. requirements for radiator flow area are increased.

Situation a decreases the specific weight (pounds mass per electrical kilowatt), while situations b and c increase the specific weight. These parameters must be analyzed in detail to give an optimum minimum weight. Optimum, as used here, means that a minimum weight must be found for an accepted reliability of system components.

2. As the diameter of the condensing channel is decreased, the vapor velocity and the pressure loss are increased. The reduced pressure reduces the condensing temperature and increases the amount of radiator surface required to dissipate the heat. A reduction in the channel flow area, however, reduces the radiator weight and its vulnerability to meteors. Once the coolant channel thickness has been chosen to stop meteors up to a certain size, a search for the optimum-diameter condensing channel can be made.

3. Extended surfaces or fins attached to the condensing channel increase the heat-transfer area and reduce the number of condensing channels required.

- a. An increased fin thickness at constant fin length increases the fin efficiency. Therefore, the heat-transfer surface-area requirements are reduced, but the weight per unit of surface area increases.

- b. In general, the fin length will be longer for a high meteor-flux environment and shorter for a low meteor-flux environment.

During this study, several possible future work areas have become apparent.

1. A high-velocity vapor coursing through the radiator structure may cause vibrations that could fatigue the structure. Turbine vibrations in an unrestrained, zero-gravity environment may also be of special significance for megawatt-output-sized turbines.

2. The coolant leakage rate must be almost zero. As the power level increases, the leak tightness of the system will be increasingly harder to achieve. Some power level may be impractical to achieve because of the magnitude of size alone.

3. Corrosion products in the secondary loop may be a significant source of gamma radiation. The cost to remove the coolant gamma-producing impurities must be balanced by the permissible amount of radiation damage done to the electrical equipment. If corrosive products become very undesirable in the secondary system, then one must determine how much is to be gained by going from a direct to an indirect cycle.

4. When electric power will be required for sustaining life on interplanetary missions, a provision for repairing meteor penetrations will be necessary. The space power plant must be repairable if interplanetary space travel is to be possible..

Appendix A

AN ADVANCED 10-MW(e) MERCURY VAPOR CYCLE
SPACE POWER PLANT

The cycle characteristics of an advanced 10-MW(e) mercury turboelectric, space power plant were investigated. Inlet turbine conditions were constant for saturated mercury vapor at 1030°K and 74.8 atm. A second lithium circuit was used in the design. This binary arrangement separated the primary radioactive coolant loop from the mercury-working fluid loop, separated the power plant into nonboiling and boiling circuits, and would provide easier access to the turbine-generator if repair would be required. A 2000°F reactor outlet temperature for the lithium loop was used, and a 500°F drop was considered a reasonable temperature difference to maintain across the reactor and heat exchanger. Table A.1 lists the parameters used in the study. Results of the thermal analysis are illustrated in Figure A.1.

Table A.1

CHARACTERISTICS ASSUMED COMMON FOR 10-MW (e) SYSTEMS

Generator output	10 MW	Mechanical efficiency	95%
Turbine output	13.2 MW	Turbine efficiency	75%
Turbine inlet temperature	1030°K	Reactor outlet temperature	1370°K
Turbine inlet pressure	74.8 atm	Reactor inlet temperature	1090°K
Generator efficiency	80%

Fig. A.1. Thermal Performance of 10-MW(e)
Lithium-Mercury Vapor Cycle

The purpose of this analysis was to investigate the effects of turbine exit conditions on an entire space power plant system. When more energy is extracted per unit mass flowing by the turbine, more moisture is contained in the vapor as it passes through the turbine passages. Moisture in the vapor produces erosion and increases as the vapor passes through the turbine. Erosion destroys turbine bucket contours and causes a permanent decrease in thermal efficiency. Moisture also affects the immediate turbine efficiency by lagging behind the flow of vapor and interfering with the rotation of the buckets. This loss in power is proportional to the percentage of moisture in the vapor. The turbine is probably the most important component in the vapor-cycle system for reducing the overall weight. Total system weight depends largely upon the radiator weight; however, the radiator weight depends on the high-temperature capabilities of the turbine.

An inefficient turbine means increased heat-rejection requirements. Less energy is extracted per unit mass of working fluid; therefore, more fluid must be circulated to produce the necessary work. The increased mass-rate flow requires more heat to be rejected, and this requires more area and more weight. The trend of the radiator area requirements is shown in Figure A.2. The required radiator area must be known, and the coolant channel thickness must be determined.

A radiator cannot be designed with 100% certainty of zero penetration. Time between penetrations is a function of the radiator area as well as of the thickness and type of radiator material to be penetrated. The data in Figure A.3 are assumed as typical of what a space vehicle might encounter just outside the earth's atmosphere. A system that provides for repairing penetrations would have a greater likelihood of accomplishing its mission than one that does not expect any penetrations.

Fig. A.2. Expected Radiator Performance for 10-MW(e) Lithium-Mercury Vapor Cycle

112-1274

Fig. A.3. Material Penetration Depth of Meteors Just Outside Earth's Atmosphere

The requirements for coolant-channel thickness depended on strength and corrosion. A 3.81-cm-diameter stainless steel 304 tube was assumed for the basic radiator channel. The allowable strength of the stainless steel was assumed to be one-fifth of the stress-rupture strength for 10,000 hr at the appropriate radiator condensing temperature. For a corrosion allowance, an additional 0.0254 cm was added to the diameter of each system analyzed.

The extra weight of the working fluid to be transported with the power plant for puncture makeup can be shown to influence the choice of

operating conditions for the power plant. To show this effect, it was assumed that for each radiator puncture, the shut-off valves in the radiator system allowed only 1 kg of coolant to escape to space.

A value of 47 kg/m^2 of surface area per centimeter of channel thickness was assumed for the radiator construction. The problem of the repairable radiator is now one of simple economics. The original cost to transport the radiator weight is considered to be the fixed cost of the investment and the coolant loss (the operating expense). The total radiator weights, shown in Figure A.4, are an appreciable portion of the total plant depending on how optimistic one wants to be about the other components. A typical variation of values for turbine and generator equipment is shown in Figure A.5. From Figures A.4 and A.5, it is estimated that a system such as this may weigh in excess of 200,000 lb.

Fig. A.4

Radiator Weight Analysis for
10-MW(e) Lithium-Mercury
Vapor Cycle

Fig. A.5. Turbine and Generator
Weight vs Power Rating

Appendix B

DETERMINING A SUITABLE RESERVE
SPACE POWER PLANT RADIATOR CAPACITY

The turboelectric energy supply of a space power station is limited by the size of the radiator. If part of the radiator is removed, the power would have to be cut at least proportionately. However, the load cannot be zero for very long. For space duty, the power plant must always maintain a minimum survival load. Omnipresent meteors greatly increase the probability that a power plant will not operate its fully expected mission without incident. A system is presented here, based on simple probability theory, that indicates a confidence level for a survival load of a radiator bombarded by meteors.

The coolant-carrying-type radiator section is composed of channels that are vulnerable to puncture by meteors. A single penetration could incapacitate the entire power plant if the section were not isolated and repaired. This loss may only be partial and temporary if the section can be repaired by a robot during the functioning of the other radiating sections. For example, if a leak were detected in one section of a four-section radiator, that wing could be temporarily removed from service until repaired with loss of only one-fourth of the total radiating capacity or power. If greater protection is required, the sectionalizing could be increased.

Elementary probability theory can help in selecting a reserve capacity for a continuous-plant operation of a minimum electrical survival load. If the probability of a forced outage of a unit is $Q = q/(q+p)$, the ratio of hours outage, q , to the number of hours in the year, $q + p$; and if the operating rate for a unit is $P = p/(q+p)$, the ratio of hours running, p , to hours in the year, $q + p$, then for any unit i ,

$$P_i + Q_i = 1.$$

The probability that all units are in operation during any given period is the product $P_1 P_2 P_3 P_m$. Similarly, the product $Q_1 Q_2 Q_3 Q_m$ equals the probability of all the units being out of service during any given period. To find the probability of partial service combinations of units in service simultaneously, note that from the expression $(P_1 + Q_1)(P_2 + Q_2) \dots (P_m + Q_m) = 1$, the term $P_1 Q_2$ is the probability of having section No. 1 in service and section No. 2 out of service simultaneously; $P_2 Q_1$ is the converse; etc.

If all the units, taken to be m in number, have identical service and outage probabilities P and Q ,

$$(P + Q)^m = 1.$$

Therefore, the general expansion for this relationship would be:

$$P^m + mP^{m-1}Q + (1/2)m(m-1)P^{m-2}Q^2 + \dots + mPQ^{m-1} + Q^m = 1.$$

The first term relates the probability of a complete success, i.e., no outage at all. The second term gives the probability of $(m-1)$ units being successful; the third gives the probability of $(m-2)$ successful units; ...; the last term gives the probability that all units will be down at one time.

A typical example would be a stainless steel radiator that rejects 10 MW (2.5 MW per section), with a specific vulnerable area capability of $10 \text{ m}^2/\text{MW}$. With a thickness of 0.02 cm, and an appropriate value of $4 \times 10^4 \text{ m}^2\text{-hr}^1$ between penetrations, a coolant loss puncture can be expected every 1600 hr. Furthermore, for a repair time of 16 hr, expected values of Q and P are ~ 0.01 and ~ 0.99 , respectively.

To calculate the probability of heat-removal capacity of this sectionalized radiator, expand

$$(P+Q)^4 = (0.99+0.01)^4 = 1$$

$$P^4 + 4P^3Q + 6P^2Q^2 + 4PQ^3 + Q^4 = 1$$

		Probability										
Heat removal capacity lost due to meteor penetration, MW	0	0.9	6	0	5	9	6	0	1			- P^4
	2.5	0.0	3	8	8	1	1	9	6			- $4P^3Q$
	5.0	0.0	0	0	5	8	8	0	6			- $6P^2Q^2$
	7.5	0.0	0	0	0	0	3	9	6			- $4PQ^3$
	10.0	0.0	0	0	0	0	0	0	1			- Q^4
		1.0	0	0	0	0	0	0	0	0		

The figures above show that the probability of no loss of full radiator heat-removal capabilities at continuous operation is 0.96059. A value 0.000001 per cent for all of the units out at the same time means that a maximum coolant loss outage will occur one day in 274,000 yr. This example demonstrates the optimistic viewpoint that four sections may provide, for all practical purposes, some continuous duty.

Although the probability of complete outage decreases with an increasing number of units, a large number of units would not be compatible with repair equipment capabilities. Other aspects of isolation such as instrumentation to locate geometrically and detect large and small leaks and the ability to quickly and remotely repair a damaged radiator, will influence the degree with which radiator sectionalization can be achieved.

¹Carter, L. J., Realities of Space Travel McGraw-Hill Book Co., Inc., New York (1957), p. 226.

Appendix C

CARNOT CYCLE ANALYSIS

Unpractical as the Carnot cycle may seem at times, it is an excellent tool for beginning the analysis of an engine cycle. From Figure C.1, note that the work W from the Carnot cycle is given by

$$W = Q_R (T_2 - T_1) / T_1, \quad (C.1)$$

where T_2 represents the inlet temperature to a turbine, T_1 the isothermal radiator temperature, and Q_R the heat that will be rejected. At first,

Fig. C.1. Temperature-Entropy Diagram for Space Carnot Cycle

equation (C.1) seems to indicate that a low T_1 will be desirable from a maximum work consideration. However, work is not the only concern. The weight of the power plant is important, and this consideration is governed by Q_R , the rejected heat. A simple relationship for this quantity is

$$Q_R = \sigma F_E F_A A (T_1^4 - T_S^4), \quad (C.2)$$

where σ is the Stefan-Boltzman constant, T_S the sink temperature, and F_E and F_A are emissivity and geometry factors that theoretically correct for practical nonidealities. Equation (C.2) implies that a high T_1 would also be a good choice to reduce the area (weight) required. A dilemma now exists.

To obtain the best choice of T_1 and T_2 for a given T_S , substitute equation (C.2) into (C.1) to obtain

$$W / (\sigma F_E F_A A) = (T_1^4 - T_S^4) [(T_2 - T_1) / T_1] \quad (C.3)$$

This result can be differentiated with respect to T_1 and then solved for T_2 , resulting in

$$T_2 = 4T_1^5 / (3T_1^4 + T_S^4) \quad (C.4)$$

When T_S equals zero, the best choice of T_1 is made when

$$T_1 = 3T_2 / 4. \quad (C.5)$$

The major theoretical index of performance for a space power plant is to obtain maximum work per unit of total power plant weight. In large space power plants, this condition is reduced to finding the minimum-weight radiator for the system; that is, it is more economical to obtain a minimum-weight system than to obtain the most efficient one when the transportation of the fuel is part of the system. The incentive to know the thermal efficiency is now only an aesthetic pursuit.

The Carnot efficiency, as a result of the effort to satisfy the requirement that W/A be a maximum, is found by substituting equation (C.4) into

$$\eta = (T_2 - T_1)/T^2. \quad (C.6)$$

The result is

$$\eta = (1/4)[1 + (T_S/T_1)^4]. \quad (C.7)$$

Again, when T_S is zero, the resultant Carnot efficiency is 25%. The trend of the relationship is shown on Figure (C.2). This curve shows the efficiency to be approximately constant at 25% from $T_S/T_1 = 0$ to $T_S/T_1 = 0.5$.

Fig. C.2. Carnot Efficiency vs Sink-to-Radiator Temperature

For $0.5 < T_S/T_1 < 1.0$, the efficiency for maximum work per unit of radiator area increases rapidly to 50%. However, when T_S approaches T_1 , the practicability of this plan breaks down because the radiator of a 50% efficient plant becomes infinite.

A more practical idealized cycle would be a Rankine cycle. The efficiency for this cycle, with temperature conditions $T_1/T_2 = 0.75$, and $T_S = 0$, is approximately 10%. The Rankine-cycle efficiency is not constant since it is a function of the working fluid properties. The analysis of the Rankine-cycle for a space power plant becomes increasingly more difficult as more nonidealities enter the problem.

Two important consequences of this Carnot analysis are as follows:

1. The desirable radiator-to-turbine inlet temperature ratio will be approximately 0.75.
2. The sink-to-radiator temperature ratio may be as great as 0.5 before T_S becomes a serious consideration in the design of the power plant.

Appendix D

RADIANT HEAT TRANSFER OF AN EXTENDED SURFACE

Radiant Heat Transfer from a Fin

Because the probability of meteor puncture is high, extended surfaces are important in the consideration of radiators for space power plants. A typical radiator would have several coolant tubes in a straight line, with extended surfaces (fins) parallel to the center line of the tubes. Finning reduces the number of coolant tubes required, and this reduces the probability that a vital coolant-carrying tube will be punctured. Protection from a high meteor flux will require extensive finning; a low meteor flux will require less finning. A fin may take the form of a rectangle, a triangle, or a surface of any polynomial description.

Analysis of the radiative power of a fin requires knowledge of the temperature profile in the fin, and the temperature profile is a function of fin geometry, sink temperature (variable), emissivity (variable), and thermal conductivity (variable). The mathematical representation of such a fin leads to a nonlinear, second-order, differential equation.

Mathematical Model

If $y(x)$ describes the surface of the fin, the amount of heat conducted into a small slice dx thick is

$$dq = \frac{d}{dx} \left[2Wk(T)y(x) \frac{dt}{dx} \right] dx, \quad (D.1)$$

or

$$dq = 2Wk(T)y(x) \frac{d^2T}{dx^2} dx + 2Wk(T) \frac{dt}{dx} \frac{dy(x)}{dx} dx + 2Wy(x) \frac{dk(T)}{dt} \left(\frac{dT}{dx} \right)^2. \quad (D.2)$$

(The meanings of the symbols are given at the end of this Appendix.)

Equation (D.2) is based on the assumption that the temperature gradients in the y and z directions are negligible. This assumption is valid for a long, thin fin (the actual situation).

Since the only heat loss is through radiation, dq must also be described by the equation

$$dq = 2W\sigma E(T)[T^4 - T_s^4(x)]ds. \quad (D.3)$$

If ds is assumed to be very nearly equal to dx , the general differential equation for T is represented by

$$\frac{d^2T}{dx^2} + \frac{1}{y(x)} \frac{dT}{dx} \frac{dy(x)}{dx} + \frac{1}{k(T)} \frac{dk(T)}{dT} \left(\frac{dT}{dx} \right)^2 - \frac{\sigma E(T)}{k(T)y(x)} [T^4 - T_s^4(x)] = 0. \quad (D.4)$$

Once the temperature distribution along the fin is found, the rate of heat dissipation from the surface of the generalized fin can be computed according to

$$q = \int_0^L 2W\sigma E(T)[T^4 - T_s^4(x)]dx. \quad (D.5)$$

A PACE analog computer was used to solve an example problem to show the gain in heat removal per pound of fin material as a function of geometry; results are shown in Table D.1. Factors common to this analysis were

$$k(T) = 25 \text{ (Btu/ft/}^\circ\text{F)},$$

$$E(T) = 1.0,$$

$$T_s(x) = 0^\circ\text{R},$$

$$T_0 = 2000^\circ\text{R},$$

$$\rho = 500 \text{ (lb/ft}^3\text{)},$$

$$H = 1.25 \times 10^{-3} \text{ (ft)},$$

$$h = 0.25 \times 10^{-3} \text{ (ft)},$$

$$L = 0.25 \text{ (ft), and}$$

$$\left. \frac{dT}{dx} \right|_{x=L} = 0.$$

Table D.1

SUMMARY OF THERMAL ANALYSIS FOR
EXTENDED SURFACES OF EXAMPLE PROBLEM

Fin Geometry (type)	Surface Description $y = f(x)$	Heat Loss per Unit Length (Btu/hr/ft)	Heat Loss per Unit Mass (Btu/hr/lb _m)
Rectangular	$y = H = h$	15,630	50,000
Wedge	$y = -\left(\frac{H-h}{L}\right)x + H$	15,180	80,900
Parabolic	$y = \left(\frac{H-h}{L^2}\right)x^2 - 2\left(\frac{H-h}{L}\right)x + H$	14,430	98,900

Symbols Used for Radiant Heat Transfer of a Fin

ds = differential element of a line

h = fin-end height

H = fin-base height

$k(T)$ = thermal conductivity, defined as a function of fin temperature

L = length

T_0 = absolute temperature at the fin base

$T_s(x)$ = sink temperature, defined as a function of fin distance

W = width

x = variable distance parallel to abscissa

$y(x)$ = variable distance parallel to ordinate

σ = Stefan-Boltzman constant

$E(T)$ = emissivity, defined as a function of temperature of the fin

ρ = density

Appendix E

RADIATOR AREA REQUIREMENTS

Introduction

The area requirements of a single-phase coolant radiator differ substantially from those for a two-phase coolant radiator. In the two-phase radiator, a near-isothermal condition exists. In the single-phase coolant, the temperature is reduced while passing through the radiator. This reduction in temperature means a loss in the heat-removal capabilities according to a fourth-power relationship.

Single-phase Radiator Area

Single-phase radiators are used for cooling a gas or for subcooling a liquid to prevent cavitation. The derivation is basically a matter of equating the heat-rate equations,

$$dq = U(T_b - T_w)dA_s, \quad (\text{convection}) \quad (\text{E.1})$$

$$dq = mc_p dT_b, \quad (\text{mass flow}) \quad (\text{E.2})$$

and

$$dq = \sigma E(T_w^4 - T_s^4)dA_s; \quad (\text{radiation}) \quad (\text{E.3})$$

where

dq = heat rejected (Btu/hr),

dA_s = sensible heat radiator area (ft²),

U = overall coefficient of heat transmission (Btu/ft²/hr/°R),

T_b = bulk mean temperature of coolant (°R),

T_w = outside surface temperature of coolant tube (°R),

m = mass flow rate of coolant (lb_m/hr),

c_p = specific heat of constant pressure (Btu/lb_m/°R),

σ = Stefan-Boltzman constant (0.1713×10^{-8} Btu/hr/ft²/°R),

E = emissivity (unitless), and

T_s = effective sink temperature (°R).

A quantity dT_b is found by equating equations (E.2) and (E.3) and differentiating the result. This expression is then substituted into the equation resulting from equating equations (E.1) and (E.2). The result is

$$dA_s = - \frac{mc_p}{\sigma E} \left[\left(\frac{dT_w}{T_w^4 - T_s^4} \right) + \frac{4\sigma E}{U} \left(\frac{T_w^3 dT_w}{T_w^4 - T_s^4} \right) \right]. \quad (E.4)$$

Integrating equation (E.4) from the inlet wall temperature, T_{wi} , to the outlet wall temperature, T_{wo} , and collecting terms,

$$A_s = mc_p \left\{ \underbrace{-\frac{1}{U} \ln \frac{T_{wo}^4 - T_s^4}{T_{wi}^4 - T_s^4}}_F + \underbrace{\frac{1}{4\sigma E T_s^3}}_{R_s} \right. \\ \left. \times \left[\underbrace{\ln \frac{(T_{wi} - T_s)(T_{wo} + T_s)}{(T_{wi} + T_s)(T_{wi} - T_s)}}_{B_1} - \underbrace{2 \arctan \left(\frac{T_s T_{wi} - T_{wo}}{T_s^2 + T_{wi} T_{wo}} \right)}_{B_2} \right] \right\}. \quad (E.5)$$

Fig. E.1. Values of F vs T_1/T_0 for Evaluating Single-phase Radiator Requirements

Values of F , R_s , and $(B_1 - B_2)$ are illustrated on Figures E.1, E.2, and E.3, respectively, to expedite the evaluation of sensible heat-rejection area requirements. The assumptions implied in equation (E.5) are as follows:

1. The radiator is in steady-state operation.
2. The quantities U , c_p , σ , E , and T_s are constant.
3. Conduction of heat in the axial direction is negligible.
4. The coolant channel wall thickness is small.

Equation (E.5) and the following equation can be used to evaluate the required area for a radiator:

$$(\text{Heat rejected}) = mc_p \Delta T_b. \quad (E.6)$$

Fig. E.2. Values of R_s vs T_s for Evaluating Single-phase Radiator Requirements

Fig. E.3. Values of $B_1 - B_2$ vs T_1/T_0 for Evaluating Single-phase Radiator Requirements

Two-phase Radiator Area

For a vapor-condensing radiator, the heat removed from the coolant is

$$Q = m\Delta h_x, \quad (\text{E.7})$$

where

m = vapor mass flow rate through turbine (gm/sec), and

Δh_x = latent heat of vaporization remaining after turbine expansion to quality X .

The heat radiated from the condenser walls is

$$Q = \sigma E(T_w^4 - T_s^4)A_L, \quad (\text{E.8})$$

where

- A_L = latent heat radiator area (ft^2),
- T_w = radiating temperature (assumed equal to condensing vapor) ($^{\circ}\text{R}$),
- T_s = sink temperature ($^{\circ}\text{R}$),
- E = emissivity, and
- σ = Stefan-Boltzman constant ($0.1713 \times 10^{-8} \text{ Btu/hr/ft}^2/^{\circ}\text{R}$).

Since Equations (E.7) and (E.8) are equal, the area required for a condensing vapor is

$$A_L = \frac{m\Delta h_x}{\sigma E(T_w^4 - T_s^4)}. \quad (\text{E.9})$$

ACKNOWLEDGMENTS

For his confidence and encouragement, Dr. R. A. Budenholzer deserves special acknowledgment.

Credit for the analog solution and constructive suggestions for the analysis presented in Appendix D are due to L. C. Just of the Applied Mathematics Division of ANL.

Appreciation for the numerical calculations of A , R_s , and $(B_1 - B_2)$ presented in Appendix E is due to Robert E. Grench of the ANL Reactor Physics Division.