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REACTOR, SYSTEM, AND COMPONENT ENGINEERING

REACTOR DESIGN

Reactor design efforts continued along general technology lines exemplified by reactors for the Rankine cycle reference power system designs SPR-4 (300 kW_e) and SPR-6 (10 MW_e). A number of new concepts have been proposed but few calculations have been carried beyond the point of showing general feasibility.

The larger reactors (exemplified by SPR-6) have a central cavity through which coolant is returned, surrounded by a region in which fluid control elements are placed. The latter are discussed elsewhere in this report. In the smaller, reactivity-limited reactors (exemplified by SPR-4), it is not clear that the central region is available for this purpose. This is not simply a matter of criticality but of control kinetics as well.

A reactor design typical of the SPR-6 size is shown in Fig. 30. This drawing is intended only to clarify certain features for analytical purposes and is not meant to bear inspection in detail. The reactor is supported within the pressure vessel on the downstream side by a hemispherical structure. The structure consists of welded tungsten rods of approximately equal length which are predominantly loaded in compression.

A thin membrane extends up from the skirt of this support, fitting closely around the active-core fuel-element bundle. A radially inward hydrostatic

pressure transmitted through this membrane stiffens the bundle into a rigid structure. A preliminary examination indicates that a rather low pressure is sufficient to prevent fuel element bowing.

The upstream end of the reactor is positioned within the pressure vessel by a spline structure that accommodates both radial and axial core dimensional changes. This spline is shown in Fig. 30 as a set of radial fingers attached to the pressure vessel through close-fitting slots in a slightly heavier extension of the core membrane just forward of the upstream core face. In this arrangement, rubbing is involved and might lead to self-welding. Another type of spline capable of bending in the radial and axial directions, but not in the circumferential direction, is being studied.

The operational loads are low, and no strength problems are foreseen except for launch loads which can presumably be supported by a removable auxiliary structure (perhaps, e. g., solidified lithium). Stiffness requirements are being studied from a control standpoint.

For the larger reactors, the most promising fuel elements are the hybrid type. For smaller reactors, the elements will be either pin-type B or internally cooled tubes (see Progress Report No. 5).

The multiple tube sheets shown in Fig. 30 permit circulation of interstitial and fuel bonding fluids; this may be required to remove fission gasses, replenish free nitrogen, etc. This

Fig. 30. General configuration of a 60 MW_{th} space power reactor designed for 2% burnup in 10,000-hr lifetime. The outside diameter of the pressure vessel is 65 cm.

Fig. 30 continued.

tube-sheet arrangement should only be considered representative of the concepts.

Pressure vessel cooling is being studied and is discussed elsewhere in this report. This will involve trade-offs in weight between the vessel and other components. These trade-offs are tied to the control problem due to the interference of vessel, coolant, and possible insulation with side-reflector control.

Means of achieving fine reactor control have been studied, and it appears that this must be accomplished by mechanically coupled external reflectors. It is expected that coarse control will be accomplished internally by isotopic mixtures, fluid-vapor interchange, etc., either in the central region shown or throughout the core proper. Dynamic coarse-control studies are just beginning. An analog reactor-control simulator was brought into operational status. This simulator models a spatially uniform temperature reactor with six delayed neutron groups.

Reactor Pressure Vessel Cooling

Radiation cooling of the reactor pressure vessel may be precluded due to the magnitude of the gamma and neutron heating and the likely presence of a reflector external to the pressure vessel, which would act as a radiation shield. However, a large amount of cooling is available with the subcooled potassium at radiator exit conditions.

Two concepts have been considered for cooling the pressure vessel. These concepts are discussed in terms of typical conditions for a large power system. In design A (Fig. 31) a portion of the lithium leaving the boiler separates from

the main stream and is cooled by the subcooled potassium in a preheater. The cooled lithium is used to cool the reactor pressure vessel and is then added to the main lithium stream prior to its passing through the reactor.

In the case shown in Fig. 31, the pressure vessel is cooled to a maximum temperature of 1410°K. At this temperature a tantalum alloy could be used. Thin tungsten cladding would be used internally to prevent contact with the lithium. A large weight savings in the pressure vessel would result. If the pressure vessel were uncooled, its temperature would be approximately 1800°K and a tungsten alloy weighing approximately 4000 kg would be used. The tantalum alloy pressure vessel and the required preheater together weigh ten times less.

Design A involves some thermal inefficiency in transfer of heat from the relatively hot lithium to the cooler potassium. For the case considered in Fig. 31, 2.6 MW is added to the lithium in the pressure vessel (due to the assumed internal heating), but 11.7 MW must be removed from the lithium in the preheater. The power handled by the preheater can be reduced to 2.6 MW by addition of another pump and a separate coolant loop. In this case the pressure-vessel coolant flow and the reactor coolant flow would not be mixed.

A more thermally efficient design would be to pass the subcooled potassium directly through the pressure vessel and then through the boiler, as shown in Fig. 32. Unless the potassium and lithium pressures in the reactor were equalized, a pressure vessel separating the two fluids would be required. Optimum operating

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Fig. 31. Design A for cooling the pressure vessel of the reactor. Typical conditions for large reactors are assumed.

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points and weights have not been determined for this design.

Reactor Control Concepts

Side-Reflector Control Concepts for High Power Reactors — Several methods of side-reflector control and devices for providing driving power were investigated for use with the central pipe configuration reactor (high power).

No attempt was made to examine the effect of any concept on reactor kinetics, control systems, or other components in the power system (i. e., boiler, shield, etc.); instead the attempt was only to investigate schemes which might, after further analysis, provide a feasible method of control. The results thus far from this investigation are summarized below:

Reactivity requirements that must be controlled by any mechanism selected are:

- 1) Ground safety.
- 2) Temperature or reduced reactivity effect.
- 3) Operational shutdown and reserve.
- 4) Operational losses.

If a single mechanism is chosen to meet these requirements, a wide range of adjustable control would be necessary, demanding the use of thick side reflectors. Thicker reflectors are heavier, difficult to cool, and require heavier shields. It has been proposed, therefore, that the largest overall reactivity requirement, ground safety, be controlled by initially filling the fuel core with a strong neutron absorber, Li^6 , and subsequently mixing this isotope with the reactor coolant, Li^7 , or the remainder of the primary loop just prior to startup in orbit. On the ground

Fig. 32. Design B for cooling the pressure vessel of the reactor. Typical conditions for large reactors are assumed.

and during launch, the lithium would be solid and would not mix. The use of a movable side reflector could augment this method of ground control by remaining in the least reflective position during all ground operations.

Provision for vernier control to provide for operational reactivity losses, startup, and reserve reactivity can be

incorporated into a side-reflector movement.

The adhesion problem present with metal-to-metal contact in a high vacuum environment is acknowledged. Detailed consideration of the problem was avoided however in this preliminary investigation. Pressurization of the environment in which the control components move (and lubrication) is assumed to be feasible at this point.

Two representative methods of providing reactivity control through reflector motion are shown in Figs. 33 and 34.

Figure 33 illustrates the use of axially moving reflector segments in two con-

centric layers. Both would be fully retracted from the core for shutdown and could be (independently) closed to various degrees for control during operation. The degree of reliability provided by this independent action of separate segments could be further enhanced by splitting the segments into a number of azimuthally separate sections.

Figure 34 shows a more familiar configuration with reflector segments rotating toward or away from the core to regulate neutron leakage.

Liquid Reactor Control System - The interest in high operating temperatures

Fig. 33 Split concentric-reflector system with axial motion.

in space reactors points to the possible use of liquids for control. Li^6 is an excellent neutron absorber for the control of high-temperature fast reactors because of its absorption peak at 1/3 MeV. Further advantage may be taken of the Li^7 isotope. Different mixtures of Li^6 and Li^7 , occupying about 20% of the core volume, sway k_{eff} by 30%. Predetermined or time-dependent mixtures can thus be applied to a variety of control situations that would be impossible with solids.

The geometric temperature effect on reactivity can be decreased by a proper selection of the lithium isotopic composition in the interstitial* and bonding fluids. The bonding fluid is the heat-transfer medium between the fuel and cladding. It experiences a displacement relative to the fueled core that results from differences in the thermal-expansion coefficients for itself, the fuel, and the cladding.

Operational losses in reactivity due to fuel burnup, etc., can likewise be decreased by a proper selection of the Li^7 enrichment in the bonding fluid. Here it is the swelling of the fuel during operation that displaces lithium from the core, causing partial self-control.

Operational shutdown or scram of the reactor in space is readily accomplished by injection of Li^6 into tubular voids within the fueled core. Removal of this lithium for restarting in space can be achieved by means of compressed gas or evaporation.

Vernier control of the reactor may be needed only for positive and negative

Fig. 34. Rotary segment reflector.

reactivity amounts of less than \$1.00 because of partial or total compensation of miscellaneous reactivity losses, as described. The mechanism is identical to that suggested above for operational shutdown, except that both Li^6 and Li^7 are used in separate channels for negative and positive control.

One disadvantage of this control system is the reactivity hazard resulting from total expulsion of all liquid media in the fueled core. To circumvent this difficulty, the overall concentration of Li^6 should be required to be above a specific value in each reactor; this would not always be possible.

Applicability of this concept depends in large measure upon the neutron energy spectrum of the reactor. System kinetics require detailed attention.

The advantages of no moving parts and minimum reactor size (minimum shield size) indicate that further attention should be given to this concept.

* This discussion applies to large central-pipe reactors having hydride-type fuel elements.

Two-Phase Fluid Control Concept -

A method of fluid reactor control which appears to minimize the need for moving parts and seals is presented schematically in Fig. 35. It would operate as follows: The control fluid in the liquid phase passes

Fig. 35. Loop for two-phase fluid-control concept.

into a duct in the control element. The control element in this scheme corresponds to an unfueled core element or an external reflector element. The fluid flows through a porous matrix and evaporates at some point in the matrix. The heat of vaporization is obtained by γ heating of

the liquid itself, by conduction from the γ -heated matrix material, and by transfer into the control element if the temperature gradient is favorable. The vapor leaves the porous material and passes out of the control element through a duct. The vapor is then condensed and the resulting liquid is pumped back into the control element to complete the cycle.

The rate of evaporation is determined by the rate of heat generation in the control element and the rate of heat transfer into the control element. If liquid is pumped into the control element faster than it is evaporated, the proportion of liquid will increase and, conversely, if liquid is pumped in more slowly than the rate of evaporation the proportion of liquid in the control element will decrease. Control of the thickness of liquid can be obtained in two ways:

- 1) By varying the flow rate of liquid into the control element. This can be accomplished by varying the power input to the pump or by a bypass line around the pump with a variable valve.
- 2) By varying the amount of heat transferred into the control element from its surrounding environment and thereby controlling the rate of evaporation. This can be accomplished by varying the temperature of the control element, which in turn is controlled by the pressure in the control element since a two-phase fluid is present. The pressure in the control element can be controlled in two ways: a) The temperature, and hence pressure, of the condenser remaining fixed, the degree of pressure drop through an adjustable control valve (see Fig. 35) will determine the pressure in the control element, or b) The pressure drop in a

fixed valve being determinate, the regulated temperature, and hence pressure, of the condenser will determine the pressure in the control element.

Now consider the stability of the device. If the liquid in the control element is used as a reflector, an increase in reactor power will increase the rate of heat generation in and heat transfer to the control element, thereby increasing the rate of evaporation and decreasing the thickness of the reflector. Conversely, a drop in reactor power will decrease the rate of evaporation, increasing the thickness of the reflector and increasing the criticality of the reactor. Thus the system has a negative power coefficient of reactivity.

On the other hand, suppose the liquid in the control element is a heavy absorber.

Similar reasoning leads to the conclusion that a poison-type control element would not be inherently stable.

Accidents such as pump malfunction, a leak of liquid from the piping, or a loss of cooling in the condenser would lead to reduced liquid in the control element and would thus be safe for a reflector element and unsafe for a poison element. Other accidents can be hypothesized which would be unsafe for a reflector element, such as a run-away pump or a leak in the vapor piping.

An evaporating liquid control system is attractive in concept, particularly if used as an internal moderator-reflector element for vernier control. Further evaluation of its practicability awaits detailed design calculations.

SYSTEM ANALYSIS

System studies of high and low power Rankine cycle power plants were continued. The availability of liquid-metal transport and thermodynamic properties in a usable form permitted a preliminary comparison of working fluids. The results of these studies are presented in Fig. 36.

A more detailed listing of some of the characteristics of the systems at specific temperatures appears in Tables XIV and XV. It can be seen from this table that increasing radiator mass causes high system mass at lower temperatures while increasing containment vessel mass (boiler, e. g.) causes the increase in system specific mass at higher temperatures.

These results must be considered preliminary since extrapolated working fluid

Fig. 36. Specific mass vs temperature for Rankine cycle power system. Results are shown for 300-kW_e and 10-MW_e systems using cesium and potassium as working fluids. System parameter values are listed in Table XIV.

Table XIV. Parameter values defining the power systems of Fig. 36.

Output power	300 kW _e	10 MW _e
Payload radius	125 cm	2.5 m
Distance, reactor to payload	12.8 m	100 m
Reactor burnup	0.0075	0.02
Enrichment, U ²³⁵	0.8	0.4
Meteoroid non-penetration probability	0.90	0.99
Shield thickness (neutron)	65 cm LiH	118 cm LiH
Turbine rpm	12,000	12,000
No. turbines	1	4
Pumps	2 electric centrifugal	2 electric centrifugal
Loops	2	2
Auxiliary cooling and radiator	1	1
Reactor outlet-to-turbine inlet temp difference	130°K	130°K
Turbine temperature ratio	0.75	0.75
Radiator pressure drop	20%	20%
Radiator armor	Be	Be
Materials	TZM, pure tungsten	TZM, pure tungsten
Turbine blades	Sylvania A strength	Sylvania A strength
Operation time	10,000 hr	10,000 hr

properties were used. Also, some components in the system are not properly modeled over large power and temperature ranges. Changes in the model and related assumptions can cause significant variation in the values shown. A preliminary conclusion, however, is that minimum system specific mass is less if potassium, not cesium, is the working fluid.

Turbines are critical components in all systems. Cesium turbines have fewer stages than potassium turbines. In the 300 kW_e systems shown, no turbines are overstressed. In the 10 MW_e systems, all cesium turbines are overstressed, and potassium turbines in systems with reactor outlet temperatures greater than 1560°K are overstressed. Such conditions

can be alleviated by: 1) Reducing speed from 12,000 rpm to 7700 rpm, which results in a system specific mass of 7.6 kg/kW at 1800°K, 2) increasing the number of turbines from 4 to 10, which results in a system specific mass of 7.9 kg/kW at 1800°K, or 3) incorporating stronger materials.

The turbine model upon which the system studies are based is optimized for potassium and is applicable to the study of cesium turbines only as a first approximation. Turbine efficiency, number of stages, velocity ratios, inlet temperature reduction, and losses will be studied in more detail.

System studies and detailed component analyses require reasonably accurate,

Table XV. Characteristics of 300 kW_e and 10 MW_e systems using potassium as the working fluid. Specific masses which are omitted are ≤ 0.1 kg/kW_e.

Power, MW	0.3	10	0.3	10	0.3	10
Reactor outlet temp, °K	1420	1420	1640	1640	1835	1835
Specific mass, kg/kW _e						
Reactor	0.8	0.3	1.0	0.3	1.3	0.4
Shield	1.2	0.6	1.1	0.4	1.0	0.3
Li reactor coolant	0.2	—	0.2	—	0.2	—
Boiler	—	—	0.2	0.2	5.8	3.1
Turbine	0.8	0.2	1.0	0.4	1.8	0.9
Alternator and radiator	4.6	0.8	4.6	0.8	4.6	0.9
Li pump	—	—	—	—	—	—
K pump	—	—	—	—	0.2	—
Structure	1.1	1.0	0.7	0.5	0.5	0.3
K working fluid	0.2	0.8	—	0.2	0.2	—
Power distribution	—	—	—	—	—	—
Controls	0.3	—	0.4	—	0.4	—
Primary piping	—	—	0.2	—	1.1	0.2
Secondary piping						
Radiator	<u>1.8</u>	<u>9.2</u>	<u>0.8</u>	<u>3.8</u>	<u>0.5</u>	<u>2.1</u>
System (Total)	11.4	13.3	10.4	7.0	17.8	8.7
K flow rate, kg/sec	0.9	29.9	1.1	32.4	1.3	34.3
Li flow rate, kg/sec	4.3	136	4.9	143	5.5	148
Reactor power, MW	1.94	61.7	2.21	64.2	2.5	66.2
Cycle efficiency, %	15.5	16.2	13.6	15.6	12.0	15.1
Radiator area (M ²)	91	2700	54	1600	37	920
Number turbine stages	9	4	12	6	13	7
Turbine efficiency	0.76	0.78	0.70	0.79	0.66	0.80
Turbine inlet pressure, bars	6.8	6.8	20.5	20	42.3	42.3
Turbine exit pressure, bars	0.5	0.5	2.4	2.5	6.5	6.5
Turbine inlet temp, °K	1290	1290	1510	1510	1705	1705
Li pressure, bars	0.8	5.8	1.8	6.9	4.5	9.5
Turbine blade stress	OK	OK	OK	Over stressed	OK	Over stressed

consistent, and accessible property data for all working fluids and coolants under consideration. Two computer subroutines were prepared to provide a central data

source for power system studies, both on and off the computer. One subroutine (called EVAP) generates thermodynamic property data for liquid metals in the

saturated and supersaturated regions. Subroutine EVAP is a revision (using improved data) and extension of subroutine KVAP, which was described in Progress Report No. 1. The purpose of the second subroutine (called TRANP) is to generate transport and other miscellaneous property data for all the alkali metals.

TRANP provides data on viscosity, thermal conductivity, specific heat at constant pressure, density, surface tension, electrical resistivity, and sonic velocity for saturated liquid and vapor conditions. The materials currently included are lithium, sodium, potassium, rubidium, cesium, mercury and NaK-78. Temperature is input and a specified property value is computed.

Temperature-property correlations were obtained by least-squares fits to tabular property data, the primary source of which was the compilation by Weatherford *et al.*²⁰

Since very little experimental work has been done on the determination of liquid-metal properties above 1500°K the system studies depend in large measure on extrapolated property data at temperatures above that value.

RADIATOR ANALYSIS

Direct-Condensing Tube-Fin Radiator

The radiator type considered thus far in the system studies is a direct-condensing tube-fin radiator with a flat rectangular configuration. The radiator consists of a central tapered inlet duct, a return duct on either side of the inlet duct, and a large number of finned tubes connecting the inlet and return ducting. The tubing is protected from meteoroid

bombardment by a shell barrier. The working fluid exhausts from the turbines into the inlet duct, where it is distributed among the tubes. The fluid condenses and is somewhat subcooled in the tube-fin array. The subcooled liquid is collected in the return ducts.

TOAST is a computer subroutine used to aid in the design of a minimum-mass tube-fin radiator. Calculations performed by TOAST yield optimum values for tube diameter, tube length, fin width, and fin thickness. In the version of TOAST used for the 300-kW_e SPR-3 reference design, the meteoroid barrier could be ignored in the thermal analysis. Essentially, the fin radiated all the heat, and its peak temperature was the fluid temperature. The meteoroid barrier thickness, t , scales with the vulnerable area, A , and the survival probability, P , in the following way:

$$t \propto \left(\frac{A}{-\ln P} \right)^{1/4} \quad (19)$$

The barrier thickness must be taken into account for high power, high survival probability systems, and the simplifying assumptions cannot be made for the SPR-6 system. TOAST was rewritten to include the heat rejected by the barrier, the temperature drop in the barrier-covered portion of the fin, and an exact calculation of the tube and fin view factors for this more dominant barrier case.

The revised code resulted in a minimum mass for the SPR-6 radiator of 56,000 kg with no tube-to-barrier contact. (Heat transfer from the fluid-carrying tube to the shell barrier is by radiation only). It is possible to design a considerably lighter radiator by specifying perfect tube-to-barrier contact. For this

case, the code yielded a minimum mass of about 21,000 kg. However, it was discovered that the perfect contact case resulted in a barrier temperature at which a significant amount of the barrier would vaporize during the mission. The code was further revised to allow for this material vaporization. The final code yielded a minimum mass of 28,000 kg for the perfect contact case. (The mass of the no-contact case was not affected by the vaporization revision because of the low barrier temperature.)

The high radiator mass calculated for SPR-6 is primarily due, either directly or indirectly, to the thickness of the meteoroid barrier. The barrier mass makes up approximately 60% of the total tube-fin barrier mass for the perfect-contact case and 70% of the total for the no-contact case.

Radiator Configuration, Flat vs Conical

Figure 37 shows the physical arrangement of a cylindrical reactor, a shield, a

radiator, and a spherical payload. In order to protect the payload from radiation, the shield must place both the payload and the radiator in the shadow from the reactor. The shadowing of the radiator is necessary to prevent radiation scattering. Two possible configurations for the radiator which result in geometrically different shields are a flat trapezoid and a truncated cone.

The radiator masses and the shield masses for the two configurations were compared for radiators of equal thermal performance. Typical calculations indicate that the flat configuration results in a lower system mass.

Briefly, the analysis is as follows: The surface area of each radiator is calculated. The effective radiating area of the flat radiator is assumed equal to its total two-sided surface area. The effective radiating area of the conical radiator is calculated as $A_c (1 + F)$, where A_c is the surface area of the outside of the truncated cone and F is the view factor

Fig. 37. Geometric model for radiator mass comparison.

from the inside of the cone. Comparisons are made between radiators with equal effective radiating areas. This equality can be achieved in any number of ways since the radiator length, h , or the radiator half width, c , may differ for the two configurations. The two cases which were considered in this analysis are:

$$\text{Case 1: } h_{\text{flat}} = h_{\text{cone}}$$

$$\text{Case 2: } c_{\text{flat}} = c_{\text{cone}}$$

The radiator masses are approximated by setting mass proportional to actual surface area (i. e., a radiator structure of fixed thickness and average density). The axially varying cross sections of the required shields are determined and the shield volumes calculated. The shield for the conical radiator has circular cross sections; the shield itself is a truncated cone. The cross sections of the flat radiator shield consist of a circle (to shadow the payload) with semi-circular-ended rectangular wings (to shadow the radiator). The shield masses are assumed proportional to their volumes.

Calculations were performed with typical proportions for a high power system, as indicated in the following (refer to Fig. 12):

$$\begin{aligned} \ell/r &= 2.4 \\ s_1/r &= 1.0 \\ t/r &= 4.0 \\ s_2/r &= 10.0 \\ r_p/r &= 5.0 \\ p/r &= 200.0 \end{aligned}$$

The results of these calculations are shown in Fig. 38. The radiator mass ratio (flat over conical) always favors the flat configuration and can approach a value of 0.5 when the radiator becomes long and slender (high h/r value, small radiating

Fig. 38. (a) Ratio of shield mass for flat radiator to that for a conical radiator of the same area, and (b) similar ratio of radiator masses. The mass ratios are plotted as a function of radiating area (relative to reactor size) for various fixed values of radiator length, h , or width, c (relative to reactor size).

area), since the inner cone surface becomes useless as a radiator. The shield mass comparison tends to favor the flat radiator except in the case of a long and slender shape, but even here the advantage of the cone is slight.

Comparisons made on the basis of equal (and large) values of h/r are felt to be more meaningful than those for

equal c/r, since presumably the design would take advantage of as much length as possible for either shape in order to minimize the radiator width and hence the shield mass.

Heat Pipe Radiator Concept

One of the principal problems in the design of space power systems is the dissipation of waste heat by radiation into space. This problem is especially severe at high power levels such as are considered in SPR-6. Even with favorably high radiation temperatures, very large surface areas are required.

Radiator arrays which use a single circulation system, i. e., ones in which there is no segmentation and all fluid carrying tubes have a common source, must be designed with extremely high structural integrity since high (e. g., 99%) probability of survival is mandatory. In addition to the predicted meteoroid hazard, ordinary structural failure modes (corrosion, fatigue, etc.) must be considered.

Radiator designs considered at LRL and elsewhere have almost always revolved around two approaches. The first of these is the classical tube and fin model in which vapor out of the turbines or

energy converter is caused to flow through a multiplicity of tubes where heat is rejected by: 1) Direct radiation from the tubes to space, and 2) conduction through the fins and then radiation to space. A typical fin-tube geometry is shown in the following sketch at approximately half scale (dimensions are given in cm).

Pertinent characteristics of this radiator obtained from a TOAST calculation are listed as follows:

Heat rejected	50 MW
Inlet fluid temperature	1250°K
Number of tubes	497
Tube length	6.5 m
Radiator width	13 m
Radiator length	32 m
Area (projected)	410 m ²

Masses

Inlet duct	1,440 kg	5% of total
Outlet duct	1,240	5
Barrier	15,500	56
Fins	9,430	34
Tube	70	
Fluid (K)	8	
Total radiator	27,700 kg	100%

The mass for meteoroid protection is not explicit in the data on masses. In addition to the 56% barrier mass, there is 14% fin mass to conduct heat from the

tube to the outside diameter of the barrier, giving a total of 70% attributable to meteoroid protection.

To gain some mass reduction, a second approach to radiator design has been used.²¹ This is one in which the solid fin is replaced by what is termed a vapor fin. A representative geometry for a vapor-fin radiator is shown in the following sketch.

Haller et al. show that for systems in the power range 500 to 1000 kW_e designed for 500-day life, substantial radiator mass savings (as much as 50%) may be realized with the vapor-chamber fin-tube radiator. The mass saving is stated to be less for higher power systems.

An alternative to finned radiators is the heat pipe radiator. In this concept, the inlet and outlet ducts originally used for vapor delivery and fluid return become the primary leg of a condenser/heat exchanger. Meteoroid protection of this ducting is inexpensive because its exposed surface area is low. One end each of a large number of heat pipes is inserted into this ducting. These heat pipes operate in parallel, function independently from one another, and form the separate secondary leg of the heat exchanger. Because many independent heat pipes are involved, they can be designed at minimum mass for a specified probability that a certain fraction will remain unpunctured during

the mission.²² This arrangement can be described in simplified terms as a large U-shaped tube out of which protrude the heat pipes in a plane array, as shown in Fig. 39.

A preliminary estimate of the mass of a heat-pipe radiator was obtained with the following assumptions:

Heat rejected: 50 MW

Rejection temperature: 1250°K

Max radiant heat flux: $Q_r = 10 \text{ W/cm}^2$

Max condensing heat flux, primary fluid-to-heat pipe: $Q_c = 50 \text{ W/cm}^2$

Max axial heat transport in heat pipe: $Q_a = 1000 \text{ W/cm}^2$

Heat-pipe diameter: $D = 8 \text{ cm}$

From the last two assumptions, one finds that the total energy transport in each heat pipe is 50 kW, so that 1000 heat pipes are required.

The length, ℓ , of the heat pipe end within the ducting was determined by equating the radial condensing heat transfer to the maximum axial heat transfer in the heat pipe:

$$Q_c (\pi D \ell) = Q_a \left(\frac{\pi D^2}{4} \right), \quad (20)$$

yielding for the assumed case $\ell = 40 \text{ cm}$.

It was assumed that the heat pipe of external length L radiates from both sides of a flat plate of width D and length L . The required length, L , was determined by equating the radiant heat transfer to the maximum axial heat transfer in the heat pipe:

$$Q_r (2DL) = Q_a \left(\frac{\pi D^2}{4} \right), \text{ yielding } L = 314 \text{ cm} \quad (21)$$

Fig. 39. Heat pipe radiator concept.

The advantage of the heat pipe is seen in comparing the latter two lengths. A heat transfer amplification of eight can be achieved and this at nearly isothermal conditions.

At 1250°K the vapor pressure of potassium working fluid would be about 5 atm. If the heat pipe is made of niobium alloy (allowable stress of 5000 psi), a pipe wall thickness of 0.060 cm is required. The density of niobium is 8.4 g/cm³. With these assumptions, the mass of the heat pipe is:

Structure	1136 g
Capillary	3040
Fluid	<u>1232</u>
Total	5408 g

To the mass of the heat pipes must be added the mass of the heat exchanger shell. It is conservatively estimated that the mass of the shell is equal to the mass of the heat pipes.

The shell mass includes meteoroid protection of the shell and a short length of the heat pipes (to protect the shell against oblique shots through the heat pipe penetrations). The heat-pipe wall thickness is increased inside the shell to prevent buckling of a heat pipe by the primary fluid in the event that a heat pipe vents.

Estimated radiator mass values of the solid-fin and vapor-fin types for the same thermal performance are compared

with those for the heat pipe radiator in the following table:

	Heat pipe radiator	Vapor fin radiator	Solid fin radiator
Specific mass, kg/kW	1.06	1.95	2.78
Relative mass	1	1.84	2.62

In addition to the potential savings of mass in the heat pipe radiator, there is a reliability advantage because each pipe is independent. The heat pipe scheme can provide a very redundant system, so that a means of detecting leaks and isolating damaged segments is not required.

TUNGSTEN HEAT PIPE FABRICATION TECHNIQUE

It has been demonstrated analytically and experimentally^{23,24} that the design of the wick structure in a heat pipe plays a very significant role in the overall heat transport capability of the pipe. In the wick it is desirable to provide small-diameter pores to give high pressure drop at the meniscus and large-diameter liquid flow passages to minimize drag forces. This is probably best done with a two-component structure since the two requirements are diametrically opposed. The two-component structure might consist of open channels covered with layers of fine mesh screen.

Investigations at LRL have been directed toward the more difficult of the wick fabrication problems: that of putting deep channels on the inside of small-diameter (1 cm) pipes of significant length (≈ 35 cm). The end objective of

the study is to be able to produce a tungsten alloy heat pipe which can be operated at $\approx 2000^\circ\text{K}$ as an emitter of a thermionic diode; such a pipe will contain a large number of channels separated by thin fins. The optimized groove structure can of course be used to equal advantage at lesser temperatures for other applications.

A cylindrical molybdenum mandrel was fitted with slotted wheels at each end. The slots in the periphery of the wheels have parallel sides separated by a distance equal to two fin thicknesses and one channel width. The lands left on the wheel are one channel-width wide. Each wheel slot is fitted with two tungsten ribbons and one molybdenum ribbon running the length of the mandrel from wheel to wheel. Shorter molybdenum ribbons are placed opposite the wheel lands in a similar manner. The assembly consists of alternate tungsten and molybdenum ribbons tightly packed (slightly looser at the outer diameter) around the Mo cylinder. The assembly is shown in this condition in Fig. 40. In the next step, tungsten will be vapor deposited on this assembly to obtain the desired tube wall thickness. Subsequently, the molybdenum components will be dissolved. End pieces will be attached, also by vapor deposition.

A preliminary evaluation of the proposed method was made using copper to

Fig. 40. Tungsten-molybdenum ribbon assembly to be used in fabrication of tungsten heat pipe.

represent tungsten and aluminum for molybdenum. Electrolytic deposition was used in place of vapor deposition. The test was highly successful. Figure 41 shows selected views of this copper heat pipe. There are 90 fins, each 0.0125 cm thick by 0.1 cm high. The groove width is 0.0225 cm.

CORROSION LOOP PRELIMINARY ENGINEERING DESIGN

An outline of the liquid metal loop program and a discussion of its objectives were given in the "Materials" section, p. 32. The preliminary engineering de-

sign of the tungsten alloy^{*}/lithium corrosion loop is summarized below. The shape of the loop, shown in Fig. 42, was selected to be consistent with available material forms, and fabrication and test facilities. All heavy components of the loop are supported in a vertical column above the pump electrode. The loop is free to expand from that point of support. Guide slots are provided elsewhere to constrain expansion to the loop plane only. The return leg, the only overhanging component,

^{*}In the initial tests, W-25Re-30Mo will be used.

(a) Before etching of aluminum mandrel and aluminum filler strips

(b) After etching of aluminum

(c) Finish machined

Fig. 41. Copper-aluminum electroplated heat pipe made to evaluate proposed method of vapor-depositing tungsten on molybdenum.

is a single length of preformed tubing comprising less than 2% of the total loop mass.

Heat is supplied to the loop by a concentric radiant heater and is rejected to the vacuum system walls mainly by

Fig. 42 Pumped corrosion loop.

radiation but in part by conduction at the pump bus bars. Heat rejection is controlled by secondary "bias" heaters, which vary the sink temperature apparent to the rejection leg of the loop. Pumping of the fluid in the reference design is done electrically by a straight-through conduction pump. Ability to fill, drain, and

pressurize the loop safely is provided by tanks which are supported from a column extending from the upper elbow. Argon pressurization lines and lithium transfer lines connect the loop to outside systems through these tanks.

In operation, the loop will be evacuated, filled from an external source, and pressurized through both tanks so that the fluid loop is a simple hydrostatic system operated under externally controlled pressure. The loop is brought up to temperature isothermally, and the required temperature difference is then established by control of the bias heater.

Heat Source

The heat source selected for the preliminary design is a radiant heating element in the form of a split cylinder entirely surrounding a loop vertical section. This loop heat addition(test) section is composed of a thick-walled tube, 0.5-in. o. d., 0.135-in. i. d., and 24 in. long. The heater element when assembled is 2 in. in diameter and 25.5 in. long. Because of the high length-to-diameter ratio of the heater element, precautions are taken to minimize uneven expansion and warping. The element is constructed symmetrically with the two halves mechanically interlocked and tension bands wrapped around the heater at assembly. The element, shown in Fig. 43, is supported at the upper arms with the lower end guided but free to expand. The electrical connection of the lower end is provided by stacked tungsten-foil straps. A wire mesh design is being carried as an alternative, pending performance tests on the reference design.

The shield pack surrounding the heater element is a free-standing 10-layer

Fig. 43. Heater electrode for pumped corrosion loop.

structure of half cylinders interleaved together, plus upper and lower shield packs to limit axial heat losses. Heater element arm pass-throughs and visual access holes for pyrometer measurement are provided.

With the 10-layer shield pack, the element was calculated to deliver 10 kW thermal power to the loop at 1900°K with an input of only 17.5 kW electrical power to the element at 2520°K. Data

from tungsten element manufacturers shows expected lifetimes in excess of 1000 hr at temperatures up to 2800°K.

Pump

The pump selected for the preliminary loop design is a straight-through conduction pump and is shown in Fig. 44. An indentation was made in the vacuum vessel wall so that the pole faces of an external magnet could be brought into

Fig. 44. Corrosion-loop pump body. The material is W-25Re-30Mo alloy.

close proximity to the pump section. Tungsten alloy electrodes are diffusion bonded to a length of 0.5-in. o. d. tubing to form the pump body. Vertical extensions of the electrodes engage the electrical pass-throughs, which support the entire loop. Holes drilled in the electrodes redirect the current and minimize thermal conduction losses from the loop.

Shielding the pump against radiant heat losses is complicated by the requirement that the pump, its shielding, and the vacuum vessel wall must all fit within the

1-in. air gap of the magnet. Gold-plated shields and vessel walls will be used to minimize heat transfer in this region. Calculations have shown a minimum heat loss thru the pump on the order of 3 to 4 kW, which is one-third of the thermal input power of the loop.

While the efficiency of this simple conduction pump is expected to be very low, only about 4-watts pumping power would be required to obtain the desired performance at 100% efficiency. The conduction pump was chosen since it can be fabricated per existing tungsten alloy techniques. A more extensive pump investigation program is described elsewhere and will provide the basis of an alternative selection should the performance of the conduction pump prove inadequate.

Heat Rejection

Heat is rejected from all surfaces except the heated section. Temperature control is achieved by various means of limiting this rejection. The problem of the pump electrode was already discussed. Of the total heat added in the heated section, about one-third is rejected at the pump, one-third in the bias control section, and one-third in other sections of the loop, such as upper and lower crossovers, elbows, loop guides, etc. Rough control of heat rejection is attained by adding or removing tungsten foil shields from such structure as necessary to bring the loop thermal balance within the controlling range of the bias heaters.

The bias heater elements are an open-mesh structure of about 40% porosity. By varying the power to the elements, more or less heat is allowed to be radiated from

the loop rejection leg to the vessel walls through the bias element. Element temperatures of 2230°K and power of 35 kW were calculated as sufficient to totally block heat flow from the control section of the loop. Because of the critical nature of the thermal control function of these heaters, two separate elements with parallel electrical connections are specified to allow normal shutdown should one fail. Replacement of failed elements has been kept a simple operation.

Junction Boxes (Elbows)

The necessity of providing one or more pipe tees to allow filling and draining of the loop required design of a fitting to permit weld joining of components without undue strain at the joints. Considerable simplification of loop fabrication is achieved by the use of the massive but versatile three-axis J-boxes shown in Fig. 45. Ample mechanical rigidity is obtained to relieve stresses on the welds, which are of the electron-beam standing edge type. The access ports are closed by seal welded plugs and backed up by mechanical fittings. The J-boxes are machined from solid billets of extruded and forged tungsten ternary alloy. Two of these joints were required by the fill-drain system selected.

Tanks and Piping

The fill-drain and expansion tanks, Fig. 46, are diffusion-bonded from the nominal loop alloy. Integral weld necks allow later attachment of connecting tube fittings by electron beam welding.

The height of the tanks was determined by expected density- and flow-induced

Fig. 45. Corrosion-loop junction box.

head differences plus incidental dynamic level variations of ± 2.5 in. without entrainment of cover gas. The volume of each tank was sized to hold the entire lithium charge plus 10% to minimize losses in the event of an accidental blowdown. The volume of the Li charge is approximately three times the net circulating system

Fig. 46. Corrosion-loop expansion tank.
volume (12.2 in.³). This results in a normal static fill level of from one-fourth to one-third the height of the tanks.

The diameter of connecting lines is sized so as to provide dynamic damping and to minimize both thermal and fluid interchange between the circulating and fill systems.

Figure 47 is a diagram of the loop with auxiliary components and interconnecting piping.

Loop Containment and Test Facility

The entire elevated-temperature loop assembly will be supported from an instrumentation spool within a 24-in. diameter ion-pumped vacuum chamber. The loop chamber hardware being prepared is compatible with both the 6000 liter/sec horizontal and 2400 liter/sec vertical vac-ion pumps shown in the center and upper field of Fig. 48. The vacuum test area shown will be used for a variety of high-temperature tests, including the corrosion loop.

Low Temperature NaK Test Loop

A room-temperature NaK loop was put into operation during this quarter and was used to test one model of the rotating dc-field induction pump. Figure 49 shows the complete system, with the 0.95 cm o.d. tube helix in the pump (lower left).

Figure 50 is a diagram of the various NaK loop components. Eutectic grade NaK is used (freezing point 262°K) with an argon cover gas. Maximum system pressure is 3.5 bar; above this pressure the relief valve allows the NaK to flow into the dump can. The cylindrical sodium-potassium reservoir (with glass end-windows) is normally half-full during loop operation. During pumping, the flow path is from the bottom of the reservoir through the pump

Fig. 47. Corrosion-loop lithium supply system.

and back into the reservoir through the top line. This gives a visual indication of flow and also serves to trap oxide formations in the reservoir.

NaK charging of the loop is done only after the system has been cleaned, evacuated, and flushed with argon gas a number of times. Some oxides form over long periods of time, but these are pumped into the reservoir and have not caused blocking of the flow path. Operation to date has been essentially free of problems as far as contamination is concerned.

The system instrumentation consists of a 0.43-cm orifice and three pressure transducers. Flow is detected by monitoring the pressure drop across the orifice and across the pump. Pump output pressure is monitored directly. Initially, strain-gauge pressure transducers compatible with NaK were installed. Operation of the loop soon indicated the desirability of being able to change these transducers without opening the loop to air. Flexible diaphragms (bellows assemblies) were tested and found to be suitable for isolating

Fig. 48. High vacuum test facility. It is planned to install the high temperature corrosion loop atop system GE-3.

Fig. 49. NaK test loop with dc helical induction pump.

the transducer element from the liquid metal through some intermediate fluid. This also eliminates the necessity of using transducers compatible with NaK and allows more precise pressure measurement. These diaphragm units have been installed.

The following prototype pump ducts were fabricated from stainless steel for testing in the rotating dc field pump:

- 1) 0.95-cm o. d. tubing. A 9-1/2-turn double helix (Fig. 51). Wall thickness 0.1 cm.

- 2) 1.25-cm o. d. tubing. A 7-1/2-turn double helix. Wall thickness 0.1 cm.

- 3) 1.6-cm $\times$ 0.32-cm rectangular channel. A six-turn double helix (Fig. 52). Fabricated from 0.038-cm sheet stock.

The analytical output curves for these ducts were shown in Progress Report No. 5. Best results were expected with the rectangular duct.

The 0.95-cm o. d. tube helix yielded lower than expected performance, showing a maximum pressure rise of about 2 psi and a maximum flow rate of about 1 gal/min. Preliminary tests showed that performance of the two other configurations was also below expectation, but time was not available for careful measurements

Fig. 50. Room temperature NaK loop. The various numbered and lettered components are valves.

before this experimental setup had to be dismantled in order to prepare for testing of conduction pumps of the type selected for the high-temperature corrosion loop (already discussed).

Two types of dc conduction pumps were selected for testing in the NaK loop. These are shown in Fig. 53. The simplest consists of a simple straight tube with conducting sidebars and is the basic design proposed for the 1800°K corrosion loop. This unit will be tested in conjunction with the dc magnet shown in Fig. 54, and will

provide a design guide for the high temperature configuration, and serve to indicate the range of current and flux density required for the corrosion loop. Space and material restrictions in the corrosion loop greatly limit the range of design for this type of pump.

The second dc pump is a hairpin type which could fit into a particular flanged housing on the corrosion loop pressure vessel. This configuration has some installation advantages, although no plans exist at present for incorporating it into

Fig 51 Fluid duct coil for dc helical induction pump The outside diameter of the tubing is 0.95 cm

Fig 52 Built-up fluid duct coil Duct has a square cross section 1.6 x 0.32 cm

Fig. 53. Two types of dc conduction pumps to be tested.

the corrosion loop. The dc current is supplied through a center bus bar and returned through two outer side bars. Its

performance is not expected to be any better than that of the straight-section pump.

Fig. 54. Conduction pump dc magnet.

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