**AIAA
Specialists
Conference on
RANKINE
SPACE
POWER
SYSTEMS**

Volume 1

held at
NASA Lewis Research Center, Cleveland, Ohio
October 26-28, 1965

Division of
Technical
Information

DISCLAIMER

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

DISCLAIMER

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

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Commission, nor any person acting on behalf of the Commission:

A. Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights; or

B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

As used in the above, "person acting on behalf of the Commission" includes any employee or contractor of the Commission, or employee of such contractor, to the extent that such employee or contractor of the Commission, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment or contract with the Commission, or his employment with such contractor.

This report has been reproduced directly from the best available copy.

Printed in USA. Price \$8.00. Available from the Clearinghouse for Federal Scientific and Technical Information, National Bureau of Standards, U. S. Department of Commerce, Springfield, Virginia 22151.

COPY PRICES

H.C. \$ 8.00; MN 3.00

CONF - 651026
REACTOR TECHNOLOGY
(TID - 4500)

AIAA Specialists Conference on RANKINE SPACE POWER SYSTEMS

Volume 1

held at

NASA Lewis Research Center, Cleveland, Ohio
October 26-28, 1965

**American Institute
of
Aeronautics and Astronautics**

ATOMS INTERNATIONAL
A Division of North American Aviation, Inc.
Cannon Park, California

BLANK

INTRODUCTION

This volume contains those papers of the AIAA Specialists Conference on Rankine Space Power Systems which are unclassified. A companion volume has also been published of the classified papers.

The conference was held at the NASA Lewis Research Center on October 26, 27 and 28, 1965. It consisted of six sessions on the following subjects:

1. Rankine Cycle Power Systems

Chairman: E. B. Baumeister, Atomics International, Division of North American Aviation, Inc., Canoga Park, California.

2. Reactors and Shields

Chairman: D. J. Cockeram, Director, Reactor Development Department, Atomics International, Division of North American Aviation, Inc., Canoga Park, California.

3. Liquid Metal Pumps

Chairman: A. G. Grindell, Oak Ridge National Laboratory, Oak Ridge, Tennessee.

4. Boilers

Chairman: P. J. Berenson, AiResearch Manufacturing Company, Los Angeles, California.

5. Turboalternator

Chairman: D. L. Southam, TRW Inc., Cleveland, Ohio.

6. Radiators and Condensers

Chairman: S. Lieblein, NASA Lewis Research Center, Cleveland, Ohio.

BLANK

TABLE OF CONTENTS

Volume I (Unclassified Papers)

Rankine Cycle Power Systems

Mercury Rankine Program Development Status and Multiple System Application	3
R. L. Wallerstedt and D. B. Miller	
Mercury Rankine System Test Experience	52
F. C. Wimberly, F. W. Poucher and P. M. Gresho	
SNAP 8 Development Status	103
R. Gordon and H. O. Slone	

Reactors and Shields

Aerospace Nuclear Safety	141
V. E. Blake, Jr.	
ORNL SNAP Shielding Program	153
C. E. Clifford and Joseph Lewin	

Liquid Metal Pumps

Development of SNAP 8 Pump for Mercury Service	171
E. S. Chalpin, J. R. Pope and C. L. Foss	
Design and Development of Pumps for High Temperature NaK Service	186
E. S. Chalpin, C. P. Colker and C. L. Foss	
Selecting Electromagnetic Pumps for Rankine Cycle Space Power Systems	201
R. G. Rhudy and J. P. Verkamp	
Development of the MRP Thermoelectric NaK Pump	222
Milton A. Perlow, Harry H. Hanlin and Jack M. Hochman	
Experience with Turbine-Driven Potassium Boiler Feed Pumps at ORNL	258
L. V. Wilson, H. C. Young and A. M. Smith	

Boilers

Design and Development of the Mercury Boiler for the SNAP 8 System	285
A. H. Kreeger, A. J. Sellers and J. N. Hodgson	
Effects of Oil Contamination on Heat Transfer Performance of Forced Convection Mercury Boilers	307
Robert S. Carey, Bruce Farwell and George Thur	
Multiple-Tube Potassium Boiler Development	327
J. A. Boppart, K. O. Parker and P. J. Berenson	
High Performance Once-through Boiling of Potassium	357
J. R. Peterson	

Turboalternator

Organic Working Fluid Cycle: Development Status	391
Donald Malohn	
MRPC (SNAP 2) Turboalternator	415
R. C. Biering and Otto Decker	
Design and Development of a Turbine Alternator Assembly for the SNAP 8 System	477
M. G. Cherry and G. Oiyé	
Potassium Turbine-Generator Development	498
J. H. Dannan and L. E. Chadbourne	
Performance Test of a Two-Stage, 200-Horsepower Turbine in Wet Potassium Vapor	521
R. J. Rossbach, E. Schnetzer, H. E. Nichols and S. E. Eckard	

Radiators and Condensers

An Experimental Study of the Condensing Characteristics of Mercury Vapor Flowing in Single Tubes	553
D. Namkoong and J. Albers	
Mercury Rankine Radiator Condenser Development Experience	582
V. H. Heiskala, R. C. Smith, E. A. Elliott and R. T. Lancet	

Design and Development of the SNAP 8 Multi-Tube Condenser Alan H. Kreeger and J. N. Hodgson	641
Thermal and Fluid Dynamic Performance of Potassium During Condensing Inside a Tube S. C. Sawochka	669
Potassium Condensing Tests of Horizontal Multi-Tube Convective and Radiative Condensers Operating at Vapor Temperatures of 1250° to 1500°F O. A. Gutierrez, N. J. Sekas, L. W. Acker and D. B. Fenn	694
Design and Development Tests of Direct-Condensing Potassium Radiators A. P. Fraas	716
Results of Hypervelocity Impacts into Space Radiator Materials Nestor Clough and J. H. Diedrich	737

RANKINE CYCLE POWER SYSTEMS

Chairman: E. B. Baumeister

Atomics International
Division of North American Aviation, Inc.
Canoga Park, California

BLANK

MERCURY RANKINE PROGRAM DEVELOPMENT STATUS
AND MULTIPLE SYSTEM APPLICATION

by

R. L. Wallerstedt and D. B. Miller
Atomics International,
a Division of North American Aviation, Inc.

ABSTRACT

The Mercury Rankine (MR) space power conversion concept has arrived at the threshold of utilization as a working spacecraft subsystem. Component and system feasibility has been verified and the program is now in a reliability improvement phase. The rotating machinery and other prime components have demonstrated greater than 3000 hr life with a relatively high degree of reliability. Development emphasis has now been shifted to verifying the expected life to 10,000 hr. System characteristics such as design point performance, startup, thermal interactions, structural responses, etc. have been analyzed and verified in an comprehensive system test program.

System design for applications has developed the multiple power conversion concept to increase the electrical output and reliability from 3 to 40 kwe. Multiple 3 to 5 kwe power conversion subsystems are combined to provide any power level or degree of redundancy in that range. Potential missions have been studied and the resulting criteria introduced into the requirements as applicable to a general development program.

I. INTRODUCTION

The present AEC Mercury Rankine Program (MRP) has evolved from the SNAP 2 Flight Test Program which was initiated in 1957. Originally planned as a 3 kwe flight test with a 1 yr life, space mission planning has been so ambitious as to require extensive broadening of the power, life, and reliability capabilities. Feasibility of the MR system concept has been established to the point where it is a logical candidate for missions requiring a significant electrical power level. The MR system now envisioned can provide a 3 to 40 kwe capability at reliability levels required by manned vehicles in extended life missions.

The prime objective of the MRP is to develop the components and system technology (exclusive of the reactor or isotope heat source) for a nuclear, turboelectric space powerplant capable of covering the power range of 3 to 25 kwe with current hardware. The immediate objective is to develop and prequalify initially for 2500 hr and later for 10,000 hr, the components for a basic 3 to 5 kwe power conversion subsystem (PCS)

module. These modular components can then be utilized to build up any specific space power application in the 3 to 25 kwe range by using multiple PCS modules. In addition, a power capability up to 40 kwe can be provided by utilizing the same multiple PCS components; with the only modification required being the development of a NaK pump with a higher pressure-flow rating.

The MR powerplant has inherent advantages of high efficiencies and low radiator area that yields a low weight-volume system package requiring a minimum of thermal power. Component development have demonstrated the design performance objectives can be met and endurance testing is now underway. A life capability of at least 10,000 hr can be achieved based on our present experience, and operational periods of 2 to 3 yr are potentially possible with further development. Three separate system test installations have verified system performance, orbital startup capabilities and structural integrity. Table 1 summarizes the operational experience on a few critical areas that the program thus far has achieved.

The MRP today stands at a crossroads since no specific application has as yet been defined for a reactor type MRP system. In the range of power requirements of 15 kwe and greater, the MRP reactor multiple PCS system has significantly better weight, radiator area and reactor power advantages as compared to other power conversion concepts. With very little relative performance decrease, redundant PCS's can be added to achieve higher reliability.

Recent government agency funded isotope application studies have shown the desirability of the MRP multiple PCS approach. In particular, the advanced Apollo and Mars studies have indicated that the MRP multiple PCS isotope system would be very competitive with respect to other powerplant systems. For this application, the basic advantages of the MRP system are related to the relatively high cycle efficiency and the low radiator area requirements. With respect to the 2500 hr Apollo extended systems (AES) life requirement, the MRP has already demonstrated this endurance capability with three CRU V machine tests. Minimum changes in the MRP development objectives are needed to be compatible with isotope system applications.

II. PROGRAM SUMMARY

Based on the more than 14,000 hr of testing thus far achieved, and in light of the recent comprehensive design review analysis, the prototype design CRU V turbomachinery is found to be a sound machine with high intrinsic reliability. It is easily adaptable to design improvements as they become desirable. Certain marginal defects which initially decreased its short-term operational reliability have been, for the most part, already

eliminated. There remains today one unresolved known deficiency (journal bearing erosion-cavitation), which possibly could make the machine marginal for an endurance life of >3000 to 5000 hr. However, there are known corrective design measures that can be incorporated to significantly increase the bearing life if required. A developmental CRU IVM machine, has previously operated for 7000 hr with no major damage noted. CRU V testing has recently been successfully resumed after a comprehensive four-month design review. In the more than 3000 hr of testing accumulated on 2 machines since the review, no machine failures have occurred. Supplemental CRU V evaluation-demonstration testing is planned for FY 1965 with emphasis on testing conditions that can simulate the actual expected long-life orbital environments.

The status of the associated MRP components is as follows; the present Model 5C boiler, based on some 20,000 hr of test experience, is structurally acceptable and its thermal performance appears to be adequate for long life operation if (1) the air-in-leakage is kept low, (2) contaminants (such as oil traces) are kept out by proper pre-operational processing, and (3) the Hg loop is not subjected to shutdowns and repairs. The conditions affecting boiler conditioning are under investigation but as yet the fundamental technological phenomena are not fully understood. Various boiler conditioning studies are now in progress aiming primarily to establish better wetting and improved 2-phase boiling flow conditions. The thermoelectric NaK pump program has accumulated over 17,000 total test hours and its performance has surpassed the initial design objectives. The copper-constantan bond quality needs further improvement and the throat duct needs to be strengthened to improve its thermal cycle capabilities. The start components (pressure regulator and injection tanks) are in an early stage of development with the only remaining problem being reliable fabrication processes of the metal bellows. Both the honeycomb and aluminum fin radiator-condensers (RC) (7 RC assemblies have been fabricated to date) have demonstrated good performance and structural characteristics. The RC program has been recently oriented toward development of a basic aluminum finned RC tube element that is applicable to most any configurational constraint foreseen. This RC concept appears to be adequate, thermally and structurally; however, more structural development testing is required to fully verify the detail characteristics of the tube fin element. Also the aluminum fin to Haynes 25 (H-25) tube bonding process (titanium braze or diffusion bonding) needs further development to achieve the near 100% bond quality desired. The other associated components (expansion compensator, valves, parasitic control, etc.) have indicated no fundamental problems and are progressing slowly commensurate with the funding level available.

With respect to the operation of integrated systems, the orbital startup transient characteristics have already been reasonably simulated and verified with no apparent fundamental problem areas. Automatic "push button" startup has been recently

accomplished with a non-nuclear system installation. Stability of operation over a wide range of conditions has been verified. Long life integrated system endurance operation beyond 500 hr has not yet been accomplished. The remaining system endurance characteristics still in question are the long life associated performance degradation, the control interactions, Hg corrosion deposition, metallurgical processing effectiveness, the simulated vacuum effects, and non-condensibles and gas entrainment. There is now little question about the components operating together as designed. Today's difficulty lies with accurately predicting the more complex endurance interactions. A repetitive shut-down and restart capability is believed to be required for most manned system applications that have been studied, either reactor or isotope. A concentrated effort is accordingly underway to develop a basic system with a reliable start and restart capability. Multiple PCS installations should next be fabricated in order to verify the integrated operational characteristics. Specifically, the NaK loop and the electrical power distribution interactions for multiple PCS installations must be investigated thoroughly.

A system schematic is shown in Figure 1. In the power cycle of the basic MR system, the compact SNAP reactor is used to heat a NaK-78 fluid in a sealed recirculating loop. The hot NaK is circulated through a boiler to evaporate and superheat Hg. The Hg vapor is then expanded within the turbine which drives a permanent magnet alternator and a Hg boiler feed pump. These 3 components are all mounted on one rotating shaft and are supported by Hg lubricated bearings; the shaft rotating at 36,000 rpm. This assembly is called the CRU. Hg is then condensed and subcooled by a RC and is pumped back to the boiler by means of a high pressure boiler feed pump. The alternator generates alternating current electrical power at 120 v, 1800 cps, 3 to 5 kwe (maximum). The NaK primary loop working fluid is pumped by means of a thermoelectric (TE) pump (a direct current conduction pump whose electrical energy is supplied by thermocouple elements). The hot NaK provides the thermocouple hot junction and the cold Hg provides the thermocouple cold junction.

The isotope MR system would operate similarly. The heat source would consist of isotopic fuel capsules radiating to a surrounding forced convection Hg boiler-superheater assembly. The boiling temperature would probably be somewhat higher than the reactor system in order to achieve a greater overall cycle efficiency. Most probably, a supplementary NaK coolant loop would be utilized to reject the isotope heat source decay thermal energy.

The pertinent features of the MR cycle include:

- 1) a high heat rejection temperature, meaning low radiator area and weight,
- 2) a completely sealed system with no operating seals to space or between fluids,
- 3) extreme simplicity with a minimum of operating auxiliaries,

TABLE 1
MERCURY RANKINE PROGRAM SINGLE SHAFT CRU CONCEPT EXPERIENCE
(Hg LUBE BEARINGS-HERMETICALLY SEALED)

	STARTUP CYCLES	ACCUMULATED HOURS
PERFORMANCE CAPABILITY Hg LUBE BEARINGS TEST OPERATION		>110,000
OPERATION OF INTEGRATED CRU MACHINE (22 MACHINES FAB.)		
MRP MACHINERY		>24,000
PROTOTYPE CRU V MACHINERY	> 250	>14,000
ONE CRU	>70	>6,500
TOTAL COMPONENT TEST HOURS (EXCLUSIVE OF CRU)		
NaK-Hg BOILER		>18,500
NaK PUMP		>19,000
RC		>3,350
LOAD CONTROL		>22,000
INTEGRATED SYSTEM OPERATION (NONNUCLEAR)		
PSM-1 FLIGHT CONFIGURATION	>25	>550
PSM-3 ORBITAL SIMULATION	>70	>650

FIGURE 1. PROCESS FLOW SCHEMATIC

4) a reasonably high turbine inlet temperature yielding good cycle efficiencies; however, the temperature is not high enough to impose serious reliability penalties,

5) a simple single shaft rotating machine concept yielding high reliability.

Advantages over other space power systems include:

1) long life (10,000 hr) with no maintenance in the space environment,

2) no orientation requirements,

3) insensitive to space radiation levels,

4) no deployment required,

5) extreme flexibility in power output and configuration,

6) low orbital drag, hence minimum propellant penalties,

7) insensitive to sun-shade variations, and no battery storage required,

8) growth potential.

As noted before, the existing equipment can be assembled in multiple modules to provide a wide power range capability. In addition, higher cycle efficiency can be achieved by operating at higher pressure ratio conditions. The major consideration to this date has been Hg corrosion. Refractory metal boilers and condensers can be developed within the next few years to provide this advantageous capability.

The following papers prepared for this Conference present some detailed discussions of the design and development of the program components and ground test systems. They will provide further insight into areas of specific interest generated by this overall program status report.

1) F. C. Wimberly, et al, "Mercury Rankine System Test Experience."

2) R. T. Lancet, et al, "Mercury Rankine Radiator Condenser Development Experience."

3) R. C. Biering, "MRPC (SNAP 2) Turboalternator."

4) M. A. Perlow, et al, "Development of the MRP Thermoelectric NaK Pump."

5) L. M. Maki and D. G. Mason, "Design and Operating Capabilities of SNAP Zirconium Hydride Reactors and Shields."

6) R. O. Williams and J. H. Roecker, "SNAP 8 Experimental Reactor Tests."

III. STATUS

A. COMPONENTS DEVELOPMENT

1. CRU

Due to its significance in the program the rotating machinery (CRU) has received the major development emphasis to this point. The CRU is being developed under subcontract to Atomics International (AI) by TRW, Cleveland, and consists of a turbine driven alternator and Hg pump on a single shaft in a sealed unit. The two models undergoing test

this past year were CRU IVm and CRU V. The latter is the flight prototype design that has evolved from the experience of five years of CRU testing.

The CRU V turboalternator, designed to the following criteria:

- 1) 3 to 5 kwe power output,
- 2) isothermal hermetically sealed housing,
- 3) Hg lubricated bearings,
- 4) permanent magnet alternator,
- 5) single shaft assembly,
- 6) vapor alternator cooling,

has been verified as a feasible, practical and reproducible concept capable of achieving high intrinsic reliability. Greater than 14,000 operating hours have been accumulated by CRU V machines in the three years since test initiation. No fundamental potential failure mode exists for machine life up to the 3000 to 5000 hr range and extrapolation to the 10,000 life objective requires no technological breakthrough. Machinery malfunctions, to date, have been identified as design deficiencies and have been or are in the process of being corrected. A recent four-month design review, undertaken jointly by TRW and AI, has shown the machine to have the performance, load and clearance margins necessary for achievement of a high intrinsic CRU reliability. Specific development areas needing further investigation are shaft dynamics, startup and restart capabilities, and journal bearing cavitation-erosion.

A total integrated test time of more than 14,000 hr has been accumulated on the CRU V since the conceptual design was initiated in November of 1961. During 2 yr of testing, 5 test units have operated for over 1000 hr and 3 test units have operated for more than 2500 hr. The longest run to date has been 3154 hr.

Other concept milestones verified during this period were

- 1) a machine high power capability of 5.6 kwe,
- 2) repetitive startup capability,
- 3) shock and vibration capability for the launch environment.

Concurrent with this period, the CRU IVM, an earlier generation design, was phasing out its developmental testing. This CRU design verified the feasibility of the concept and served as a basis for the CRU V design. Over 10,000 hr were logged on all machines with one CRU accumulating more than 6500 hr.

The CRU V design is shown in Figure 2.

The unit consists of a single shaft assembly operating at 36,000 rpm. The 2-stage axial flow turbine drives the 3-phase 6-pole permanent magnet alternator and a centrifugal Hg pump, and is cooled by the turbine exhaust vapor. The shaft is supported by two journal bearings and the double acting thrust bearing that are Hg lubricated. The enclosing housing is hermetically sealed. The power output is 3 to 5 kwe.

8-27-65C

7636-0920A

FIGURE 2. TURBO ALTERNATOR ASSEMBLY-CRU V

Nominal criteria used as a basis for design and testing has been as follows:

Net Power (kwe)	3.5
Speed (rpm)	36,000
Bearing Flow (lb/min)	20
Pump Inlet Pressure (psi)	6.0
Bearing Coolant Inlet Temperature (°F)	400
Turbine Inlet Temperature (°F)	1150
Turbine Inlet Pressure (psi)	115
Turbine Exhaust Pressure (psi)	9.0

Two turbine configurations with different aerodynamic designs have been utilized in CRU testing. One designed for optimum high power performance and one for the lower power range (see Figure 3). As they are essentially interchangeable the CRU can be tailored to a broad range of power applications. Testing has verified a machine turbine efficiency of about 40% for gross power outputs of 2 to 6 kwe.

The CRU V alternator has a wide operating range and substantial performance margins over its design objectives. Using 60 mfd/ $\emptyset$ circuit capacitance the alternator is capable of delivering 2 to 6 kwe over a wide power factor range at an efficiency above 85% and voltages between 100 and 115 v. Figure 4 is a performance map of the present component.

The alternator stator windings are sealed from the Hg vapor with a ceramic bore seal. As a backup protective measure, the stator is evacuated to space through a control orifice that limits the inventory loss to a negligible value. This assures that the liquid Hg that could leak into the stator through hairline cracks in the bore seal is not trapped in the windings.

The CRU Hg pump has performed generally satisfactorily in both CRU IVM and CRU V. However, improvements have been made to increase cavitation margins, increase flow capacity and raise the output head. The improvement in performance margins plus the range of operational conditions are shown in Figure 5. The bearing pressure limits are a result of a reduction in pump output head at high flowrates (i.e., power).

As a result of extensive component and CRU testing with a variety of bearing geometries the 3-pad bearing was selected as the reference design for the 2 journal bearings. This bearing represents the optimum tradeoff of parameters influencing performance, fabrication and endurance. This design represents a hybrid (hydrodynamic-hydrostatic) type bearing. Figure 6 shows a typical journal bearing. Over 50,000 bearing test hours in 17 CRU units have been accumulated on this configuration with only one primary bearing failure. Other bearing failures have been experienced but always as a secondary effect from another primary mode such as running interferences, turbine wheel fracture, etc. The 3-pad journal bearings have apparently large operating margins over the normal load conditions. The basic problem area remaining is that of long life (> 3000 hr)

9-17-65C

7636-01045

FIGURE 3. CRU TURBINE PERFORMANCE CHARACTERISTICS

8-31-65C

7636-01041A

FIGURE 4. CRU V ALTERNATOR PERFORMANCE MAP

FIGURE 5. CRU V Hg PUMP PERFORMANCE MARGINS

FIGURE 6. TYPICAL JOURNAL BEARING

cavitation-erosion damage and although no 3-pad failures have been attributed to this phenomenon it represents a potential long-life endurance limitation. An earlier attempt was made to decrease the cavitation-erosion problem. Three-sector journal bearings were utilized in three CRU V exploratory machine tests but the 3-sector bearing was determined to have marginal load carrying capability. Other solutions do exist to prevent cavitation-erosion and are being studied intensively. Bearing flow-pressure and temperature effects, surface finish, clearance ratios, dynamic loading and clearances will be adjusted to attain the required endurance.

A double acting hydrostatic thrust bearing is now utilized in CRU V. Initially a hydrodynamic bearing was used, and discarded because of the marginal load carrying capability during startup.

The significant accomplishment of the program in recent months has been the increased understanding of the CRU shaft-bearing dynamic and thermal interactions. Analog and digital computer studies using the wealth of test data have simulated the effects of conical whirl, resonance effects, shaft bending, bearing damping and various shaft loadings. Thermal maps of the CRU have been completed for steady state and startup conditions. These skills will be invaluable for future improvement programs.

The CRU requires a certain number of auxiliaries for a flight system or ground tests. Flowmeters, filters, check valves, flow regulators or cavitating venturis, and speed control equipment are required to operate the machine. These auxiliaries have been developed concurrently and are in an advanced state of development. A typical module schematic is shown in Figure 7. The following Figure 8 shows actual CRU and module hardware.

In summary, the developmental progress of CRU program continues to increase in successful testing experience and acquisition of the fundamental analytical tools. Current machines under test demonstrate a relatively high degree of reliability. Failures, rebuilds, redesigns, component testing, etc. in the early development models have yielded the criteria necessary to reach this status. Figure 9 shows the steady increase including the setbacks of machine endurance through the feasibility and development phase, a design review and finalization period and now the current demonstration phase. The short 3-sector bearing investigative phase is also included to round out the test history.

2. Boiler

The current MR boiler has been developed within the present program objectives which utilize a reactor as a system thermal source. The component development has been a successful one with the exception of one remaining deconditioning problem area. As the applications for the MR system may very well be an isotope powered system using a radiation boiler the design and technology of an isotope boiler-superheater is directly extrapable from the current experience.

8-31-65

7636-01028 A

FIGURE 7. CRU V PCS MODULE SCHEMATIC

FIGURE 8. TYPICAL CRU V MODULE

The boiler is a NaK to Hg single pass, counterflow heat exchanger. Hot NaK flows through an annular chamber counter to the Hg flowing in tubes coiled in the chamber. The Hg is preheated, boiled and superheated during its passage. The Model 5C boiler represents the reference design now in use and is illustrated in Figure 10. The boiler consists of inner and outer shells of 316 stainless and coiled H-25 boiler tubes. Inserts and swirl wire devices are employed to enhance heat transfer and boiling stability.

The tube coil reverses the helix at the midpoint to balance the angular momentum forces of the flowing Hg. The boiler is 76 in. in length and has an O.D. of 4 in. The inner can has an I.D. of 3.5 in. Four 0.335 I.D. Hg tubes 29 ft length are coiled in the annulus. The boiler has a dry weight of 30 lb total.

The Model 5C boiler has been performance tested for more than 14,000 hr in both component and system test rigs. The steady state performance parameters have been verified and are summarized on Table 2.

TABLE 2
MODEL 5C BOILER NOMINAL PERFORMANCE
CHARACTERISTICS

NaK Flow (lb/min)	69
NaK Inlet Temperature (°F)	1200
NaK Outlet Temperature (°F)	1026
NaK Pressure Drop (psi)	0.24
Mercury Flow (lb/min)	20
Mercury Outlet Temperature (°F)	1175
Mercury Inlet Temperature (°F)	750
Mercury Outlet Pressure (psi)	120
Mercury Inlet Pressure (psi)	210
Quality	95%

The 95 to 100% vapor quality factor is based on current test experience with a conditioned boiler. Design criteria provides about 250°F of superheat but actual boiler tests have indicated less than 100% vapor conditions even with superheating. The vapor quality has not been directly measured but obtained from system heat balances or CRU performance. It is expected that some minimum quality specification will be re-defined in the future based on mission lifetime and moisture droplet sizes. A summary of the boiler testing is also shown in Figure 10.

Successful testing notwithstanding a major problem area remains with inconsistent behaviour of some of the boilers. A phenomenon described as "deconditioning" has occasionally degraded performance of these boilers, initiating a vigorous study and test

9-23-65C

7636-01104

FIGURE 9. CRU V TEST PROGRESSION

UNIT	TEST TIME (hr)	APPLICATION	REMARKS
MODEL 5C-1	20	COMPONENT TEST	
-2	1000	COMPONENT TEST	
-3	-	SYSTEM TEST - STRUCTURE	
-4	1300	SYSTEM TEST	SEVERE DECONDITIONING (~80% qual)
-5	1000	COMPONENT TEST	INITIALLY PARTIALLY DECONDITIONED
-6	350	SYSTEM TEST	INITIALLY PARTIALLY DECONDITIONED
-7	6100	PCS TEST	RECENT MARGINAL PERFORMANCE
-8	1500	PCS TEST	DECONDITIONING NOTED
-9	2200	COMPONENT TEST	90-day PREQUALIFICATION
-10	0	SPARE	
-11	>400	SYSTEM TEST	PERFORMANCE SATISFACTORY
MODEL 7-1	415	COMPONENT TEST	RECONDITIONED BY H ₂ TREATMENT
-2	225	COMPONENT TEST	DECONDITIONING NOTED.

10-18-65C

7636-01178

FIGURE 10. MRP COMPACT BOILER EXPERIENCE

effort to find a solution. "Boiler deconditioning" is a low quality heat transfer boiling characteristic that results in significant amounts of liquid Hg (up to 5 to 20 wt.%) being carried over by the Hg vapor into the CRU turbine. The temperature profile of a deconditioned boiler is shown in Figure 11. The total energy available in the Hg is thereby less (for a constant Hg flowrate) and the turbine power output is significantly reduced (approximately 1.2% reduction in turbine efficiency for each 1% reduction in vapor quality). Also, this entrained liquid can cause accelerated CRU turbine degradation through two possible mechanisms. First, corrosion products, generated in the boiler, can be entrained in the liquid Hg droplets and deposited on the turbine blades and nozzles during the vapor expansion process. Secondly, the liquid droplets, of unknown size, distribution and frequency, through impact and erosion processes can significantly alter the critical leading edges of the turbine blades. These mechanisms also cause degradation of the turbine efficiency and power output. Thus, liquid carryover from the boiler into the turbine is detrimental and an understanding of the fundamental causes and their elimination is essential.

Several hypotheses have been advanced that attempt to explain the observed phenomena and to provide guidance to the development activities. These hypotheses include:

- 1) Inadequate Wetting Hypotheses. Inadequate wetting at the boiling heat transfer surface by the liquid Hg would result in film boiling in lieu of nucleate boiling. The greatly reduced heat transfer rate of film boiling would make the present boiler area insufficient to transfer the heat necessary to produce high quality, superheated vapor.
- 2) Superheated Liquid Slug Flow Hypotheses. This hypotheses assumes that liquid Hg can be superheated due to lack of nucleation sites at either the tube wall-liquid interface or within the liquid. The liquid is superheated above its saturation temperature and at some critical value (possibly the point of maximum liquid superheat), nucleation occurs and flash vaporization at the boiling surface-liquid interface occurs. Since the point of maximum liquid superheat can occur a considerable distance behind the liquid-vapor interface, large slugs of saturated liquid are trapped between vapor slugs. This process is repeatable, thus resulting in a cyclic slug flow of liquid in the boiler tubes.
- 3) Multiple Tube Instabilities. The deconditioning phenomena may be exaggerated by the multiple tube boiler concept currently utilized by the MRP. The multiple tube boiler design requires that the pressure drop across each boiler tube be equal. If the heat transfer coefficient of each tube is equal, then the fluid flowrate and the exit quality of each tube are also equal. However, it is quite easy to postulate conditions whereby the heat transfer performance of one tube could be different than the other three and to satisfy the ΔP requirements, the Hg flowrate through the tube would be automatically altered. Thus, one tube, if the flowrate were high enough and/or the heat transfer characteristics significantly impaired, low quality vapor could be provided to the boiler plenum and subsequently to the turbine.

1-14-65

7569-C1898

FIGURE 11. TEMPERATURE PROFILE OF A DECONDITIONED BOILER

Efforts to combat any or all of these hypotheses include new cleaning and pickling processes, pool boiler experiments to study oil and air contaminated Hg and surface conditions, and single tube boilers to study flow characteristics and the behaviour of non-condensable gases.

Preliminary results of these experiments suggest the following tentative conclusions:

- 1) A vapor velocity (in the low quality region of the boiler) which is high enough to induce swirl flow is necessary but not a sufficient condition for conditioned operation; another requirement being adequate wetting.
- 2) In general, the loss of wetting and/or non-wetting decondition theory appears to be reinforced by the experimental data gathered to-date. However, additional tests are required to more completely understand the basic mechanism.
- 3) The effect of the parallel tube design may aggravate the deconditioning problem and must be more fully explained.
- 4) The Model 5C boiler appears to be adequate for current test system requirements if a leak tight, sealed system is employed.
- 5) Corrosion and/or corrosion product deposition (i.e., variable surface chemistry) and their subsequent effect on wetting, rate of absorption of contaminants and the basic boiling process kinetics may be important to long time boiler deconditioning.

With the solution of this problem the next step to a prototype flight boiler will be a straightforward engineering program. The changes required are a more compact configuration, improved design for assembly and a more reliable NaK-Hg interface. Of the three the NaK-Hg interface will present the greatest potential change. As the interface reliability constitutes a major factor in the overall multiple PCS system specific attention must be devoted to insuring a highly reliable NaK-Hg containment within the boiler.

The isotope boiler employs essentially a constant flux radiant heat source in place of the NaK. Basic Hg flow and heat transfer characteristics should be quite similar particularly with respect to the deconditioning factor.

3. Radiator-Condenser (RC)

RC technology has progressed to the point where a lightweight reliable component can be designed for a wide variety of application configurations. Three radiator concepts have been developed on the MRP. A copper fin brazed on a stainless steel honeycomb shell with the tubes imbedded in the honeycomb was the initial concept and the second approach was to braze or diffusion bond a H-25 tube to an aluminum fin tying the whole semi-monocoque structure together with steel frames. Initially there were certain questions of feasibility (bonding, structural capability, etc.) which have since been disproved and the aluminum finned tube concept is now the current reference design for the program.

The flat RC used in PSM-3 required a third design to be developed. A steel tube with copper fins on both sides was fabricated for zero gravity system testing. Fins radiated to both sides in order to reduce the volume of the vacuum containment vessel.

Three honeycomb, two aluminum steel RC's of a truncated cone geometry and two flat RC's have been fabricated and tested for performance, static and dynamic structural loading and aeroheating capabilities. Four of these components are shown in Figure 12. All units have successfully demonstrated the feasibility of design. Present efforts are concentrating on improving the aluminum fin tube bond by fabricating individual tube fin elements that can be joined at the ends with manifolds and configured to any geometry required. Again intermittent frames attached internally would provide the integral structure.

The current tube fin element under development is shown in Figure 13. The picture shows the electron beam welded tube, aluminum fin, frame clips and doublers brazed into a subassembly ready for integration into any configuration. Diffusion bonded elements are also under experimental investigation.

The evolution of the tube fin design is shown in Figure 14 for the developmental components and the ultimate design. Tapered tubes with thin walls can now be fabricated from H-25, 15 PH-7 and 405 stainless steel in a variety of sizes. Nominal limits of 12 ft in length and a minimum 0.125 I.D. appear to be adequate at this time.

Fin effectiveness has a limited significance when applied to a large RC of this sort. The heat flow is two-dimensional and results in a different temperature distribution from that of the one-dimensional fin effectiveness model. The RC model has a fin base temperature and fin length varying down the length of the tube fin element. However, the effectiveness can be helpful in comparing RC concepts of the same geometry. When computed at the midpoint the typical honeycomb RC has a value of 0.94 and the aluminum steel 0.75. These values of course can be changed by varying fin thickness or alloy composition.

A major advantage of the RC concept is that even when optimized for minimum weight its inherent structural abilities need only be supplemented slightly to provide an efficient spacecraft structural skin. Used as a nose cone or as the spacecraft shell, the RC can substitute its equal area in structure through the launch environments of acceleration, shock and vibration, aeroloading and aeroheating.

Figure 15 presents a qualitative summary of the launch environments of an Atlas-Agena boost which has been used as reference structural design criteria. Peak values of this trajectory present the most severe launch environment of any boost system today making the criteria applicable for Titan or Saturn. These criteria and combinations of same produce the structural test program required for flight qualification. After the vehicle passes through Mach 1, the product of dynamic pressure and angle of attack becomes

**RC-10 AL-STEEL
PROTOTYPE &
ANALYSIS
37 HOURS
- THERMAL PERFORMANCE
TESTING**

I-10-65

**AL-STEEL PANELS
RC-13
2160 HOURS
- AL-STEEL JOINT
THERMAL ENDURANCE TEST**

**HONEYCOMB
RC-6
PSM-1
1150 HOURS
- THERMAL PERFORMANCE
TESTING**

**HONEYCOMB
RC-11
PSM-2**

7636-0578 A

FIGURE 12. RADIATOR CONDENSER DEVELOPMENT

FIGURE 13. ALUMINUM STEEL TUBE-FIN ELEMENT

5a. R/C - 6 NONTAPERED ROUND TUBE

5b. R/C-10 TAPERED
TUBE CROSS-
SECTION

5c. R/C-11 RECTANGULAR
TAPERED TUBE

5d. R/C-13 RECTANGULAR
TAPERED TUBE

5e. R/C-14 RECTANGULAR
TAPERED TUBE

5f. ULTIMATE
DEVELOPMENT
OBJECTIVE

1-25-65

7569-01948

FIGURE 14. TUBE-FIN DESIGN

maximum resulting in the critical hoop and bending stresses in a nose cone geometry system. The end boost condition produces the maximum dynamic pressure that must be combined with the most severe aerodynamic heating skin temperature. Simulation of these combined conditions has been successfully accomplished in the RC test program to verify the design. Performance check, then structural test and finally a steady state performance and thermal cycle test has established the capability of the aluminum steel and honeycomb concept to endure the launch phase prior to orbital operation.

Application studies have utilized a shared fin concept where multiple and/or redundant PCS systems are used for reliability and greater than 5 kwe power outputs. The shared fin requires that each fin element has a separate condensing tube from each PCS and the fin is sized for the number of active elements. When a PCS fails, a standby redundant PCS is started up keeping the fin heat load equal. The approach requires more complex manifolding and closer tube packing but yields significant weight savings and reliability increases by the virtue of having a completely separate RC for each PCS.

Stability has been intensively studied to define the acceleration and vibration limits for the condensing operation. Vibration may create surface waves on the vapor-liquid interface causing the interface to bridge the tube or throw droplets back upstream. A vibration induced change in condensing length can produce an instability if at the new condensing length the rate of change of pressure drop with condensing length is negative. Negative accelerations can cause a change in condensing length and may cause runback type instabilities. If these runback instabilities become large enough, the downstream pressure fluctuations may cause a loss of pump prime. The present RC have designed these instabilities out or have increased their resistance to these phenomena to levels believed to be consistent with spacecraft orbital accelerations during maneuvering.

Improvements in the design efficiencies of other components are expected to result in an appreciable increase of electrical power output capability with only a small increase in Hg flowrate. This would increase the thermal power load and in the case of a higher system boiling temperature improvement suggest an increase in condensing temperature. It is felt that the present aluminum steel concept has a capability up to a maximum of 800°F condensing temperature allowing an approximate 250°F increase in boiling temperature for system growth.

In summary, it is felt that this technology development is equal to immediate application for a wide range of manned and unmanned space flights.

4. Start Components

Included as part of the PCS start subsystems are the necessary auxiliary Hg components such as the injection tank, pressure regulator, valves, and flow regulators.

The current PCS concept requires a pressure regulator tank that fixes the condensing pressure within a narrow range through a combined metal bellows-spring action. This assures a uniform condensing pressure during any inventory fluctuation, such as sun-shade transients by providing independent inventory control. The regulator is also used during startup to store a large transient excess Hg inventory. The current design is illustrated in Figure 16.

Pressurized Hg for startup is achieved through the use of a metal bellows injection tank. High pressure gas is expanded into the injection tank forcing the Hg into the bearings and into the boiler simultaneously. Orbital start and restart are provided by the equipment shown in Figure 17. After operation is achieved, the injection tank is recharged through the check valve shown in the schematic after the gas is bled off. The excess recharge Hg is bled off directly from the Hg pump.

Both the pressure regulator and injection tanks use the same metal bellows concept. Problems have been encountered in the welding and/or forming of leak tight bellow assemblies. As both the regulator and injection tanks are affected by the materials welding difficulties, efforts to solve the H-25 weld procedures or to substitute another material are being pursued. Full size welded H-25 bellows have been recently fabricated and thoroughly tested. AM-355, a material used successfully in the NaK expansion compensator bellows is being studied particularly with respect to its Hg corrosion resistance. Its properties are probably adequate for low temperatures ($< 350^{\circ}\text{F}$) but the desired use temperature (600°F) requires materials testing to verify its suitability.

The auxiliary components comprising the remainder of the start subsystem such as valves, flow regulation, piping, filters, crud removal, and etc. are being utilized throughout the component and system test program and their necessity in ground testing has kept their development status concurrent with the major components. This start subsystem concept can be considered as a feasible and even an attractive approach for the ultimate use. All of the component operating functions and requirements have been verified. Methods are available today to make it serve as a reliable subsystem. However, due to the significant relative weight of the Hg inventory and startup components a flight program would most certainly motivate an intensive weight optimization effort for these components.

5. Thermoelectric Pump

The TE pump as shown in Figure 18 converts thermal energy into electrical energy which, in turn is converted into potential fluid energy in terms of an increase in pressure. The pump removes thermal energy from the hot NaK leg and rejects thermal energy to the cold Hg stream or space. The pump preheats the cold junction Hg stream thus conserving rather than wasting the thermal energy. Chromel constantan TE elements are used because of their basic structural capabilities. The pump contains

7636-0115

FIGURE 15. TYPICAL TRAJECTORY LOADING

FIGURE 16. PRESSURE REGULATOR

no moving parts representing the ultimate in potential reliability. Two generations of pumps have been designed and tested for the program. Prototype I has been the developmental component to test feasibility. Prototype II is essentially the same pump with the addition of a beefed up NaK throat and NaK radiating fins.

The thicker NaK throat was added to increase the thermal cycle life of the pump. The fins have resulted from the use of redundant PCS's with a single heat source loop. Each PCS has a TE pump but the idle PCS pump does not have Hg flowing through the cold junction. This could allow reverse flow through the idle loop by the pumping action of the active TE pumps short circuiting a portion of the total NaK flow. This penalty is averted by attaching radiating fins to the cold junction providing a temperature differential sufficient to generate a small flow thus preventing the reverse flow. The fins also increase the steady state performance by about 10%.

Over 18,000 test hours have been accumulated on 14 pumps with one unit exceeding 6000 hr in endurance. The last two units were Prototype II models with fins.

Figure 19 shows the progressive increase in performance of the development phase units as testing and analysis provided improvements. With this data it is felt the TE pump has demonstrated its adequacy for use in space applications utilizing 3 to 5 kwe multiple PCS's. The pump does become limited at about 20 kwe because of the steadily increasing primary loop pressure drop.

6. Expansion Compensator and Miscellaneous NaK Components

The compensator is used to compensate for changes in NaK volume, maintain a void free system, and set the NaK system operating pressure. The component design is similar to the Hg regulator tank, using a spring-bellows combination to vary volume and pressure. Figure 20 illustrates the current hardware. This technology is based on the compensator used in the SNAP 10A flight system requiring only an increase in size to serve this system. Testing of welded AM-355 bellows and structure has verified the adequate high temperature strength, corrosion resistance and fabricability to be used here. Determining a size with the maximum applicability for a wide spectrum of missions remains the only major unknown.

The remaining NaK components are also shown on the figure and have been developed to further the ground test effort and in some cases will be required for flight.

The Reactor Simulator Heater is used in development testing to provide the system thermal steady state and transient power. The problems of remote testing a reactor powered system seriously constrain a system test when the emphasis is on the power conversion and heat rejection interaction characteristics. The electrical heater is

FIGURE 17. MULTIPLE HIGH PRESSURE GAS SUPPLIES

FIGURE 18. NaK THERMOELECTRIC PUMP TEST EXPERIENCE

PUMP NO.	ATMOSPHERE	INITIAL PERFORMANCE (gpm at psi)	PRESENT FLOW (gpm)	TOTAL ACCUM HOURS	APPROVED FLOW DEGRADATION (%)
6	AIR	11.5 at 0.99	9.7	4654	-15.7
8	AIR	11.8 at 1.03	10.8	5590	- 8.5
10	AIR	12.0 at 1.07	10.9	2960	- 9.2
11	VAC	12.5 at 1.17	12.3	1000	1.6
12	VAC	12.6 at 1.17	12.5	356	<1.0

9-8-64

7636-0483

FIGURE 19. MERCURY RANKINE PROGRAM THERMOELECTRIC PUMP (PROTOTYPE I) INITIAL PERFORMANCE

EXPANSION COMPENSATOR

EXPANSION JOINT

PARASITIC LOAD HEATER

REACTOR SIMULATOR HEATER

FIGURE 20. NaK COMPONENTS

computer controlled to simulate the reactor thermal behaviour. The internal geometry and pressure drop is identical to the SNAP 2 reactor. High heat flux density calrod heaters are used to provide the power.

The parasitic load heater dissipates the load control waste electrical power into the NaK stream. The concept uses a sheathed resistance heater coiled and brazed around the primary loop NaK pipe. Many hours of testing has been accumulated in the component program and greater than 7000 hr has been accumulated on an individual heater assembly. Two heaters are required in the flight system, one in the hot and one in the cold leg of the NaK loop. This method simply raises the average NaK temperature reducing the reactor power required.

Expansion joints have been developed to help realize the close coupled piping and equipment configurations shown in the system design sections. Formed bellows connect the piping and an internal sliding sleeve guides the piping geometry. The components have been fabricated and tested at 1200°F temperatures and introduce a pressure drop only 1-1/2 times the equivalent pipe length (approximately 9 in.). Piping loops to relieve the expansion stress are considered undesirable due to the added volume and weight and to the fact angular momentum forces are difficult to balance within the system geometry itself.

7. Reactor

Reactor development capability has progressed consistent with the MRP goals. The SNAP 2, 8 and 10A reactor programs can show a reasonably demonstrated capability of 50 to 1000 kw, a range completely covering the projected thermal power needs. The successful SNAP 10A flight test, extensive nuclear ground testing of the SNAP 2/10A reactor and the recent ground test endurance run of SNAP 8 has established the reactor as one of the more advanced component technologies and will most certainly not limit the power conversion system availability. This paper will not discuss the details of the reactor development, but it is mentioned here to establish its status.

B. SYSTEM DEVELOPMENT

A comprehensive ground test system program has been carried out concurrent with component development. The systems were designed to test component interaction and system integration. Three systems were built and tested for performance (PSM-1), structural (PSM-2), and orbital startup (PSM-3) verification.

1. PSM-1

The PSM-1 system was the first system to be fabricated to the configuration of a flight system. The system was complete except for the TE pump and the substitution of a reactor simulator. The test objectives were to provide (1) steady state system performance data in a vacuum, (2) component performance under system operation constraints, (3) initial test of the proposed startup concept, and (4) an off design performance map. Figure 21 shows the system installed in the test facility.

The system operated for 1100 hr of which 550 hr was with a CRU IVM. A successful system operation was demonstrated and the objectives met with the exception of a planned endurance run at the end of the program. Of particular note was that a Hg system could indeed be built to operate within the imposed configurational restraints.

The system provided a great amount of important design information. The problems of structurally integrating the system within defined space constraints were overcome. NaK and Hg pressure drop data was obtained. Certain questions relating to thermal interactions between components were answered. Flow stability over a wide range of conditions was also verified. Items other than performance criteria gained from the program were:

- 1) The boiler "deconditioning" problem was experienced in a system test for the first time.
- 2) System leak tightness was marginal and was probably contributing to the boiler problem.
- 3) The CRU displayed an amazing durability in operating 500 hr with very low quality vapor.
- 4) The available industrial Hg equipment (filters, valves, instrumentation, pump, etc.) was generally not adequate and needed specific development attention.

As discussed above the first system test had both successes and failures but was an important milestone in system design, fabrication and test experience.

2. PSM-2

The major objective of the PSM-2 test system was to subject the SNAP 2 flight test configuration to an Atlas-Agena launch environment. Static dynamic and acoustic loads from the established Atlas-Agena Payload Test Specification were used to verify the structural suitability of the design. Detail objectives were to establish system and component responses to shock and vibration, determine individual component loading, measure prime structure damping levels and measure the effects of aeroheating and pressure. The specific objectives were to determine the margins of the design so as to use them for further structural optimization. The system and some of test data is shown in Figure 22.

APPROXIMATELY 1000 HOURS (550 w/CRU) OF TEST OPERATION ACHIEVED
STABILITY OF COMPLETE INTEGRATED SYSTEM VERIFIED
PREDICTED ANALYTICAL THERMAL AND HYDRAULIC PERFORMANCE WAS VERIFIED
OPERATION OF A SPACE CONFIGURED SYSTEM WITHIN A VACUUM WAS DEMONSTRATED
ADEQUACY OF INJECTION STARTUP AND GROUND SHUTDOWN SHOWN
SYSTEM LEAKAGE REQUIRES IMPROVED DESIGN AND FABRICATION TECHNIQUES
BOILER CONDITIONING PROBLEM INDICATED
CRU RUGGEDNESS SHOWN THROUGH SEVERE OPERATIONAL CONDITIONS

FIGURE 21. MAJOR PSM-1 TEST RESULTS

2PSM2A CONFIGURATION

**AERODYNAMIC
PRESSURE TEST SETUP**

**LONGITUDINAL
VIBRATION SETUP**

**TEST OBJECTIVES: PROVE STRUCTURAL INTEGRITY
AT QUALIFICATION LEVELS AND
DETERMINE RESPONSE CHARACTERISTICS**

TEST & RESULTS:

LATERAL DECAY:

- a) DEFLECTION WITHIN 3 1/2 % OF PREDICTED
- b) EQUIVALENT VISCOUS DAMPING 3.3%
- c) APPROXIMATE FIRST MODE LATERAL FREQUENCY
18.7 cps; PREDICTED 19.2 cps

LATERAL SINUSOIDAL VIBRATION:

MEASURED RESONANT FREQUENCIES 13.8 & 58.0 cps
PREDICTED VALUES 19.2 AND 109.1 cps
(Fixture problems being investigated)

LONGITUDINAL SINUSOIDAL VIBRATION:

MEASURED FREQUENCIES 82.0 & 151.0 cps
PREDICTED VALUES 98.9 & 395.6 cps

CRU VIBRATION EXCITATION:

NO ADVERSE RESPONSE

MECHANICAL SHOCK:

NO ADVERSE RESPONSE

STATIC AERODYNAMIC PRESSURE:

HOOP STRESSES LOWER THAN EXPECTED; OTHER STRESS
LEVELS AS PREDICTED.

FIGURE 22. PSM-2 STRUCTURAL TEST SYSTEM

The system was tested for shock, vibration and pressure loadings successfully. Testing was accomplished in rigs especially designed for this utilization. A considerable amount of data was achieved over a wide range of expected conditions. Due to the cancellation of the flight test program, some of the latter simulated launch aeroheating tests were deleted.

There had been concern about the package configuration and mass distribution being a problem due to excessive response to the spacecraft dynamic input; i.e., applying qualification loadings that previously applied to short high density packages to a long flexible structure (RC) with a heavy mass cantilevered on the end (reactor and shield). These problems did not appear in the dynamic testing probably due to the large RC damping values. In fact little or no damage to the system or components was noted throughout the complete test series.

The test proved that the basic structural design was adequate for one of the most severe booster launch environments. Since this package design is still applicable for some missions the configuration can be adopted with confidence.

3. PSM-3

The PSM-3 is a zero gravity system to verify the orbital start and operational concept. The schematic of the arrangement is shown in Figure 23. A photograph of the installation is also shown on the figure. The flat RC is installed on the same centerline as the CRU to negate the gravity effect. The test program has been conducted in several phases, starting with a RC and ending up with a completely integrated system utilizing a CRU IVM unit. A third phase is planned to incorporate a CRU V into the system. Test operations were initiated early in CY 1963.

The present system has a completely automatic startup system simulating a reactor startup. Automatic "push button startup" has been accomplished repeatedly. The CRU presently in the system has over 6500 hr running time, the majority being accumulated in component testing. Two other machines have been in the system to total 620 hr of CRU running time in the test program. The nature of the startup testing is a stringent test of the machine durability. The system has been started up 86 times with 76 successful attempts. The installation is representative of a complete system except for the current use of a TE pump. It is expected that a TE pump will be installed late this year.

A large mass of data has been accumulated both at steady state design point and at off design conditions. In particular, operational stability has been verified for all intended modes of operation. Transient characteristics during both startup and shutdown are being thoroughly studied. No technological aspect has been as yet defined that would seriously affect actual space operations. In fact, the startup system has already been made less complex by certain equipment deletions.

SYSTEM STABILITY IN BOTH STARTUP AND STEADY-STATE OPERATION VERIFIED—WITHOUT THE AID OF GRAVITY

FLAT RC DEMONSTRATED STABILITY AND ADEQUATE FLOW DISTRIBUTION

STARTUP ANALYTICAL MODEL DEFINED

Hg STARTUP INVENTORY REQUIREMENT IS 10% LESS THAN PREDICTED

CRU AND RC COUPLED SATISFACTORILY

TE PUMP SIMULATOR PROVED FEASIBILITY OF OPERATION

APPROXIMATELY 700 HOURS OF TEST OPERATION AND 60 STARTUPS WITH AN INSTALLED CRU HAVE BEEN ACHIEVED THUS FAR

PSM-3 STARTUP TEST INSTALLATION

FIGURE 23. MAJOR PSM-3 TEST RESULTS
(AS OF JAN. 15, 1965)

The major objectives have been conclusively demonstrated. The startup concept has had a significant number of proof tests with a CRU in the loop, and the understanding of the behaviour of the startup and steady state condensation process under simulated zero gravity has been immeasurably improved as well as the behaviour of Hg inventory migration. Further efforts will be to investigate more fully the RC transients, the CRU drainage characteristics, and overall integrated stability factors. In particular, continued emphasis will be on achieving a higher reliability for startup. Special tests are also planned for verifying automatic restart and to investigate partial preheat considerations.

IV. SYSTEM DESIGN CONSIDERATIONS

System design and test experience presents the final proof of concept feasibility. Component testing is the logical initial phase of a development program but can only demonstrate a practical system is possible. When system integration is attempted new requirements such as component performance and thermal interactions, system reliability, configuration and structural restraints, operational procedures, and component tradeoffs introduce development problems at the least equal to the initial component feasibility period.

The unfortunate tendency of these requirements is to penalize the theoretical component performance if the system design is not both thorough and imaginative. Total component tolerances combine to degrade an individual components performance and endurance. Operational requirements such as startup and shutdown, safety, etc., add auxiliary systems reducing system reliability. Configuration and structural restraints can require less than ideal orientation or environments for the component. Compromises in one component's performance to benefit another can in many cases yield a better total performance. The scope of redundancy may actually decrease reliability if carried too far. These and many other problems must be analyzed, tested and solutions specified.

The MRP is well into this effort with a substantial amount of system design and three system test operations completed or underway. System design has been carried out in support of the component development, flight test program and a variety of application studies. The projected weight, area and efficiency estimates utilized herein take into account such considerations.

1) Multiple PCS Systems. System studies have developed the multiple PCS approach for greater power and reliability capability. The basic 3 to 5 kwe PCS is used in multiple building blocks to cover a power range of 3 to 40 kwe. Combinations of active and redundant modules can be assembled to provide a variety of discreet power levels or reliability within that range.

The PCS modules would be completely independent of each other with a common NaK loop in a reactor system and a combined source boiler for the isotope concept. Figure 24 shows the schematic of a multiple PCS reactor system which is also basically applicable to an isotope system.

The multiple PCS concept covers the projected power range for isotope systems completely. Highly reliable systems from 3 to 20 kwe are possible based on the hardware under development today. Redundant standby systems can be used to increase life as well as reliability. The present one year life objective could be extrapolated to two or three years by adding additional PCS's in the initial package.

The 3 to 5 kwe module is an optimum PCS power up to 25 kwe but can be used to about 40 kwe in a reactor system before the complexity becomes excessive. As the power levels grow, the NaK flowrate and pressure drop increases and these parameters must be varied to accommodate the available system pumping power. Reactor models under development cover this power range and are utilized as flowrate and pressure drop constraints require. At about 25 kwe, an advanced NaK pump is necessary to either replace or at least supplement the TE type pump.

2) Startup. Ground and orbital (simulated) startup feasibility has been demonstrated by comprehensive testing but still remains a significant factor in the overall system development planning. The startup subsystem concept constitutes a major reliability factor and a significant portion of the system weight.

The reactor start system is depicted in Figure 25. The primary considerations involved are:

- a) supplying adequate bearing flow,
- b) boiling-superheating the injected Hg flow,
- c) limiting severity of thermal shock to the turbine,
- d) providing adequate drainage for the bearings,
- e) priming the Hg pump,
- f) coupling the CRU alternator and parasitic load control with the power conditioning equipment,
- g) controlling reactor power and flow characteristics,
- h) establishing flow stability within the RC.

An isotope system would remove the reactor and TE pump and use the boiler as a source radiation boiler. The isotope system startup process would be basically very similar and probably simpler. The current start system inventory location history is also shown in the figure for a single 5 kwe PCS with the proposed restart capability. The injection tank inventory is shown depleting in 2-1/2 min and then being refilled for a restart capability within about 14 min. RC preheat is obtained by condensing and initially collecting the startup vapor inventory until at 600°F the 6 psia condensing

pressure is reached to prime the pump. The RC profile shows the injection boiler condensation filling the tubes until pump prime and then draining to the regulator tank. The pressure regulator receives the RC and bearing drain flows and then discharges the liquid to the pump to achieve steady state operation and refilling of the injection tank.

Startup testing has been a primary objective for the PSM-3 system and has highlighted the complex component interactions, condensing stability and inventory migration inherent in this process. Based on more than 80 starts, it has been demonstrated conclusively that the Hg injection concept selected can quite reliably accomplish the complete startup process.

The condensing, filling and draining stability of the RC has been consistently as predicted and the necessary bearing lubrication has always been adequate for a successful spinup. The capability for complete automatic "push button" system start has also been recently demonstrated.

Remaining problems include inventory reduction and development of reliable hardware. Inventory reduction will be for the most part a function of the specific hardware design used. The bearing lube supply requirements must be better defined to assess the inventory that must be available during spindown. The capability for automatic shutdown and subsequent restart are now under development. Continued emphasis on improving the reliability will be carried on. Specific advances have already been made in reducing the number and complexity of required startup equipment. Hardware processing will require bellows development as discussed in the component status section. A reliable controller, check valve and gas pressurization equipment will be required. The CRU must also be parametrically tested to ascertain if preheat is required and to test its ability to startup in a partially flooded condition.

3) Configuration Flexibility. The many vehicle configurations and mission plans under consideration for manned and unmanned, reactor and isotope fueled missions require a variety of power system configurations but must minimize the component changes from one application to the other. Application studies for reactor systems generally tend to stay with the truncated cone geometry identified with the earlier SNAP 2 flight test configurations. This geometry is the minimum shield weight for any application and is applicable for manned or unmanned missions.

A typical space configured system for a manned mission using the module approach is shown in Figure 26. The system uses a radiation shield split into two sections with a gallery between. The NaK boilers and TE pumps are mounted in the gallery as it is necessary to shield the radioactive NaK from the manned spacecraft. The first shield should be as short as possible for minimum shield weight but thick enough to prevent activation of the Hg that flow up to the boilers and TE pumps from the PCS area below.

The RC can be stacked as shown, or a more preferable method is the shared fin approach described earlier. The tube fin geometries are identical but the CRU's are located at the bottom of the RC and their vapor manifolds split the flow to each cone segment. All of the PCS components would be structurally mounted to the RC.

The electrical control and instrumentation equipment with the exception of the transducers and wiring would be in the manned environment or in the case of an unmanned application, at the bottom of the RC protected by a thermal barrier.

Extension booms and cables are utilized wherever possible to provide both separation and a minimum cone angle for lower shield weights. As the boom and electrical cable weights increase with separation distance and the shield decreases an optimum separation length must be defined. In addition, an extended mass can be a mission attitude control or maneuvering fuel penalty and must be included in the optimization if applicable.

Although 4π shields are felt not to be required for most manned missions, their use may be mandatory in a few cases. The large protected area for 4π or even 2π shields allows a large amount of flexibility in system configurations, and due to the extreme weight of the shield configuration penalties are insignificant.

Isotope systems also must provide a wide range of configuration flexibility. The lower source radiation levels allows an integral installation with a manned spacecraft for a minimum of shield weight. This approach then brings in other considerations such as volume constraints, thermal isolation and maintenance access not present in a separated system. In addition, control of the source through launch, steady state operation and re-entry or disposal becomes an overriding design parameter. A typical isotope system integration scheme is shown in Figure 26.

However, the flexible geometry capability of the RC, the low volume required for the PCS module packages, the availability of low volume/weight radiation insulations and minimum electrical and control requirements allow the MR concept to be configured for these applications with a minimum of performance penalty.

4) Stability. Stable operational performance has been demonstrated in the PSM development system test installations. No system instabilities have been observed with respect to the basic NaK loop and the PCS. Some oscillations have been seen in the alternator-parasitic load control and electrical power load circuitry. Also, interaction between the RC, pressure regulator and Hg pump have been noted at times. The above noted deficiencies were corrected but further study and investigation is being carried on. In general, the integrated operation of a MR system has been found to be quite stable.

The question of whether a stable condenser can be designed for zero gravity operation has had an intensive investigation during the development period. In the absence of the earth gravitation field it is necessary to rely on the liquid and vapor momentum and drag forces to maintain a stable flow path. The most stable configuration is a single tube

FIGURE 26. MULTIPLE PCS SYSTEM
UNSHIELDED SYSTEM CONFIGURATIONS FOR SEVERAL POWER LEVELS

FIGURE 27. MERCURY RANKINE-RDEPS CSM INTEGRATION

but this approach yields such a severe weight penalty in large RC's that parallel-tube condensing must be utilized. Tapered tubes offer significant advantages in stability due to the higher average vapor velocities and in addition reduce the startup inventory. In the six full scale RC's fabricated and tested, no major instabilities have been observed.

The major influences on stability have been studied analytically and in single tube and RC tests to define the operational limits of the concept. Interface zero gravity liquid leg, force field liquid leg and runback instabilities all must be understood in order to assure tube-to-tube stability, limit the vehicle maneuvering accelerations or to configure the RC to take advantage of the gravity forces. Single tube condensing tests have been operated using angles of inclination from -5° to $+10^\circ$ from the horizontal. Non-condensibles were injected to study the influence of gas bubbles at the interface and their nature of entrainment.

General conclusions are that a stable RC can certainly be built for space flight. Some considerations from the studies are as follows:

- a) the MR flat radiator test proves that the multiple tube RC is stable in a horizontal orientation,
 - b) the condensing length must be in the order of 20 times the effective tube diameter for zero gravity liquid leg stability,
 - c) the MRP design can tolerate approximately 0.16 negative gravity against the flow path,
 - d) a wetted tube surface may enhance runback instabilities and lower the 0.16 value.
- General areas of future experimental work that will be more thoroughly explored are:
- a) pressure drop and stability studies of the RC tapered tubes,
 - b) verification of zero gravity liquid leg stability analysis,
 - c) wetting versus non-wetting stability and pressure drop studies.
- 5) Hg Corrosion. Corrosion, erosion and product materials deposition represent potential failure modes for the system and solutions are being sought by two methods. The major area of concern is with the possible "crud" deposition in the Hg lubricated bearings and flow control devices and/or in the turbine nozzle. Over the last seven years, the basic materials Hg corrosion resistance properties have been tested and new material with improved properties continuously are being evaluated. The second approach is to develop components that can remove the corrosion products from the liquid and/or vapor. In this respect, a great percentage of the boiler generated crud is fortunately deposited within the superheater section where it causes no trouble.

Figure 28 shows the corrosion rates of the typical materials used today in the MR systems at the design temperature. The materials such as 15 PH-7 and H-25 show an early accelerated corrosion that decreases with time to a rate that could be acceptable in practical systems. A minimum one year life capability appears definitely attainable

based on both corrosion loop and actual CRU test experience. The second curve on the figure shows the effectiveness of various separator test loops for removal of the corrosion products. Centrifugal separators, columbium wool traps, filters, etc., are being evaluated in this type of testing for the brute force approach. Continual improvement in both the filtering and crud separation techniques should further improve the situation. An area of concern is associated with the deconditioning problem. The experience has been that, when liquid carryover is evident from the boiler (deconditioning), a significant increase in turbine nozzle crud deposition occurs.

Refractory materials such as columbium look very attractive for future systems where longer than one year life or higher operational temperature requirements are imposed. The refractory alloys are a Class I material and the corrosion problem would appear to be negligible with their use. Stainless steel 316 columbium duplex tubing is available today plus it is felt the availability of refractory materials in a wider range of shapes and workability will improve at a rate compatible with the program development schedule.

6) Reliability. The increase in reliability associated with multiple PCS approach has been the major advance in gaining the reliability levels required for manned missions. Of course, individual component improvement is still the major emphasis today and must be continued. However, the multiple system approach allows a realistic PCS reliability development objective to be selected with confidence in its ultimate attainment.

The effect of PCS redundancy on total system reliability versus individual PCS reliability is shown in Figure 29. The significance of the plot is the fact that practical system reliabilities of interest can be attained with individual PCS reliabilities of 0.8 and above with less than 100% redundancy. The PCS design objective of 0.96 is thought to be a realistic number making the 25% redundant system practical for future planning. Figure 30 illustrates the effect of weight for various levels of PCS redundancy necessary to achieve a 0.95 system reliability.

Another advantage gained by the multiple module design is the high partial power reliability inherent in the concept. Take home or standby power is vital for many missions to provide crew safety, emergency resupply, etc., making this feature a very desirable one. Figure 31 shows the partial power advantages of multiple PCS's for a fixed 20 kwe output.

7) Electrical Output. System electrical power output characteristics must be matched to the mission power mix and conditioning scheme. However, using a state-of-the-art alternator somewhat fixes the raw power criteria and in most cases requires the conditioning equipment to provide the flexibility in power type.

TEST SUMMARY

CORROSION PRODUCT REMOVAL SUMMARY

FIGURE 28. MERCURY CORROSION

FIGURE 29. EFFECT OF PCS REDUNDANCE ON TOTAL SYSTEM RELIABILITY 20 Kwe SYSTEM

FIGURE 30. SYSTEM WEIGHT AS A FUNCTION OF PCS RELIABILITY
(TO ACHIEVE GREATER THAN 0.95 SYSTEM RELIABILITY)

FIGURE 31. EFFECT OF PCS REDUNDANCE ON PARTIAL POWER RELIABILITY

Typical characteristics of a 3.5 net kwe CRU V include a line to line voltage of 115 v, power factor of 1.0 to 0.8 (lagging), 1800 cps frequency and a frequency regulation of $\pm 1\%$. The efficiency of the permanent magnet alternator versus power output is illustrated in Figure 32. The short circuit capacitors currently being used are 40 to 60 μ fd/0. Gross electrical power is 4.1 kwe before degradation and control allowances.

Power conversion methods studied heretofore have only considered conversion of 1800 cps alternating current to 28 v, direct current and 400 cps alternating current. Attempts to define the approximate characteristics have been completed and typical equipment specifications defined. The following criteria must be evaluated in the design of conversion equipment specifically for the permanent magnet and generally for other types of alternators.

- a) good voltage regulation must be provided in spite of a ± 10 to 15% variation from the power supply side,
- b) allowances for a high harmonic content power source,
- c) allow for interactions with the parasitic load control,
- d) considerations of the conversion equipment power factor into the power supply,
- e) provide adequate short circuit and overload protection,
- f) provision for mixing outputs from several converters on a common bus and in addition the ability to balance electrical loads and reactive loads,
- g) provide efficient regulation in voltage and frequency with a good wave form,
- h) allow for a long transmission line where extension methods are employed,
- i) provide battery charging methods where direct current is used,
- j) must be efficient, lightweight, minimize radio interference, and be radiation tolerant in some configurations.

It is felt that direct paralleling of the 3 to 5 kwe permanent magnet alternators into a single 1800 cps system is not practical at this time. Such a system would require synchronization, load, and reactive power sharing provisions. Also, permanent magnet machines operating in parallel with capacitor outputs are probably inherently unstable. Auxiliaries such as Hg throttle valves, voltage control, and synchronization equipment could help solve the problem but would decrease reliability by increasing system complexity.

Brushless electromagnetically excited type alternators proposed for advanced PCS concepts do not have this disadvantage and in theory could be used in parallel designs.

Figure 33 shows possible output conversion schemes for 400 cps and 28 v direct current outputs. Mixtures of these types and 1800 cps in a single system could be accomplished by the techniques illustrated. Where 400 cps is generated directly, the alternating current conversion is eliminated. A static 400 cps converter can be constructed by means of a cycle converter frequency changer technique using silicon

FIGURE 32. ALTERNATOR OUTPUT CHARACTERISTICS

A. ALL dc SYSTEMS

B. ALL 400 cps SYSTEMS

FIGURE 33. ELECTRICAL POWER OUTPUT DISTRIBUTION SCHEMATIC

controlled rectifiers for power switches. The direct current system employs a transformer-rectifier-filter system using silicon controlled rectifiers to attain voltage and load sharing control.

In conclusion, it is felt that adequate flexibility exists to provide a number of power output types for multiple PCS systems and as specific requirements become available, the power output could be tailored to the needs.

8) Structures. The advent of high temperature nuclear power systems in space has introduced a new condition into the designers normal launch environment criteria. In addition to surviving the shock, vibration and accelerations of launch the system must operate at high temperatures for a long mission life. When launched cold the system structure must then absorb the thermal expansion of a 70 to 1000°F temperature differential and in many cases the designs preclude ideal structures for the launch condition.

As discussed in the component section the RC is the primary structural component where in one case it is the overall geometry with components tied to it and in the other serves as the spacecraft structural skin. As both a structure and high temperature component detail design becomes complicated with thermal isolation and expansion connections at its periphery. Using the tubes as a primary load path requires care so as not to jeopardize the tube fin braze or manifold connections with launch loads.

Thermal isolation of the high temperature components from low temperature gear such as electronics requires new design concepts, in particular if the connection is a significant load path. Materials such as fiberglass must serve as structure and insulation where heat flows are to be low. Complex geometries can also be used to provide high conduction resistance paths for the thermal leakage.

System leak tightness is also of primary concern to system performance. The in-leakage of the atmospheres during the ground test period can infuse the system with non-condensibles and contaminates. This can contribute to boiler deconditioning RC inefficiencies, or to depriving the pump and must be avoided at all costs. Both the initial component structural design and the fabrication procedures must prevent thermal stress or structural loads that could lead to small cracks or holes during the ground test period. Of course, it is obvious that catastrophic failures must be avoided.

With this general system experience being fed into the component development it is felt that the component performance is more meaningful in that all of the system requirements are being tested under representative conditions expected in the near future.

V. APPLICATION CONSIDERATIONS

The wide range of potential applications for a MR power system has introduced many new requirements and changed the scope of existing specifications. Realization that the program must be continually cognizant of the final use requirements has required extensive study of future mission plans and equipment to guide the current planning.

Manned and unmanned missions using a reactor or isotope thermal source have generated a bewildering array of unique requirement dependent on whether the mission is weight, reliability, volume or cost oriented. Impossible as it is to optimally satisfy all of these variables, flexibility in system power output, configuration, reliability, and operational procedures have been established as always being general requirements. Unique tradeoffs such as radiator area versus efficiency or thermal power, shield weight versus extension mechanism or spacecraft attitude control weight, maintenance versus redundancy, etc. must be evaluated within each mission's constraints, but the ability to vary these system characteristics is a general requirement.

This has motivated an examination of all the component and system development objectives in an effort to build this flexibility into the MR concept within the constraints of the budget and aims of a general technology effort. The result of this examination reveals the basic system now under development has the inherent capability to meet these needs with a minimum of change or added complexities.

With these studies has come a general picture of what the weight, performance and configuration characteristics would be in a space mission. Assuming normal operational and vehicle constraints, system weights for a wide power range can be specified for mission planning. Shielding is the exception to this as the weight is strictly a function of the unique mission configuration.

Figure 34 shows the reactor system weights to 40 kwe and shows the discreet jumps in weight as multiple PCS's are added and a larger reactor is required. The component weights that comprise these totals are based on an extrapolation to the ultimate program performance and reliability objectives. In most cases existing hardware has been used as a basis for the extrapolation and the weight range can be used with confidence for preliminary mission planning.

10-22-65

7636-01196

FIGURE 34. REACTOR MULTIPLE PCS WEIGHT SUMMARY

MERCURY RANKINE PROGRAM SYSTEM TEST EXPERIENCE

P. M. GRESHO, Senior Research Engineer
F. W. POUCHER, Dynamic Systems Development Group Leader
F. C. WIMBERLY*, Systems Development Project Engineer
Space Systems Department

Atomics International
Division of North American Aviation, Inc.,
Canoga Park, California

ABSTRACT

Considerable test experience (over 2300 hours) has been accumulated at Atomics International in the area of complete Mercury Rankine systems. This testing included a space-configured system operating in a vacuum chamber, a structural mockup system for vibration testing, and a system designed to simulate the effects of zero gravity condensing. Each of these system tests has contributed significantly toward a better understanding of the operation of a Mercury Rankine system and the combination of systems has demonstrated the feasibility of ground startup, launch survival, simulated orbital startup, and simulated steady-state operation in space. In this paper the basic objectives and overall accomplishments of these systems are discussed, the experience gained in test operations is presented, and the significant test results are summarized and discussed.

I. INTRODUCTION

This report summarizes the experience and discusses test results of the ground system testing of the Mercury Rankine Program. This program has as its objective, technology development leading to a broad capability of space power supplies utilizing a reactor or isotope-powered Mercury Rankine System. The basic system has been developed in a modular form which permits use of one or more modules tailored to suit a wide variety of potential applications or test requirements. A single module is capable of generating 3-5 kwe at 120 volts, 1800 cps. Multiples of this basic module can be feasibly utilized to the 30-40 kwe range.

Figure 1 is a schematic representation showing the prime components of the basic system. The elements of the system shown in the figure form the functional arrangement of each of the ground test systems built and operated to date in the program. However, each system had a configuration as necessary to accomplish its specific objectives.

*Current address: Westinghouse Atomic Power Division, Pittsburgh, Pa.

Figure 1. Mercury Rankine Basic Schematic

Upon initiation of the SNAP 2 program (later redirected as the Mercury Rankine Technology Program) a comprehensive ground test program was planned as a major element in the system qualification. Although the SNAP 2 flight test objective was cancelled early in 1964, the first, or Prototype System Mockup (PSM), phase of SNAP 2 system testing was continued as part of the Mercury Rankine Technology Program.

The PSM program was comprised of three test systems each with specific objectives:

A. PSM-1

The PSM-1 test was designed to verify system operation in an environment which simulated space vacuum. The system was in a configuration which was representative of the flight system design and contained components which were the best prototype representation of the ultimate flight components. The system was contained within a vacuum chamber maintained at an absolute pressure of 10^{-2} mm Hg during system operation. The general objectives of the PSM-1 program included verification of analytical techniques, collection of operating data during startup, steady state, and shutdown, observation of system degradation during long term operation, and observation of thermal and dynamic interactions of components.

B. PSM-2

The PSM-2 system was a structural mockup of the reference flight system. The test program consisted of subjecting this mockup to the stress levels which would be encountered on the launch pad and from the instant of launch to placement in orbit.

C. PSM-3

The PSM-3 was a dynamic system used to verify the adequacy of the startup process while subjected to the effects of zero gravity. This was accomplished by use of a special flat radiator condenser which was installed in the horizontal plane of the Combined Rotating Unit (CRU) centerline. Thus startup testing was possible for the two critical zero gravity effects, i. e., zero g condensing and absence of a gravity head at the CRU mercury pump suction.

II. SUMMARY

A. TEST SCHEDULE

Testing of the PSM-1 and PSM-2 systems has been successfully completed. The PSM-3 system testing is continuing at the present time with an interim CRU installed (CRU-IVM). It is planned to continue system testing with this machine until about mid Fiscal Year 1966. The system will then be shutdown, a flight type CRU (CRU-V) will be installed and system testing will be reinitiated.

PSM-1 system testing was started in September of 1963 with a nozzle desuperheater installed in place of the CRU. By November of the same year, the integrated system performance had been adequately verified, thus allowing the installation of a CRU. System testing with the CRU was reinitiated in January, 1964, and was concluded in July, 1964.

The PSM-2 system testing was conducted in two phases with Phase I being a mass mockup of the system which was subjected to vibration testing. This testing phase was completed in mid calendar year 1963. Phase II PSM-2 was fabricated in the first half of calendar year 1963 and utilized developmental components. The environmental shock and vibration test phases were successfully accomplished in 1963. Static structural testing was completed in the first quarter of calendar year 1964.

The testing of PSM-3 was broken into several basic interrelated test phases. Phase IA testing, which was essentially a component test of the first horizontal radiator-condenser, was conducted early in calendar year 1963. Phase IB was the acceptance test of the power conversion system (PCS). The PCS acceptance test was conducted by TRW in the first half of calendar year 1963. Concurrently with the testing of the PCS at TRW, the CRU mercury pump was tested in Phase IC coupled with the horizontal radiator-condenser. Phase ID was a test of the complete PSM-3 system with the exception of a CRU which had been replaced with a nozzle desuperheater. This testing was conducted in the first half of calendar year 1964 and was followed in the second half of that calendar year by Phase IE testing which continued until January, 1965. Phase IE testing included the entire PSM-3 system with a CRU IVM installed. Phase IF was initiated in June, 1965, and is continuing at the present time. The major change for Phase IF was the incorporation of an automatic or "pushbutton" start system.

B. GENERAL RESULTS

The PSM-1 test system has been instrumental in defining the operational characteristics of a Mercury Rankine powerplant. Despite problems encountered in boiler outlet vapor quality and leak tightness, the 1100 hours of test operation verified the performance, startup, and stability characteristics of the system. Major test results were:

- 1) Approximately 1100 hours (500 with CRU) of test operation.
- 2) Stability of integrated system verified.
- 3) Predicted analytical thermal and hydraulic performance verified.
- 4) Operation of a space configured system within a vacuum demonstrated.
- 5) Adequacy of injection startup and shutdown verified.
- 6) System leakage encountered - identified requirement for improved design and fabrication techniques.
- 7) CRU ruggedness demonstrated through severe operational conditions.

The PSM-2 system testing was conducted with accomplishment of all objectives except that of aerodynamic heating. The heating phase of testing was not conducted due to funding limitations and the deletion of the flight test objective from the program.

The system proved to be a very successful test vehicle and little or no damage was noted during the entire test sequence. Based upon the PSM-2 test program, the structural design of a flight system can be performed with a high degree of confidence.

PSM-3 system testing which is still in progress has also been very successful. More than 1200 hours of system operation and over 70 successful orbital-type starts have been conducted. Zero gravity condensing has been verified and the adequacy of the orbital start technique has been demonstrated over a wide range of transient conditions. The feasibility of remote startup has been demonstrated by Phase IF operation which requires no manual action after the pushbutton start activation from a cold NaK system, through preheat and injection and finally to operation of the system at steady state.

III. TEST EXPERIENCE

A. PSM-1 SYSTEM

Early demonstration of the feasibility of a Mercury-Rankine system in a flight configuration and operating in a simulated space heat transfer environment was the prime objective of this ground test. Thus it was necessary for fabrication of PSM-1 to utilize prototype components in their early phases of development and simulated components in some cases. Because of the use of these components and the lack of test experience of a complete system operating in a vacuum environment, the test program was performed in two phases. The first phase contained a nozzle-desuperheater and separate mercury pump combination which replaced the CRU to allow the majority of the operating characteristics of the system to be determined without the added complications of operating a CRU for the first time. In the second phase, the nozzle-desuperheater was replaced by the CRU-IVM-5. The major objectives of this second test phase were: (1) obtain steady-state performance verification of a space configured system, (2) obtain and evaluate the performance data from components integrated into a space configured system, (3) develop a startup method for ground test systems, (4) obtain the off-design performance of the system in ranges typical of the allowable system performance tolerances, and (5) verify the NaK system heat losses during orbital startup and steady-state operation.

The test system consisted of a void-free NaK heat transfer loop, a mercury power conversion loop, and a breadboard mercury injection system — all installed in a vacuum chamber. Figure 2 is an isometric drawing of the installed system and Figure 3 shows the actual test system. The major NaK system components included a 100 kw simulated reactor NaK heater, a bellows-type NaK volume expansion compensator to insure void-free operation, a full-flow NaK dc conduction pump, and an electromagnetic flowmeter. The major mercury system components were a radiator-condenser, a combined rotating unit (CRU), a NaK-to-mercury boiler, and a breadboard CRU speed-controller. The radiator-condenser was a conical, copper-clad steel honeycomb design with 110 ft² of heat transfer surface area designated R/C 6-2. The CRU (see Figure 1) consisted of an alternator, a two-stage mercury turbine, and mercury pump on a common shaft. The NaK-to-mercury boiler consisted of four Hg tubes coiled within a five inch diameter NaK shell. The entire system was located within an 8 foot diameter by 16 foot high vacuum chamber with water-cooled walls in which the pressure was maintained below 10 microns Hg to eliminate natural convection heat transfer. One mil thick gold foil wrapped around the NaK piping provided a low emissivity surface to

Figure 2. PSM-1 Test System Isometric

Figure 3. PSM-1 Test System (Installed on Vacuum Chamber Base)

minimize radiation heat losses from the NaK system. The radiator-condenser and inside of the vacuum chamber were coated with a high emissivity coating to provide the surface required to radiate waste heat from the condenser surface to the chamber wall. The mercury injection system consisted of three mercury injection tanks that had bellows which were pressurized on one side with argon for injection, two pressure regulating tanks for storage of the excess startup mercury inventory from injection and for maintaining a constant inlet pressure to the mercury pump, and a radiator-condenser liquid-level control valve (LLCV) for ground operation. The LLCV was used to eliminate the gravity head due to the liquid level in the vertical radiator-condenser. The system design, fabrication, and installation was based upon the CRU-IVM piping configuration. The CRU-IVM was an interim unit used for non-nuclear system development. Figure 4 shows a typical CRU-IVM module as received after acceptance testing with the insulation applied.

System testing with the nozzle-desuperheater was initiated in September 1963 and continued through November 1963. In general system performance was as predicted. The void-free NaK loop was successfully fabricated and operated in a flight configuration and within the vacuum chamber. The simulated reactor (NaK heater) and control system maintained an essentially constant NaK boiler inlet temperature during startup and steady state conditions as would exist in the reactor system. The radiator-condenser liquid-level control valve and its control system were able to maintain a constant radiator-condenser pressure during steady-state operation and performance during startup was within the predicted range. The only major problem encountered was the inability of the NaK-to-mercury boiler to produce superheated vapor. Various combinations of NaK and mercury flow and temperature conditions were employed in an attempt to improve boiler performance but were not successful. Since the need for obtaining information from a complete system operating with a CRU was of paramount importance, small amounts of magnesium were finally added to the mercury system to promote wetting; design point performance was subsequently obtained. Also, during this test phase, the NaK system heat loss was determined, radiator-condenser performance was obtained at various mercury flow rates, the NaK system pressure drop was determined, and the mercury side boiler pressure drop was measured. At the time of shutdown for installation of the CRU-IVM-5, 613 total hours of system operation had been accumulated.

Operation of the system with the CRU-IVM-5 was initiated in January 1964. Design point operation of the system was demonstrated and a total of 563.5 hours of system operation was accumulated with the CRU-IVM-5. Operation of the integrated system with the CRU-IVM-5 verified the stability of the Mercury-Rankine System. No significant oscillations or resonant conditions between system

components were detectable. The boiler and CRU startup transients were as predicted and the parasitic load-speed control demonstrated that injection startup of the CRU was feasible. The ground startup and shutdown methods were demonstrated and showed that the planned startup and shutdown methods were adequate. CRU-IVM ruggedness was demonstrated through severe operational conditions. Although subjected to numerous starts, including three which were without CRU preheat, and severe liquid entrainment in the mercury vapor for most of the 563.5 hours of operation, the CRU demonstrated successful operation at rated power in a complete system. In addition, the CRU performed successfully with its alternator stator winding area evacuated. This area had previously been pressurized to prevent mercury entering the stator winding area due to a history of cracks in the bore seal between the alternator cavity and the stator windings.

The PSM-1 system was shut down in July 1964 due to a bearing failure of the CRU-IVM-5 while performing unpreheated system startup tests. However, it had demonstrated that operation of a complete Mercury-Rankine System within a vacuum environment could be accomplished successfully. Table I shows a summary of the design points for the nozzle-desuperheater and CRU phases of operation and the actual test results obtained. As can be seen from the table, a wide range of test results was obtained with the CRU-IVM-5 installed in the system. This range is attributed to less than 100 percent vapor being produced in the boiler for the majority of the operating period even though magnesium was added to the system.

In addition to proving the feasibility of a flight configured system, PSM-1 served to update the state-of-the-art for future integrated system designs, fabrication and operation. NaK and mercury components with little previous operating history were proven or upgraded during testing to allow confidence in their use for PSM-3. Melting of the NaK D. C. conduction pump power leads occurred with operation in a vacuum and resulted in larger capacity leads being installed. Solenoid valves which were installed for system flexibility were observed to leak, indicating a need for additional valve development and the desirability of minimizing auxiliary requirements for a reliable system. The mercury bellows-type injection tanks were found to be leaking early in the testing and were satisfactorily replaced with free-surface type tanks until further bellows tank development could be performed. The mercury boiler flow regulator was incapable of surviving repeated injection transients but the manual bypass valve provided satisfactory flow control for system startup and steady-state operation. Problems with mercury leaks in welds and chipless connectors indicated that a more stringent quality control program should be established to decrease system shutdown time, increase

TABLE I
PSM-1 DESIGN POINT VS TEST RESULTS

	Nozzle-De- superheater Design Point	Nozzle-De- superheater Test Results with Poor Boiler Performance	Nozzle-De- superheater Test Results with Good Boiler Performance	CRU IVM ⁽¹⁾ Design Point	CRU IVM-5 Test Data Range
Gross Power Output, kwe	-	-	-	3.0	1.2 - 3.2
Power In- put to NaK, kw	52	41	55	44.0	37.4 - 45
Reactor Simulator Outlet Temp., °F	1215	1215	1215	1215	1180 - 1215
NaK Flow Rate lbs/min	71	71	71	71	70 - 73
Mercury Flow Rate lbs/min	20	20	20	17	16.2 - 20.0
Calculated Mercury Vapor Quality, %	100	82.5 avg	100	100	85 - 95
Nozzle or Turbine Inlet Temp., °F	1150	900 - 950	1205	1150	1080 - 1095
Nozzle or Turbine Inlet Pres- sure, psia	119	107	114	90	74 - 95
Mercury Condensing Temperature, °F	600	560	560	600	584 - 620
Subcooler Outlet Tem- perature, °F	150	150	150	250	167 - 240
Turbine Ideal Power, kw	-	-	-	8.7	5.9 - 9.8

TABLE I
PSM-1 DESIGN POINT VS TEST RESULTS (continued)

	Nozzle-De- superheater Design Point	Nozzle-De- superheater Test Results with Poor Boiler Performance	Nozzle-De- superheater Test Results with Good Boiler Performance	CRU IVM ⁽¹⁾ Design Point	CRU IVM-5 Test Data Range
Shaft Speed, rpm	-	-	-	36,000	36,000
Overall System Efficiency, % (2)	-	-	-	6.8	2.8 - 7.3
Machine Efficiency, % (3)	-	-	-	34.4	17.7 - 35.6

NOTE: (1) based on CRU IVM acceptance test data;
(2) over-all efficiency = Gross Electrical Power/Power into NaK
(3) machine efficiency = Gross Electrical Power/Turbine Ideal Power

system reliability, and decrease the possibility of in-leakage of contaminants which were one of the suspected causes of the poor boiler performance. Numerous unplanned system shutdowns were encountered which were the result, in many cases, of a very conservative approach to automatic system protection. Based on system operating experience it was possible to minimize the automatic shutdown functions. Also, as operating experience was gained it was possible to reduce the number of operating personnel required to start up the system. This philosophy was later reflected in the PSM-3 design where many of the PSM-1 manual operations were changed to automatic functions.

B. PSM-2 SYSTEM

The PSM-2 system was designed to verify the structural analysis of a flight design and to supplement component structural testing by supplying system structural interactions for all phases of the launch environment. The test program included two phases with completely separate test vehicles for each phase. The first vehicle (Phase I shown in Figure 5) was used to establish test information for design as early as possible; mass mockups were used for the internal components, while the structural components, such as the radiator-condenser and brackets, simulated the stiffness of the flight design at that time. The second vehicle (Phase II shown in Figure 6) incorporated design improvements from the first phase

Figure 4. Typical CRU-IVM Module (As Received After Acceptance Testing with Insulation Applied)

Figure 5.
PSM-2, Phase I Structural Mockup

Figure 6.
PSM-2, Phase II With Nose Cone Removed

**TEST OBJECTIVES: PROVE STRUCTURAL INTEGRITY
AT QUALIFICATION LEVELS AND
DETERMINE RESPONSE CHARACTERISTICS**

TEST & RESULTS:

LATERAL DECAY:

- a) DEFLECTION WITHIN 3 1/2 % OF PREDICTED
- b) EQUIVALENT VISCOUS DAMPING 3.3%
- c) APPROXIMATE FIRST MODE LATERAL FREQUENCY 18.7 cps; PREDICTED 19.2 cps

LATERAL SINUSOIDAL VIBRATION:

MEASURED RESONANT FREQUENCIES 13.8 & 58.0 cps
PREDICTED VALUES 19.2 AND 109.1 cps
(Fixture problems being investigated)

LONGITUDINAL SINUSOIDAL VIBRATION:

MEASURED FREQUENCIES 82.0 & 151.0 cps
PREDICTED VALUES 98.9 & 395.6 cps

CRU VIBRATION EXCITATION:

NO ADVERSE RESPONSE

MECHANICAL SHOCK:

NO ADVERSE RESPONSE

STATIC AERODYNAMIC PRESSURE:

HOOP STRESSES LOWER THAN EXPECTED; OTHER STRESS LEVELS AS PREDICTED.

2PSM2A CONFIGURATION

**AERODYNAMIC
PRESSURE TEST SETUP**

**LONGITUDINAL
VIBRATION SETUP**

Figure 7. Mercury Rankine Program SNAP 2 PSM-2 System Test

testing and contained structural components which were considered flight items and internal components which were prototypes or accurate mass-geometry mockups. The specific PSM-2 test objectives were:

- 1) Establish system resonant responses
- 2) Determine individual component loadings
- 3) Measure radiator-condenser damping
- 4) Determine response to shock loadings
- 5) Measure force transmissibility from the CRU to other system components
- 6) Verify effects of aerodynamic heating in a simulated launch
- 7) Establish the structural effects of dynamic launch pressures.

All of the above test objectives were successfully accomplished except for those for aerodynamic testing which was deleted with the cancellation of the flight objective.

Figure 7 briefly summarizes the test results. The test results confirmed the structural integrity of the system and the data confirmed, in general, the analytical methods used. A summation of the design, fabrication and assembly efforts; and an outline of the test program is given in Reference 1. A summary of the structural dynamic analysis, structural dynamic tests and data analysis, and structural static tests and data analysis is given in Reference 2.

C. PSM-3 SYSTEM

The PSM-3 test program was directed toward providing an experimental evaluation of zero gravity effects on the transient and steady-state performance of the Mercury Rankine System. The NaK heat transfer and mercury power conversion systems are similar for both the PSM-1 and PSM-3 system tests. However, the PSM-3 radiator-condenser was developed on a horizontal plane and was located within a vacuum chamber at the same elevation as the centerline of the CRU mercury pump. This was done to minimize the gravity effects on system operation to simulate orbital operation. The PSM-1 system, which utilized a vertical radiator-condenser, required a control valve to eliminate the gravity effects on CRU mercury pump operation due to liquid mercury accumulating in the radiator-condenser.

Figure 8 shows the PSM-3 as it exists at the present time, Phase IF. Also included on the figure is a brief summary of the operations through September 27, 1965.

In February 1963, test operations were initiated for the Phase IA facility, shown in Figure 9. The primary component of this system was the horizontal radiator-condenser, RCH-1. This unit, as shown in the figure, was installed in the midplane of a specially constructed rectangular vacuum chamber. Test operations were performed to obtain the steady-state and startup characteristics of the condenser in order to integrate it with a power conversion system. Test results confirmed the adequacy of the radiator-condenser design and operation and provided significant information about the thermodynamics of the condensing mercury vapor during system startup. In addition, this system provided critical radiator-condenser pressure and temperature transient test data that were utilized for designing an advanced horizontal radiator-condenser, RCH-2.

With the assurance of stable condenser operations, the Phase IC facility was then constructed to provide information on the interactive effects of the radiator-condenser and the CRU-IVM mercury pump during startup transients. A mercury pump utilizing an air-driven turbine and mercury lubricated bearings was fabricated for this purpose (Figure 10) and the pump was installed in the PSM-3 system at the same elevation as the radiator-condenser. Transient tests during this phase demonstrated the capability of the CRU mercury pump to be integrated with the horizontal radiator-condenser and operate stably under simulated zero-gravity startup conditions. This information enabled design of a complete power conversion system, Phase IE, to be finalized and in October 1963 the Phase IC facility was shut down in order to fabricate a complete Mercury Rankine power conversion system. The power conversion system, designated PCS-II-3 and shown in Figure 11, had been fabricated and acceptance tested (Phase IB) concurrently with the Phase IA and IC. Throughout Phase IA and IC the system operating problems and experiences provided valuable training for test personnel. In addition, as a result of failures of several auxiliary components during testing, a formal operations review board was instituted. This review board, composed of design, test and management personnel, examined the adequacy of the system design for all possible operating modes. An operations analysis was presented and all risks to either critical test items or operating personnel were examined and evaluated. This mode of operation has continued through the later phases and has greatly contributed to successful test operation.

The design and fabrication of the Phase ID facility included a breadboard NaK system designed to duplicate the hydraulic and thermal transient characteristics of the reference flight system design. The configuration of the test components

ORBITAL STARTUP DEMONSTRATION SYSTEM

TOTAL SYSTEM OPERATING TIME AS OF 9/27/65 (hr) 1190

TOTAL SYSTEM STARTS WITH CRU 76

	STARTS	HOURS
WITH CRU-IVM-2B	32	120
WITH CRU-IVM-4A	29	249
WITH CRU-IBM-3B	15	259

LONGEST UNINTERRUPTED RUN WITH CRU (hr) 76

MOST CONSECUTIVE SUCCESSFUL STARTS 29

SYSTEM IS PRESENTLY OPERATING WITH CRU-IVM-3B
INSTALLED

START SYSTEM IS AUTOMATED FROM PREHEAT TO STEADY
STATE

Figure 8. PSM-3 System Isometric and Test Summary

Figure 9. PSM-3 Phase IA Test Facility

Figure 10. PSM-3 Phase IC - CRU IVM Mercury Pump Module

was arranged in order to duplicate the transient system characteristics. The radiator-condenser was the only component operating within a vacuum environment. Previous test experience verified the capability of the vacuum system to maintain pressures of 1 to 10 microns within the vacuum chamber and of the adequacy of the chamber cooling water system to remove the heat from the radiator-condenser at the design point condensing temperature of 600°F. In addition, since the Phase IC test data confirmed the stability of the radiator-condenser, regulator tank, and CRU mercury pump, the primary objectives of the Phase ID system were limited to:

- 1) Full power checkout of the NaK system
- 2) Verification of adequate boiler performance
- 3) Operational checkout of the auxiliary systems
- 4) Training of test personnel.

The Phase ID test system design incorporated a nozzle, desuperheater, and auxiliary mercury pump combination which replaced the CRU. This was similar to the first phase of the PSM-1 system test. The facility included a NaK system operating at temperatures up to 1250°F within the same enclosure as the mercury power conversion system. The NaK system contained an electrical cartridge heater, identical in design to the heat source for the PSM-1 system, an expansion compensator which allowed void-free NaK system operation at all temperatures, an E.M. flowmeter; a d.c. NaK pump, and a Model 5C-5 NaK-to-mercury boiler. Figure 12 shows the overall Phase ID facility including the enclosure which contained the NaK and mercury loops and the radiator-condenser vacuum chamber. Figure 13 shows some of the actual installed components.

In April of 1964, test operations were initiated for the Phase ID system. For the next three months a total of 20 startup runs were made and 555 hours of integrated testing time was accumulated. At the end of the three month period the system was considered to be performing satisfactorily for CRU operation and Phase ID testing was terminated.

With the successful completion of the Phase ID system and the successful acceptance testing of the CRU-IVM-2B unit, the prime uncertainties concerning component and subsystem capabilities were resolved. On September 8, 1965, testing of the complete Mercury Rankine system with the CRU-IVM-2B unit was initiated (Phase IE). For this system test the alternator stator was evacuated prior to and during system operations. This mode of operation, which closely approximated flight conditions, was successful. Figure 14 shows the CRU installed in the test system.

Figure 11. PSM-3 Phase IB - PCS II-3

Figure 12. PSM-3 Phase ID Test Facility

Figure 13. PSM-3 System (Showing NaK Heater, EM Flowmeter, DC Conduction Pump and Lower Portion of Boiler)

Figure 14. PSM-3 Phase IE Test System (Showing CRU Module Installed)

Initial CRU operations in PSM-3 duplicated the test methods previously used for acceptance of the CRU-IVM-2B. However, as the testing program progressed the full capabilities of the system and its components were investigated. One of the primary aims of the Phase IE testing program was to reduce the ground auxiliary components used for CRU operations to an absolute minimum, thereby simulating a flight system startup. During installation of the CRU module an excessive leak in the casting of the CRU turbine housing was found. This leak, attributed to a porous CRU housing casting, was considered detrimental to sustained system operations because of: (1) possible resultant boiler deconditioning problems, and (2) the possibility of non-condensables displacing the liquid mercury within the horizontal condenser and causing the CRU mercury pump to cavitate. The CRU housing was electroplated in an attempt to seal the leak. After this operation a system vacuum of 15 microns was achieved with a 50 micron/hr leak-up rate. This repair proved to be inadequate since, after several startups, system leak-up rates of 700 microns/hr were observed. The system, however, successfully started and the CRU mercury pump coupled despite this high leak-up rate. In addition the 5C-5 boiler improved in its performance—with continued system testing the boiler superheat lengths reached 20 feet. Test operations continued with the CRU-IVM-2B unit through mid-November 1964. During this period a total of 32 completely successful startups was accomplished. Six reduced preheat starts and six unpreheated starts were successful through initial spinup of the CRU. Four of these reduced preheat tests were terminated by underspeed scrams shortly after spinup. Each of these CRU startup tests was aimed at further optimization of the system and, as a result, no attempt was made to demonstrate the maximum number of startups by repetitive system operations. These parametric tests demonstrated the capability of the system to successfully start up under the following conditions:

- 1) Dual radiator-condenser injection of liquid mercury from the bearing drains and mercury vapor from the CRU turbine.
- 2) Elimination of CRU module bearing drain valves at startup and shutdown.
- 3) Startups with a radiator-condenser exit valve both closed and open during radiator-condenser preheat.
- 4) High turbine back pressure tests (11 psia).
- 5) System startups with leak-up rates of 700 μ /hr (non-condensables were found to travel through the radiator-condenser when condensing pressures of 9.5 psia were reached).

- 6) Startups with an evacuated stator.
- 7) Programmed NaK flow and temperature transients similar to calculated nuclear system transients.
- 8) Startups under varying injection tank pressurization modes.
- 9) Adequacy of an inline non-condensable removable system for the radiator-condenser liquid mercury outlet.
- 10) System restarts with the CRU and radiator-condenser flooded with mercury.
- 11) System startup and operation with a flow regulator upstream of a thermoelectric pump thermal simulator.
- 12) Spindowns with various electrical loads on the CRU.
- 13) Adequacy of system startups with CRU preheated to within 400-600°F.

During the seventh unpreheated startup the CRU-IVM-2B failed to spin and the spare assembly, CRU-IVM-4A, was installed in its place. Test operations with the CRU-IVM-4A module excluded startups at CRU preheat levels less than 400°F because of the previous system test results with the 2B unit.

Initial test operations with the CRU-IVM-4A demonstrated the reliability of system startup capabilities by accomplishing 10 consecutive complete system startups within a two-day period. A total of 29 consecutive startups were accomplished with the 4A module installed in the system. Only one startup attempt was aborted during this testing period.

During startup tests utilizing a throttle valve to control the liquid mercury flow rate, oscillations in boiler flow were observed during the initial few minutes of a startup. These oscillations were not present when a flow regulator was subsequently installed in the boiler inlet mercury line. These oscillations are discussed in detail in the test results section of this report.

Parametric testing of the system continued with the CRU-IVM-4A unit until mid-January. During this period a wide variety of system parametric tests were performed, including: (1) stable system operation without a speed control for speed changes of 32,000 to 38,000 rpm, (2) CRU spindown tests, (3) boiler parametric tests to determine conditions for stable system operation and (4) single actuation startup tests to verify the design adequacy of the follow-on automatic Phase IF system.

The Phase IE system operations with the CRU-IVM-4A module were terminated in January 1965, when the Model 5C-5 boiler developed a NaK-to-mercury leak in

one of its tubes. The amalgams formed as a result of this boiler failure contaminated both the NaK and mercury systems, thereby necessitating a complete system rebuild prior to continuation of the testing program with the Phase IF system. A summary of operations of Phase IE is shown in Table II; Table III shows the successful operation of the system as compared to the system design point.

TABLE II
OPERATIONS SUMMARY PSM-3 PHASE IE

CRU IVM		-2B	-4A
STARTS	Attempted	36	30
	At Speed	34	30
	Coupled - CRU mercury pump supplying system flow	32	29
Run Time (Hrs.)		120	249
Power Generated (kw-hr)		372	822
Average Power (kw)		3.1	3.2
Total Coupled Starts on CRU-IVM Units 9-1-64 thru 1-18-65 (end of Phase IE)			61
Total Run time on CRU-IVM Units in PSM-3 (hrs) 9-1-64 thru 1-18-65 (end of Phase IE)			369
Total Megawatt-hrs produced with PSM-3 9-1-64 thru 1-18-65 (end of Phase IE)			1.2
PSM-3 Mercury System Includes Nozzle-Desuperheater Operation thru 1-18-65 (end of Phase IE)	Total Hours	931	
	Total Cycles	89*	
PSM-3 NaK System Includes Nozzle-Desuperheater Operation thru 1-18-65 (end of Phase IE)	Total Hours	1210	
	Total Cycles	87*	
Maximum consecutive Starts (CRU-IVM-4A)			29

*The mercury system cycles are greater than the NaK system cycles since during some periods the NaK system was held at steady state between mercury injections.

TABLE III
PSM-3 PHASE IE DESIGN POINT VS. TEST RESULTS

	Normal ⁽¹⁾ Design Point	Test Data Range	Typical Run
Gross Electrical Power Output, kwe	3.0	2.4-4.1	3.1
Power Input to NaK, kw	44.0	37-59	44.6
Reactor Simulator Outlet Temp, °F	1215	1225-1235	1225
NaK Flow Rate, lb/min	71	71-75	72
Mercury Flow Boiler Inlet, lb/min	17	16.8-22.2	20.0
Mercury Vapor Flow, lb/min	17	15-22.1	17.5 ⁽²⁾
Avg. Condensing Temp. °F	600	575-614	585
Avg. Condensing Pressure, psia	6.7	5.0-9.1 ⁽³⁾	5.6 ⁽³⁾
Turbine Inlet Temp. °F	1150	1085-1190	1125
Turbine Inlet Pressure, psia	90	80-104	94
Subcooler Outlet Temp. °F	250	113-169	125
Turbine Ideal Power, kw	8.7	7.1-11.1	8.5
Shaft Speed, rpm	36,000	36,000	36,000
Over-all System Efficiency, % ⁽⁴⁾	6.8	6.5-7.0	6.9
Machine Efficiency, % ⁽⁵⁾	34.4	33.5-39.3	36.0

Note: (1) Based on CRU IVM acceptance test data.

(2) 17.0 lb/min vapor would correspond to 3.0 kw nominal design point.

(3) Average condensing pressure is based on average condensing temperature

(4) Over-all efficiency = Gross Electrical Power/Power into NaK.

(5) Machine efficiency = Gross Electrical Power/Turbine Ideal Power.

Fabrication and installation of the Phase IF system was completed in June 1965. The system is now fully automated to simulate orbital operation from the start of NaK system preheat through mercury injection, steady-state operation, shutdown, and mercury draining for restart. Figure 15 shows the control system logic for system startup and shutdown. In addition, the new horizontal radiator-condenser (RCH-2) is installed in the system as shown in Figure 16. Three startups were made with the CRU-IVM-1D installed but the unit failed to reach design speed each time. A fourth attempt at startup was unsuccessful in that the unit failed to rotate. The unit was removed and replaced with the CRU-IVM-3B which had

Figure 15. PSM-3 Automatic Startup Logic Diagram

previously operated at TRW for about 6300 hours with about 40 startups. A successful startup with the 3B unit installed in PSM-3 was made on June 25, 1965.

Operation of the Phase IF system has been directed toward "pushbutton" type startup demonstrations, the performance of specific tests to better define system interactions, and defining the RCH-2 characteristics. Table IV is a current summary of operations for Phase IF with the CRU-IVM-3B through September 27, 1965.

TABLE IV
OPERATIONS SUMMARY
PSM-3 Phase IF
(Through 9-27-65)

CRU IVM		3B
STARTS	Attempted	20
	Coupled - CRU mercury pump supplying system flow	15
Run Time (hrs)		259
Power Output (avg.) kw		2.9
Power Generated (kw-hr)		746
Total Coupled Starts on CRU-IVM units 9-1-65 thru 9-27-65		76 out of 86 attempts
Total Run Time on CRU-IVM units in PSM-3 (hrs) 9-1-64 thru 9-27-65		628
Total Megawatts - hrs produced with PSM-3 9-1-64 thru 9-27-65		1.9
PSM-3 Mercury System (Phase IF thru 9-27-65)	Total Hours	259
	Cycles	20
PSM-3 NaK System (Phase IF thru 9-27-65)	Total Hours	320
	Cycles	26
Totals for CRU IVM-3B Including operation at TRW	Run Time (hrs)	6559
	Spinups	60
	Coupled Starts	45

Figure 16. PSM-2 Horizontal Radiator-Condenser (RCH-2)

Figure 17. Model 5C-4 Boiler Profile; PSM-1

IV. DISCUSSION OF TEST RESULTS

A. PSM-1 SYSTEM

1. Boiler Performance

Data from the 5C-4 boiler were compared with other 5C boiler data to determine performance. Figure 17 shows typical boiler profiles based on the NaK side temperatures for, (1) operation during component acceptance testing, (2) operation in PSM-1 with the nozzle-desuperheater but without magnesium, (3) operation in PSM-1 with the nozzle-desuperheater after the addition of 200 ppm of magnesium, and (4) operation in PSM-1 with the CRU IVM-5 after the addition of 80 ppm of magnesium. All four profiles are for a mercury liquid flow rate of 20 lbs/min and a NaK flow rate of 71 lbs/min. The pertinent data for the four profiles are tabulated below:

	Preheat Length (ft)	Boiling Length (ft)	Superheat Length (ft)	NaK Temp. (°F)
1. Acceptance Test	4	4	21	195
2. N-D Test Before Magnesium	3	26	0	157
3. N-D Test After Magnesium	5	4	20	164
4. CRU IVM Test	4	6	19	160

While the shape of profiles (1), (3) and (4) is similar to what is normally experienced for a boiler producing 100 percent quality vapor, the total heat transferred from the NaK in the PSM-1 boiler with the CRU is about 16 percent lower than the heat transferred during component acceptance test. Based on boiler heat balance calculations and calculations utilizing the measured turbine nozzle coefficient and liquid mercury flow rate, the boiler exit vapor quality averaged about 88 percent for the majority of the operating period.

2. CRU IVM-5 Performance

A comparison of CRU IVM-5 operation in PSM-1 with the predicted performance based on data accumulated from performance, endurance, and acceptance testing of several CRU IVM units is shown in Figure 18. Based on a least squares linear curve fit to the test data, a gross output power discrepancy of 500 to 700 watts is indicated for the same ideal available turbine power. By definition, the ideal turbine power is, $KW_I = W_V \Delta h \times 17.57 \times 10^{-3}$ (KW)

where:

W_v = actual turbine vapor weight flow (lbs/min)

h = ideal enthalpy difference ($h_{inlet} - h_{exit}$) from Mollier diagram
assuming isentropic expansion through the turbine (BTU/lb).

Vapor weight flow, W_v , is determined from the nozzle calibration of the CRU and is given by the equation

$$W_v = \frac{7.59 \times P}{\sqrt{T}}$$

where

7.59 = nozzle coefficient for CRU IVM-5

P = nozzle inlet pressure (static) (psia)

T = vapor inlet temperature ($^{\circ}R$)

Inlet enthalpy is taken from the mercury Mollier diagram at the point corresponding to P and T . Exit enthalpy is taken from an isentropic expansion from the inlet point to exhaust pressure. The low power output in PSM-1 is attributed to low quality mercury vapor entering the turbine. The effect of liquid carryover into the turbine is to decrease the power output of the CRU due to lower overall turbine efficiency and increased machine losses.

3. Radiator-Condenser Performance

Figure 19 shows a typical radiator-condenser temperature profile. Also included on the figure is a typical profile during acceptance testing of the radiator-condenser at the same mercury flow rate. Both profiles are similar in condensing length and exit temperature for a flow rate of 20 lbs/min. However, the condensing temperature is about 40 to 50 $^{\circ}F$ lower for PSM-1 operation. This is attributed to the fact that the heat losses from the radiator-condenser in PSM-1 to the bottom of the vacuum chamber were greater than during the acceptance testing of the radiator-condenser since PSM-1 did not contain a thermal barrier at the bottom of the radiator-condenser while the acceptance test installation contained an aluminum plate thermal barrier. Also, it may be observed in Figure 19 that (1) the condensing interface is not as well defined for the PSM-1 profile as for the acceptance test profile and (2) the subcooler profile indicates that more heat addition occurred in the subcooler region during acceptance testing than during PSM-1 testing. The first observation may be explained by the presence of noncondensables in the PSM-1 system. The second observation may be explained by the fact that the subcooler region was insulated with aluminum foil during PSM-1 testing but not during acceptance testing.

Figure 18. CRU IVM-5 Performance in PSM-1

Figure 19. Radiator Condenser (RC-6) Temperature Profiles

4. Nak System Heat Losses

The PSM-1 Nak system piping and boiler was insulated with 0.001-inch thick gold foil. Figure 20 shows the NaK system heat loss from the heater outlet to the heater inlet as a function of length of NaK piping. As can be seen from the curve, the heat loss from the boiler was so small that, within the instrumentation accuracy, the boiler heat loss was effectively zero. The heat loss data presented are for a radiator-condenser average temperature of 188°F and the vacuum chamber wall average temperature of 91°F. Because of the pump and flowmeter in the line, there were some sections of piping which did not have gold foil, therefore, increasing the losses. The measured NaK system heat loss was 4.55 kw. This heat loss was 1.15 kw greater than the predicted value for the flight system. However, there were no uninsulated components or piping in the flight system.

B. PSM-3 SYSTEM

1. Demonstrated Feasibility of a Single-Actuation, Orbital Type (Zero-Gravity) Startup

The primary objective of the PSM-3 system was to demonstrate feasibility of the reference orbital startup method. This objective remains firm despite the evolution of new improved startup methods and/or modifications to the current reference method.

The current reference startup method is described, for restartable multiple systems, in Figure 21. The current PSM-3 system approaches the reference system with the following exceptions:

1) The reactor and thermoelectric pump in the primary loop are replaced by an electrically-powered reactor simulator heater and a dc conduction pump. (Future plans include the addition of an analog computer into the heater control system to simulate reactor transients and the replacement of the conduction pump by a prototype thermoelectric pump.)

2) The gas bottle injection system and bellows injection tank are replaced by a facility gas system and a free-surface injection tank.

3) The bellows and spring-controlled pressure regulator tank are replaced by a free-surface, gravity regulator tank.

4) The interim CRU (IV-M) replaces the flight article (CRU V); current plans include the installation of a CRU V early in 1966.

The above differences, though significant with respect to flight hardware, do not significantly bias the fact that the current PSM-3 system has adequately

Figure 20. PSM-1 NaK System Heat Losses

MULTIPLE UNIT ORBITAL STARTUP & RESTART

1. BATTERY POWER SUPPLIED TO MAINTAIN 12" FLOW DURING INITIAL STARTUP
2. COMMAND SIGNAL STARTS CONTROL DRUMS STEPPING IN TO PREHEAT NaK LOOP
3. WHEN REACTOR OUTLET TEMPERATURE SENSOR REACHES 1185°F, A SIGNAL---
4. ACTUATES AN INJECTION VALVE WHICH PRESSURIZES THE INJECTION TANK VIA ONE OF THE GAS BOTTLES
5. INJECTED MERCURY FLOWS THROUGH A CHECK VALVE AND SIMULTANEOUSLY INTO THE BEARINGS AND THROUGH THE THERMOELECTRIC (T/E) PUMP INTO THE PREHEATED BOILER, CAUSING NaK FLOW TO INCREASE TO DESIGN VALUE
6. MERCURY FROM THE BEARING DRAINS FLOWS INTO THE BOTTOM OF THE RADIATOR CONDENSER (R/C)
7. MERCURY VAPOR FROM THE BOILER SPINS UP THE COMBINED ROTATING UNIT (CRU), AND IS CONDENSED AND COLLECTED IN THE R/C
8. CONDENSING MERCURY PREHEATS THE R/C UNTIL THE CONDENSING PRESSURE REACHES 6 psia, AT WHICH TIME
9. THE MERCURY PUMP PRIMES THE INJECTION TANK IS EXHAUSTED, AND THE CRU PUMP SUPPLIES THE BOILER. THE MERCURY SUPPLY COMING FROM THE EXCESS INVENTORY IN THE R/C. THE REMAINDER OF THIS EXCESS INVENTORY IS COLLECTED IN THE REGULATOR TANK AS THE R/C APPROACHES EQUILIBRIUM.
10. AS TIME GOES ON, THE INJECTED GAS IS BLED FROM THE INJECTION TANK THROUGH A FIXED ORIFICE
11. WHEN THE GAS PRESSURE IN THE INJECTION TANK FALLS BELOW THE MERCURY PUMP DEVELOPED PRESSURE, A CHECK VALVE OPENS ALLOWING THE EXCESS INVENTORY IN THE REGULATOR TANK TO FLOW BACK INTO THE INJECTION TANK, THUS PRIMING IT FOR A SUBSEQUENT STARTUP. NOTE THAT AFTER THE COMPLETION OF STEP 3, ANY REDUNDANT UNIT MAY BE STARTED (IF REQUIRED) SIMPLY BY INITIATING MERCURY INJECTION (STEP 4).
12. SYSTEM SHUTDOWN IS ACCOMPLISHED BY CLOSING THE VALVE IN THE BOILER SUPPLY LINE. THE STEADY STATE MERCURY INVENTORY REMAINS IN THE SYSTEM DURING SHUTDOWN PERIODS. RESTART IS ACCOMPLISHED IN AN IDENTICAL MANNER TO THE FIRST STARTUP EXCEPT THAT THE T/E PUMP DOES NOT NEED BATTERY POWER TO SUPPLY LOG FLOW SINCE THE NaK REMAINS HOT

*STEPS 4 THROUGH 11 APPLY SIMULTANEOUSLY TO ALL ACTIVE UNITS

Figure 21. Multiple Unit Orbital Startup and Restart

demonstrated the required feasibility of the orbital startup method. Automatic startup of the PSM-3 system is accomplished as follows:

- 1) The startup signal (push-button) initiates NaK heating at low power, actuates the dc pump to provide low NaK flow (10-15% of design), and actuates heaters for the PCS components (CRU, injection tank, etc.).

- 2) When the desired NaK temperature is attained, a sensor on the heater outlet line -

- a) Increases NaK flow to 100%

- b) Changes the heater operation from constant power to outlet temperature control

- c) Initiates mercury injection into the boiler and the CRU bearings by pressurizing the injection tank.

- 3) From this time on, no further actuations are required and the system approaches steady-state, full power operation in less than one-half hour.

Thus, the primary objective of the PSM-3 system has been realized and future testing is expected to demonstrate the compatibility of a reactor system, thermoelectric pump, and CRU V in the overall startup method.

2. Demonstrated Overall System Stability

a. General System Stability

Another significant result of the PSM-3 testing has been the demonstration of basic system stability, both with and without the CRU speed control equipment, over a fairly large range of operating conditions. Stable system operation has been achieved over a range of mercury flow from 14 to 22 lb/min (design value - 19), of NaK flow from 60-80 lb/min (design value - 72 lb/min), of NaK heater outlet temperature from 1150 to 1220°F (design value - 1200°F), and of R/C condensing pressure from approximately 5.4 to approximately 6.6 psia (design value - 6.0 psia).

b. Boiler Instabilities

There was also encountered, however, a certain mode of system operation in which boiler instabilities were apparent. This mode of operation employed a fixed-position valve instead of a flow regulator to control boiler mercury flow. The instability was consistently observed during startup and occasionally during steady-state operation. It consisted of a sinusoidal oscillation of the following parameters: boiler flowrate, turbine inlet pressure (180° out of phase with boiler flowrate), and boiler outlet NaK temperature. Neither NaK flow nor NaK boiler

inlet temperature were varying because of the dc pump and the fixed temperature-control mode of heater operation respectively. The natural frequency of the oscillations observed at steady-state was approximately 4 cpm (Note: NaK loop transport time $\cong 0.4$ min). These oscillations were normally self-induced and usually died out without requiring operator action. During the first 20-30 sec of mercury injection, a different pattern of oscillations was observed with inlet boiler flow only; a high-frequency (approximately 1-2 cps), low amplitude oscillation was modulated by a lower frequency (0.1-0.2 cps), higher amplitude oscillation.

These oscillations appear to be caused by a combination of the coupling of boiler pressure drop to boiler mercury inlet flowrate (via the fixed-resistance valve) and the particular thermal characteristics of that boiler. The cause of the perturbations is thought to be variable heat transfer characteristics in the boiler and the sensitivity of the boiler pressure drop to these thermal characteristics. These variable thermal characteristics at times, pushed the boiling region back into the tube inserts (first 5 ft of tubes, used to promote heat transfer in preheat region), instead of several feet past the inserts into the tubes (normal operation). Insert boiling further contributed to the instability by significantly increasing the boiler pressure drop (this type of behavior, in fact, was also occasionally observed with a boiler flow regulator employed instead of a valve and is attributed to the fact that, with a very large boiler pressure drop, the ΔP across the flow regulator is reduced to beyond the point at which regulation is possible).

Based upon these observations, the following conclusions may be drawn:

- 1) A boiler flow regulator is required in the system to ensure stable system performance in the event of varying boiler performance.
- 2) Boiling in the insert region of the boiler tubes should be avoided since it contributes both to a high pressure drop and to possible erosion damage of the tubes (the boiler in question failed in a mercury tube in the insert region). This can be easily accomplished in the flight system by proper initial setting of NaK flow.
- 3) The flow regulator should be designed to regulate over as large a ΔP range as possible, to further damp the effects of boiling variations and perturbations.

c. Speed Control Instabilities

During the course of PSM-3 testing, it was observed that there was a dynamic interaction between the speed controller and the alternator. This interaction

appeared as a 40-60 cps amplitude modulation of the basic 1800 cps voltage waveform as shown in Figure 22; the upper scale shows the top of the voltage waveform (i. e. , the total amplitude in the picture is 5-10% of the alternator voltage) with the modulation present and the bottom scale shows the basic 1800 cps alternator waveform.

This instability was determined to be caused by the feedback of phase variations of the load voltage wave into the discriminator portion of the speed control circuit (see Figure 23). By increasing the capacitance of the short-circuit capacitors, it was determined both analytically and experimentally, that the phase variation could be eliminated (or reduced to a negligible amount). The result of increasing this capacitance from 40 to 60 μ f per phase is shown in Figure 24 which shows the waveform to be smooth and essentially free of the modulation.

Thus, although further analysis is continuing, it has been determined that an increase from 40 to 60 μ f per phase in the short-circuit capacitors will eliminate this feedback instability while still providing adequate protection of the permanent magnet against short-circuit loads. Also, for applications which require conversion to dc power, this amplitude modulation does not appear to present any problems.

3. Demonstrated R/C Stability

Another purpose in testing a horizontal radiator-condenser was to study R/C stability, with multi-tube condensing, in a zero gravity environment. Two types of horizontal radiators have been tested in PSM-3; the first one (RCH-1) had 44 circular, non-tapered tubes with a larger-than-critical diameter (0.26 in.) and the second, and current one (RCH-2), has 40 tapered rectangular tubes (0.270 x 0.218 in. to 0.270 x 0.125 in.) to simulate a flight system. The critical diameter is that diameter below which the interface is unconditionally stable for any disturbance and in any orientation; this diameter varies with g-level, being small on the surface of the earth and large in low-g orbits.

The types of instabilities which have been defined analytically and from single tube tests include both single tube (interface and runback instability) and multi-tube (zero-g and liquid leg instability) instabilities; these types are discussed in detail in Reference 3. Suffice it to say here that for all normal (i. e. , near the design point) operating modes, both startup and steady-state, there were observed no instabilities in either RCH-1 or RCH-2.

Early parametric testing with RCH-1 did, however, reveal a system instability at very low (approximately 5 lb/min) mercury flow (component-type parametric tests have not yet been run with RCH-2). This instability was manifested as an

Figure 22. Voltage Instability
(40 μ f Capacitors) in PSM-3

Figure 23. Schematic of Speed Control Circuit

Figure 24. Stability Regained with
60 μ f Capacitors

oscillation (non-sinusoidal) of R/C pressures and temperatures on about a 3 min period. It was determined that this instability was not present for flowrates above approximately 7 lb/min. Since this is so far removed from the design point (i. e. , 20 lb/min), the importance of the oscillations is questionable. However, future parametric testing (with RCH-2) is planned for this low flow range and if this instability arises again it will be closely observed, recorded, and analyzed for detailed causes.

4. Demonstrated Thermoelectric Pump Startup Performance

Prior to system testing, it was hypothesized that there could be a serious startup problem depending on the mercury-side heat transfer characteristics of the thermoelectric pump. This potential problem may be described as follows: (See Figure 21)

- 1) The preferred location of the mercury boiler flow regulator is upstream of the thermoelectric pump (thus allowing it to operate at approximately 500°F instead of approximately 800°F if it were downstream).

- 2) In this preferred location of the flow regulator, the thermoelectric pump may be subjected to low-pressure operation (just after mercury injection) until the mercury pressure builds up in the boiler.

- 3) Since the thermoelectric pump is very hot just prior to injection (1100-1200°F), there would exist a very large ΔT between the cold junction wall and the mercury. Also, the pressure being low would cause boiling of the mercury in the pump.

- 4) If the boiling mechanism were such as to cause very low heat transfer rates due to film-type boiling, the tube wall (which is essentially the cold junction of the thermoelectric pump) would cool slowly.

- 5) Analyses of the startup transients at injection indicated that at the (conservatively postulated) low heat transfer rates associated with film boiling the pumping action of the thermoelectric pump would be seriously delayed, i. e. , the ΔT between hot and cold junctions would increase slowly causing the NaK flowrate to also rise slowly.

- 6) Inadequate NaK flow after mercury injection could result in boiler flooding, spilling, and a subsequent startup failure.

It is seen that, in order to assess the likelihood of occurrence of such a chain of events, a system test is required.

Since a thermoelectric pump was not available, system testing was accomplished in PSM-3 with a thermal simulator of the thermoelectric pump installed downstream of the boiler flow regulator. Test results from this testing indicated that the aforementioned mode of operation apparently did not occur. The simulated thermoelectric pump cold junction cooled very rapidly following mercury injection thus indicating a high rate of heat transfer. Subsequent testing of a thermoelectric pump in a component loop also substantiated these simulator results and showed a sharp increase in NaK flow as soon as mercury reached the thermoelectric pump.

It has thus been concluded that the thermoelectric pump may be placed downstream of the flow regulator without impairing system startup reliability.

5. R/C Preheat and Drain Transients

The first experimental data obtained during the transient operational mode associated with R/C startup were obtained from PSM-3. The test results indicated that, although condensation preheat of the R/C is a good, simple, and feasible startup concept, the early analytical methods employed for predicting R/C preheat were in error. The data also revealed the reasons for the error and led to an improved analytical model which did correlate very well.

The early model employed an assumption of a uniform condensing region throughout the condenser with the result that the condensing temperature and pressure increased uniformly with time until the final temperature (approximately 600°F) and pressure (approximately 6 psia) were attained. The results of this model are shown in Figure 25.

The PSM-3 startup transients for RCH-1 are shown as the solid lines in Figure 26; the dashed lines represent the new analytical model which is discussed shortly. The observed startup transients are seen to be radically different from those predicted from the early model. The principal difference arises from the fact that the mercury flow becomes choked at the tube inlet, thus forcing the pressure to rise to a level necessary (approximately 2 psi) to pass the required flow. Resulting from this behavior is the fact that there is a very large condensing drive force ($T_{\text{sat}} - T_{\text{wall}}$) as the vapor enters the cold tube. The heat transfer coefficient for condensation is so large that, with the large ΔT available, only a very short length (approximately 1 ft) is required to condense all of the vapor. Thus, it is observed that preheat occurs over a short length of R/C instead of throughout the tubes and as the tube wall in this short length heats up, the condensing region progresses further down the tube until eventually the tube downstream of this condensing region fills with liquid. When this occurs, the

conditions for uniform condensing are finally attained since the entire condensing region is hot; the remaining portion of preheat then consists of the time required to uniformly raise the condensing temperature from its initial value of approximately 500°F (approximately 2 psi) to its final value of 600°F. It is also observed, from the data in Figure 26, that the area of rapid condensation is also the area of large pressure drop; i. e., the pressure upstream of this region is approximately 2 psi and downstream is only a small fraction of 1 psia.

The preceding discussion forms the basis for the new analytical model: the condensing temperature is constant and is approximated by the saturation temperature at the tube inlet choking pressure. The rate at which the condensing region moves down the tube (i. e., rate of exposing "cold tube") is controlled by the rate of vapor flow, the radiation heat losses, and the variation in the total RC heat capacity. The results of this new model are shown in Figure 27 and the dashed lines in Figure 26. Also, a detailed discussion of this new model may be found in Reference 3. The agreement of the new model with PSM-3 results has been very good with RCH-1; it is currently being compared with the startup transients in RCH-2.

After RC preheat is complete (6 psi), the RC begins to drain its excess startup inventory into the pressure regulator tank as equilibrium conditions are approached. A similar thermal model, coupled with the transient fluid dynamics of the RC and regulator tank are used to predict RC drain rate. The details of these calculations are presented in Reference 3, and only the results will be shown here. In Figures 28 and 29 are shown analytical and PSM-3 experimental values of RC pressures during startup and liquid level transients during draining. The agreement between experimental and analytical values, while not perfect, is judged to be adequate since predicted RC preheat inventory and drain time are within 5 - 10% of the observed data.

It is thus felt that the current method of predicting R/C startup transients is adequate and may be extended to flight-type R/C designs. The importance of an accurate prediction of R/C preheat inventory is explained as follows: too high a prediction results in excess system weight (both mercury and tankage) while too low a prediction could result in a startup failure. Thus, it is seen that an accurate prediction will minimize weight penalties (i. e., contingent mercury supply) and assure adequate startup inventory.

6. CRU IV-M Startup Transients

The major items of interest in a system test of this nature are the various interactions during an orbital-type startup; this applies especially to the effect of

Figure 25.
Temperature Distribution in
RC During Injection

Figure 26.
Experimental vs Predicted
RC Preheat Temperatures

Figure 27. Mercury Rankine Program
SNAP 2 PSM-3 System Test

Figure 28. Experimental vs Predicted
RC Preheat Pressures

Figure 29. Experimental vs Predicted
RC Draining Transient

these interactions on CRU startup. Figure 30 shows the major system startup transients with a CRU IV-M in PSM-3; these transients are typical of a fully preheated startup. System startup transients with no CRU preheat are shown in Figure 31. These two figures form the basis for the following discussion, which will more or less "follow" the mercury path (i. e. , starting at the boiler and ending in the R/C).

These graphs are started with the beginning of mercury flow to the boiler for convenience only; this flow begins approximately 15-20 sec after the injection signal and flow to the CRU bearings is available when boiler flow starts.

The boiler used for these tests was performing very well as is evinced in the figures by a very rapid build-up of boiler pressures and in general by a short boiling length (approximately 4 ft), long superheat length (approximately 22 ft), and high exit quality (approximately 90% - this quality is used more as a comparative result than as an absolutely accurate value, since it is calculated by a boiler heat balance). The boiler was more "conditioned" during the test of Figure 30 than in the earlier test (Figure 31) as may be seen by a much larger pressure spike at the boiler inlet (approximately 300 psia in Figure 30 and approximately 200 psia in Figure 31).

CRU spinup begins as soon as the turbine inlet pressure reaches 10-20 psia in both cases (i. e. , for hot and cold starts) but the spinup characteristics are notably different for the two startups. The CRU spins up very rapidly and uniformly for the hot start, requiring only 13 sec to attain full speed. For the cold start, however, spinup time is delayed by the presence of liquid mercury and full speed is not attained for 47 sec. The major cause of the variation in spinup rate is flooding of the alternator rotor-stator gap and of the high-pressure labyrinth seal near the turbine. This flooding is caused by two factors, both condensation in the CRU (for cold starts) and by a gravity effect which tends to push liquid into the labyrinth seal. Unfortunately, the magnitude of the two effects cannot be separated and it is thus not known whether spinup in a zero gravity cold orbital startup would be closer to a cold or hot startup in PSM-3. It is felt, however, that the major cause of drag is due to gravity and that spinup time in a zero gravity cold start will be close to 15 sec. In any event, this reduced pre-heat testing combined with more severely "flooded" startups (in which case both the RCH and liquid lines contained mercury prior to startup) did indicate that reduced CRU preheat is feasible, although not yet perfected for a CRU IV-M (this particular CRU IV-M failed during cold starts; this failure was due to overloading the hydrodynamic thrust bearing beyond its capacity while turbine exhaust pressure and shaft speed were low). Further reduced preheat testing with a

Figure 30. Transients for Typical, Preheated Startups;
PSM-3 Phase IE, Run 434

Figure 31. Transients for Typical, Unpreheated Startups;
PSM-3 Phase IE, Run 139

CRU V machine (which was designed for zero preheat) is therefore required in order to fully demonstrate the feasibility of cold startups.

Another effect of a cold start is the delay in the rise of turbine exhaust pressure. A venturi is located in the CRU vapor exhaust line which chokes during startup to raise the cavity pressure. This rise in cavity pressure is intended to accomplish two things; (1) prevent flooding within the CRU cavity by increasing the pressure on the vapor side of the bearing seals above that on the liquid side, and (2) reduce the thrust bearing load during startup. In Figure 30, there is little or no condensation in the CRU exhaust cavity because of the hot startup, thus causing the turbine exhaust pressure to "follow" the turbine inlet pressure to the choking point (4.7 psia at this flowrate). In the cold start (Figure 31), there is a delay (approximately 25 sec) between the rise in turbine inlet and exhaust pressures due to condensation in the turbine exhaust cavity and exit line. It is this pressure delay which contributes to the slower spinup as discussed earlier.

The electrical output of the CRU (gross alternator output, in Figures 30 and 31) may be seen to follow the CRU speed during spinup, level-off when the exit venturi chokes, and drop-off as the turbine exhaust pressure increases to design value. This behavior is normal and expected; however, the power dip in Figure 31 requires an explanation. This dip is caused by a "hump" in the characteristic curve of the power versus speed in the speed controller at approximately 30,000 rpm; i. e., as the shaft spins up through this speed, the parasitic load momentarily "comes-on" and unloads as soon as this speed is passed. This occurs too quickly to be noticed in a fast spinup, thus, it is observed only in Figure 31 where the spinup is slower. This speed controller "hump" in the power versus frequency curve has been biased out in the more recent CRU V designs.

Also shown in Figures 30 and 31 are the significant R/C pressures which affect CRU performance; viz., liquid manifold pressure (or pump inlet pressure and bearing back-pressure) and vapor manifold pressure. The liquid manifold pressure causes the CRU pump to prime (at approximately 0.1 psia inlet) and later to couple to the system and provide boiler and bearing flow (2-3 psia). The vapor manifold pressure rise causes the venturi exhaust pressure to unchoke, and rise to design conditions. During the course of PSM-3 testing with many (approximately 70) startups and of several different types it has been shown conclusively that the CRU pump will prime and couple to the system with a high reliability.

If the R/C pressure transients in Figures 30 and 31 are compared both with the earlier RCH-1 pressure transients (Figure 28) and with each other, there are observed certain differences; most of which are explainable. In the test phase

resulting in Figure 28, there was an exit valve at the outlet of the R/C which was opened at the end of R/C preheat while in the later testing (Figures 30 and 31) this valve was always open allowing the R/C to collect the bearing drains. It is primarily this difference which results in the pressure differences (vapor and liquid manifolds) between Figures 28 and 30. Also, the line from the CRU exhaust to the vapor manifold was fully preheated (approximately 600°F) in the earlier testing and only partially preheated in the later phases (resulting in the relative delay and slower rise rate of the vapor manifold pressure in Figure 30). The difference between R/C pressures in Figures 30 and 31 can only be partially explained by a combination of two factors. It is relatively certain that the R/C was not empty at the time of startup in Figure 31; i. e., some inventory (approximately 20-50 lbm) remained in the R/C after the previous shutdown. Also, the vapor line leading to the R/C was not hot. These two factors tend to explain why the two pressures (vapor and liquid manifolds) rise at about the same rate (the R/C was almost full of liquid when vapor flow started) but do not explain the fact that the liquid manifold pressure rises first. This phenomenon has been also observed in recent testing in which the R/C has had a large (30-50 lbm) initial inventory, thus correlating the pressure behavior with a pre-loaded R/C. Current data analysis is directed toward the explanation of this anomaly, with special emphasis on the characteristics of the pressure instrumentation.

7. NaK System Startup Transients

The NaK loop in PSM-3 closely simulates the "flight design" type NaK loop regarding line size and length, total pressure drop (~ 1 psia), and a NaK heater which simulates (thermally and hydraulically) the SNAP 2/10A reactor. Thus, the overall performance of the NaK system, both transient and steady-state, can be accurately verified by PSM-3 testing. Also, original plans called for the installation of a thermoelectric pump and an analog computer which would drive the NaK heater to duplicate reactor transients. Neither the thermoelectric pump nor the analog computer have yet been integrated into the system for reasons of availability, schedule, and funding; current plans include the installation of the computer into PSM-3.

Nevertheless, certain transient data have been obtained by manually operating the NaK heater to approximate reactor-type transients, the results of which are shown in Figures 32 and 33. For this type of run, the NaK heater controller was changed from temperature control (fixed outlet temperature-normal control mode) to manual control. From the analytically-predicted transients of reactor power and NaK flow during PCS startup (Figure 32), the heater power and NaK flow were manually adjusted to simulate these transients. The resulting boiler temperature

transients (inlet and outlet temperatures) are shown in Figure 33. Although the simulation was not perfect, the general effect was to cycle the NaK temperatures at approximately the correct frequency and amplitudes thus accomplishing, at least preliminarily, the desired input effect on the mercury system (see Figure 33). The results of these tests, as determined by mercury system performance, indicated that there are no adverse effects (such as serious suppressed boiling, boiler spilling, and variations in CRU performance) on the mercury system due to the expected NaK transients at startup.

Figure 34 shows typical boiler (Model 5-C) transient profiles during mercury injection. The NaK system was on temperature-control for this startup, thus the boiler inlet temperature changes very little. This is not considered significant however, when it is observed (from the figure) that the boiler transient response is fast enough to closely follow the expected variation of boiler inlet temperature. In this figure, it is also seen that the boiler is operating with ~13 feet of suppressed boiling, ~6 feet of boiling region, and only 7 feet of superheat length. This mode of operation corresponds to a NaK inlet temperature of ~1160°F, a NaK flow of 69 lb/min and a mercury flowrate of 20 lb/min. The steady-state boiler profile has been found to be easily adjustable (though not so easily predicted, repeatably) by varying NaK flowrate and inlet temperature. It has been learned from such adjustments, that a "conditioned" boiler delivers approximately constant mercury vapor conditions (i. e. , temperature and quality) independent of the superheat length; the only effect of superheat length is that on boiler pressure drop (i. e. , a long superheat length causes a large pressure drop). Thus, it is concluded that (1) the boiler transients during mercury injection are well-behaved with superheated vapor being produced throughout injection with approximately a constant distribution of suppressed boiling, boiling, and superheat lengths and (2) the Model 5-C boiler is somewhat over-designed with regard to the effect of superheat length on mercury vapor exit conditions (Model 7 boilers, now being built at TRW, have only ~19 feet of mercury tubing compared to ~26 feet in the Model 5-C).

8. CRU Spindown

A recent objective in the MRP has been directed toward providing orbital shutdown and restart capability. A direct consequence of this goal involves a detailed knowledge of the system behavior during shutdown. The reference method chosen for shutdown is simple in concept, but requires feasibility verification in several critical areas. One of these areas is the method of providing CRU bearing lubrication during spindown. The reference method of shutdown involves, (1) closing the boiler shutdown valve to starve the turbine and, (2) loading the

Figure 32. Simulation of Reactor Transients During Mercury Injection

Figure 33. PSM-3 NaK Temperature Transients Resulting from Simulated Reactor Transients During Mercury Injection

alternator to quickly decelerate the CRU. The CRU pump is expected to be capable of supplying adequate bearing flow during spindown (preliminary test data, both at TRW and AI tend to confirm this) if the pump inlet pressure is high enough to sustain pump prime. Thus, it is desirable to program the alternator loading to assure that the spindown will occur fast enough that the pump does not lose prime during spindown due to loss of inlet pressure. The rate of inlet pressure decay is governed by the R/C shutdown characteristics (discussed later) which require that spindown be completed within ~1 minute from shutdown (based on the conservative assumption that the pump inlet pressure must be 1 psia or more during spindown—preliminary data indicate that loss of prime is experienced only when the inlet pressure drops to several tenths of one psia). Since a normal spindown (i. e. , no special alternator load other than a fixed resistance) takes from 2 to 6 minutes, some method of shutdown-loading of the alternator is desirable to reduce this time to under one minute.

In addition to component-level CRU testing at TRW, system testing in PSM-3 is required in order to assess the feasibility of a fast spindown. Effort in this direction has thus far been limited to spindown tests with fixed-resistance loads and no short-circuit capacitors (these capacitors tend to unload the alternator during spindown due to their increasing impedance at low frequency). The purpose of this type of spindown test is to determine the electrical characteristics of the alternator (e. g. , internal winding inductance, resistance, etc.) which would then allow the incorporation of a programmed load into the total equivalent circuit.

The results from a typical spindown test in PSM-3 are shown in Figure 35, in which a fixed-resistance load is used instead of the normal manual load, parasitic load, and short-circuit capacitors. The voltage, current, and shaft speed (frequency) data are currently being used to attempt to construct an equivalent circuit for the alternator. The major problem at present is to define an equivalent alternator circuit that describes the machine both at speed (with capacitors and various loads) and during spindown with the capacitors removed. It is also apparent, from Figure 35, that the boiler pressure decay characteristics have a significant effect on the spindown time; the boiler pressure is important for at least 40 seconds in this run, thus further emphasizing the necessity of a good shutdown load. It is felt, however, that the "chugging" characteristics observed in the PSM-3 boiler (see Figure 35) will not occur in a zero g environment since these pulses are probably due to gravity draining of the liquid line to the boiler (i. e. , the integrated pulses have about the same inventory as does the line between the shutdown valve and the boiler inlet). Alternatively, the location of

Figure 34. Typical Boiler Transient Profiles During Mercury Injection

Figure 35. CRU and Boiler Transients for Typical Shutdown-PSM-3, Phase IE, Run 426

the shutdown valve may be changed to be closer to the boiler inlet (this is also being considered for PSM-3).

Thus, although more testing and analysis are required in this area, it is felt that the proposed shutdown method is feasible and that the spindown time will be reduced to less than ~1 minute.

9. R/C Shutdown Transients

In the above section on CRU Spindown, the general system shutdown problems were discussed. This section deals specifically with the shutdown characteristics of the RC and the resulting decay of pump inlet pressure.

The PSM-3 system has been used to obtain the decay rate of pump inlet pressure as it is affected by system interactions such as regulator tank inventory, RC cooling rate, and boiler flow decay. Since the reference shutdown and restart method employs a low regulator tank inventory (~15 lb) during normal operation and since it is expected that the decay of vapor flow will be rapid (e. g. , 10 sec time constant), the RC temperature decay rate (3-5 minute time constant) may be expected to govern the rate of pressure decay.

Both detailed analysis (Reference 3) and PSM-3 tests have confirmed this reasoning, as is shown in Figure 36. This figure compares the analytically predicted RC pressure to that observed in PSM-3 (with RCH-1) and shows good agreement. The boiler flow decay, for the analytical prediction, was taken from typical shutdown data (see Figure 35), with a smooth curve approximation.

Although these tests must be repeated with the tapered-tube (RCH-2) planned for current testing, it is reasonably certain that the analytical method of predicting RC shutdown transients is adequate. Future effort in the shutdown and restart studies should be directed toward optimum alternator loading and further verification that the CRU pump provides adequate bearing lubrication during spindown.

REFERENCES

1. W.B. Gruenberg, NAA-SR-9906, "Snap Mercury Rankine Program SNAP 2 Structural Evaluation Prototype System (PSM-2)," December 1, 1964.
2. R.A. Castle, R.D. Galletly, and M.G. Weisman, NAA-SR-10560, "Analysis of SNAP 2 PSM-2A Test Data," (to be released).
3. V.H. Heiskala, R.C. Smith, E.A. Elliott, and R.T. Lancet, "Mercury Rankine Radiator-Condenser Development Experience," presented at the AIAA Rankine Cycle Space Power System Specialist Conference, October 26-28, 1965.

Figure 36. Experimental vs Predicted RC Shutdown Pressure

SNAP-8 DEVELOPMENT STATUS - SEPTEMBER 1965*

by

R. Gordon, Manager
SNAP-8 Division
Von Karman Center
Aerojet-General Corporation
Azusa, California

and

H. O. Slone, Manager
SNAP-8 Project
Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio

ABSTRACT

The status of the development of the SNAP-8 Program is summarized. The actual weights and measured performance of subsystems and components are listed. The development problems are described and the total amount of operating time and operating experience for each component are listed. The rated power loop (RPL-2) has operated for over 800 hours; and the PCS-1 (power conversion system 1), Phase I test operations, have been completed. Results from these are included.

I. INTRODUCTION

The SNAP-8 is a turboelectric nuclear space power system using a mercury Rankine cycle. The system comprises three major subsystems: a nuclear system consisting of a reactor, reactor controls and shielding; a flight radiator assembly consisting of radiator heat exchangers required to remove heat from the liquid cooling loops; and the power conversion system consisting of turbine-alternator assembly, boiler, condenser, mercury and NaK pump-motor assemblies and necessary controls, piping and structure.

*The information appearing in this paper is based on work performed under National Aeronautics and Space Administration Contract NAS 5-417.

The SNAP-8 Program was started by NASA in 1960 after an industrial competition which was won by Aerojet. The initial design, based on the technology developed on the SNAP-2 Program, included only 2 loops; a NaK circuit between reactor and boiler; and a mercury circuit from the boiler to a direct-condensing radiator.⁽¹⁾ The operating conditions, temperatures and sizes proved sufficiently different from the then current state-of-the-art that difficult development problems arose. After a thorough evaluation by NASA and Aerojet, the system was redesigned in late 1962 in order to separate and to simplify the problems.

The resulting four-loop system uses three liquid-metal loops in the power conversion cycle. A liquid sodium-potassium alloy (NaK) loop transfers heat from the reactor to the boiler. The Rankine-cycle secondary loop, using mercury as the working fluid, converts thermal energy into mechanical energy as mercury is expanded through a turbine. The mercury is subsequently condensed and returned to the boiler. A third loop, containing NaK, transfers the heat from the mercury condenser to the radiator where it is rejected to space.

The fourth loop contains an organic fluid which lubricates the bearings of the main rotating components and provides necessary cooling for the alternator, electric motors, and controls. There is also an auxiliary loop which functions only during the startup sequence. It provides for transferring heat directly from the primary to the third loop to minimize reactor transients, and to warm the radiator. However, development of the radiator is not part of the current program. The SNAP-8 system schematic diagram is shown in Figure 1. The most significant system parameters are shown on the diagram. A conceptual flight configuration of SNAP-8 is shown in Figure 2.

This electrical generating system (EGS) is designed to operate continuously for 10,000 hours in space after a remote automatic startup. The net electrical output of the SNAP-8 system is 35,000 watts, sufficient power to sustain many contemplated long-term manned and unmanned space explorations.

Superscript numbers in parentheses refer to similarly numbered References listed following the text of this paper.

The status of this program was last presented to the AIAA in September 1964.⁽²⁾ Much progress has since been made. The purpose of this paper is to provide a current status of the technical developments. As a result of the omission of SNAP-8 from NASA's FY 66 budget, the current efforts constitute a phase-out program directed towards the objective of demonstrating 10,000 hours of operation on the first power conversion system, PCS-1.

II. COMPONENT TESTING HISTORY

A. NUCLEAR SYSTEM

1. Reactor

The SNAP-8 reactor shown in Figure 3 is being developed by Atomics International. The reactor is compact, lightweight (600 lb), and utilizes hydrided zirconium-uranium fuel-moderator rods clad with Hastelloy N. It is described in detail in another paper⁽³⁾ at this meeting. The SNAP-8 experimental reactor (S8ER) was developed to demonstrate and measure performance of the reactor core (such as the fuel and internal-reflector characteristics, the core hydraulics and heat transfer) and the nuclear performance of the reflector assembly. As illustrated in Figure 4, the S8ER core and beryllium reflector assembly are suspended from a concrete and lead shutdown shield. The reactor is controlled manually through commercial electrical motors mounted on top of the shield and connected to the drums by long shafts.

The reactor went wet critical in June 1963. Since then, it has demonstrated sustaining power operations capability, and static and dynamic stability. Analyses of control-drum worths, critical mass, and temperature coefficients of reactivity have been verified. The reactor and test facility were permanently shut down on 15 April 1965 after an accumulated total of 12,000 test hours. One hundred days of testing were at a power level of 600 kw, and a NaK outlet temperature of 1300°F. The longest continuous run was 5000 hours.

2. SNAP-8 Development System (S8DS)

The S8DS, which was scheduled to be incorporated in the first SNAP-8 nuclear power system, is comprised of a reactor, shield, and associated reactor controls.

Fabrication of the components for this reactor is nearing completion. Delivery of the reactor-shield assembly is scheduled for Spring 1966 for installation in the nuclear test facility (Figure 5) and initiation of shakedown testing. The arrangement shown in Figure 5 is designated as a "loosely coupled" system. Combined testing is not part of the current phase-out program.

B. BOILER

1. Tube-in-Shell Boiler

The SNAP-8 boiler is a once-through evaporator of cross-flow design where the mercury is preheated, boiled, and superheated in a single pass through the heat exchanger. Figure 6 illustrates the design. The mercury flows through four parallel tubes of equal length which are coiled on two diameters and wound together forming two double-lead helices. The NaK flows through an annular passage formed by inner and outer cylinders around the tubes containing mercury, in a cross-flow arrangement. The overall configuration is 24 in. in diameter and 58 in. long, and weighs 810 lb, including fluid inventories. Details of this boiler are given by Kreeger, Sellers, and Hodgson⁽⁴⁾ in another paper at this meeting and will not be repeated here.

Initial tests on the boiler (in RPL-2), which had become contaminated with oil, indicated that a 600-hr operating period was insufficient to condition it to full performance. The chief problem in obtaining design performance of the boiler immediately on startup is ascribed to the lack of full mercury wetting on the inside wall of the 9M steel (9Cr-1Mo) tubing. This is a complex problem involving the metal surface condition, mercury purity, and the thermohydrodynamics of two-phase flow. Although not completely understood on a fundamental basis, there are enough engineering data available to permit the design of specific units. This problem is treated in greater detail by Carey, Farwell, and Thur in another paper.⁽⁵⁾

As an alternative to a conditioning period in the boiler, trace amounts (less than 1,000 ppm) of rubidium have been added to the mercury to improve heat transfer performance. Rubidium was selected as an additive because of its low vapor pressure at

SNAP-8 temperatures permitting it to circulate through the mercury loop. Figure 7 shows the boiler performance parameters as a function of time. The left-hand portion of the figure is the performance after 600 hours of operation whereas the right-hand side shows the effect of the additive.

As long as the rubidium concentration was maintained, the boiler performed satisfactorily. As soon as the rubidium concentration dropped, because of oxidation or other losses, the boiler stopped performing properly. However, subsequent operation of the tube-in-shell boiler in the RPL-2, starting in early May 1965, resulted in the boiler conditioning itself, without rubidium, within three hours. After this, operation in RPL-2 has been without rubidium. The tube-in-shell boiler has been removed from the RPL-2, and is being cut apart for detailed metallurgical examination. This unit has had 1500 hours of operation and will be extensively analyzed.

Another tube-in-shell boiler has been tested during the PCS-1, Phase I test series. Made to the same prints as the previous boiler, this unit had not been heavily contaminated with oil. It became conditioned after some 24 hours of operation, as shown in Figure 8, without the addition of rubidium. Subsequent starts were made in which the boiler conditioned immediately, but the degree of conditioning varied.

The operation of the tube-in-shell boiler has been generally satisfactory with the exception of the inconsistent conditioning. In addition, the vapor entering the turbine has been found to contain approximately 5 to 15% liquid in the form of fine droplets even though there is over 200°F of superheat. Although we are not satisfied with the performance of this boiler, it could be used, if necessary.

2. Tube-in-Tube Boiler

Because of the inconsistent conditioning of the tube-in-shell boiler and the inability to evaluate its internal performance because of the tube-in-shell construction, a backup boiler was designed. The design of this boiler is based upon small-scale, single-tube tests which were conducted at Aerojet-General Nucleonics, San Ramon, California.

The results of these tests are best illustrated by the NaK profiles shown in Figure 9. Curve 1 shows the profile after 10 hours of operation and Curve 2 shows it after 670 hours of operation, in both cases with a 0.8 fps inlet liquid-mercury velocity. Very little improvement is shown in the NaK profile although the mercury outlet conditions did improve and finally reached superheated vapor. The boiler inlet plug was then changed to give an inlet liquid mercury velocity of 6.0 fps. There was a substantial improvement in the NaK thermal profile, indicating good boiler performance, immediately upon startup. Curve 3 indicates the results four hours after startup. Curves 4 and 5, which were taken later in time, indicate that the profile was getting better and that the boiler was still conditioning. The original 0.8 fps mercury velocity plug was then replaced and Curve 6, which is representative of a conditioned boiler tube, was then obtained. From this it appears that, once a mercury boiler surface is conditioned by high velocity, the boiler is in a "wet" condition on subsequent runs and then becomes insensitive to the inlet liquid mercury velocity and the plug insert design, provided the system remains clean and is not contaminated by hydrocarbons.

Subsequent tests conducted at AGN on small-scale, single tubes have furnished much insight into the phenomenon of conditioning and deconditioning. For example, the mix-4P3E (polyphenyl ether) lubricant used in the SNAP-8 was progressively injected into the mercury. In one series of tests, 1 gram, 2 grams, 3 grams, and 6 grams were successively injected. Although the outlet temperature showed fluctuations for a short period of time, it continued to produce fully rated vapor conditions. Once shut down, however, it would not restart and reproduce the last operating point. In another series of tests, a boiler was completely deconditioned by injecting 25 grams of lubricant. When rubidium was progressively added, definite improvement was noted with each successive addition, but this improvement decayed in time between the injections. These tests are still in progress and results to date are not completely understood. However, they are very illuminating and indicate the complexity of the problem.

Based on the NaK profile tests described above and SNAP-2 heat-transfer pressure correlations, the tube-in-tube boiler (Figure 10) was designed and built. The

tube-in-tube boiler consists of seven parallel tubes nested in a single large tube in a pure counter-flow arrangement. The boiler length has been decreased from 60 to 30 ft, based upon extrapolation of the performance on small-scale, single-tube tests. The entry to this boiler is a 5-ft straight section to facilitate changing the inlet plugs should a need for this be demonstrated. This boiler has been installed in the PCS-1. Tests were begun in August. This boiler, incidentally, weighs 480 lb, including liquid inventories, as compared to 850 lb for the tube-in-shell boiler. Three such boilers have been built - one for RPL-2, one for PCS-1, and one for our W-1 loop here at Lewis. Performance has been disappointing.

C. CONDENSER

The SNAP-8 condenser is a NaK-cooled tube-in-shell heat exchanger where the mercury vapor discharging from the turbine is condensed and subcooled. Figure 11 illustrates the design. The mercury vapor flows through the tubes while the NaK is in counter-flow in the shell. The shell is approximately 8 in. in diameter at the vapor inlet and 5 ft long. The unit weighs 120 lb with its nominal fluid inventories. This component is described in greater detail by Kreeger and Hodgson in another paper.⁽⁶⁾

While detailed component tests on the condenser have not been completed, the unit has been tested in conjunction with tests on the turbine-alternator assembly (TAA) with satisfactory results. Figure 12 shows a shell-side temperature profile generated during TAA tests. Profiles such as these are used to evaluate condenser performance with variations in NaK-side and mercury-side conditions.

D. TURBINE-ALTERNATOR ASSEMBLY

The turbine-alternator assembly was designed to produce a nominal electrical output at the alternator terminals of 51.6 kw at system design conditions.

To achieve this, the thermodynamic design of the turbine was based on equilibrium characteristics for mercury, an aerothermodynamic turbine efficiency of 60%, and an overall alternator efficiency of 87%, with system conditions of:

	<u>Inlet</u>	<u>Exit</u>
12,000 rpm speed	1250°F	685°F
11,200 lb/hr flow	265 psia	16.5 psia

The results of analytical studies indicated that the turbine should be an overhung, 4-stage, impulse machine with an equal division of available energy among the stages. The first two stages are split path, partial admission, while the last two stages are full admission. Ball bearings are used.

Turbine-alternator assembly testing in RPL-2, using a prototype boiler and condenser at rated operating conditions, demonstrated the capability of this unit to produce 54 kw. After 830 hours of testing, much of it at over-design flow rate, the turbine failed due to the escape of a locking device from the floating shaft seals (Figure 14). Post-test inspection revealed that the turbine had suffered other detail failures but had continued to operate. These test results will be described by Cherry and Oiye,⁽⁷⁾ so here we will just highlight the findings:

1. The Stellite 6B withstood the erosive effects of the wet mercury very well and had sufficient toughness to digest the numerous pieces of stainless steel which had broken off early in the tests.

2. The Stellite 6B is extremely notch-sensitive and, at the operating temperatures, undergoes a phase transformation which was not anticipated or predicted.

3. Although many cracks occurred in the interstage diaphragms and nozzle block retainer welds, the unit stayed together and operated until the locking ring came out and jammed the first-stage wheel.

The turbine performance resulting from the tests, as shown in Figure 13, indicates that the efficiency is 3% points low. This is due to the conservative design of running clearances and dynamic seal leakage, which can be easily improved by small changes, now that the basic hardware is proven by test.

The alternator (Figure 15) provided by General Electric Company, Erie, Pennsylvania, is a 4-pole, homopolar, inductor type with an ML-type insulation system. Dynamic seals at each ball bearing prevent the lubricant from flooding the rotor cavity. The alternator was designed for a rating of 80 kva at 0.75 pf and 400 cps. Performance test results (Figure 16) indicate that the efficiency is well above design, and the

winding hot-spot temperature is well below the design limit. As a result of the lower winding temperature, the rating can be increased 16%, or the life extended 4.4 times. The alternator weight is 435 lb.

The complete turbine-alternator assembly, including its mounting and instrumentation, weighs 690 lb (Figure 17).

E. NaK PUMP-MOTOR ASSEMBLIES

The SNAP-8 system utilizes an integral pump-motor assembly to circulate NaK liquid through the reactor (primary) loop at 1170°F and through the heat rejection loop at 500°F. These are described in detail by Chalpin, Colker, and Foss⁽⁸⁾ in another paper at this meeting.

Each NaK pump-motor assembly (Figure 18) has an integral assembly, a recirculation pump which circulates the coolant within the motor passages and bearings through a combination heat exchanger/cold trap, economizer, filter, and back to the motor. The weight of the pump and the motor is 169 lb. The complete NaK pump-motor assembly weight, including the pump, motor, recirculation system, and stand, is 321 lb. The same pump-motor assembly, with modified hydraulics, is used for the primary system.

The NaK pump-motor assemblies have successfully undergone bearing performance verification tests in air-motor dynamometer tests, heat-exchanger cold-trap tests, NaK simulation loop testing, and actual PMA operation in NaK at rated temperature and design conditions. The actual performance in NaK is shown in Figure 19. The total operating time in NaK at temperatures over 1100°F is over 1000 hr. More than 1215 total hours at all temperatures have been logged on the 5 different units tested.

F. MERCURY PUMP-MOTOR ASSEMBLY

This unit consists basically of a one-piece solid shaft supported by two angular-contact ball bearings mounted in a central transmission housing. The bearings are lubricated with a polyphenyl ether fluid (mix-4P3E), and mercury sealing is effected by slinger and molecular pump seals. A 400-cycle, 3-phase induction motor is overhung from the bearing housing, at one end, with a combination jet-centrifugal pump overhung on the other end. This unit is described in greater detail by Chalpin, Pope, and Foss⁽⁹⁾ in another paper at this meeting.

Six such pump-motor assemblies have been built to date. The unit, shown in Figure 20 complete with instrumentation used for development testing, has completed 120 hours of operation at 500°F suction temperature. The H/Q data, shown on Figure 21, indicate that predicted performance was exceeded. The cavitating performance (NPSH) indicates a good safety margin compared to the SNAP-8 system requirements.

The motor weight is 39.5 lb. The total weight of the complete pump-motor assembly, including instrumentation, is 186 lb.

G. LUBRICANT-COOLANT PUMP-MOTOR ASSEMBLY

This unit, produced by TRW at Cleveland and shown in Figure 22, consists of a 400-cycle, 3-phase induction motor straddle-mounted between carbon sleeve journal and thrust bearings. The pump impeller and volute housing are overhung from the motor, and the whole unit operates flooded in the pumped medium, which is mix-4P3E.

A total of seven of these pump-motor assemblies have been built, all of which have been acceptance tested. The development unit has completed 2609 hours of operation of which 1018 hours were at rated conditions at the vendor's plant with no measurable signs of wear or degradation. More than 4300 hours of operation on 4 different units have been accumulated (as of 15 September 1965). Figure 23 shows the actual performance curve. The total weight of the pump-motor assembly is 26 lb.

To determine the compatibility for the ML-insulation system with the mix-4P3E fluid, a life test was conducted at VKC for a similar motor running immersed in the same fluid at 250°F. This unit operated continuously for 20,000 hours with no appreciable change in winding resistances or noticeable degradation in lubricant-coolant fluid or insulation.

H. ELECTRICAL CONTROLS

The electrical controls shown in Figure 24 schematically can be divided into two areas, the start system and those components that operate for 10,000 hours. Except for ground testing, restart is not a current requirement. The 10,000-hour components are all hermetically sealed, filled with sulfur hexafluoride (SF_6) and mounted to conduct

their self-generated heat to a cold plate from which the heat is transferred to an actively-cooled heat sink. To achieve a ultra-high reliability, semiconductor diodes are placed in "quad" connections by paralleling resistors such that a failure of one would not prevent proper operation; derating all the components, simple circuits, strict quality control, derating and burn-in, have all been incorporated into the electrical controls.

1. Start System

A 28-volt battery of 300 ampere-hour rating supplies power to the start programmer for power and control during the startup period only.

The start programmer receives the start signal and in the proper sequence: (a) provides battery power to the reactor and the inverter, (b) controls the speed of the inverter from 95 cycles to 220 cycles, (c) transfers the pump-motor assemblies (excluding the mercury pump-motor assembly) from the inverter to the alternator, (d) controls the mercury injection valves, lubricant-coolant and the mercury flow control valves, (e) provides flashing current to the alternator field and biases the saturable reactors in the off position, (f) controls the auxiliary shutoff valve, and (g) at the end of the sequence of operations, closes the vehicle load breaker, which supplies power to the vehicle load. The electrical design of the start programmer utilizes relay logic control employing relays, diodes, resistors, RC timers, a transistor inverter frequency control, and L-C tuned valve and MTC control. The start programmer is located on the PCS, weighs 15 lb, and is designed for a maximum operating temperature of 165°F. The integrated neutron flux during the startup is approximately $10^9 \text{ n/cm}^2(\text{f})$.

The inverter developed by Westinghouse Electric at Lima, Ohio, is shown in Figure 25. At a signal from the start programmer, the inverter output frequency is increased at a maximum rate of 2 cps from the initial 95 to the final 220 cps. The voltage increases with the frequency from an initial value of 20.5 to a final value of 85 volts. This final condition is maintained until the load is transferred to the alternator output. The inverter is rated at 370 watts at 0.3 pf at 95 cps and 1700 watts at 0.28 pf at 220 cps. The unit is helium filled and hermetically sealed. It is provided with aluminum

heat sinks to absorb all its own heat losses without auxiliary cooling. The final weight is 316 lb. Total test time on one inverter is about 80 hours with the longest test period being five hours.

2. Steady-State System - 10,000 Hour Components

a. Voltage Regulator-Exciter

The voltage regulator-exciter consists of a reference circuit, a voltage sensing circuit, single-stage magnetic amplifier, stabilization transformer, and a static exciter (block diagram). The voltage regulator basically is a single-stage magnetic amplifier receiving a signal from a volts-per-cycle reference operating in conjunction with a saturating current potential transformer that regulates the output voltage by providing the correct amount of alternator field excitation power. The voltage regulator-exciter is designed to control the output voltage to 208 volts line-to-line within 3% under all operating conditions and is packaged in two hermetically-sealed enclosures. One enclosure contains the voltage regulator weighing 72 lb and located in the low-temperature controls assembly which is fluid cooled. The other enclosure contains the exciter, weighing 80 lb and located in the transformer-reactor assembly which is also fluid cooled. These are shown in Figure 26.

b. Speed Controller

The speed control consists of a stagger-tuned circuit for the frequency reference, a single-phase single-stage magnetic amplifier, a saturable reactor, and a parasitic load resistor. The speed sensing circuit output signal is amplified by the magnetic amplifier and fed into the control winding of the saturable reactor. The saturable reactor controls the load on the system by turning the parasitic load on and off, as the system demands.

The speed control is housed in four hermetically-sealed modules. The three high-temperature modules located in the transformer-reactor assembly are the saturable reactor (195 lb), the power transformer (10 lb), and the stabilization transformer (10 lb). The one low-temperature controls assembly is the speed control amplifier (30 lb). The assemblies are shown in Figure 27.

c. Parasitic Load Resistor

The parasitic load resistor, which is cooled by primary NaK, is shown in Figure 28. The resistor is three-phase wye connected with six elements in parallel per phase. It is rated at 47 kw total with a power flux of 75 watts per square inch of resistor sheath area. The weight, filled with NaK, is approximately 80 lb.

d. Harness

The harness, like all of the modules, is designed for all welded electrical connections. The control harness contains flexible, corrugated hermetically-sealed stainless steel tubing with mating flanges for the electrical components. When welded to the components, a hermetically-sealed control harness assembly will be completed. The flexible conduit provides protection for the wires while on the ground and also provides a hermetically-sealed environment containing SF_6 for space application. The weight of the harness is estimated to be 65 lb.

III. TEST FACILITIES

A. SINGLE-COMPONENT TEST LOOP

A typical loop used to develop a single component is the liquid NaK loop 3 (LNL-3). Figure 29 shows the facility less the main loop insulation. The control panel, the cover gas control panel and the temperature recorders are mounted outside the test cell. The purpose of the test facility is to provide a means to develop the NaK pump-motor assemblies, conduct an endurance run of 3000 hours at the SNAP-8 system primary loop conditions, obtain the components' thermal map, and to investigate startups and transients.

This is a simple loop in which the PMA under development is the sole means of NaK circulation. All system pressures are visually indicated and all temperatures are printed on strip recorders. The test facility is designed to operate and has been operated unattended. Critical loop functions are wired for automatic shutdown such that the failure mode can be ascertained by tell-tale lights even when no personnel are in

attendance. This has proven to be a relatively reliable test loop, simple to operate, maintain, and modify as required. Through 15 September 1965, the loop has accumulated more than 1600 operating hours.

B. MULTIPLE COMPONENT TEST LOOP

Initially, the test loop for the turbine-alternator assembly was designed solely for that purpose. However, it soon became apparent that, in order to test the turbine on mercury, it would be necessary to develop long-duration test-support equipment, such as pumps, boilers, condensers, valves, etc., which would be almost as complex as the proposed system hardware. In an evolutionary manner, then, this test loop became a multiple component test facility (RPL-2) with the mercury boiler, turbine, condenser and pump all being the final prototype equipment. This facility is shown in Figure 30.

Although much more complex, difficult to maintain and modify, four development units are being tested at once and under conditions more typical of the final arrangement. Flexibility of varying the operating conditions on any one component without violating specified operating conditions on the other is limited. If this facility had been initially designed to test the four major mercury components concurrently, flexibility provisions could have been provided.

The most interesting aspect of rated power loop 2 testing has been the stability exhibited by the components and the control system. From 13 May to 22 June, 837 test hours toward a 1000-hour component demonstration test had been accumulated. It was found that all major components operated together stably and that there was no significant drift. This is illustrated in Figure 31 which shows the output parameters during "hands-off" operations.

C. POWER CONVERSION SYSTEM TEST LOOP

The final major test loop is the one required for the nonnuclear test of the entire power conversion system (PCS-1). Because of the complexity, this first system loop (SL-1) was installed into two phases:

Phase I - Prototype boiler, condenser and mercury pump-motor assembly. The turbine is simulated by a desuperheater, and the NaK pump-motor assemblies by electromagnetic pumps. Starting is effected by test support equipment.

Phase II - The prototype turbine-alternator assembly, NaK pump-motor assemblies, and startup system are added. The only nonprototype elements left are the gas-covered expansion reservoirs and the electrical harness.

As of this time, all Phase I tests have been successfully completed, and the loop is being used to test the tube-in-tube boiler before being modified for Phase II testing.

The piping and instrumentation diagram for the Phase I tests shown in Figure 32 indicates the complexity of the test loop even for the incomplete power conversion system. The PCS-1/SL-1 for this initial series of tests consists of six loops: (1) primary, (2) mercury, (3) heat rejection, (4) radiator simulator, (5) lubricant-coolant, and (6) Dowtherm. A 750-kw gas-fired NaK heater located outside the building serves as the primary heat source in this nonnuclear system test. An electromagnetic pump circulates NaK through this heater and the mercury boiler. The facility and test units during Phase I are shown in Figure 33. It is seen that the PCS-1 is almost lost in the maze of ground test support equipment.

Following the replacement of the simulator by the turbine-alternator assembly and installation of the NaK pump-motor assemblies and the power conversion system electrical controls, full system characteristics will be explored. Testing during this phase is designed to explore system dynamic characteristics, transient response, rated power output and initial startup operations. After a thorough characterization of system behavior, a 10,000-hour endurance run at power will be initiated.

D. COMPARISON OF TEST LOOPS

It should be expected that, as test loops become more complex, their on-stream factor should decrease. However, since more units are under test, the total test time on all units should increase quantitatively as well as become more significant (because of better environmental simulation). Thus, there should be an optimum test loop.

The on-stream factors for the three loops are shown in Figure 34. As expected, LNL-3 has a very high availability whereas RPL-2 and SL-1 do not. Noticeable, however, is the very rapid increase in availability of SL-1. This is believed to reflect the experience obtained on RPL-2.

Since RPL-2 has 5 major system components (boiler, turbine-alternator assembly, condenser, mercury pump-motor assembly, and lubricant-coolant pump-motor assembly) whereas LNL-3 has only two (NaK pump-motor assembly and lubricant-coolant pump-motor assembly), it is apparent that RPL-2 produces more unit test hours monthly than LNL-3. However, the cost of RPL-2 is much greater than that of LNL-3. The relative cost effectiveness of single-purpose and multipurpose loops is still under investigation.

IV. CONCLUSIONS - TECHNICAL STATUS

A. The design and behavior of the power conversion components have received their second* experimental confirmation through operation of the RPL-2 test. Although not arranged in "prototype" configuration, all the major components (which were in prototype configuration) operated together stably and essentially within design requirements.

B. The one significant exception was the boiler, which required the addition of rubidium to the mercury before it would produce the required superheat at design flow rates. Recent laboratory tests with high-velocity plugs in a tube-in-tube configuration however, have provided evidence of a practical solution to this problem without the use of an additive.

C. The total operating time at design conditions on each of these components is shown in Table 1. It can be seen from this that the major components of the power conversion system have had approximately a 1000-hr demonstration.

D. The system design and behavior has had a preliminary experimental confirmation through operation of RPL-2. The speed control subsystem demonstrated $\pm 1/2\%$ frequency limits on the alternator. System pressure variations of $\pm 2\%$ due to boiler performance, although acceptable, should be reduced by the new tube-in-tube boiler design mentioned above.

* All units were bench tested initially to confirm basic design features.

E. The first prototype power conversion system, PCS-1/SL-1, is in fabrication. The initial series of thermal tests using a turbine simulator have been completed. The initial series of performance tests using the complete PCS-1/SL-1 is scheduled for next summer.

Thus, the SNAP-8 Program, like many others, has finished its preliminary phases and is now ready for operations which are in the "proof-of-the-pudding" nature for the complete nonnuclear system.

REFERENCES

1. P. I. Wood, D. L. Forrest, and B. M. Wilner, SNAP-8, The First Electric Propulsion Power System, ARS Paper No. 2050-61, October 9-15, 1961.
2. R. W. Powell, Status of SNAP-8 Electrical Generating System, Presented to AIAA Third Biennial Aerospace Power Conference, Philadelphia, September 1, 1964.
3. R. O. Williams, Jr., and J. H. Roecker, SNAP-8 Experimental Reactor Tests, Paper presented at the AIAA First Rankine Cycle Space Power Systems Specialists Conference, 26-28 October 1965.
4. A. H. Kreeger, A. J. Sellers, and J. N. Hodgson, Development of the SNAP-8 Boiler, Paper presented at AIAA First Rankine Cycle Space Power Systems Specialist Conference, October 26-28, 1965.
5. R. S. Carey, B. E. Farwell, and G. Thur, Effects of Oil Contamination on Heat Transfer Performance of Forced-Convection Mercury Boilers, Paper presented at AIAA First Rankine Cycle Space Power Systems Specialist Conference, October 26-28, 1965.
6. A. H. Kreeger and J. N. Hodgson, Design and Development of the SNAP-8 Multitube Condenser, Paper presented at AIAA First Rankine Cycle Space Power Systems Specialist Conference, October 26-28, 1965.
7. M. G. Cherry and G. Oiyee, Design and Development of a Turbine-Alternator Assembly for the SNAP-8 System, Paper presented at AIAA First Rankine Cycle Space Power Systems Specialist Conference, October 26-28, 1965.

8. E. S. Chalpin, C. P. Colker, and C. L. Foss, Design and Development of Pumps for High-Temperature NaK Service, Paper presented at AIAA First Rankine Cycle Space Power Systems Specialist Conference, October 26-28, 1965.
9. E. S. Chalpin, J. R. Pope, and C. L. Foss, Development of a SNAP-8 Pump for Mercury Service, Paper presented at AIAA First Rankine Cycle Space Power Systems Specialist Conference, October 26-28, 1965.

TABLE 1

CUMULATIVE OPERATING TIMES FOR MAJOR PCS COMPONENTS THROUGH 30 SEPTEMBER 1965

Component	Cumulative Hours	Cumulative Starts
1. Turbine-Alternator Assembly (all)	856	119
RPL-2	820	47
2. NaK Pump-Motor Assemblies (all)	1953	195
LNL-3	1433	71
3. Mercury Pump-Motor Assemblies (all)	1292	149
RPL-2	673	43
SL-1	494	30
4. Lubricant-Coolant Pump-Motor Assemblies (all)	4933	277
LNL-3	1940	86
RPL-2	990	35
SL-1	712	37
5. Boiler, Hg Side (all)	1901	(30)
RPL-2 (T-S)	1407	--
SL-1 (T-S)	226	20
SL-1 (T-T)	268	10
6. Condenser, Hg Loop (all)	1901	(30)
RPL-2	1407	--
SL-1	494	30

Fig. 1 SNAP-8 Electric Generating System Schematic

Fig. 2 Typical Flight Configuration of the SNAP-8 Electrical Generating System

Fig. 3 SNAP-8 Reactor

Fig. 4 SNAP-8 Experimental Reactor Shown Being Installed for Testing

Fig. 5 Ground Prototype System Test Arrangement

Fig. 6 Tube-in-Shell Boiler

Fig. 7 Tube-in-Shell Boiler Conditioning

Fig. 8 Boiler Behavior After Mercury Injection

LIQUID VELOCITY FT/SEC	CURVE NO.	RUN NO.	TIME, HRS	
.80	1	1	10	
	2	1	670	
6.00	3	2	4	
	4	2	20	
	5	3	1-1/2	RESTART
.80	6	4	1	RESTART

Fig. 9 Corrosion Loop 4 Boiler NaK Temperature Profile

Fig. 10 Tube-in-Tube Boiler

Fig. 12 Condenser Temperature Profile

Cutaway Drawing of Condenser Assembly

Fig. 11 Condenser Assembly

Fig. 13 Turbine Performance

Fig. 14 Third-Stage Nozzle Inlet Diaphragm Showing Retaining Ring Partially Out of its Placement Groove

Cutaway Drawing of SNAP-8 Alternator

Fig. 15 SNAP-8 Prototype Alternator

Fig. 16 Alternator Performance Characteristics

Fig. 17 Turbine-Alternator Assembly

Fig. 18 Heat Rejection Loop NaK Pump-Motor Assembly

Fig. 19 485° NaK Test Performance Curve

Fig. 20 Mercury Pump-Motor Assembly

Fig. 21 Performance Characteristics at 500°F - SNAP-8 Mercury Pump-Motor Assembly

Fig. 22 Lubricant-Coolant Pump-Motor Assembly

Fig. 23 Lubricant-Coolant Pump-Motor Assembly Performance Curve

Fig. 25 SNAP-8 Inverter

Fig. 24 Electrical Controls Schematic

Exciter Assembly

Voltage Regulator Assembly

Fig. 26

SPEED CONTROL POWER TRANSFORMER

SPEED CONTROL MODULE

SATURABLE REACTOR ASSEMBLY

Fig. 27

SNAP-8 Speed Control Components

Fig. 29 Single-Component Test Loop - Liquid NaK Loop 3 (LNL-3)

Cutaway of Parasitic Load Resistor

Fig. 28 Parasitic Load Resistor

Primary NaK Loop Piping with Dump Tank in Lower Left and Boiler in Upper Right (During Loop Modifications) (6 November 1964)

Hg Stand with Condenser in Center, HRL Dump Tanks and Valves in Lower Right, and Hg Dump Tank in Lower Center (6 November 1964)

Fig. 30 Multiple-Component Test Loop - Rated Power Loop 2 (RPL-2)

Fig. 31

Hands-Off Operation - Test Results from RPL-2

Fig. 32 Power Conversion System 1,
Phase I Testing Piping and Instrumentation Diagram

Fig. 33 Power Conversion System 1 in the First Integrated System Test Loop - PCS-1/SL-1

Fig. 34 SNAP-8 Test Loops - On-Stream Factor

REACTORS AND SHIELDS

Chairman: D. J. Cockeram

Atomics International
Division of North American Aviation, Inc.
Canoga Park, California

BLANK

AEROSPACE NUCLEAR SAFETY
V. E. Blake, Jr.
Sandia Corporation, Albuquerque, New Mexico

ABSTRACT

This paper gives an overall look at the space nuclear power program, emphasizing the safety problems and approaches. It provides a brief history of the Atomic Energy Commission program to introduce nuclear power supplies. A description is given of the various nuclear systems presently under development. A partial listing of isotope fuels of interest, along with some of their important characteristics, is also provided. Comparisons are drawn between the usefulness of nuclear power, and other forms of space power. A simplified hazard chart is presented showing the many combinations of events that can occur in the use of nuclear power in space. The chart illustrates how a systematic check can be made to assure no hazard path has been overlooked. An example is presented which shows how a significant reduction in calculated hazard was achieved in a nuclear power application by use of this chart and other techniques. A brief study of the hazards caused by meteorites is presented. It is suggested that this natural hazard can provide a basis for assessment of nuclear power for space, although additional work in the area will be required. The paper concludes by challenging industry to take an active interest in the safety of aerospace nuclear power supplies.

In 1958 the United States placed its first satellite in an earth orbit. The signals from this satellite were heard for about two weeks, the limit of the battery supplied power. Since then the United States has put forth a major

This work was supported by the
United States Atomic Energy Commission

effort in space. Satellites weighing many tons have been orbited which have required electrical power plants producing as much as several thousand watts. It became apparent early in the space program that the operation of electrical payloads in space for extended periods of time would dictate that some form of solar or nuclear energy would ultimately be required.

The Atomic Energy Commission, in the early 1950's studied a number of nuclear energy power source concepts from the viewpoint of establishing their feasibility for use in future spacecraft. Both reactors and radioisotope generators were considered. Although ground-based reactors are inherently large and heavy and require massive shielding, it appeared possible that smaller compact versions could be designed for space use. Likewise, the high specific power available from several of the radioisotope materials made it probable that reasonable power levels could be obtained from isotope power sources.

The first space atomic power source was demonstrated in 1959. Called the SNAP-3, this isotopic generator used Polonium-210 as a fuel, weighed four pounds and produced in excess of $2\frac{1}{2}$ watts of power for 90 days. The total power available from this four pound unit was equivalent to nearly one ton of nickel cadmium batteries. In 1961 a version of the SNAP-3 generator fueled with Plutonium-238 was used to power a portion of two navigational satellite systems. One of these nuclear powered systems is still operating. The condition of the other is unknown since it ceased transmitting after eight months of operation, apparently due to an electronic failure.

Starting in 1963 larger plutonium fueled isotope power supplies were flown on three navigational satellites. These power supplies were designed to develop 25 watts for five years. Two of these are still in orbit while a third was destroyed as a result of a failure of the missile to place the satellite in orbit.

This year the first reactor power supply, designated SNAP-10A, was placed in orbit around the earth. This unit produced in excess of 500 watts of electrical power for a period of 43 days. The reactor was then apparently shut down by an onboard sensor which malfunctioned. Until shutdown, the power supply performance exceeded expectations.

These systems are members of a family of atomic power supplies designed for use in space. Figure 1 shows the different systems under development. The left side of the chart shows the higher powered systems which are reactors and include units to produce heat as well as those to produce electricity. The right side of the chart shows the lower powered systems, again divided between those to produce heat and those to produce electricity. Figure 2 gives the characterization of the various systems including weight, fuel, and power level. Figure 3 is a partial listing of isotope fuels of interest along with some of their important characteristics.

There is some hazard associated with the use of radioactive materials as an energy source. This is also true of other useful energy sources, like steam or gasoline. The public has learned to accept certain safety precautions to live with these more common energy sources, and a similar pattern will evolve with radioactive materials.

The hazards with radioactive materials can arise from the close association between the material and man, or his environment. The hazard can result from inhalation, ingestion, or from direct radiation. To use nuclear power in space, the power supply designer must take positive steps to keep from involving man with the material in a manner which can be harmful.

The ability of any nation to successfully pursue the exploration of space is most certainly governed by the amount of electrical energy which can be carried into space. Figure 4 shows a spectrum of power levels versus lifetime for various space systems.

The higher levels of power can be met only by use of reactors. Clearly, there is no competition with atomic energy in this area. In the high intermediate levels, solar dynamic systems, isotope dynamic systems, or reactor systems, all using rotating machinery for energy conversion, can meet the power levels. The low intermediate power levels can be handled by either solar static or isotope static systems.

Considering the broad spectrum of capability, nuclear energy will certainly prove to be indispensable in space. Safety probably represents the greatest drawback to the extensive use of nuclear power in space thus far. Safety is not something that just happens. It must be carefully thought out and positive steps taken to achieve it. There are many ways of buying safety. It is the responsibility of the designers of nuclear space power systems to find and employ the best ways.

The next chart, Figure 5, is shown to present in the form of a multiple path chart, a simplified version of the overall aerospace nuclear safety problem. Given a launch, the chart shows that the flight can result in one of three things. Either the missile achieves a successful orbit, or a short orbit, or it aborts. By definition, all possible eventualities of an attempted launch can be made to fit into one of the blocks. In other words, the sum of the probabilities in this row is one. Since the chart is a simplification, the actual number of blocks in this row can be as many as desired as long as the sum of probabilities totals one. The second row shows what can happen as a result of reentry. Again, every eventuality is covered in these three blocks. The third row shows the form the resulting exposure may take. The final row is a safety index which consists of three items to be evaluated; the number of people involved, the probability of exposure, the severity of exposure. More will be said later about this safety index. This chart is used to show that each system can be described in a systematic manner which will cover all the possible consequences of a flight. For most systems, the number of paths will be more than

represented by this array. It is important to note that the probability of exposure is never zero along any path so that some evaluation will always be necessary. For example, a power supply designed to burnup during reentry has some finite possibility of returning intact and it may in fact develop that the greatest potential hazard exists along this unexpected path. Conversely, a power supply designed to return intact has some probability of burning up.

When the critical path or paths are found, then attention can be given to these areas to try to reduce the hazard by reducing the number of people involved, the severity of the exposure, or the probability of occurrence (the three items in the safety index). A plutonium-238 power supply was analyzed recently which illustrates how this evaluation plan can work. A systematic analysis of a similar chart revealed that all the important hazard paths occurred in the short orbit path chain. The critical hazard could occur if incomplete burnup produced particles that would fallout over a limited area producing an inhalation hazard. The probability of a short orbit occurring was calculated to be .03. The study further showed that almost all short orbit situations resulted from failure of the second stage engine to restart to circularize the orbit. A Hohmann transfer orbit was being used for the mission where the spacecraft is first injected into an elliptical transfer orbit with a low perigee and a 700 NM apogee. On the first pass through apogee the second stage engine is restarted for a short period of time circularizing the orbit. Failure to restart results in an orbit of only a few months lifetime.

It is also possible to achieve a 700 NM circular orbit by flying a ballistic path in place of the elliptical transfer orbit. In this case the missile flight profile is altered so that a ballistic path is flown with a maximum altitude of 700 NM. The second stage engine is restarted at this altitude and burned until circular orbit velocity is achieved. If the second stage engine fails to restart, the

spacecraft simply falls back to earth. The impact point for the mission can be selected so that the radioactive material, if it fails to completely burn up will fall harmlessly over the ocean. By using this ballistic ascent into orbit, the probability of a short orbit is reduced from .03 to .0004 or a decrease of about two orders of magnitude. Since the short orbit was the cause of the greatest hazard, the probability of any hazard occurring was reduced by a factor of one hundred. The ballistic ascent to orbit is not achieved without penalty, however. On this flight it required about 500 pounds of additional fuel on the second stage to fly the altered mission profile since this profile is a less efficient way of achieving orbit. However, the actual mission was not penalized since the missile had a payload capability of over 3000 pounds and the payload for the proposed flight weighed only 1600 pounds. In other words, at no cost to the mission the probability of a hazard was reduced by a factor of over a hundred.

Another possibility for improved safety would be to decrease the number of people who could be exposed by intentionally designing the power supply to reenter intact, rather than burning up. By using this design approach, the number of people exposed is significantly reduced; however, the level of exposure is, at the same time, raised. A logical question to be asked is, how does one compare these two quite different hazard potentials to determine which is the more acceptable. Unfortunately, an answer to this question does not now exist. However, nature has provided an everyday hazard which may be used to gain some insight into the problem. This is the natural hazard arising from the random reentry of meteorites.

Figure 6 is a set of curves relating to iron meteorites showing their size, frequency of arrival, and resulting lethal area. It can be seen, for example, that five meteorites weighing ten tons each reenter the earth's atmosphere each year. Picking another size, it can be seen that a meteorite weighing 100,000 tons can be expected to strike the earth at a rate of about 5×10^{-4} per year or one every 2000

years. The well known crater in Arizona was made by a meteorite of this size. The resulting crater was 1.2 km in diameter. It was estimated that all life within a diameter of 10 km was destroyed as a result of the impact. The newspapers recently noted that an asteroid named Ictharus will pass within $4\frac{1}{2}$ million miles of the earth in the summer of 1968. This asteroid, which revisits the earth every 19 years, is about one mile across and will weigh in the vicinity of 10 billion tons. The chart shows that the crater from this meteorite, should it ever strike the earth, would be perhaps 100 km in diameter while the lethal area would cover a diameter in the range of 1000 km. There are two craters in Africa that are 25 and 40 km in diameter, providing evidence that large meteorites do strike the earth. A book on the subject of meteorites states that evidence indicates the earth has been bombarded throughout geological time by meteorites and there is no reason to believe it will not continue.

From these curves, and the population distribution on the earth, the curve shown in Figure 7 was derived. It shows the relation between the number of people that could be killed from a given size meteorite striking the earth and the probability of that event happening. It shows, for example, that the probability of one death per year is about one in 300 while the probability of as many as eight million people being killed is about one in one billion. In spite of the fact that about ten meteorites strike the surface of the earth each day, there apparently has never been a death attributed to a meteorite. An individual's personal hazard is remote, being about one chance in 10^{-12} of being struck in any given year.

Referring back to the last row of the multiple path chart, the three items suggested for consideration in the safety index, were the number of people involved, the probability of exposure, and severity of exposure. As can be seen, Figure 7 provides a form of a safety index guide for the most severe possible exposure, from a naturally occurring hazard. It seems reasonable to assume that this could also prove to be an acceptable guideline for a man-made hazard. It should be possible to draw

other curves on this chart which can represent other levels of exposure (severe injury, slight injury, 100 rem, 25 rem, maximum permissible body burden, etc). If the curves for other levels of exposure can be added, the chart could form a basis for safety design as well as evaluation.

The future of atomic power in space is clear. Without it our ability to successfully explore and scientifically examine space will be severely limited. There are hazards associated with these radioactive power systems. However, the designer, by a systematic analysis, can identify the areas of significant hazard. Once identified, positive design action can be taken to reduce these hazards to acceptable levels. Industry should not delegate this responsibility to government, but should take the initiative and lead the way by providing the safest possible systems for nuclear power in space.

SPACE NUCLEAR POWER SYSTEMS FIG. 1

SNAP, POODLE AND ROVER DATA

SNAP	WEIGHT (lbs)	APPROXIMATE SIZE O. D. x HEIGHT(in.)	FUEL	POWER OUTPUT	LIFE/MISSION TIME
2	1470	----	U-235	3 KWe	1 YEAR
3	4	5 x 6	Po-210/Pu-238	3 We	90 DAYS
8	1500-4500	30 x 50	U-235	30/60 KWe	1 YEAR
9A	27	20 x 10	Pu-238	25 We	5 YEARS
10A	1000	60 x 120	U-235	500 We	1 YEAR
11	30	20 x 12	Cm-242	25 We	90 DAYS
17	30	10 x 36	Sr-90	25-30 We	5 YEARS
19	30	20 x 11	Pu-238	30 We	5 YEARS
25	38	----	Pu-238	75 We	5 YEARS
27	30	17 x 18	Pu-238	50 We	5 YEARS
POODLE	30	4 x 17	Po-210	5 KWT	----
ROVER/ NERVA	8500	52 x 96	UC ₂	1000MWt	----

NUCLEAR POWER SYSTEM CHARACTERISTICS FIG. 2

ISOTOPIC POWER DATA

ISOTOPE	POWER DENSITY WATTS/GM	RADIATION	HALF LIFE YEARS	SHIELDING
PLUTONIUM 238	0.56	ALPHA	89.0	MINOR
STRONTIUM 90	0.90	BETA	28.0	HEAVY
CERIUM 144	25.6	BETA-GAMMA	0.78	HEAVY
CURIUM 242	120.0	ALPHA	0.45	MINOR
CURIUM 244	2.8	ALPHA	18.0	MODERATE
POLONIUM 210	141.0	ALPHA	0.38	MINOR

SANDIA CORPORATION

ISOTOPE FUEL CHARACTERISTICS FIG. 3

SPACE POWER SYSTEM CHARACTERISTICS FIG. 4

AEROSPACE SAFETY MULTIPLE PATH CHART FIG. 5

IRON METEORITE CHARACTERISTICS

FIG. 6

METEORITE HAZARD EXPECTANCY

FIG. 7

CONF-651026-2
28: 5183

THE SNAP SHIELDING PROGRAM AT ORNL*

C. E. Clifford and J. Lewin
Oak Ridge National Laboratory
Oak Ridge, Tennessee

ABSTRACT

The space reactor shielding program at Oak Ridge National Laboratory is concerned with the development of advanced techniques for calculating accurately the intensity of the radiation transmitted through shields designed for both manned and unmanned missions, and also for calculating, by iterative processes, a minimum-weight design for these shields within the radiation level constraints for a given mission. The development program includes improving and coupling Monte Carlo and multigroup (S_n) transport codes, as well as performing experimental work to provide a test of the accuracy of the transport codes. Experimental results to be obtained include simple geometry neutron scattering experiments, studies of the deep penetration of neutrons in slabs, and an engineering test utilizing a SNAP reactor and shield. A significant portion of the theoretical work is in the design and error evaluation of the more complicated experiments. The complex geometry of shields for space systems and the need for a high level of confidence in the predicted radiation intensities in space systems requires the extension of shielding technology to the rigorous Monte Carlo techniques made possible by the advancement in computer size and speed.

The SNAP shielding program at Oak Ridge National Laboratory is only one phase of a program of theoretical analysis, calculation, and experimentation related to the radiation shielding problem. Obviously, shielding requirements have become more stringent and exacting as nuclear energy has been applied to mobile systems. As part of the departed but not forgotten ANP program, the Tower Shielding Facility was constructed at Oak Ridge to make possible in-air large-scale experiments in support of the shielding program. Despite the demise of the ANP program, a great deal of valuable technology, including shielding data, was generated. The Tower Shielding Facility has since then been

*Research sponsored by the U. S. Atomic Energy Commission under contract with the Union Carbide Corporation.

used in checking calculations for vehicle shielding and in establishing the shielding effectiveness of various structures. It will in the near future be used to verify experimentally the most advanced calculational methods for predicting neutron transport in lithium hydride shields. A hydride moderated SNAP reactor, currently available from Atomics International will be used as the neutron source, but the results will contribute to the perfection of methods for calculating the shielding configurations for all types of SNAP reactors. The general aims of the SNAP shielding program are the development of accurate techniques for calculation of the dose delivered through and around the shield of any SNAP reactor, and optimization of shield shape and weight.

The Tower Shielding Facility at ORNL is shown in Fig. 1. Control room and counting rooms are shielded by concrete and earth fill. The towers are about 300 feet high and can elevate a load of 55 tons in the plane of the two nearer towers to a height of about 125 feet. The reactor shown here and in Fig. 2 is the third reactor to be used here and, as the first two reactors, has a water-moderated and cooled core. Figure 3 shows the flexibility of experimental arrangements achievable with this installation. The reactor is now enclosed in a collimator so that a beam of radiation may be directed over a range of directions. The nearer sphere is a 10-ft detector collimator, shown in Fig. 4, that, when filled with lead and water, attenuates neutron dose by a factor of 10^5 and gamma dose by a factor of 10^6 , thus permitting discrimination of spatial distribution of incoming radiation. This facility then we plan to use for some work with a SNAP reactor, which will be described later.

Is shielding research really necessary? In 1957 a shielded reactor was flown in an airplane and measured neutron dose rates were found to be within a factor of 10 of calculated predictions in most cases. The simplest answer is that factors of 10 are not good enough in space applications. However, in addition there is the problem of uncertainty, for some calculations, especially when departure is made from the simplest geometries, often differ widely from measurements. The dose rate at any given point within a shielded volume consists of several components each of which is contributed by a distinct set of physical processes. These component parts of the total dose rate can vary sharply with change of geometry, materials, and radiation source. As a matter of fact, knowledge of the physical

Fig. 1. Tower Shielding Facility at ORNL.

Fig. 2. Tower Shielding Reactor II in Spectrum Modifying Shield.

Fig. 3. TSR-II Reactor in Source Beam Collimator.

Fig. 4. TSF Detector Collimator.

processes themselves is constantly being supplemented; that is, interaction cross sections are still under investigation.

In the current SNAP system the principal concern, so far as shielding is concerned, has been the protection of instruments from radiation damage. The damage threshold in instrument payloads is usually reached earlier in the course of accumulation of neutron dose than of gamma-ray dose. For this reason, most attention has been focused on the neutron component of radiation emanating from the reactor core.

The shielding material that gives the lowest weight shield for neutrons is lithium hydride. It is relatively light, having a fabricated density of 0.74 to 0.77 at room temperature and a macroscopic cross section of about 0.15 per cm when the thicknesses considered are not over 2 ft. A few years ago the value of the various microscopic cross sections that make up the average value of macroscopic cross sections were very much in doubt. Some cross sections had just been declassified, and to verify calculations in which geometry played a minimum role, some experiments on the transmission of neutrons through slabs of lithium hydride were conducted at the Tower Shielding Facility. The 12-ft diameter spherical collimator was used to obtain beams of neutrons 6 in. in diameter and 15 in. in diameter that were directed through slabs of lithium hydride of various thicknesses. Calculations were done using the Monte Carlo technique as incorporated in the O5R code developed at Oak Ridge. The results are shown in Fig. 5 and were interpreted as confirming the relative accuracy of cross sections and the adequacy of the number of histories selected for calculation.

This was one of the first tests of the Monte Carlo method code, known as O5R for shielding, at the Oak Ridge National Laboratory. Monte Carlo is the most promising method presently available for calculation of radiation transmission through material, in three dimensions, and it is the only method that is not limited by geometry of the source-shield combination. However, there are still problems in using this method in that good accuracy of dose prediction requires a sufficient number of histories to be run with useful biasing techniques along with an evaluation of the statistical error inherent in the calculation. Also consistent errors in programming are not difficult to make and very difficult to discover, once made. The problem of

transmission through slabs, for example, required many hours of machine time on a CDC-1604 computer to evaluate biasing techniques and to reduce the statistical deviation for 25,000 neutron histories.

There are, of course, other less meticulous methods of calculation. Some of them are comparison methods in which attenuation from a point source to a point receiver is written as a function of distance on the basis of data from experiments. To this class, with various refinements, belong the various kernel methods which yield good results so long as the geometry and materials involved in the actual case are not very different from the experiments on which the kernel is based. Some others, such as the S_n method, are purely mathematical and solve the Boltzmann transport equation with various mathematical representations of the angular flux distribution and scattering cross section variation. Dr. Cain and others at ORNL are at present developing these methods into two dimensional capabilities and are working toward a coupling of the O5R Monte Carlo code with S_n codes in order to decrease the machine time required to obtain the source term for secondary gamma-ray generation in the shield. This will be of greatest interest in manned systems.

A Monte Carlo code for gamma-ray transport, called OGRE, has been developed at Oak Ridge and in combination with O5R will be used to obtain secondary gamma-ray doses for manned systems.

The dose predictions for the SNAPSHOT experiments — the orbited SNAP-10A reactor of Atomics International — were made by the point-kernel method. However, the dose at the payload plane was not dominated by the neutrons that penetrated the shield but by the neutrons scattered around the shield by structural components of the system. Even so, the predictions obtained were quite good for a small unmanned system, varying from 10% below the measured dose to 300% above. In the past, calculations of dose rates for the ANP program using kernel methods yielded results that were from 50% too high to a factor of 2.7 too low.

The scattering of neutrons by structure was illustrated at ORNL in a series of experiments utilizing the TSR-II reactor. Figure 6 shows the scheme of the experiments. The truncated cone below the TSR-II core is a water tank, representing a SNAP system

Fig. 5. Fast Neutron Dose Rate 16.5 ft from Source for 6-in. Beam and 18.5 ft from Source for 15-in. Beam.

Fig. 6. Scatter Shield - Configuration II; Neutron Scattering Experiment.

shield. A beryllium block, simulating a SNAP reactor control drum is mounted adjacent to the reactor core as shown in Fig. 7. In addition to the beryllium, a small cylinder of nickel was mounted above. The beryllium could be swung away from the reactor vessel so as to come out farther from the shield shadow cone, as shown in Fig. 8.

The detector collimator traversed horizontally from the center line of the shield to about 8 in. to the right of the edge of the shield. In other words, it moved from the shadow of the shield out to where it looked directly at the beryllium. Some of the results are shown in Fig. 9. Positions 0, 1, 2, and 3 are positions of the beryllium progressively farther out from the reactor vessel. The rise in dose rate clearly indicated that the directions of neutrons are changed in collision with beryllium atoms and increase the dose received even well within the shadow of the shield. Figure 10 shows the same results but with tighter collimation, 3-in. collimator as compared to an 8-in. collimator in the previous results. The total dose rate per watt of reactor power is down as would be expected, but the change in dose rate with beryllium position is greater. Figure 11 shows the increments of dose for each position of the beryllium blocks. The background has been subtracted so that these dose rates represent increments due to scattering only. The effect of the very small, 1-in. diameter by 1-in. high nickel cylinder is clearly visible above background. These experiments gave an indication of the total effect to be expected from structural scattering and in the SNAPSHOT flight the scattering effect was the dominant dose contributor over most of the dose plane.

Over the next couple of years the program at ORNL is oriented toward development of advanced mathematical techniques for calculating accurately neutron transport through shields and for calculating by iterative processes a minimum weight design for a given mission profile. The program includes improving the O5R code, including amplification of the cross-section library and coupling of this code with the S_n code to achieve shorter machine time for a given calculation. The program also envisages development of a two-dimensional S_n code and coupling of both O5R and OGRE codes to the S_n codes for rapid calculation of secondary gamma-ray generation and the resultant dose contribution.

Fig. 7. TSR-II with Simulated SNAP Shield and Beryllium Control Drum.

Fig. 8. Beryllium Block Positions for Neutron Scattering Experiment.

Note: Numbers in circles correspond to "position numbers" of data plots. Dimensions are in inches.

Fig. 9. Control Drum Mockup Experiment - Dose Rate (Long Counter Data Normalized to FND Dose) of Fast Neutrons Scattered from Beryllium Block.

Fig. 10. Control Drum Mockup Experiment - Dose Rate of Fast Neutrons Scattered by Beryllium Block 3-in. Collimator.

First problems to be undertaken by these mathematical programs will be related to a SNAP 2/10A system and to provide experimental verification of the calculations, a SNAP reactor will be installed at the Tower Shielding Facility. The experimental program will be divided into three major parts. First will be the mapping of the leakage flux from the core as shown in Fig. 12.

The reactor will be suspended on a lead screw at the end of a boom fixed to a rotatable column. In the position shown in Fig. 12, with the reactor at not over one watt power, a collimator for a neutron spectrometer will rotate about the core while maintaining its focus on one point in the reactor. This procedure will yield a source spectrum for use in subsequent calculations.

The second part of the program is represented by Fig. 13. Here the dose and spectrum transmitted through a lithium hydride shield will be mapped. The detector collimator will be placed in a dry chamber that will be shielded from air-scattered and structure-scattered radiation by concrete cover slabs. The shield itself will be surrounded by a polyethylene and iron collar to exclude air-scattered neutrons which could be ten times the transmitted dose if the collar were not used. The design of this collar and the validity of this experimental configuration is at present the subject of both theoretical and experimental study.

It is, of course, recognized that the collar itself introduces a source of error into the experiment, for neutrons that in a space environment would leave the system can be scattered back toward the detector by the collar. A calculation was done about two years ago in the geometry shown in Fig. 14. The shield and a collar also made of lithium hydride were assumed to be in a vacuum and the collimator was assumed to be perfect: that is, only those neutrons that penetrated or were scattered into the collimator angle reached the detector. The source for this Monte Carlo calculation was assumed to be a disk, and the shield was taken to be 50.8 g/cm^2 thick. The ratio of dose received at the detector position with collar to dose received with shield only was calculated for three thicknesses of collar, 38.8, 44.8, and 50.8 g/cm^2 . The corresponding results were 1.73, 1.20, and 1.15. From this, it was concluded that a meaningful experiment of this type could be designed.

Fig. 11. Control Drum Mockup Experiment - Dose Increments from Scattered Neutrons.

Fig. 12. Leakage Spectrum Mapping for SNAP-10A Reactor Core.

Fig. 13. Transmitted Dose Mapping for SNAP-10A Reactor Core - SNAP 2 Development Shield.

Fig. 14. Shield-Collar Calculation Model.

In practice we could not use a lithium-hydride collar, but a laminated collar made basically of iron and polyethylene is practical, and some calculations of such a configuration, as shown in Fig. 15, have been made. By changing the order and arrangement of iron and polyethylene, the contribution from the collar was reduced to 16% of the total dose. Final design calculations, utilizing a volume source that is a good representation of a SNAP-2 reactor are now in progress and we feel that the collar will be effective in preventing distortion of shield performance in space by the earth-bound environment. However, an experiment utilizing the TSR-II will be conducted shortly to verify this. The configuration is shown in Figs. 16 and 17.

It is recognized that there are other sources of error in the experiment: leakage through the collimator, air scattering in the collimation cone, collimator material scattering, surrounding ground scattering. By careful design it is felt that the sum of all the perturbations can be reduced to acceptable levels and that neutron physical dose can be measured to an accuracy of 30 to 50%.

The third phase of the program will be concerned with structural scattering contributions as shown in Fig. 18. A view of the overall facility for the reactor installation is shown in Fig. 19, and a general view of the reactor is shown in Fig. 20. The reactor has been designed and is being built by Atomics International. It is hoped that the installation will be flexible enough to allow some extension of the program into thicker shields and eventually to allow verification of shield shaping calculations that may in the future contribute to weight saving in manned as well as unmanned space systems.

Fig. 17. ORNL SNAP Shielding Program; Collar Evaluation Experiment.

Fig. 18. Neutron Scattering Evaluation for SNAP-10A Reactor Structure.

Fig. 19. Facilities Arrangement.

Fig. 20. SNAP Reactor for TSF Installation.

LIQUID METAL PUMPS

Chairman: A. G. Grindell

Oak Ridge National Laboratory
Oak Ridge, Tennessee

DEVELOPMENT OF A SNAP-8 PUMP FOR MERCURY SERVICE*

by

E. S. Chalpin, J. R. Pope, C. L. Foss
SNAP-8 Division
Von Karman Center
Aerojet-General Corporation
Azusa, California

ABSTRACT

A mercury boiler feed-pump-motor assembly was designed, built, and tested for the SNAP-8, 35-kilowatt mercury Rankine-cycle, nuclear turboelectric space power system. The design incorporates, on a single shaft, a centrifugal pump and an ML-insulated, conventional 400 cps, 3-phase induction motor. The rotor is supported by angular-contact ball bearings of M-50 material. Polyphenyl ether (mix-4P3E) fluid is circulated through the motor housing, space-seal heat exchanger, and bearings for lubrication and cooling. A visco pump and a molecular pump are used in series to minimize the loss of mercury to space. A slinger and a molecular pump in series are used to reduce leakage of mix-4P3E to space and to preclude leakage of the lubricant to the motor cavity during operation. Carbon-face static seals are incorporated to minimize leakage of the process fluids while the assembly was shut down, and to preclude interchange of the fluids. A liftoff device is incorporated to lift the carbon seals from the rotating shaft during operation.

A jet-pump is incorporated upstream of the centrifugal pump to provide positive suction head to the centrifugal impeller at very-low-flow start conditions. The pump-motor assembly is all-welded for positive sealing.

Over 1,000 hours of testing have established the design performance and integrity of the pump-motor assembly.

Basic problems resolved were a space-seal system to minimize loss of process fluids to space, special bearings designed for 10,000-hr operation, an electrical insulation of polyimide (ML) material for resistance to mix-4P3E at an operating temperature of 250°F,

*Information appearing in this paper is based on work performed under National Aeronautics and Space Administration Contract NAS 5-417.

and resistance to a radiation environment in space. Instrumentation for accurately measuring NPSH was also a basic problem.

I. INTRODUCTION

The mercury pump-motor assembly (PMA) was designed and developed for pumping mercury in the SNAP-8 35-kw nuclear turbo-electric space power system. One pump is required per system as shown on the SNAP-8 system schematic diagram, Figure 1.

Tested performance characteristics of the PMA are shown in Table 1. Operational envelopes shown in Figures 2, 3 and 4 describe the areas in which the assembly can operate for an expected life of 10,000 hours, with no detrimental effects.

II. DESCRIPTION OF THE PUMP-MOTOR ASSEMBLY

The pump-motor assembly is a self-contained, hermetically-sealed unit (with the exception of the seal-to-space vent ports) incorporating on a single shaft a centrifugal pump, low-leakage dynamic seals, an induction motor and angular contact ball bearings lubricated with a polyphenyl ether (mix-4P3E), which is also used as a motor and space seal coolant. Integral with the assembly are a jet booster pump and lubricant-coolant valves with associated plumbing. A photo of the PMA is shown in Figure 5. Figure 6 is a cross section of the pump and motor. The main features of the assembly are described below.

A. JET-CENTRIFUGAL PUMP

The jet-centrifugal pump combination includes a jet nozzle placed upstream of the centrifugal impeller which is of the semiopen-vane type incorporating 90° back vanes for hydraulic axial thrust balance. Approximately 45% of the centrifugal pump flow is directed through the jet pump.

The purpose of the jet pump is to suppress cavitation during PMA startup when system available net positive suction head (NPSH) is 0.16 ft at 0.65 gpm. Even though this extremely low NPSH requirement took precedence over efficiency considerations, the

combination jet-centrifugal pump at a specific speed of 430 attained an efficiency of 30% at the design point. To obtain the NPSH value, the jet pump is operated toward shutoff from its best efficiency point. The efficiency of the centrifugal pump with back vanes was 50% at the design point.

A partial volute-circular casing housing was used to minimize pump radial imbalance over a wide range of pumping conditions.

Both the centrifugal pump and its housing were designed using empirical relationships used for water pumps which are well established in industry. Special attention was given to the choice of materials for mercury service and relatively large impeller-to-housing clearances were used, because of the low specific speed of the pump, without significant detriment.

B. MOTOR

The 3-phase, 208 volt, 400 cycle squirrel-cage induction motor was designed for reliability as well as efficiency. A scavenging slinger is employed to maintain an evacuated stator cavity to decrease power consumption and excessive winding temperatures.

The problem of compatibility between the polyphenyl ether (mix-4P3E) lubricant-coolant and its long-life reliability with the motor insulation system was solved by using insulation materials impregnated with an aromatic polyimide polymer (ML).

The motor is fitted with hermetically-sealed, ceramic electrical and thermocouple terminals.

C. BEARINGS AND LUBRICATION SYSTEM

The PMA rotating assembly is supported by two angular contact bearings using a conventional spring-loaded back-to-back mounting arrangement. Material used for the bearing balls, outer and inner races is triple vacuum melt, M-50 tool steel, the retainer being a one-piece design of iron-silicon-bronze material piloted on the outer race. The bearing material selection was made primarily for the 10,000-hour life requirements at 99.5% reliability.

The lubricant-coolant (mix-4P3E) was chosen for its radiation resistance. This fluid is used for three purposes. First, it lubricates the bearings. Secondly, it acts as a coolant for the motor stator. Thirdly, it acts as a coolant in the heat exchanger surrounding the mercury seal-to-space system.

A lubricant-coolant inlet valve, outlet valve, filter and associated plumbing complete the system.

D. LOW-LEAKAGE SEALS

Since mercury is the fluid pumped and mix-4P3E is the fluid used for lubrication, it follows that a means of keeping these fluids apart had to be devised. Also, a means of preventing both fluids from escaping to a space environment had to be developed. The visco-pump/molecular-pump combination satisfies both conditions. Because of the 10,000-hour design-life criterion, it was evident that face-type seals were totally unacceptable. The visco-pump is used on the mercury side only. It consists of a helical screw designed to pump the mercury back into the pump and maintain a liquid-to-vapor interface between the visco pump and the molecular pump. The visco pump is surrounded by a mix-4P3E heat exchanger which assures a 300°F mercury temperature at the interface. The molecular pump is similar in design to the visco pump, its function being to pump mercury vapors, which are in the molecular flow regime, back to the visco pump liquid-to-vapor interface.

On the lubricant side, a dynamic slinger is used to develop a stable liquid-to-vapor interface, and a molecular pump, similar to that used on the mercury side, restricts losses in the molecular regime. The unique feature of these designs is that they minimize leakage of process fluids to space to an expected 1.76 lb/10,000 hours for mercury and 0.081 lb/10,000 hours for mix-4P3E.

Two very comprehensive papers⁽¹⁾⁽²⁾ have been given that fully describe the visco pump and molecular pump theoretical and design philosophies.

Superscript numbers in parentheses refer to similarly numbered references listed following the text.

Since the dynamic seals do not function when the PMA is shut down and are ineffective at speeds below 6000 rpm, a static seal system had to be incorporated. This was done by using two bellows-type carbon face seals, sealing against opposite surfaces of a rotating ring. If these seals were allowed to rub continuously during operation, they would increase the PMA power losses and would wear out rapidly. Therefore, a bellows device was provided to lift off the seals at 6000 rpm PMA speed. The bellows are ported to the discharge of the mercury pump. As the pump discharge pressure increases beyond 200 psia, the device "lifts-off" both seals from their mating rotating face. At PMA shutdown, the seals reseal and prevent process fluids from escaping.

III. OTHER SIGNIFICANT PUMP-MOTOR ASSEMBLY DESIGN FEATURES

The materials used in the construction of the PMA are shown in Table 2. Materials in direct contact with mercury were chosen for their resistance to attack by the liquid metal as well as their physical properties in a high-temperature and creep environment.

The PMA was designed for applications or upgrading of the SNAP-8 system which might require increases of 11% in pump capacity and 21% in head rise above existing design requirements as can be seen in Figure 7.

The PMA requires only electrical power and a supply of lubricant coolant with no maintenance over an expected life of 10,000 hours.

IV. DEVELOPMENT TEST PROGRAMS

Various test programs were completed to obtain feasibility, performance and reliability data.

A bearing test rig is shown in Figure 8. The purpose of testing was to obtain basic data on the bearings and lubricant. Figure 9 shows some representative results of the testing.

A seal test rig is shown in Figure 10. The purpose of testing was to verify analytical theory and demonstrate the capability of the visco pump/molecular pump to meet the leakage limits noted above.

A motor insulation test rig operating in polyphenyl ether is shown in Figure 11. An ML insulated motor was operated in mix-4P3E for 20,000 hours at temperatures of 250°F and 300°F to determine the rate of degradation of the insulation, if any. No perceptable degradation was noted.

A jet-pump test program was completed to obtain data on the feasibility and performance of the design. Water and mercury were used in this test series. Figure 12 shows the test set-up.

A liquid mercury loop was designed and built as shown in Figure 13. The facility incorporated a mercury loop capable of attaining 600°F temperature, a lubricant-coolant loop and a vacuum system to simulate a space environment at the PMA seal-to-space port. Such parameters as head-capacity, cavitation, thermal mapping, lubricant-coolant pressure drop and heat balance, power losses determination and pump pressure profile mapping to ascertain pump radial and thrust forces over the full range of head and flow were established. The head vs capacity performance is plotted in Figure 7, and the cavitating performance is shown in Figure 14. The PMA tested had acquired 120 hours and 72 starts with no problems other than moderate cavitation damage of the centrifugal impeller in over 30 hours of extensive cavitating operation. However, SNAP-8 system design requirements call for only a maximum of 10 minutes in a cavitating regime, during the start cycle.

A second PMA is currently installed in a rated-power test loop and has operated in excess of 680 hours with no problem other than a leak in the static seal "lift-off" bellows which was attributed to a poor bellows convolute weld.

A third PMA is currently installed in a prototype power conversion system test facility. It has operated in excess of 200 hours with no problems.

V. PROBLEMS ENCOUNTERED AND RESOLVED DURING TEST

As noted previously, the problem of the failure in the lift-off bellows occurred on two occasions. Both units exhibited failed bellows welds. After the first failure, it was deduced that the bellows were subjected to "water hammer" when the PMA was operated in a "dead headed" condition during rated-power loop testing. A needle valve was placed

between the pump discharge and the lift-off bellows for the second test series. The pump was then run as before but the needle valve was kept closed until full PMA speed was attained. The needle valve was then opened. The lift-off bellows failed again, indicating that the pump discharge pressure of 490 psia was excessive for the design. As an interim measure, the bellows are being actuated with an auxiliary gas source to a pressure of 200 psia.

Other design concepts utilizing a diaphragm between the pump and the lift-off device, a method of actuation employing the centrifugal or rotation effects of the PMA, temperature, electrical power or valves are being considered.

VI. CONCLUSIONS

The mercury pump-motor assembly developed for the SNAP-8 program is a proven component except for the startup seal lift-off device. However, this problem is not complex.

The PMA is unique in that it can operate at extremely low suction pressures with no deleterious effects. It can also operate in an outer space environment for 10,000 hours with extremely small leakage of the system fluids.

REFERENCES

1. R. L. Lessley and J. N. Hodgson, Low Leakage Dynamic Seal to Space, ASME Paper 65-GTP-14.
2. J. N. Hodgson, Designing a Molecular Pump as a Seal to Space, ASME Paper 65-GTP-15.

TABLE 1
MERCURY PUMP-MOTOR ASSEMBLY PERFORMANCE DATA

<u>Components and Parameters</u>	<u>Design Point</u>	<u>Test Results</u>
<u>Pump Jet/Centrifugal</u>		
Flow, lb/hr	11,500	11,500
Speed, rpm	7,800	7,830
Inlet temperature, °F	505	505
Inlet pressure, psia	10.5	10.5
Pressure rise (ΔP), psi	390	487
Head, ft	69.5	87
Hydraulic power, kw	0.30	0.38
Efficiency, %	22.0	29.0
<u>Total Hydraulic Losses, Watts</u>	1,190	1,150
<u>Motor Scavenge Slinger Loss, Watts</u>		650
<u>Motor-Induction Type</u>		
Total electrical losses, watts	280	428
Motor electrical efficiency, %	90.2	87.8
Motor Efficiency, % (electrical + hydraulic)	48.3	36.8
Input power, kw	2.84	3.53*
Power factor	0.68	0.77
Overall PMA efficiency, %	10.55	10.75

* Additional power required by motor power scavenging slinger not in original design concept.

TABLE 2
MERCURY PUMP MOTOR ASSEMBLY
LIST OF MAJOR MATERIALS

Shaft	AISI 4340	Alloy Steel
Dynamic Seals	AISI 4340	Alloy Steel
Bearings	M50-TVM	Tool Steel
Bearing Housing	9Cr-1Mo	Alloy Steel
Pump Housing	9Cr-1Mo	Alloy Steel
Motor Housing	9Cr-1Mo	Alloy Steel
Pump Jet Nozzle	9Cr-1Mo	Alloy Steel
Pump Suction and Discharge Lines	9Cr-1Mo	Alloy Steel
Pump Impeller	9Cr-1Mo	Alloy Steel
Motor Stator Laminations	M19	Silicon Steel
Motor Rotor Laminations	M19	Silicon Steel
Rotor Conductor Bars	Silver	
Motor Stator Windings	Copper	
Motor Insulation System	Polyimide "MI"	Organic

Fig. 1 SNAP-8 System Loop Schematic Showing Location (Crosshatching) of the Mercury Pump-Motor Assembly

Fig. 2 Operational Envelope Showing Head vs Capacity of the SNAP-8 Mercury Pump-Motor Assembly

Fig. 3 Operational Envelope Showing v/f vs Frequency of the SNAP-8 Mercury Pump-Motor Assembly

Fig. 4 Operational Envelope Showing NPSH vs Capacity of the SNAP-8 Mercury Pump-Motor Assembly

Fig. 5 SNAP-8 Mercury Pump-Moter Assembly

Fig. 6 Cutaway Drawing Showing Design Details of the SNAP-8 Mercury Pump-Motor Assembly

Fig. 7
Performance Characteristics
at 500°F - SNAP-8 Mercury
Pump-Motor Assembly

Fig. 8 Bearing Test Fixture - SNAP-8 Mercury Pump-Motor Assembly

Fig. 9 Average Steady-State Temperature vs Inlet Flow Rate - SNAP-8 Mercury Pump-Motor Assembly Bearing Testing

Fig. 10 Seal-to-Space Test Fixture Used During Development Test of the SNAP-8 Mercury Pump-Motor Assembly

Fig. 11 Motor Insulation System Testing. Motor is Operating Submerged
in Polyphenyl Ether

Fig. 12 Jet-Pump Test Fixture - SNAP-8 Mercury Pump-Motor Assembly
Development Tests

Fig. 13 Instrumented SNAP-8 Mercury Pump-Motor Assembly Installed in a Liquid-Mercury Test Loop During Development Testing

Fig. 14 SNAP-8 Mercury Pump- Motor Assembly Performance During Development Tests in Liquid-Mercury Loop

DESIGN AND DEVELOPMENT OF PUMPS FOR HIGH-TEMPERATURE NaK SERVICE*

by

E. S. Chalpin, C. P. Colker, and C. L. Foss
SNAP-8 Division
Von Karman Center
Aerojet-General Corporation
Azusa, California

ABSTRACT

The SNAP-8 35-kilowatt nuclear turboelectric space power system uses a sodium-potassium alloy (eutectic NaK-78) to transmit heat to and from the mercury Rankine-cycle loop. Two pump-motor assemblies of like configuration were design, built, and tested for service in the NaK heat-input loop and in the heat-rejection loop. The units are self-contained, all-welded, and hermetically sealed. They incorporate on a single shaft a centrifugal pump, a hermetically-sealed (canned) drive motor, an internal NaK lubricant-coolant circulating pump, and NaK-lubricated bearings. Included with the pump-motor assembly are a heat exchanger-cold trap combination utilizing a polyphenyl ether fluid as the coolant, a NaK economizer, and a NaK filter. The NaK lubricant-coolant circuit is independent of the process pumping circuit, attained through the use of a shaft-to-housing annulus. No static or dynamic seals are used. The bearings are of the tilting-pad journal and thrust design.

Over 1800 hours of testing have established the design performance and integrity of the pump-motor assembly. Basic problems resolved were positive containment of NaK without leakage, operable bearings utilizing NaK as the lubricant-coolant, and a motor which is operable in a space and radiation environment.

Plots of H-Q, efficiency, power, and NPSH are presented. A comparison to show correlation of performance data obtained using water and NaK is also presented.

* Information appearing in this paper is based on work performed under National Aeronautics and Space Administration Contract NAS 5-417.

I. INTRODUCTION

The NaK pump-motor assembly (PMA) was designed and developed for pumping a sodium-potassium liquid metal (eutectic NaK-78) in the SNAP-8 35-kw nuclear turboelectric space power system. Two pump-motor assemblies are required in the SNAP-8 system - one in the reactor loop and the second in the heat rejection loop. The schematic diagram, Figure 1, of the SNAP-8 system shows the locations of the PMAs. The PMAs are identical in design, thus increasing reliability and decreasing system complexity.

The tested performance characteristics of each application of the PMAs are shown in Tables 1 and 2. Operational envelopes shown in Figures 2, 3, and 4 describe the areas in which the assembly can operate for an expected life of 10,000 hours with no detrimental effects. The design point is also shown on the plots.

II. DESCRIPTION OF THE PUMP-MOTOR ASSEMBLY

The pump-motor assembly is a self-contained, hermetically-sealed unit incorporating on a single shaft a centrifugal pump, a hermetically-sealed motor, an internal lubricant/coolant recirculation pump, and NaK-lubricated bearings. Integral with the assembly is the recirculation loop containing a "cold-trap" system, heat exchangers, and a NaK filter.

Since assembly design life of 10,000 hours with no leakage of NaK is a criterion, no static or dynamic shaft seals are used. Isolation between the main-loop NaK and the recirculation loop NaK is achieved by the incorporation of a close clearance annulus about the shaft between the pump and the motor. The unique features of this design concept are that a high thermal gradient exists between the pump and the motor; therefore the recirculation fluid remains relatively uncontaminated by the main-loop fluid, which can contain a much higher oxide level than that tolerated in the bearings and motor. Any migration of oxides through the annulus is "trapped out" by the recirculation loop. A photo of the PMA, including its recirculation system, is shown in Figure 5. Figure 6 is a cross section of the pump and motor. The main features of the assembly are described below.

A. CENTRIFUGAL PUMP

The centrifugal pump was designed using empirical relationships for water pumps that are well established in industry. Special attention was given to the choice of

materials and relatively large impeller-to-housing clearances because of the high-operating temperatures (500°F and 1160°F). The impeller is of the semiopen vane type incorporating 90° back vanes for hydraulic axial thrust balance. The 1600 specific speed pump has an efficiency of 68.5%, which is slightly lower than that expected for this design because of the large impeller-to-housing clearances.

A two-splitter-volute housing design was used to minimize radial imbalance over a wide range of pumping conditions. Fabrication of the housing consisted of three-dimensional machining of two halves, then welding them together at the volute flow passage centerline. Since the annular section between the housing and the motor is small in diameter (thus acting as a thermal barrier), three load-support arms were designed to carry the thermal and system line loads. The support arms are attached to mating arms from the pump housing by means of "slip-fit" pins. The pins allow relative thermal expansion but resist the moments and loads. This feature is very important when one considers that the pump housing can operate at 1160°F while the motor normally operates at 350°F .

B. MOTOR

The 3-phase, 208-volt, 400-cycle squirrel-cage induction motor was designed for reliability as well as efficiency. Since NaK was used as the lubricant-coolant, the rotor and stator were canned with thin cylinders of Inconel.

Inorganic insulation (ceramic) was used providing maximum life and reliability in a radiation environment as well as permitting the motor to operate satisfactorily at temperatures up to 600°F , even for the 10,000-hour expected life.

Since the stator can is pressure-formed into the ceramic core, the stator housing is evacuated and hermetically sealed to prevent reverse pressure gradients which could result in buckling of the can.

The motor is fitted with hermetically-sealed ceramic-insulated electrical and thermocouple terminals.

C. BEARINGS

The rotor shaft is supported on two tilting-pad journal-bearing assemblies of four pads per bearing. This design was chosen because of its ability to accept

misalignment, its inherent stability, and its low frictional characteristics. Conventional design philosophy was used with the exception that metal facing involving plating or standard bearing material was not employed.

The thrust bearing consists of a rotating thrust plate or "runner" with a thrust bearing assembly on each side of the runner to support axial loads in either direction. Each bearing consists of six pads pivoted on spherical segments, that in turn are supported by a pair of gimbal plates. The combination of tilting pads, spherical pivots and gimbal plates allows the accommodation of a high degree of misalignment that may result from manufacturing tolerances. The thrust Runner is "coned" such that the apex is on the shaft centerline and the pads are flat. This was done for ease of manufacture and to prevent "wringing" of the pads to the thrust runner, especially during shutdown in the presence of thin layers of NaK oxides on the surfaces.

D. RECIRCULATION LOOP COMPONENTS

The recirculation loop is quite simple in design, yet vitally important to the reliability and life of the assembly. The system consists of a small, full-open centrifugal pump mounted on the main shaft at the rear of the motor, a combination heat exchanger and cold trap, an economizer, a filter and associated plumbing. The heat exchanger cold trap combination is a canister whose center section is packed with a stainless steel mesh to trap out oxides and an outer annular chamber through which the SNAP-8 lubricant/coolant, polyphenyl ether, passes. As the NaK passes through the mesh and is cooled, the NaK oxides come out of solution and deposit on the mesh, thus maintaining a 20 ppm purity desired for the bearings.

The economizer is a counter-flow device that transfers the heat from the incoming NaK to the outgoing NaK from the "cold trap." This assures that the filter will not act as a cold trap.

The filter is constructed of sintered stainless steel that has a 5-micron rating. The purpose of this device is to filter foreign particles other than NaK oxides that pass through the cold trap.

The recirculation flow is 1.6 gpm. The temperature of the NaK entering the motor assembly is 300°F and 350°F leaving. The cold-trap temperature is maintained at 250°F.

III. OTHER SIGNIFICANT PUMP-MOTOR ASSEMBLY DESIGN FEATURES

The materials used in the construction of the PMA are shown in Table 3. Materials used in direct contact with NaK were chosen for their resistance to attack by the liquid metal as well as for their physical properties in a high-temperature and creep environment.

The PMA was designed for applications or upgrading of the SNAP-8 system which might require increases of 10% in pump capacity and 20% in head rise above existing requirements, as can be seen in Figure 2.

The PMA can be used for pumping fluids or liquid metals other than NaK with little or no modifications.

Since the PMA is an integral, hermetically-sealed system, it requires only electrical power and a supply of coolant for the NaK cold trap. If the fluid pumped and used for the recirculation system does not require cold trapping and if a source for radiating the motor losses is available, then electrical power leads will remain as the only interface.

IV. DEVELOPMENT TEST PROGRAMS

Development testing of the PMA has been accomplished in water and NaK. Preliminary tests were made using inhibited water to prove out the design concepts of the pump and the motor. Water was used because of the ease of handling and inspecting the components and the ease of relating the data to expected performance in NaK. This approach accelerated the program since care required in sealing a water system was not nearly that required for a NaK system.

A facility was built to test the PMA with NaK at SNAP-8 system temperatures of 500°F and 1160°F. Such tests as head-capacity, cavitation, thermal mapping, starts and stops, speed-torque confirmation, heating and cooling rates, varying system contamination levels, maximum motor temperature tests and endurance were performed.

Five PMAs have been tested to date. The last unit is still operating for a projected 3,000 hours of endurance at 1170°F.

Performance curves plotted from data obtained in water and NaK are shown in Figures 7, 8, and 9, respectively.

V. PROBLEMS ENCOUNTERED AND RESOLVED DURING TEST

The pump housing of the PMA was originally a Type 347 stainless steel casting. It was found acceptable by X-ray, dye penetrant, pressure test, and helium leak test. However, for 3 cases in which castings were tested in NaK at temperatures between 500°F and 1170°F for 30 to 70 hours, they began to leak. Post-test inspection revealed that the NaK had permeated areas of microporosity that were not evident during inspection. It was concluded that castings of 347 stainless steel were unacceptable for NaK service, especially at temperatures where leaching would be accelerated. The solution was to use a wrought 304 stainless steel material (plate stock). As noted previously, the change to this material necessitated a 3-dimensional machining technique and welding of the two halves at the volute radial centerline.

The thrust bearing runner and pads were originally made extremely flat and highly polished. Testing in water indicated that "wringing" resulted to such an extent that motor torque was insufficient to overcome this force. Also, this condition caused further locking in the gimbal joints in that the convex portion of the gimbal would ride up on its mating concave portion - similar to the "Sprag clutch" principle. Resolution of the bearing problem was obtained by "coning" the thrust runner such that the apex of the cone was on the shaft centerline as noted previously. This approach results in line contact between the runner and the pad instead of surface contact. A thorough analysis was made to determine if "coning" the runner had the load-carrying capacity that could be obtained by the conventional method of "crowning" the pads or shoes. It was found that "coning" the runner was equivalent to the latter method when the existing shoe design was used. Another important factor that influenced the design choice was the overwhelming saving in cost and time over modifying the individual pads.

The "locking" of the gimbals was easily solved by inserting a pin in each joint, allowing freedom of gimbal but restricting relative rotation to 0.002 inch.

The effect of bearing materials becoming oxide free in a NaK reducing atmosphere and thereby resulting in an increase of the friction coefficient has not been fully explored. Testing to date indicates there is some increase in break-away torque but not sufficient to cause detrimental effects in the operation of the PMA. More testing is required to analyze time, temperature, and rate of torque increase.

Fretting of the bearing pad pivots has not been detected to date and will have to be evaluated following much longer periods of operation since this is a time-dependent phenomenon.

No doubt, anyone who has worked with NaK systems has been plagued with leaks and problems associated with cleaning and welding NaK-contaminated materials. Positive, reliable techniques still remain elusive, especially in complex or intricate configurations.

VI. CONCLUSIONS

The integral NaK pump-motor assembly and recirculation system developed for the SNAP-8 program is a proven component (better than 1800 hours of operation to date). It is unique in that it is completely self contained, needs no other motor lubricant/coolant than that which is being pumped, can be operated at motor temperatures as high as 600°F, and will function reliably over a wide performance range.

TABLE 1

PRIMARY LOOP NaK PUMP-MOTOR ASSEMBLY PERFORMANCE DATA

Components and Parameters	Design Point	Test Results
<u>Pump, Centrifugal</u>		
Flow, lb/hr	45,900	45,900
Speed, rpm	5800	5815
Inlet temp, °F	1100	1110
Density, lb/ft ³	45.8	45.8
Inlet pressure, psia	28.6	30.8 ⁽¹⁾
Pressure rise (ΔP), psi	31.5	28.3
Head, ft	99	89 71.6 ⁽¹⁾
Outlet pressure, psia	60.1	59.1 ⁽¹⁾ 53.6 ⁽¹⁾
Hydraulic power, hp	2.29	2.06
Hydraulic power, kw	1.71	1.54
Pump efficiency, %	61	64.2
Shaft input power, hp	3.77	3.21
Shaft input power, kw	2.81	2.39
<u>Motor, Induction</u>		
Motor elec. efficiency, %	70.2	74.4
Motor overall efficiency, %	61.8	53
Input power, hp	6.1	6.06
Input power, kw	4.71	4.52
Power factor	0.53	0.555
PMA overall efficiency, %	36.3	34.1
SUMMARY OF LOSSES:		
<u>Electrical (watts)</u>		
Stator iron	199	110
Stator copper	273	312
Rotor copper	111	102
Rotor can	16	20
Stator can	329	158
NaK eddy	344	389*
Stray load	133	65
Total electrical losses (watts)	1405	1156
<u>Mechanical (watts)</u>		
Rotor hydraulic	290	--
Bearings	95	--
Auxiliary pump	105	--
Total mechanical losses (watts)	490	975
Heat from pump to motor	130	130*
TOTAL HEAT LOSSES TO COOLANT (WATTS)	2025	2261

* Calculated values.

(1) System requirements.

TABLE 2

HEAT REJECTION LOOP NaK PUMP-MOTOR ASSEMBLY PERFORMANCE DATA

Components and Parameters	Design Point	Test Results
<u>Pump, Centrifugal</u>		
Flow, lb/hr	40,200	40,200
Speed, rpm	5800	5810
Inlet temperature, °F	495	495
Density, lb/ft ³	50.8	50.8
Inlet pressure, psia	15.0	15.0
Pressure rise (ΔP), psi	41.8	39.8 [†]
Head, ft	118	113 [†]
Outlet pressure, psia	56.8	55.5
Hydraulic power, hp	2.40	2.30
Hydraulic power, kw	1.79	1.715
Efficiency, %	63.4	68.5
Shaft input power, hp	3.79	3.36
Shaft input power, kw	2.83	2.51
<u>Motor, Induction</u>		
Motor elec. efficiency, %	70.4	75.5
Motor overall efficiency, %	60.0	54.6
Input power, hp	6.33	6.17
Input power, kw	4.72	4.6
Power factor	.53	0.555
Overall PMA efficiency, %	38.0	37.3
SUMMARY OF LOSSES:		
<u>Electrical (watts)</u>		
Stator iron	199	112
Stator copper	273	323
Rotor copper	111	110
Rotor can	16	20
Stator can	329	154
NaK eddy	344	351*
Stray load	133	60
Total electrical losses (watts)	1405	1130
<u>Mechanical (watts)</u>		
Rotor hydraulic	290	--
Bearings	95	--
Auxiliary pump	105	--
Total mechanical losses (watts)	490	960
Heat from pump to motor	30	30*
TOTAL HEAT LOSSES TO COOLANT (WATTS)	1925	2120

* Calculated values.

† Trimmed impeller.

TABLE 3

SNAP-8 NaK PUMP-MOTOR ASSEMBLY MATERIALS

<u>Component</u>	<u>Material</u>
Motor housing	9Cr-1Mo
Pump housing	304 SS
Journal bearings	
Sleeves	Tungsten T-1
Pads	Moly-Tung M-2
Thrust bearing	Carboloy 44
Impeller	410 SS
Shaft	9Cr-1Mo
Rotor and stator cans	Inconel 600

Fig. 1 Diagram of SNAP-8 Electrical Generating System Showing Location (Cross Hatching) of the NaK Pump-Motor Assemblies

Fig. 2
Operational Envelope -
Head vs Capacity -
SNAP-8 NaK
Pump-Motor Assembly

Fig. 3 Operational Envelope - Net Positive Suction Pressure vs Capacity - SNAP-8 NaK Pump-Motor Assembly

Fig. 4 Operational Envelope - v/f vs Frequency - SNAP-8 NaK Pump-Motor Assembly

Fig. 5 SNAP-8 NaK Pump-Motor Assembly in Foreground Showing the Heat-Exchanger Cold Trap (Upper Right), Economizer (U-shaped Tube), and Line Filter Center

Fig. 6 Cross-Section Drawing of the SNAP-8 NaK Pump-Motor Assembly

Fig. 7 Performance Curve for SNAP-8 NaK Pump-Motor Assembly Operating in 495°F NaK (Extrapolated from Water Tests)

Fig. 8 Performance Curve for SNAP-8 NaK Pump-Motor Assembly Operating in 1170°F NaK

Fig. 9 Performance Curve for SNAP-8 NaK Pump-Motor Assembly Operating in 495°F NaK

ELECTROMAGNETIC PUMPS FOR SPACE POWER SYSTEMS*

R. G. Rhudy and J. P. Verkamp

General Electric Company

ABSTRACT

This paper constitutes a broad appraisal of the suitability of electromagnetic (EM) pumps for space power application. A general expression for EM pump efficiency is developed in terms of pump design parameters, fluid properties, and frequency of the power utilized. Although approximate, this expression applies to a wide variety of pump types. It is used to indicate parameter ranges promising for the design of pumps of high efficiency. Based on preliminary designs for space power application, performance characteristics and weight of EM pumps of several types are tabulated. The weights of these and other preliminary pump designs for space power application are compared to those of EM pumps which have operated satisfactorily in service. A specific pump design for space power application is described in detail. EM pumps presently operating with high reliability in facility service are discussed. It is concluded that, in EM pumps designed for space power application, it is feasible to achieve pump weights an order of magnitude less than those of contemporary facility pumps for similar hydraulic requirements. Potential reliability is excellent.

I INTRODUCTION

Present concepts of Rankine cycle space power systems require pumps for transferring liquid metals in closed loops. Both mechanical and electromagnetic pumps have been considered for this function. In mechanical pumps, mechanical power, typically in the form of an impeller rotating in the fluid or a piston moving in a cylinder filled with fluid, is converted to hydraulic power. In electromagnetic (EM) pumps, the conversion is from electrical power to hydraulic power, without intermediate stages. EM pumps, in common with d-c motors, induction motors, and many other electromechanical devices, operate by virtue of the Ampere force, in which a current carrying conductor in a magnetic field experiences a force. In an EM pump, the fluid pumped constitutes the conductor, hence such pumps are

*Work supported by the National Aeronautics and Space Administration as part of contract NAS3-2543.

limited in practical application to liquid metals. The force on the fluid in an EM pump is a body force, the development of which does not require moving boundaries of the pumping region. Thus EM pumps may be completely free of moving parts - bearings, seals, valves, etc. It is the outstanding reliability made possible by the absence of moving parts which makes EM pumps attractive for space power applications.

Inasmuch as the force on the fluid in an EM pump is proportional to the product of the current and the magnetic field in the fluid, the pump output may be controlled by varying the current or the field.

II TYPES OF EM PUMPS

As indicated above, electromagnetic pumping requires the establishment of an electric current and a magnetic field in the fluid. The electric current may be conducted into the fluid by way of electrodes or it may be induced in the fluid by means of a time varying magnetic field. Accordingly, EM pumps are classified as conduction pumps or induction pumps. This is the primary basis of the classification scheme shown in Fig. 1. Conduction pumps may utilize either direct or alternating current. The magnetic field in the pumping region may be direct or alternating but must be the same as the current supplied to the fluid. In induction pumps, both the magnetic field and the current in the fluid are alternating. The alternating magnetic field may be produced by alternating current in windings on a static stator structure or by moving magnets, which may be either permanent or direct current excited electromagnets. In those types of EM pumps where moving parts are not required, there is wide freedom of choice in shape and arrangement of pump ducts. A variety of flat and cylindrical configurations have evolved.

III EM PUMP EFFICIENCY

Since all EM pumps operate by virtue of the Ampere force, interesting and useful generalizations may be made despite the variety in appearance and arrangement. In the following development, pump efficiency is related to the characteristics of the fluid, the pump design, and the frequency of the power utilized by the pump. The effects of duct curvature, hydraulic loss, iron loss, and losses due to end and side effects are neglected. These assumptions are discussed further in the references. The nomenclature follows.

Nomenclature:

A	Sheet current density
B	Magnetic induction
C	Constant
P	Developed pressure
Q	Flow rate
V	Voltage
W	Power
Y	Duct parameter

a, b, c	duct dimensions (Fig. 2)
f	frequency
g	magnetic gap
j	current density
k ₁	winding factor
r ₁	winding resistance, specific
s	slip
t	duct wall thickness
v	fluid velocity

η	efficiency
λ	pole pitch
μ	magnetic permeability
ρ	electrical resistivity

Subscripts:

d	duct
f	fluid
o	output
s	synchronous
1	pertaining to winding
m	magnetizing

For the elementary pump duct shown in Fig. 2, in consistent units, the following relationships apply

$$P = Bjc \quad (1)$$

$$Q = vab \quad (2)$$

The voltage applied across the duct is

$$V = (j\rho_f + Bv)b \quad (3)$$

Defining synchronous speed, v_s , as the fluid velocity at which the current density in the fluid becomes zero,

$$V = Bv_s b \quad (4)$$

The current density in the fluid may be written, from equations (3) and (4),

$$j = \frac{Bv_s s}{\rho_f} \quad (5)$$

where slip, s , is defined as

$$s = \frac{v_s - v}{v_s} \quad (6)$$

The current density in the duct walls is

$$j_d = \frac{Bv_s}{\rho_d} \quad (7)$$

Power output, power loss in the fluid, and power loss in the duct are given by

$$W_o = PQ = \frac{B^2 v_s^2}{\rho_f} \left(\frac{s}{1-s} \right) (abc) \quad (8)$$

$$W_f = j^2 \rho_f (abc) = \frac{B^2 v_s^2}{\rho_f} \left(\frac{s}{1-s} \right)^2 (abc) \quad (9)$$

$$W_d = j_d^2 \rho_d (2tbc) = W_f \frac{Y}{s^2} \quad (10)$$

where

$$Y = \frac{2t}{a} \frac{\rho_f}{\rho_d} \quad (11)$$

The i^2r loss in the pump winding is developed for a distributed winding which provides both excitation and armature reaction compensation. This arrangement is characteristic of induction pumps. Conduction pumps are frequently uncompensated, and compensated conduction pumps usually have concentrated exciting windings and distributed compensating windings. For this reason, the development should be regarded as approximate, particularly in application to conduction pumps. Ideally, the compensating current distribution is the same as the sum of the current distributions in the fluid and the duct walls. Thus, where k_1 is the pitch and distribution factor for the winding ($k_1 < 1.0$),

$$A_1 = \frac{1}{k_1} (a j + 2t j_d) \quad (12)$$

$$A_1 = \frac{1}{k_1} \frac{Bv}{\rho_f} \left(\frac{sa}{1-s} \right) \left(1 + \frac{Y}{s} \right) \quad (13)$$

where A_1 is the sheet current density (as amperes/inch) in the compensating winding. Similarly, where λ is the pole pitch of the exciting winding and μ the permeability of the magnetic gap, including the fluid,

$$A_m = \frac{\pi g}{\mu k_1 \lambda} B \quad (14)$$

where A_m is the sheet current density in the exciting winding.

Since A_1 and A_m are 90 electrical degrees out of phase, the total winding loss, W_1 , is proportional to the sum of their squares. Thus

$$W_1 = r_1 (A_1^2 + A_m^2) (bc) \quad (15)$$

where r_1 is the electrical resistance of the winding per unit area of stator surface.

The efficiency of the pump is

$$\eta = \frac{W_o}{W_o + W_d + W_f + W_1} \quad (16)$$

Substituting equations (9), (10), and (15) in equation (16), and simplifying, yields

$$\eta = \frac{s(1-s)}{(s+Y) + \frac{ar_1}{\rho_f k_1^2} \left[(s+Y)^2 + \left(\frac{2\pi \rho_f f g}{\mu v^2 a} \right)^2 (1-s)^4 \right]} \quad (17)$$

$$\text{or} \quad \eta = \frac{s(1-s)}{(s+Y) + C_1 [(s+Y)^2 + C_2(1-s)^4]} \quad (18)$$

$$\text{where} \quad C_1 = \frac{ar_1}{\rho_f k_1^2} \quad (19)$$

$$\text{and} \quad C_2 = \left(\frac{2\pi\rho_f fg}{\mu v^2 a} \right)^2 \quad (20)$$

It is useful to examine this expression for the limiting case when

$$C_1 \ll 1.0; \quad C_1 C_2 \ll 1.0 \quad (21)$$

This represents the case of negligible power loss in the pump windings.

$$\text{Then} \quad \eta = \frac{s(1-s)}{s+Y} \quad (22)$$

It may be shown that, for this limiting case, the maximum efficiency occurs at a slip

$$s = Y \left[\sqrt{1 + \frac{1}{Y}} - 1 \right] \quad (23)$$

and is given by

$$\eta = 1 - 2Y \left[\sqrt{1 + \frac{1}{Y}} - 1 \right] \quad (24)$$

This efficiency is shown as a function of the duct parameter Y in Fig. 3. Since the effect of non-zero C_1 and C_2 is to increase the denominator in equation (18), it is apparent that Fig. 3 gives the upper limit of the efficiency as a function of the duct parameter Y .

These qualitative observations may be made from inspection of equation (17)

- 1) Efficiency is improved by an increase in the fluid velocity, v .
(Hydraulic losses, neglected in this development, limit the range over which this is true.)
- 2) Efficiency is improved by a reduction in the ratio of the magnetic gap to the fluid height, $\frac{g}{a}$.
- 3) Efficiency is improved by a reduction in the frequency of the power supply.

In an induction pump of reasonable proportions, frequency and fluid velocity are inter-related so that it is not possible to vary each independently of the other over a wide range. Thus, observations (1) and (3) must be qualified at extremes of frequency and

velocity. Nevertheless, detailed design studies bear out these observations, and frequencies in the order of thousands of cycles per second are quite unattractive for EM pump supply.

IV PUMP DESIGNS FOR SPACE APPLICATION

Detailed design calculations have been made for various pump ratings considered typical for a Rankine cycle space power system of the 500-1000 kw level. The basic hydraulic requirements are shown in Table I.

TABLE I
TYPICAL PUMP RATINGS FOR SPACE POWER APPLICATION

Pump No.	Pump Application	Fluid	Temp., F	Flow Rate, GPM	Developed Pressure, PSI
1	Condensate Boost	K	1200	16	30
2	Condensate	K	1300	96	240
3	Radiator Coolant	NaK	1200	60	25
4	Radiator Coolant	Li	1200	31	20

Preliminary pump designs were made for these ratings for d-c conduction pumps and for one or more induction pump types. By a series of design modifications, configurations which gave minimum overall system weight for the assumed weight penalty schedule were evolved. These weight penalties and the system weight contributions associated with the pump ratings are given in detail in the references. The generalizations drawn from this evaluation and comparison are:

1. D-c conduction pumps permit minimum system weight.
2. Pump specific weight decreases with increasing pump output.
3. Radiator coolant systems using lithium are significantly lighter than those using NaK.
4. Minimum pump system weight is not necessarily associated with minimum pump weight.

Although offering the highest efficiency and lowest weight of all EM pump types, d-c conduction pumps have several undesirable features. Because they require very high current at very low voltage, they are unattractive system components from a power conditioning viewpoint. The electrode to pump duct joint region and the insulating flow splitters required in pump entrance and exit transition passages pose difficult design problems and a threat to reliability.

Among induction pumps, moving magnet and polyphase induction pumps are similar in principle, differing only in the means by which excitation is provided to the fluid. Although moving magnet pumps may have excitation systems somewhat more efficient than those of polyphase induction pumps, the additional problems associated with rotation make them less attractive in space power applications. Polyphase induction pumps require frequencies in the order of a hundred cps and moderate voltages. Power conditioning is relatively easy. Pump ducts are not solidly connected to other pump parts and may be symmetrical and relatively free of stress concentrations. The most difficult problems appear to be associated with insulation system design and cooling of the windings. These problems are simplified greatly when the winding is enclosed in an hermetic envelope and charged with an inert gas.

Performance for various types of EM pumps designed for the applications given in Table I is summarized in Table II.

V EVOLUTION OF EM PUMP WEIGHT

It has been observed that, in EM pumps designed on a consistent basis*, a fairly constant correlation exists between the weights of the various pumps and a parameter defined by the product of design values of flow rate, developed pressure, and fluid electrical resistivity. This parameter has been designated the pump capability parameter. In Fig. 4 the ratio of pump weight to this parameter is plotted against the parameter for thirteen induction pumps designed during the past fifteen years and for several designs developed for space power application. The specific pump ratings shown in Fig. 4 are indicated in Table III.

*In this context, the term "consistent basis" encompasses many areas, including relative emphasis on cost, performance, size, and weight; anticipated production volume; fluid and structural material properties; and various design margins.

TABLE II

CALCULATED EM PUMP CHARACTERISTICS

<u>Pump No.*</u>	<u>Type**</u>	<u>Frequency</u>	<u>Efficiency</u>	<u>Power Factor</u>	<u>Weight</u>
1 a	HI	100	0.1	0.49	33
	HI	200	0.1	0.43	26
	HI	400	0.07	0.46	22
	b DC	—	0.14	—	9
2 c	HI	60	0.22	0.52	410
3 d	AI	50	0.13	0.43	120
	AI	100	0.15	0.39	52
	AI	200	0.12	0.39	38
	AI	400	0.1	0.31	41
	HI	50	0.13	0.41	42
	HI	100	0.12	0.38	66
	HI	200	0.09	0.46	51
	HI	400	0.06	0.49	58
	e DC	—	0.15	—	19
4 f	HI	100	0.13	0.55	33
	HI	200	0.11	0.53	28
	HI	400	0.07	0.47	26
	g DC	—	0.15	—	13

* The number designations are consistent with those of Table I

The letter designations are consistent with those of Table III and Fig. 4.

** HI Helical Induction

AI Annular Induction

DC DC Conduction

Data points 1 through 13 represent actual pumps, many of which have operated tens of thousands of hours with virtually no maintenance and with outstanding reliability. Data points a through g represent recent designs (see Table II) for space power application.

The data in Fig. 4 show that the weight of EM pumps designed specifically for space power application may be less than one tenth that of EM pumps developed for conventional applications. This weight reduction has been achieved by: 1) virtually eliminating the pump frame, 2) using somewhat higher current and flux densities in the active parts of the pump, and 3) using a magnetic flux return path inside the pump duct. To achieve low weight of the frame, cylindrical configurations have been preferred because they provide optimum utilization of structural material for rigidity and pressure containment.

The observed correlation of pump weight with pump capability parameter may be rationalized for induction pumps in the following manner. In a particular pumping configuration with a magnetic field of fixed peak amplitude moving at a fixed synchronous velocity, a conducting fluid passing through the pump duct at a fixed velocity less than that of the moving magnetic field experiences a pressure rise inversely proportional to its electrical resistivity. Accordingly, since the flow is the same for all fluids for the assumed conditions, the product of flow rate, pressure, and electrical resistivity is constant. Thus the pumping configuration has a capability related to this parameter, and the weight of a pump is a function of this parameter. This analysis neglects significant variables such as fluid temperature, density, and viscosity. It should be employed with discretion, particularly in comparing pumps for fluids with widely different characteristics or pumps designed for completely different applications or environments.

VI DESCRIPTION OF HELICAL INDUCTION PUMP

Figure 5 is a cross-sectional outline drawing of a Helical Induction pump for the condensate boost application, pump #1a, Table II. The duct is AISI type 316 stainless steel. The duct assembly comprises an outer cylinder, an intermediate cylinder in which helical passages are machined, an assembly of magnetic core laminations, an inner cylinder, end members connecting and closing the cylinders at the ends, and inlet and outlet connections. The radial wall thicknesses of both outer and intermediate duct cylinders are 0.025 in. The helical passages are square with a side width of approximately 0.34 in.; there are two such

TABLE III

EM PUMP WEIGHT CORRELATION DATA

Pump No.*; Fluid	Temp., F	Flow Rate, GPM	Press., PSI	Elec. Res., $\mu\Omega$ -in.	Capability Parameter, Q/P 10^{-3}	Weight, LB	Eff., %	Pump Type**	Service
1 Na	700	6500	53	8.5	2930	18,500	44	FI	EBR-II Secondary Loop
2 Na	700	5000	40	8.5	1700	14,000	40	FI	Argonne Nat'l Lab.
3 Na	600	3300	85	7.1	1990	11,400	43	FI	USS Seawolf
4 Na	700	1200	40	8.5	408	6,430	33	FI	KAPL Test Pump
5 Na	700	60	30	8.5	15.3	800	15	FI	KAPL/EBR-II Service
6 Na	900	500	40	10.6	212	2,250	17	HI	LASL
7 K	1600	42	150	35.0	220	1,500	10	HI	SPPS Turbine Loop
8 K	1850	200	20	37.5	150	1,500	6	HI	SPPS 300 KW Loop
9 K	1850	20	67	37.5	50.1	1,100	5	HI	SPPS 300 KW Loop
10 K	1400	50	50	29.0	72.5	1,000	11	HI	AiResearch Corp.
11 K	1400	4	120	29.0	13.9	300	4	HI	AiResearch Corp.
12 K	2200	3	75	45.0	10.1	570	1	HI	SPPS 100 KW Loop
13 K	500	4	80	10.2	3.3	250	4	HI	SPPS Brg. Seal Service
a K	1200	16	30	23.5	11	33	10	HI	Condensate Boost
b K	1200	16	30	23.5	11	9	14	DC	Condensate Boost
c K	1300	96	240	26.2	603	410	22	HI	Condensate
d NaK	1200	60	25	32.5	49	38	12	AI	Radiator Coolant
e NaK	1200	60	25	32.5	49	19	15	DC	Radiator Coolant
f Li	1200	31	20	15.4	9.5	33	13	HI	Radiator Coolant
g Li	1200	31	20	15.4	9.5	13	15	DC	Radiator Coolant

*Numbers (existing pumps) and letters (study pumps) consistent with Fig. 4.

**FI = flat induction; HI = helical induction; DC = direct current; AI = annular induction.

passages hydraulically in parallel. The axial width of the boundaries of the flow passages is 0.02 in. The inner cylinder is not linked by the magnetic field and, hence, may have any convenient wall thickness. The core is composed of one-piece laminations of cobalt-iron alloy, with the principal planes of the laminations perpendicular to the axial centerline of the duct. Because of the low flux density and low frequency, lamination thickness is not critical. A stable interlaminar insulation which is not subject to continued outgassing or reaction with the magnetic core at 1200 °F is required.

The stator assembly comprises the cooling coil assembly, which is welded or brazed to the radial periphery of the cylindrical frame; the cylindrical frame; the stator laminations; the stator winding; and the stator can. The cooling coil assembly constitutes the heat sink for the pump. It is supplied with NaK at 600 °F. Pump winding temperature is relatively insensitive to coolant flow rates of one gpm or higher.

Since the cylindrical frame must support the stator lamination assembly and contain the gas in the stator winding cavity, its design is determined by structural considerations.

The stator laminations are at a temperature of approximately 700 °F. Because an important function of the stator laminations is to conduct the losses to the heat sink, high thermal conductivity is desirable. The low silicon magnetic steels, common iron, or the cobalt-iron alloys afford a satisfactory combination of thermal and magnetic properties. Lamination thickness is not critical. Interlaminar insulation requirements are essentially the same as those discussed earlier for the core.

The stator winding is two-pole, three phase, lap wound, two coil sides per slot. Eighteen slots, 0.32 in. wide by 0.75 in. deep, are illustrated, and the individual coils are form wound using rectangular wire. (Random wound coil construction using round wire is possible and may prove preferable.) Performance calculations are based on copper windings with a resistivity of 1.75 μ -ohm-in. at 750 °F. Ground wall insulation and insulation between turns are assumed to be 0.025 in. and 0.005 in., respectively. With eight turn coils, wye connected, one circuit per phase, the line voltage is 76 volts and the line current 31.5 amperes. The voltage and current combination may be changed by changing the number of turns per coil. The stator can is austenitic steel, 0.015 in. thick. The stator winding cavity is filled with nitrogen at a pressure of approximately one atmosphere at 700 °F. Power is

supplied to the stator windings through a ceramic-to-metal seal assembly in the stator envelope.

The thermal insulation between the pump duct and the stator can consist of alternate layers of austenitic stainless steel cloth and strip that is laminated to reduce eddy current loss. Four layers of the strip and cloth combination are located in the 0.05 in. space available for thermal insulation.

Figure 6 shows developed pressure, power input, and power factor as a function of flow rate. Developed pressure is shown for four different voltages. Power input varies as the square of the voltage and is illustrated for rated voltage. Power factor is independent of voltage. Both power input and power factor are relatively insensitive to flow rate. The shape of the pressure-flow rate curves illustrates the suitability of this pump for control of flow rate by voltage variation.

Table IV shows the disposition of the power input. The predominant losses occur in the stator winding and as resistive loss in the potassium. Duct losses are directly proportional to radial duct wall thickness and inversely proportional to the electrical resistivity of the duct material. It is apparent that little improvement in efficiency will be gained by using a thinner duct wall or a material of higher resistivity.

VII FACILITY PUMPS

EM pumps have been used in alkali metal test and development facilities for the past fifteen years or more. Many of the facility pumps have attained the high reliability desired for flight components. Weight and efficiency were secondary design considerations, however, and facility pumps are too heavy and many are too inefficient for flight applications. The earliest models were small and most were of the a-c conduction type. They served to eliminate shafts and seals that had been a continuing source of maintenance problems with mechanical pumps. Some d-c conduction pumps have been used, but the low voltage, high current power requirements have limited their application. Two basically different types of induction pumps have been used in facility applications, the rotating magnet type and the polyphase induction type. The rotating magnet induction pump has been used very successfully in relatively small sizes, while the polyphase induction pump has been developed and used over a wide range of sizes.

Figure 7 illustrates a 6500 gpm polyphase pump used by Argonne National Laboratory in the Experimental Breeder Reactor II. It is a flat linear induction pump, with straight through flow and an expansion joint within the hermetically sealed frame, or casing. Figure 8 is a photograph of a 60 gpm unit used at Knolls Atomic Power Laboratory. This too is a flat linear pump with the duct connections arranged in a U return bend configuration to permit locating inlet and outlet at the same end of the pump, thus avoiding the use of an expansion joint, yet retaining an hermetically sealed frame, or casing. The flat design is best adapted to high flow rates with low to moderate developed pressures.

More recently the helical induction pump has been used widely in facilities associated with Rankine cycle space power development work. The helical design is best adapted to high pressure applications with low to moderate flow rates. To date the sizes have ranged from 500 gpm down to less than one gpm with developed heads up to about 250 psi.

The helical duct configuration is a good pressure containment vessel and is therefore well suited to high-pressure, high-temperature applications. Table III lists a number of helical induction pumps now operating in test facilities together with the design ratings of each.

A photograph of the stator for a 200 gpm, 20 psi helical induction pump is shown in Fig. 9. This pump appears as item #8 in Table III, and was one of the first of this style to be put in service. The duct for this pump is made of Haynes #25 alloy with no magnetic core. The pump has accumulated 4460 hours of operation at temperatures from 800 °F to 1850 °F. Brief excursions over 1850 °F have occurred. The performance of this pump has been very satisfactory, and no changes with age have been observed.

Figure 10 is a photograph of the pump shown as item #12 in Table III. This is a helical induction EM pump with a stator similar to that shown in Fig. 9. The significant difference lies in the duct and its insulation and containment. To accommodate the fluid design temperature of 2200 °F, the duct was made of Cb - 1% Zr alloy as was the piping system it was to serve. The entire piping system including the pump duct was protected from oxidation by containment in a vacuum chamber at a pressure in the 10^{-8} TORR range. To accomplish this, the pump duct was contained in a stainless steel thimble protruding from the side of the vacuum chamber as illustrated in Fig. 10. The annulus between duct and thimble was filled with

a number of layers of Cb - 1% Zr foil to block radiant heat transfer. Performance of this pump over the presently accumulated life of 6203 hours has been very satisfactory. Unfortunately, pressure instrumentation was not adequate to provide quantitative data on pump performance.

Some of the facilities referred to have included two phase alkali metal systems. In such service pumps may be subject to low NPSH conditions. Since the means of imparting energy to the fluid are very different from that in mechanical pumps, cavitation in EM pumps may not be analogous to that in centrifugal pumps. Nevertheless, the characteristically sharp loss of pump performance at low NPSH associated with the onset of cavitation in centrifugal pumps has been observed in EM pumps. In conduction pumps, this condition has led to failure of the duct. Induction pumps have demonstrated a tolerance of cavitation. The degree of tolerance has not been established, and further exploration of this area is planned.

REFERENCES

1. Verkamp, J.P., Electromagnetic Alkali Metal Pump Research Program, Quarterly Progress Report 2, National Aeronautics and Space Administration Contract NAS3-2543, NASA-CR-54019, February 17, 1964, General Electric Company.
2. Verkamp, J.P., Electromagnetic Alkali Metal Pump Research Program, Quarterly Progress Report 3, National Aeronautics and Space Administration Contract NAS3-2543, NASA-CR-54036, May 22, 1964, General Electric Company.
3. Rhudy, R.G. and Verkamp, J.P., Electromagnetic Alkali Metal Pump Research Program, Final Report, National Aeronautics and Space Administration Contract NAS3-2543, in publication, General Electric Company.

TABLE IV
POWER DISTRIBUTION FOR CONDENSATE BOOST PUMP*

Design Point Data		
	KW	Percent Input
Pump Output	0.20	10
Stator Winding Loss	0.69	34
Secondary (Fluid) Loss	0.68	33
Iron Loss	0.2	10
Duct Wall Loss (0.025 in.)	0.13	6
Stator Can Loss	0.05	3
Hydraulic Loss	<u>0.08</u>	<u>4</u>
Total Power Input	2.03	100

*See Fig. 5

CLASSIFICATION OF EM PUMPS

Figure 1. EM PUMP CLASSIFICATION

Figure 2. ELEMENTARY PUMP DUCT

Figure 3. MAXIMUM EFFICIENCY VS. DUCT PARAMETER

Figure 4. SPECIFIC WEIGHT RELATIONSHIP EM PUMPS

Figure 5. CROSS-SECTIONAL OUTLINE
HELICAL INDUCTION PUMP
CONDENSATE BOOST APPLICATION

Figure 6. PERFORMANCE OF HELICAL INDUCTION EM PUMP

Figure 7. FLAT LINEAR INDUCTION PUMP
 SODIUM, 6500 GPM, 53 PSI
 3 PHASE, 60 CPS, 440V, WATER COOLED

Figure 8. FLAT LINEAR INDUCTION PUMP
SODIUM, 60 GPM, 30 PSI
3 PHASE, 60 CPS, 220V, WATER COOLED

Figure 9.
HELICAL INDUCTION PUMP STATOR AND DUCT PARTS
SODIUM/POTASSIUM 1850F, 200 GPM, 20 PSI
3 PHASE, 60 CPS, 440V, AIR COOLED

Figure 10. HELICAL INDUCTION PUMP
 POTASSIUM 2200F, 1.25/5 GPM, 100/20 PSI
 3 PHASE, 60 CPS, 440V, AIR COOLED

The Development of the MRP Thermoelectric NaK Pump

by

Harry H. Hanlin (Atomics International)

Jack M. Hochman (Atomics International)

and

Milton A. Perlow (Atomics International)

Unclassified

ABSTRACT

A thermoelectromagnetic pump was developed to circulate the heat transfer fluid in a Rankine cycle, space power system. This device combines a dc conduction pump and thermoelectric generator into a single, all-metallic unit. The thermoelectric elements, chromel and constantan, extract heat from the fluid being pumped, convert a portion of it directly into electricity to supply the conduction pump, and reject the waste heat to the secondary working fluid, mercury, as it enters the boiler. This pump type offered the major advantages of no moving parts and complete sealing of the liquid metals for highest reliability and complete utilization of rejected heat for highest system efficiency. The design performance objective of 1 psi, 12 gpm NaK-78 at 1200°F was achieved and an endurance run to 6000 hours was demonstrated. This report describes the complete development program, detailing the design method and equations, laboratory procedures and all important test results.

I. INTRODUCTION

A liquid-metal-cooled nuclear reactor - mercury (Hg) Rankine cycle power system for space applications is currently under development for the Atomic Energy Commission (AEC) by Atomics International (AI), a division of North American Aviation. The Mercury Rankine Program (MRP)⁽¹⁾ has as its objective the development of modular power packages capable of delivering 3 to 5 kw of electric power, while unattended, for periods greater than 10,000 hr in a zero-gravity (g) space environment. These packages can be integrated to cover power ranges up to 20 kwe and with minor modifications, 25 kwe and greater. This is a power increase and modification of the former 3-kwe (SNAP 2) system.

The system is comprised of a 60-kwt reactor heat source and a Hg vapor turbo-electric power conversion subsystem (PCS). Figure 1 illustrates the complete system and operation of its major components. The pump which circulates the liquid metal in the primary reactor-boiler loop has undergone extensive development at AI. The development of this components as part of the 3-kwe system is the subject of this report. Development is continuing for application in the advanced systems of the MRP.

History of Development and Advantages of Concept

Initially, the liquid-metal pump was a rotating member on the single shaft of the combined rotating unit (CRU). This is the turbine driven alternator unit. CRU development is being performed by Thompson Ramo Wooldridge (TRW), Inc., subcontractor for the PCS. In early 1963, a decision was made to develop the NaK pump as a separate component at AI. The type selected to replace the onshaft design was the thermoelectromagnetic (TEM) pump. Figure 2 is a photograph of Prototype-I, which was the final development model during the 3-kwe program phase. This type of pump employs the metals chromel and constantan as the thermoelectrics (TE's) and has the following major advantages:

- 1) No moving parts
- 2) Complete liquid-metal sealing

3) Built-in electromagnetic (EM) flowmeter

4) Acts as a preheater to the Hg boiler.

This last advantage is accomplished by employing the Hg entering the boiler as the coolant fluid. This economizer effect, coupled with the unloading of the CRU shaft increased the (SNAP 2) system output by 14% (from 3.6 kwe to 4.1 kwe).

The TEM pump development progressed through three separate designs, dictated by changing system requirements.

The first TEM pump utilized Pb-Te as the TE's and was initiated when CRU-NaK pump operational problems became severe; development was conducted in parallel with the TRW effort. This design was a direct extension of the SNAP 10A TEM pump;⁽²⁾ Hg coolant tubes were merely substituted for the radiator fins. Since an upper limit of 1000°F existed for Pb-Te, this design could only employ the lower temperature NaK return line as the heat source. Unstable system startup characteristics were encountered with this scheme.

In early 1963, re-evaluation of the 3-kwe system hydraulic losses changed the pump pressure requirement from (nominally) 2 psi to 1 psi. Preliminary calculations showed that these requirements could be met by a TEM pump utilizing chromel and constantan as the TE's while employing the higher 1200°F NaK hot leg as the heat source. Chromel is an alloy of 90% nickel and 10% chrome; constantan is an alloy of (nominally) 55% copper, 45% nickel. Development of the Pb-Te pump was dropped in favor of this pump.

In May, 1963, following successful operation of several experimental models, the chromel-constantan TEM pump was chosen for the primary development effort, replacing the onshaft pump. Fabrication and testing of the Prototype-I pumps have continued from August, 1963 to the present. Achievement of all major 3-kwe system objectives has been demonstrated. A modified version of these pumps is now under development and testing for utilization in the advanced MRP system.

Objectives

The objectives and environmental limitations for the TEM pump (as finally established for the 3-kwe system)⁽³⁾ were:

- 1) Provide a flowrate of 71 lb/min (11.75 gpm) NaK-78 at 1200°F against a system pressure of 1.05 psi.
- 2) Reject the waste heat to the Hg upstream from the boiler, limited to 3.5 kw maximum. The Hg flows at 20 lb/min at an inlet (to the pump) temperature of 510°F.
- 3) Perform the auxiliary functions of:
 - a) Pre-reactor startup circulation of NaK at 3.5 lb/min (0.522 gpm) at 40°F against a system pressure of 0.016 psi. The current required shall be less than 40 amp. This is to prevent freezing.
 - b) Provide means for measuring the NaK flowrate within $\pm 10\%$ accuracy in the 1 to 10 mv range.
- 4) The pump shall perform these functions in a space environment for a period of 1 year.
- 5) The pump shall have a minimum weight design objective.
- 6) Degradation in flow attributable to the pump shall be limited to less than 10% over one year.

II. PUMP DESCRIPTION

Figure 2 is a photograph of the TEM pump, utilizing chromel and constantan as the TE's which was the final development model during the 3-kwe system phase of the program. This design was designated Prototype-I. The most important feature of this pump is its monolithic structure; the entire unit is an integral, all metallic mass relatively insensitive to damage from thermal and/or mechanical shock. This primary advantage of the chromel-constantan pump was more important than the weight penalty incurred by the low figure of merit of chromel vs constantan. Figure 3 is a photograph of the Prototype-1 pump with the magnet system removed to show the details of the webbed bus plates. Also shown are the auxiliary current lugs and

flowmeter leads. Table 1 summarizes the important dimensions and characteristics of this design.

A. Pump and Magnet

This configuration was selected as most amenable to:

- 1) High TE ΔT
- 2) Low electrical losses
- 3) Simple magnetic geometry
- 4) Ease of fabrication

A high TE ΔT was obtained by placing the TE elements as close to the hot and cold bodies as possible. Figure 4 shows a typical temperature profile. The ΔT available to the TE's is in the order of 70 to 80% of the total ΔT . Major ΔT drops occur unavoidably in the hot and cold body fluid films.

Electrical losses were minimized by keeping the circuit as short as possible. Interference between the magnetic and electric circuits was compromised by routing the return bus through slots in the magnetic pole shoes. The slot width and number of slots were chosen such that only a small decrease in magnet system effectiveness resulted.

Horseshoe shaped permanent magnets generate the magnetic field which is conducted to the centrally located working air gap by high permeability pole shoes. The magnet arrangement had a low leakage factor because the air gap is close to the high potential surfaces (permanent magnet ends). Good mechanical integration was effected by the slotted pole shoes and laminar bus plates, no external brackets were required for alignment of the magnet system to the pump throat assembly. This simple geometry also enhanced ease of assembly and permitted independent modification of each subassembly. Finally, as a practical matter, this arrangement also enabled the use of stock magnets for the initial test work; a considerable savings in terms of development time and cost. It was later found that stock magnet sizes existed such that only small weight savings would be gained with tailored magnets, hence, stock magnets become part of the Prototype-1 design.

B. Fabrication

The primary effort during the pump fabrication program was to obtain acceptable braze bonding of the 71.5 square inches of joining surfaces throughout the pump. Brazing was accomplished in two steps as shown in Figure 5.

Hydrogen was selected as the most suitable brazing atmosphere on the basis of maintaining oxide-free surfaces. The throat element was nickel plated to promote wetting of the throat-TE junctions. Early tests revealed that oxides contained in the constantan TE elements were broken down by the hydrogen and combined with the chrome in the 316 SS to form a chrome oxide on the throat surface. This was the primary reason for nickel plating the throat ducts. The braze alloys, Nicoro for the first step and Incoro 60 for the second step braze operation, were ultimately chosen from the results of an extensive braze materials evaluation program. This program is discussed in detail in Reference 4.

III. DESIGN AND ANALYSIS

A. Electrical Design

The high electrical conductivity of NaK makes it most amenable to EM pumping.⁽⁵⁾ The simplest and most direct EM pump is the Faraday type dc conduction pump (Fig. 6).

Force is developed on the liquid metal contained in the thin-walled throat duct by the interaction of the magnetic field and electrical current. This is a vector system, and when mutually orthogonal, the magnitude of the force per unit area (pressure) on the liquid metal is given by

$$P = \frac{F}{A} = \frac{K_1 B I}{b} \quad \text{.....1}$$

Where:

P = Pressure

F = Force

A = Area

K_1 = A constant

B = Magnetic field intensity

I = Current

b = Throat height between magnet poles

The electric circuit is depicted in Figure 7. The total current (I_T), which is supplied by the driving source, is comprised of the useful component (I') and a loss component (I_W), the current conducted through the walls.

The total electrical power (W_{IN}) delivered to the pump is

$$W_{IN} = I_T E_W \quad \text{.....2}$$

The portion of this power appearing as hydraulic power is

$$W_{Hydraulic} = I' E_B \quad \text{.....3}$$

where E_B is the back emf. The pump under flow conditions suffers two pressure losses:

$$P_{Hydraulic} = \frac{2fDV^2L}{gd} + K_H \frac{DV^2}{2g} \quad \text{.....4}$$

$$P_{\text{Eddy current braking}} = \frac{K_2 k_B^2 Q}{\rho_N^b} \quad \text{.....5}$$

Equations 4 and 5 are described in Table 2.

The hydraulic loss arises from friction between the liquid metal and the duct walls within the pump and in the transition from the round system tubing to the square or rectangular throat. The eddy current braking pressure loss results from the movement of the liquid metal through the fringe magnetic field at each of the pump ends.⁽⁶⁾ Eddy currents are generated within the liquid metal and interact with the magnetic field, resulting in a braking effect which appears as a loss in pressure.

A summary of the principal power distributions in a dc conduction pump is given by Equation 6. (An additional loss resulting from current fringing around the electrode ends and not linking the magnetic field is treated by some authors,

References 5 and 6. Significant overhang of the magnetic field in the pumps discussed herein, effectively couples these fringe fields and has reduced this loss to a negligible value.)

$$W_{IN} = E_W IT = P_{NET} Q + P_E Q + P_H Q + I^2 W_W R + I^2 R_N \quad \dots\dots 6$$

An effective pump design is one which minimizes the loss components while meeting the constraining power source and weight requirements.

B. Thermal Considerations

The generation of TE power, simply treated, results from the establishment of a temperature gradient across two dissimilar metals comprising a closed electrical loop containing a suitable electric power absorbing element (load).⁽⁷⁾

The temperature gradient is established by placing the junctions in thermal contact with a heat source existing at T_1 and a heat sink existing at T_0 (Fig. 8). Heat will be transferred from the hot to cold body principally by conduction (q_c) described by:

$$q_c = (T_1 - T_0) (K_P + K_N) \frac{A}{t}$$

Where

T_1, T_0 = Hot and cold body temperatures

K = Thermal conductivity

A = Heat transfer area

t = Thickness of medium

A smaller amount of heat, the Peltier heat, will also be transferred, due to the Peltier effect. Heat is absorbed at the hot junction T_1 when a current (I) passes through it. Most of this energy is liberated at the cold junction T_0 by current passage in the reverse direction. The difference in the absorption and liberation rates of this thermal energy is that which has been made available as electrical power. Part of this is immediately converted again to thermal energy by virtue of the electrical resistance of the TE materials. The remaining portion is the useful energy, capable of doing work at the load element. The amount of useful TE power thus transformed is given by Equation 8.

$$W_{out} = I(S_N + S_P) (T_1 - T_0) - I^2 (R_N + R_P) \quad \dots\dots 8$$

Where

S = Seebeck coefficient (volts/°temp.)

or more simply

$$W_{out} = SI \Delta T - I^2 (R_N + R_P) \quad \text{.....9}$$

C. Development of Design Equations

The output impedance of a TE generator and input impedance of a dc conduction pump are typically very low, in the order of micro-ohms. This fact suggested the direct coupling of the two devices. The fluid to be pumped constitutes a heat source which enabled the pump and converter to be integrated into a single unit, thus eliminating the necessity of transmission elements. The most straightforward integration of these two devices is shown by Figure 9.

The heat flows from the pumped hot fluid, through the TE's, to the coolant fluid. This heat conducted through the TE elements creates a temperature difference (ΔT) across each of the N and P elements. The resulting voltage causes current to flow in the closed electric circuit. Application of the magnetic field at right angles to the current in the liquid metal produces a force on the fluid, thus completing the requirement for pumping the hot fluid.

The equations describing this system would be relatively simple were it not for the fact that a nonlinear temperature gradient exists ⁽⁸⁾ along the length of the pump. This condition is illustrated by Figure 10. As the coolant fluid flows through the pump, heat is absorbed by the fluid causing its temperature to increase exponentially. At the same time, the primary fluid temperature decreases in the reverse (counter-flow) direction as heat is extracted. (The primary fluid (NaK) temperature profile was linearized for most of the pump design and analysis work as shown because of its relatively high flowrate and heat capacity.)

The derivation of the pressure equation for a TEM pump is most simply accomplished on an incremental pump length (dx) basis. Figure 11 depicts the incremental pump and Figure 12 is its equivalent electric circuit. The TE derived voltage (E_G) operates on the closed electric circuit comprised of the TE elements, bus and throat duct. The throat duct appears as a variable impedance parallel

circuit; one brach being the priamry (pumped) fluid and the other the wall (r_S) path. The impedance of the fluid path varies as the flow rate since E_B , the back emf term, is a function of fluid velocity. The current which links the magnetic field to develop the pressure is that which flows in the primary fluid, di' in Figure 12.

$$di' = \frac{r_S}{r_S r_T + r_N r_T + r_N r_S} E_G - \frac{r_T + r_S}{r_S r_T + r_N r_T + r_N r_S} E_B \quad \dots\dots 10$$

It is observed from Table 2 that the length differential, dx , appears singularly in the denominator of each resistance term and can be factored out to obtain,

$$di' = \frac{r'_S}{r'_S r'_T + r'_N r'_T + r'_N r'_S} E_G dx - \frac{r'_T + r'_S}{r'_S r'_T + r'_N r'_T + r'_N r'_S} E_B dx \quad \dots\dots 11$$

Where r'_S , r'_T , etc., are the per unit length resistances of each element. Since they will appear as constants in the following intended integration the following substitutions can be made:

$$N_1 = \frac{r'_S}{r'_S r'_T + r'_N r'_T + r'_N r'_S} \quad \dots\dots 12$$

$$N_2 = \frac{r'_T + r'_S}{r'_S r'_T + r'_N r'_T + r'_N r'_S} \quad \dots\dots 13$$

hence,

$$di' = N_1 E_G dx - N_2 E_B dx \quad \dots\dots 14$$

Substituting the Table 2 value for E_G

$$di' = N_1 S \eta_T \Delta T_x dx - N_2 E_B dx \quad \dots\dots 15$$

Now ΔT_x , the fluid-to-fluid ΔT at length x is developed as shown in Figure 13

The heat energy absorbed by the differential volume of coolant along dx , is

$$dq_A = WC_P dT \quad \dots\dots 16$$

The heat transferred to the differential volume is

$$dq_T = \frac{1}{R} dx (T_H - T_x), \text{ where } T_H \text{ is the source temperature } \dots\dots 17$$

(T_H is assumed constant along the entire pump length)

These two heats must be equal,

$$dq_A = dq_T \quad \text{.....18}$$

$$\therefore WC_P dT = \frac{1}{R'} dx (T_H - T_x) \quad \text{.....19}$$

or

$$\frac{dT}{T_H - T_x} = \frac{dx}{WC_P R'} \quad \text{.....20}$$

Integrating,

$$\Delta T_x = C \epsilon \exp \left(- \frac{x}{WC_P R'} \right) \quad \text{.....21}$$

where, $\Delta T_x = T_H - T_x$

$$\text{at } x = 0, \Delta T_x = \Delta T_o \quad \text{.....22}$$

$$\therefore \Delta T_x = \Delta T_o \epsilon \exp \left(- \frac{x}{WC_P R'} \right) \quad \text{.....23}$$

Substituting this expression for ΔT_x into Equation 15,

$$di' = N_1 S \eta_T \Delta T_o \epsilon \exp \left(- \frac{x}{WC_P R'} \right) dx - N_2 E_B dx \quad \text{.....24}$$

Let

$$WC_P R' = L_o \quad \text{.....25}$$

then

$$di' = N_1 S \eta_T \Delta T_o \epsilon^{-\frac{x}{L_o}} dx - N_2 E_B dx \quad \text{.....26}$$

The above expression describes the useful pumping current in the differential elements as comprised of two components, a forward current derived from the TE generator, variable with element location and reverse current derived from the back emf generator, constant with element location. The total useful pumping current existing in a pump L inches long is the integral of the above equation,

$$I' = \int_{x=0}^{x=L} di' = N_1 S \eta_T \Delta T_o \int_{x=0}^{x=L} \epsilon^{-\frac{x}{L_o}} dx - N_2 E_B \int_{x=0}^{x=L} dx \quad \text{.....27}$$

Then

$$I' = N_1 S \eta_T \Delta T_o \left[-L_o \epsilon^{-\frac{x}{L_o}} \right]_0^L - N_2 E_B \left[x \right]_0^L \quad \text{.....28}$$

and

$$I' = N_1 S \eta_T \Delta T_o L_o \left(1 - \epsilon^{-\frac{L}{L_o}} \right) - N_2 E_B L \quad \dots\dots 29$$

This current, when linked with the magnetic field, produces pressure in accordance with the following equation,

$$P_d = K_1 \frac{BI'}{b}, \text{ psi} \quad \dots\dots 30$$

making the substitution for current and expanding E_B (see Table 1 for E_B)

$$P_d = K_1 N_1 \eta_T \frac{SB}{b} \Delta T_o L_o \left(1 - \epsilon^{-\frac{L}{L_o}} \right) - \frac{K_1 K_3 N_2 L B^2 Q}{b^2} \text{ psi} \quad \dots\dots 31$$

where

$$K_1 = 8.83 \times 10^{-8} \quad \text{a constant}$$

$$K_3 = 3.86 \times 10^{-8} \quad \text{a constant}$$

It is to be noted that a value for B exists such that the developed pressure can be maximized. This would be found by solving $\partial P_d / \partial B = 0$ for B. Substitution of this value into the original expression would yield P_d maximum. This is not a particularly useful expression, however, since the net pressure (P_{net}) rather than developed pressure is desired to be maximized. Net pressure differs from developed pressure by the flow losses (end effect and hydraulic) previously discussed in Section III.

$$P_{net} = P_d - \Delta P_H - \Delta P_E \quad \dots\dots 32$$

Substituting the expressions for each of these constituents in this equation gives the following complete expression for net developed pressure of fluid cooled TEM pump:

$$P_{net} = K_1 N_1 \eta_T \frac{SB}{b} \Delta T_o L_o \left(1 - \epsilon^{-\frac{L}{L_o}} \right) - \frac{K_1 K_3 N_2 L B^2 Q}{b^2} - \frac{2fDV^2 L}{gd} \\ - K_H \frac{DV^2}{2g} - \frac{K_2 k Q B^2}{b \rho_{NaK}} \text{ psi} \quad \dots\dots 33$$

D. Optimization of Design

The design for the minimum weight pump was initiated when the four primary considerations were established:

- 1) Basic equations
- 2) Performance specifications
- 3) Configuration
- 4) Materials properties.

A digital computer program was formulated from this information and utilized to determine the minimum weight pump. This was accomplished by varying each of the important parameters, such as throat-duct dimensions, magnetic field, etc., and determining the minimum total pump weight value for each. Essentially, a cross-sectional configuration was set into the program and a thermal heat balance was initiated. A pump length was computed which satisfied the thermal boundary conditions. From this, the temperatures were determined and the generated voltage ($S \Delta T$) was calculated. Next, the pumping current was computed from the generator voltage and system impedances. A pressure was then computed. This process was repeated until a pump length was found which satisfied the pressure requirement. Once obtained, the total pump weight was computed as the final solution.

Curve A of Figure 14 shows the effect of variation in TE thickness on total pump weight of the Prototype-I configuration. At small thickness values the ηT efficiency is low. (This is the ratio of TE ΔT to the total fluid-to-fluid ΔT .) At large thickness values the total weight continues to decrease at a decreasing rate. The design value of $t = 1$ in. was chosen primarily to enable use of the smaller size stock magnets and maintain proper clearances between the magnet and throat assemblies. Throat duct height exhibited a minimum weight value (Curve B Figure 14) Below approximately 0.8 in. the hydraulic pressure loss becomes significant. Above this value the weight increases, due principally to the increase in magnet system airgap length. Pump weight was found to minimize at a magnetic field value of approximately 14 kilolines/in.² (Curve C, Figure 14). Weight increases below this value because a longer pump is required to increase the current component in the pressure equation (Equation 1). Above this value the weight of the magnetic system begins to dominate since

$$W_M \text{ (Weight of magnet)} = \frac{FfV_g B^2}{B_M H_M}$$

.....34

where:

F and f are leakage factors

V_g = volume of the airgap

$B_M H_M$ = Energy product of the material.

The final design for the Prototype-I pump was fixed primarily upon the basis of these parameter variation studies. Several repetitions of this process were made; each succeeding run employed the previously determined best parametric values as the reference cross-section.

The following parameter values were found to establish a close to minimum weight pump design:

Throat height	0.8 in.
Throat width	0.8 in.
TE thickness	1.0 in.
Magnetic field	14 kilolines/in. ² .

The values for the other parameters such as bus length, throat wall thickness, etc. exhibited no minimum weight point and were selected primarily on strength or clearance considerations.

IV. EXPERIMENTAL PROCEDURES AND RESULTS

A. Test Loop Requirements and Equipment Description

The objective of the pump test program was to measure the performance characteristics of each pump under selected environmental and thermal conditions. The conditions selected were those that would enable evaluation of performance under ultimate use environment and also to enable evaluation of the theoretical design equations and calculations. The test apparatus and procedures were designed to reproduce or adequately simulate conditions to meet these objectives.

Each of the test facilities consisted of two horizontal counter-flow liquid-metal loops, stacked vertically as shown in Figure 15. The primary loop included the pump on test, EM flowmeter to measure flowrate, throttling valve to change the hydraulic characteristics, auxiliary dc conduction pump, heater bundle and two pressure measuring tanks located at the entrance and exit to the test pump. These tanks also served as fluid expansion tanks. The coolant loop contained a flowmeter, dc conduction pump and throttling valve. The primary loop was insulated to minimize heat losses. The coolant loop required to reject the collected heat was uninsulated. Surface heaters along the coolant loop allowed for variation in the coolant temperature. This loop was split into two parallel legs at the entrance and exit to the test pump to allow independent coolant conditions for each side of the pump. Each leg contained a flowmeter and throttling valve.

Facility No. 4 enclosed the test pump in a vacuum chamber to permit operation in a vacuum environment down to 3 microns. (Facilities No. 1, 2 and 3 did not enable vacuum environment operation.) The vacuum was drawn and maintained by a mechanical pump. The vacuum chamber walls were cooled by circulating water through copper tubes attached to the outside of the vacuum vessel. Cold traps were located in both the primary and secondary loops. These are "fingers" welded to the pipes. Refrigerated Freon was circulated through tubes surrounding the finger to maintain the liquid metal well below room temperature. Oxides collect at these lowest temperature points. In this manner, cleanliness of the NaK was maintained. Primary and coolant fluid flowrates, fluid entrance and exit temperatures and various selected pump and loop temperatures were measured and continuously recorded by the two strip chart recorders. The flowmeters were of the EM type coupled directly to the recorder. Immersion Type-K chromel-alumel thermocouples were used to measure the inlet and outlet fluid temperatures; all others were surface mounted.

All test facility controls were manual with the exception of the primary and secondary fluid temperature limiting controllers and the cold-trap refrigeration system. The fluid temperature controllers were set to automatically interrupt

heater power in the event of a temperature increase beyond allowable limits. The refrigeration system maintained the cold trap temperature at -20°F . Primary and secondary temperatures were set by variac adjustments. Cover gas pressures on both the primary and coolant loops were maintained within required limits by gas bottle pressure regulators and vent valves. Primary and secondary fluid flowrates were adjusted by manually operated throttling valves. Once established, steady-state conditions were maintained with normally minimal effort.

B. Test Procedures

Four separate and distinct tests were devised and performed during the program to determine the important performance, stability and endurance characteristics of the TEM pump. These included: prestartup NaK motoring under auxiliary power (to prevent NaK freezing) startup transient test (coolant injection), steady-state performance, and endurance (life) tests.

1. NaK Motoring Tests

This test determined the auxiliary battery supplied power requirement for maintaining the primary fluid flowrate at 5% of the full flow value during pre-reactor startup phase. Coolant fluid was not contained in the secondary coolant ducts. Conductors were attached to the electrodes on the pump throat and to a dc power supply. The pressure vs flow characteristics for constant current settings and primary fluid temperatures were measured and plotted. The fluid temperature was controlled by heater power setting and the flowrate was varied by throttling valve setting.

2. Startup Test (Coolant Injection)

The space system is started by injecting $\sim 300^{\circ}\text{F}$ Hg through the TEM pump into the boiler. With the NaK in the reactor boiler loop at approximately 1200°F , the Hg is immediately vaporized. The Hg vapor impinges on the CRU turbine and the system enters the startup phase. A severe thermal shock is sustained by the TEM pump during this phase of the startup operation. To simulate this thermal shock condition the following procedure was followed (refer to Figure 15).

- 1) The primary loop temperature is brought to 1200°F. Primary fluid flow is sustained by the test pump itself due to a positive ΔT existing across the TE elements, even without coolant. (This condition would also exist in the space environment due to radiation heat losses.)
- 2) The coolant fluid is located in the drain tank and pressurized to 20 psig. The secondary loop is evacuated to ~ 10 microns.
- 3) The oscillograph is started and the drain tank valve is opened.
- 4) The fluid flows out of the drain tank and through the test pump. Back flow through the loop is prevented by closure of the valve adjacent to the drain tank line.

3. Steady-State Performance Test

The purpose of this test was to determine the steady-state performance at design thermal conditions. This was accomplished by setting the coolant inlet temperature and flowrate and primary fluid temperature at the design thermal conditions and obtaining the pressure vs flowrate characteristic curve. The flowrate was varied by adjusting the primary loop throttling valve. Intersection of this curve with the system hydraulic plot yields the operating point of the TEM pump in the actual system.

4. Endurance Test

Several pump units, after initial performance mapping tests, were left in the test rigs at design thermal conditions for extended periods of time. The purpose of these tests was to determine degradation characteristics. The pressure-flowrate characteristic was measured periodically (weekly) and compared with the initial performance. Percent change in flowrate as would be exhibited in an actual system was the comparison standard.

C. Test Results

Ten TEM pumps utilizing chromel and constantan as the TE elements were tested during the Prototype-I development phase. The principal areas of testing were performance mapping in the steady-state condition to gain fabrication and design

feedback information, and long-term endurance at design thermal conditions to determine the stability of the materials. Sensitivity to thermal and mechanical shock and vibration was considered to be relatively unimportant owing to the rugged metallic construction of the device, hence, only limited testing was done in this area. One thermal shock test consisting of injecting room temperature NaK into a Prototype-I unit at 1200°F was accomplished. Also, on several units, thermal cycling tests were performed to determine stability. Supplementary tests were performed to determine the prereactor startup current requirements at low flows to prevent NaK freezing.

1. Prereactor Startup NaK Circulation

Figure 16 plots the performance of TEM pump SN-012 for a zero TE ΔT condition. The design temperature environment of 40°F was not tested due to test facility limitations, however, significant errors were not effected by using the room temperature (75°F) characteristic. In this mode 90 amp was required to circulate 0.52 gpm in the 3-kwe system. This current requirement is 50 amp in excess of the specified maximum value (40 amp as per the 3-kwe system specification). Methods to reduce this value were being considered when the program was redirected.

2. Startup (Coolant Injection) Test

Figure 17 plots the results of Hg injection (simulated with NaK) performed on SN-012. This was one of three injections performed on this unit as part of an acceptance test prior to delivery of the unit to a system. The pump was instrumented to monitor the two fluid inlet and outlet temperatures and the four critical pump temperatures. As expected, the most severe shocks occurred at the coolant tube - cold junction interface of the pump. Injection was accomplished with room temperature NaK and maximum shock rates of 350°F/sec to 500°F/sec were obtained for three injections. These rates were controlled by the injection valve setting. Figure 18 compares the steady-state performance of the unit before and following each of the injections;

no measurable change in performance occurred within experimental error limits.

3. Steady-State Performance at Design Thermal Conditions

Figure 19 plots the performance characteristic for each of the 10 Prototype-I units tested; 4 of these units performed at, or above, the design performance goal.

In general, the better performing units were those fabricated later in the program, thus reflecting the success of the fabrication development effort.

Endurance at Design Thermal Conditions

A total of 17,000 hr at design thermal conditions (1200°F) were accumulated on 5 test pumps during the Prototype-I development phase. Of these, 15,000 hr were performed in air and approximately 2000 hr were in a vacuum environment. Figure 20 shows the total and individual pump test hours. As noted, the last two units (SN-011 and 012) were tested for relatively short periods in vacuum. These tests were terminated early to make these units available for system test work. Of the air-tested units, degradation rates (as measured by-flowrate in a 3-kwe system) were below 10% with one exception.

The degradation rate of SN-006 increased sharply following a thermal cycle after 3000 hr of operation. It was concluded that oxidation of the copper base braze alloy weakened the joints to the point where the stress levels during the thermal cycle were sufficient to cause rupture.

Pump SN-008 failed after 6000 hr due to a throat duct leak. Subsequent investigations revealed that failure was due to metal fatigue of the throat between the TE blocks. Design changes were subsequently made to stiffen the throat in these areas. Figure 21 plots the degradation-time characteristics of these units. As noted degradation rates for vacuum tested units were well within the design requirement for the periods tested.

V. CONCLUSIONS

The performance and stability test results for the TEM pump developed during this phase (for the 3-kwe system) of the MRP demonstrated that the Prototype-I design is capable of meeting all major specified pumping requirements. Design procedures developed and used were proven to be adequate as indicated by the close to design point performance in Figure 22. A rigorous test program enabled selection of the most suitable materials and uncovered the potential operational problem areas early in the program. Effective steps have been taken for their correction in the pump design for the advanced MRP systems; primarily in the area of fatigue failure of the throat. New designs call for stiffening the throat in the failed sections. The only other problem area, the observed degradation following thermal cycling after extended time periods for the air operated units, has lead to the restriction of operation to either an inert or vacuum environment.

TABLE I
CHROMEL-CONSTANTAN TEM PUMP
DESIGN CHARACTERISTICS

Weight	Total weight-45 lb, magnet system weight-30 lb, throat assy-15 lb
Length	13.5 in. including transition pieces, 10.4 in. pump length
Throat	0.8 x 0.8 in. inside flow area, 316 SS, 0.020 in. wall thickness
TE	Chromel P and Hoskins alloy 832 (constantan), 0.8 x 1 in. cross-section
Coolant Tube	3/8 in. OD, 0.020 in. wall, Haynes-25 alloy
Magnet	Alnico 5 Arnold Engineering Co., separator magnets Part No. 3204 pole shoes, armco iron
Bus Work	OFHC copper

TABLE II
NOMENCLATURE FOR EQUATIONS
(Sheet 1 of 4)

Symbol	Description	Equation	Dimension
E_G	Seebeck voltage at element, dx	$E_G = S \eta_T \Delta T_x$	volts
S	Seebeck coefficient		V/°F
η_T	Fraction of total fluid-fluid ΔT appearing across TE element.	$\frac{1}{1 + \frac{K_{TE}}{K_{HA25}} + \frac{K_{TE} t_{ss}}{K_{ss} t_{TE}} + \frac{K_{TE} t_{cu}}{K_{cu} t_{TE}}}$ $+ \dots + \frac{1}{\frac{K_{TE}}{K_{HA25}} + \frac{t_{HA25}}{t_{TE}} + \frac{K_{TE}}{\sqrt{h_{gf}} t_{TE}}}$	
K_i	Thermal conductivity of ith element		watts/in-°F

TABLE II
NOMENCLATURE FOR EQUATIONS
(Sheet 2 of 4)

Symbol	Description	Equation	Dimension
η_N, η_H	NaK and Hg fluid film coefficient		watts/in ² -°F
t_i	Thickness of ith element		in.
ΔT_x	Fluid to fluid temperature difference at x	$\Delta T_x = \Delta T_o \epsilon^{-\frac{x}{L_o}}$	°F
ΔT_o	Fluid to fluid temperature difference at coolant inlet (x=0)		°F
ϵ	Base, natural logarithm	2.718	
x	Distance from coolant entrance point		in.
L_o	Factor	$WC_P R'$	
W	Coolant max flow rate		lb/min
C_P	Specific heat of coolant fluid		$\frac{\text{watt-min}}{\text{lb-°F}}$
R'	Per unit length thermal resistance between fluids	$\frac{1}{b} \left[\frac{1}{\eta_N} + \frac{t_{ss}}{K_{ss}} + \frac{t_{TE}}{K_{TE}} + \frac{t_{cu}}{K_{cu}} + \frac{t_{Ha25}}{K_{Ha25}} + \frac{1}{\eta_{Hg}} \right]$	$\frac{\text{in.-°F}}{\text{watt}}$
b	Lateral dimension of heat transfer elements		in.
r_T	Total generator-transmission resistance	$r_T = r_{TE} + r_b + r_{ss}$	ohm
r_{TE}	TE resistance	$r_{TE} = \frac{(\rho_N + \rho_P) t_1}{b \, dx}$	ohm
ρ_N	Resistivity of constantan		ohm-in.
ρ_P	Resistivity of chromel		ohm-in.
t_1	Thickness of TE in direction of current		in.
b	Thickness of TE perpendicular to current		in.

TABLE II
NOMENCLATURE FOR EQUATIONS
(Sheet 3 of 4)

Symbol	Description	Equation	Dimension
dx	Differential pump length		in.
r_b	Copper bus resistance	$r_b = \frac{\rho_{cu} l_{cu}}{n_2 dx}$	ohm
ρ_{cu}	Resistivity of copper		ohm-in.
l_{cu}	Effective length of copper		in.
n_2	Effective thickness of cu perpendicular to current		in.
r_{ss2}	Resistance of throat duct wall perpendicular to current	$r_{ss2} = \frac{\rho_{ss} 2t}{b dx}$	ohm
di_T	Total current in element dx		amp
r_s	Total leakage path resistance (principally through throat duct walls)	$r_s = \frac{\rho_{ss} a}{2t_2 dx}$	ohm
ρ_{ss}	Stainless steel throat resistivity		ohm-in.
a	Channel width		in.
t_2	Wall thickness		in.
di'	That portion of the total current in element dx which contributes to pumping		amp
r_N	NaK resistance	$r_N = \frac{\rho_{NaK} a}{b dx}$	ohm
ρ_{NaK}	Resistivity of NaK		ohm-in.
B	Back-generated voltage due to fluid passage through magnetic field	$E_B = B a v 10^{-8}$ or $E_B = 3.86 \frac{BQ}{b} 10^{-8} = K_3 \frac{BQ}{b}$	volt volt
B	Magnetic field intensity		lines/in. ²
v	Fluid velocity		in./sec
Q	Volume flow-rate		gpm

TABLE II
NOMENCLATURE FOR EQUATIONS
(Sheet 4 of 4)

Symbol	Description	Equation	Dimension
ΔP_H	Hydraulic pressure drop	$\frac{2fDV^2L}{gd} + \frac{K_H DV^2}{2g}$	psi
K_3	Constant	$3.86 \cdot 10^{-8}$	
D	Density of fluid		lb/in. ³
V	Velocity of fluid		in./sec
g	Gravitational acceleration	384	in/sec/sec
d	Hydraulic diameter		in.
f	Friction factor	$0.079 R_e^{-0.25}$	
R_e	Reynolds number	$d V_D / \mu$	
K_H	Number of velocity heads due to inlet & exit losses		
ΔP_E	Eddy current braking pressure drop	$\frac{3.41 \cdot 10^{-15} k Q B^2}{b^2 N^2 k}$	psi
k	a constant	0.21	

REFERENCES

1. Atomics International Staff, "The Mercury Rankine System and Potential Application," AI-65-79, April 22, 1965
2. M. A. Perlow, "SNAP 10A Thermoelectric Pump Development," NAA-SR-9488, June 3, 1964
3. "Specification for SNAP 2 Thermoelectric Pump (Chromel-Constantan)," AI Specification NR7550-41
4. H. H. Hanlin, "Development of the Thermoelectromagnetic Pump for the 3-kwe Mercury Rankine System," NAA-SR-11198
5. "Liquid Metals Handbook-Sodium NaK Supplement," 3rd Ed., Washington, D.C., the U.S. Atomics Energy Commission and the U.S. Dept. of the Navy, 1955. (Pump Section, p 297)
6. "Analyses of Experimental D.C. Pump Performance and Theory of Design," D. A. Watt, AERE R/E 2275, 1957
7. C. Zener, "Thermal Electric Power," (Westinghouse Research Report) R-94460-Z-0, August 1952
8. W. D. Leonard, "Thermoelectric Pump Design," NAA-SR-TDR 8841

8-20-64

7636-0460C

Fig. 1 - MRP Space Powerplant

Fig. 2 - TEM Pump for
3 KWE System

Fig. 3 - TEM Pump with Magnet
System Removed

Fig. 4 - Temperature Profile

Fig. 5 - Two-Step Braze Bonding

Fig. 6 - Faraday Type DC Conduction Pump

4-9-65

7636-0823

Fig. 7 - DC Conduction Pump Equivalent Circuit

4-9-65

7636-0880

Fig. 8 - TE Generator

4-9-65

7636-0826

Fig. 9 - DC Conduction Pump Integrated with
T. E. Generator

Fig. 10 - Temperature Change Along Pump Length

Fig. 11 - TEM Pump Differential Length

4-9-65

7636-0808

Fig. 12 - Simplified Differential Pump
Electric Circuit

4-9-65

7636-0807

Fig. 13 - Coolant Fluid at X

Fig. 14 - Parameter Variation
Effect on Total Pump Weight

Fig. 15 - Functional Diagram of TEM Pump Test Facility

Fig. 16 - SN-012 Pre-startup Current Requirement Data

Fig. 17 - Coolant Injection Transient Results - SN-012 (Cycle No. 2)

Fig. 18 - Performance of SN-012 Before and After Each Injection Cycle

Fig. 19 - Prototype I TEM Pump Performance Summary

Fig. 20 - Accumulated Time at 1200°F for TEM Pumps During Test Program

Fig. 21 - TEM Pump Degradation Characteristics in Life Test Program

Fig. 22 - Initial Performance History for Prototype I TEM Pumps

-4 20:5184

EXPERIENCE WITH TURBINE-DRIVEN POTASSIUM BOILER-FEED PUMPS
FOR THE MEDIUM POWER REACTOR EXPERIMENT*

L. V. Wilson
Design Engineer, Oak Ridge National Laboratory
Oak Ridge, Tennessee

H. C. Young
Development Engineer, Oak Ridge National Laboratory
Oak Ridge, Tennessee

A. M. Smith
Development Engineer, Oak Ridge National Laboratory
Oak Ridge, Tennessee

ABSTRACT

A series of potassium vapor turbine driven boiler feed pumps with potassium lubricated bearings has been designed, built and tested for a total of 2000 hr of operation at high temperatures. This paper summarizes the problems that have arisen in the course of the design, fabrication, and operation of these units. Approaches to the thermal expansion problems, provisions for lubricant flow to the bearings, speed pickups, and maintenance of moving parts installed in hermetically sealed systems are discussed. Test experience with the units operated with steam and water is related to the test experience obtained in potassium systems. The problems encountered in fabricating tungsten carbide parts, difficulties both with the brittleness of TZM molybdenum alloy and its tendency to gall and seize, together with solutions that have been found effective for coping with these problems are outlined.

Introduction

One of the basic precepts for the Medium Power Reactor Experiment (MPRE) Program was that, to obtain a system with a high degree of reliability, it is essential that a potassium vapor turbine-driven boiler feed pump using potassium lubricated bearings be developed. Work at Knolls Atomic Power Laboratory** had shown that tungsten carbide running against tungsten carbide in NaK was very resistant to scuffing, and hence should

*Research sponsored by the U.S. Atomic Energy Commission under contract with the Union Carbide Corporation.

**"Materials Compatibility Report," compiled by Bearing and Oil Seal Branch of the Bureau of Ships, USN.

be suitable for withstanding the boundary lubrication conditions that prevail in startup and shutdown in high temperature potassium. Since this point raised a basic question of materials feasibility, it evidently deserved attention early in the development program.

Other feasibility questions included turbine bucket erosion by wet vapor and cavitation damage to the impeller of a pump operating continuously in cavitation. It was clear that much insight into all of these problems could be provided by the operation of a small potassium Rankine cycle system with a free turbine-driven boiler feed pump, hence work was initiated on the smallest size of system that promised to be meaningful. The turbine pump, referred to as the ORNL turbine pump, for this system was designed and produced by ORNL. For two following larger systems called the intermediate (IPS) and large potassium systems (LPS), two other units, referred to as the Aeronutronic Mark I and Mark II turbine pumps, were designed and produced in conjunction with the Aeronutronic Division of the Philco Corporation.

A number of problems are inherent in the design of any piece of rotating machinery and, as we all know, these problems are compounded in rotating machinery for use in liquid metals. Quite often a problem that should have been recognized and resolved in the early design stages is either overlooked or its importance is underestimated, a single operating condition is specified instead of a realistic range of operating conditions, or some rationalizing assumptions are made that evade a problem. In most cases it is probably more realistic to build a prototype unit, knowing that some problem areas may be overlooked, test the unit in low temperature liquids, and make the necessary modifications before going into the high temperature hardware. Unfortunately, the schedules for other components often require simultaneous fabrication of low temperature prototype and high temperature hardware. Thus although most of the basic development problems in the turbine-pump units showed up during water tests, it was not always possible to make the implied modifications in a potassium unit phased only a little behind the water prototype.

Material Selection

Our design philosophy has been to make the mechanical design of the unit as simple as possible and yet obtain a reliable set of bearings. In approaching the bearing design

problem, tungsten carbide with 12% cobalt binder was selected for the mating bearing surfaces on the basis of the previously mentioned tests by Knolls Atomic Power Laboratory. Its use presents some major design problems since its coefficient of thermal expansion is 3×10^{-6} in/in-°F as compared to 9×10^{-6} in/in.-°F for the stainless steel used in the rest of the system. While it is difficult to design for use of an austenitic stainless steel rotor running in tungsten carbide bearings, TZM molybdenum has a coefficient of thermal expansion very close to that of tungsten carbide with 12% cobalt binder and is much stronger at high temperature than the austenitic stainless steels. Hence it was decided to match the coefficient of thermal expansion of tungsten carbide to that of a molybdenum alloy rotor in the region where clearances are most vital, i.e., in the bearings, and make use of other provisions in the less sensitive zone between the bearing supports and the casing. The use of a molybdenum rotor offered the additional advantage that molybdenum is one of the very few materials that shows promise for operation at 2000°F in potassium vapor.

Bearing Tests

The first step in implementing the development program was the operating of a bearing test rig that had been used in the Molten Salt Reactor Development Program. The most expeditious approach to the use of this rig was to mount a stainless steel journal on the existing Inconel shaft and run it against a tungsten carbide sleeve bearing. This test consisted of a series of cycles in which the shaft was started up under light load, the load was then manually increased as the speed was increased, then full load and speed were maintained for a period of an hour after which the load and speed were proportionately reduced and the shaft was brought to a stop under the original starting load. In one test, a plain stainless steel journal running against a tungsten carbide bearing in 800°F potassium showed very little change in the power required to drive the shaft after 160 cycles. On disassembly the journal was moderately scuffed. The test was repeated with a chrome-plated stainless steel journal running against a second tungsten carbide bearing. This test was run for 450 cycles at which point it was decided to shift to a continuous endurance test. The test bearing was still operable at shutdown after a total operating

time of 4085 hr, and the journal and bearing were both in excellent condition.

ORNL Turbine Pump

Concurrently with the above work on bearings, work was initiated on the construction and operation of a small potassium vapor turbine-driven feed pump employing a TZM molybdenum rotor with tungsten carbide journal bearing sleeves brazed to the shaft, and tungsten carbide sleeve bearings mounted in a stainless steel housing with provisions to accommodate differential thermal expansion between them. This pump, shown in Figs. 1 and 2, is a fairly small unit having a turbine wheel diameter of 2.75 in. and an impeller diameter of 1 in. No effort was made to design the impulse turbine for good efficiency because more than adequate vapor flow was available from the boiler. The flow passages of the pump impeller are 7/64 in. diameter drilled holes which discharge into a fully concentric collector. Design and operational criteria for the turbine pump are shown in Table 1.

The rotor is supported on two potassium-lubricated, full-journal bearings each having a diameter of 0.5 in., a length of 0.25 in., and a radial clearance of 0.0004 in. The axial loads are carried by radially-grooved, flat-plate thrust bearings. Unfiltered potassium is fed directly into the bearings from the collector. The turbine wheel, pump impeller, and shaft are fabricated from TZM molybdenum (0.5% Ti, 0.08% Zr, 0.03% C). The tungsten carbide journals are brazed to the TZM molybdenum with a nickel-boron-silicon brazing alloy. In this case, the differential thermal expansion problem was handled by mounting the tungsten carbide bearing sleeve on stainless steel cantilever beams designed to be radially "soft" but translationally stiff.

The first step in the development program for this small unit was to build a unit of conventional materials for test in a water-steam system. This was done not only to check out the performance of the pump but also the problems of "bootstrapping" a free turbine-driven feed pump of this sort in a hermetically sealed system and the general problems of stability and control that might be associated with the system. Performance tests on this prototype unit were made in an open steam system (not in a closed Rankine cycle) in order to determine some of the operating characteristics. Fig. 3 shows the head-flow curves for several speeds. The flatness of the curves stemmed from the use of a concentric

collector with a rather generous flow area. Typical performance curves for the turbine-pump unit for various turbine inlet pressures are shown in Fig. 4. By obtaining performance data in this manner it is possible to analyze the problem of the pump performance relative to the Rankine cycle requirements for a range of system power levels.

Several problems arose during the developmental water tests of the unit. It was necessary to alter the bearing lubricant paths in order to assure adequate lubricant supply and to control lubricant leakage to the turbine cavity. The outside diameter of the impeller suction shroud was reduced to reduce the axial forces on the thrust bearing. Originally the turbine pump had only unidirectional thrust capability; this arrangement was found to be inadequate for a wide range of system operating conditions. A thrust bearing for carrying reversed loads was added.

Because of the smallness of the unit and the high rotational speed it was found to be desirable to maintain extremely close quality control on the bearing surfaces. The squareness and eccentricity of these surfaces was held to about .0001 TIR or better - usually much better. We are not too sure about the importance of the bearing surface finishes, which were held to about 4 rms, since corrosion, erosion, and the abrading effects of starting and stopping probably affect the finish.

Upon completion of the prototype tests in water, the first of the turbine pumps was installed in a small potassium system. Potassium vapor from the boiler and separator passes through the turbine throttle valve, the turbine, and into the condenser. The condensate enters an inert gas trap where any non-condensables may be removed. Liquid potassium is subcooled downstream of the trap before entering the pump suction. Liquid from the pump discharge is then fed back through the preheater to the boiler. A parallel boiler feed line containing an Electrodynamic pump is used to supply the boiler feed during the system startup and is valved out of the system once the turbine pump is in operation. Procedures for startup and operation of the system are discussed in more detail in a companion paper* presented during this conference.

*R. E. MacPherson, "Potassium Rankine Cycle Operating Experience for the MPRE." Proceedings of the First Annual Rankine Cycle Space Power System Specialists Conference, AIAA, Cleveland, Ohio, October 26-28, 1965, (Confidential).

After a turbine pump is installed, the entire system is evacuated, flushed with a cleaning charge of potassium, and then filled with the operating charge of potassium. The boiler feed is supplied by the Electrodynamic pump until the desired system conditions are obtained, at which time the turbine pump is brought onto the line.

A typical cavitating condition will have the pump oscillating at a frequency of about 10 cycles per second between a 35% and 75% cavitation coefficient (C) which is defined as

$$C = \left[\frac{P_H - (P_d - P_c)}{P_H} \right] \times 100$$

where P_d = pump discharge pressure, psia

P_c = condenser pressure, psia

P_H = non-cavitating head capability of pump, psi.

A cavitation coefficient of 75% means that the pump is providing 25% of its non-cavitating head at that speed. None of the impellers have shown any evidence of cavitation damage to date although most of the test durations were short.

A total of eight ORNL turbine pumps has been installed and tested in the facility. A summary of the individual tests is shown in Table 2. Of the eight units installed, seven were successfully started and operated in potassium for various time intervals ranging from 30 minutes to 327 hr before it became necessary to terminate the test. One turbine (Unit No. 4) could not be started after installation. During the 327-hr test period of Unit No. 6, one continuous run of 147 hr was obtained without any adjustments being made to the system. Total operating time on the seven units tested has amounted to 540 hr with 129 starts and stops. All of the 72 starts for Units 1, 2, and 3 were "bootstrap" starts; i.e., without the use of the Electrodynamic pump.

Failure of the threaded portion of the molybdenum shaft on Unit No. 8 has led to the design of an elastic washer of Cb - 1 Zr alloy that will be used on future units. The purpose of this washer is to maintain a constant axial force on the turbine wheel and thus prevent the possibility of over-tightening the retaining nut during installation.

After several unsuccessful attempts were made to operate Unit 4, particulate matter was found to have locked a thrust bearing. Various attempts at filtering the potassium were not successful in succeeding units, since particulate matter apparently caused bearing seizure in Units 5 and 6. It would have been desirable to filter the bearing lubricant supply, but because of the internal flow path from the impeller periphery, this was impossible without a major redesign of the turbine pump.

Aeronutronic Mark I Turbine Pump

When the preliminary tests of the steam-water prototype of the ORNL turbine pump turned out favorably, the design of a new model of turbine pump suitable for a larger potassium system was roughed out and the design requirements were defined. Since we were very short of design personnel at the time, and since we wished to take advantage of the government investment in the development of mercury and potassium vapor turbines, a set of performance specifications was prepared and sent to the organizations working in this field. As a result of this solicitation a contract was placed with the Aeronutronic Division of the Philco Corporation for the fabrication of one unit for operation on steam and water and two units for operation on potassium. The contract did not call for any guarantees for operation on potassium; ORNL assumed full responsibility for this phase of the work.

The Aeronutronic Mark I turbine pump (see Figs. 5 and 6) was designed to meet the requirements of an intermediate boiling potassium system. It has a turbine wheel diameter of 6.3 in., an impeller diameter of 2.75 in., and operates at a design speed of 8000 rpm. The turbine is a single-stage, partial-admission, impulse type driving an open faced pump impeller. Design and operational data for this turbine pump are shown in Table 1. The rotating component is supported on two full journal bearings each having a diameter of 1.0 in., a length of 0.5 in., and a radial clearance of 0.0017 in. The thrust bearings are of a hybrid type - being pressurized at the inside diameter and outward through radial grooves extending into the flat thrust face. The grooves are dammed at the outside diameter to prevent excessive through-flow of potassium into the turbine cavity. The pads between the grooves generate hydrodynamic films.

Potassium for lubricating the bearings is taken from the pump discharge line and passed through a 10-micron sintered metal filter before entering the bearing housing. The flow enters an annulus between the two journal bearings where it is divided in two and passes through each journal bearing-thrust bearing combination in series and thence into the turbine cavity and the pump cavity.

The materials are identical to those selected for the ORNL turbine pump; i.e., the shaft, turbine wheel, and impeller are fabricated from TZM molybdenum and all bearing surfaces are tungsten carbide (12% cobalt binder). The journals are ground for a sliding fit on the molybdenum shaft rather than being brazed to the shaft as was done on the ORNL turbine pump. The resulting multiplicity of fits was not conducive to obtaining and reproducing good concentricity and squareness at assembly.

Differential thermal expansion between the turbine housing (type 316 stainless steel) and the bearing housing (molybdenum) was taken care of by allowing the bearing housing flange to slip under the heads of the bolts that hold the bearing housing in place. This was later felt to be unsatisfactory due to (1) the lack of control of slip forces, (2) the lack of concentricity of the rotary element within the housing, and (3) the possibility of damage to the mounting flange and static seal resulting in excessive flow of potassium into the turbine cavity.

Water tests were run that indicated an excessive thrust load on the turbine end thrust bearing. The pressure differential between the turbine cavity and the pump suction for a zero axial force on the thrust bearings was deduced from the rate of change in the flow of bearing lubricant into the turbine cavity as a function of this differential pressure. Fig. 7 shows typical curves of this leakage flow versus the differential pressure. The intersection of the two portions of the flow rate curve was interpreted as the differential pressure at which the axial thrust reversed directions. Vanes were machined into the back of the impeller to reduce the pressure there and consequently to reduce the thrust load to an acceptable level. Unfortunately, time did not permit this change to be incorporated in Unit No. 1, the first unit that was operated in potassium. It was, however, incorporated in Unit No. 2.

The turbine vapor supply and discharge lines enter the turbine cavity through the cover flange which has the turbine nozzle ring mounted to its inside surface. This design was found to be quite inconvenient because it required cutting and rewelding the vapor lines in order to maintain the turbine pump rotary assembly.

Typical water test pump performance curves are shown in Fig. 8. Fig. 9 shows typical water cavitation data which agrees with predicted values for the required NPSH at the point of incipient cavitation.

Vibration measurements on the Mark I turbine pump housing taken during water tests revealed a large amplitude at a frequency corresponding to one-half the shaft speed. By raising the water temperature from 80°F to 200°F and thus reducing the viscosity of the water lubricant to one-third of its original value, the amplitude of vibration was drastically reduced at 6000 rpm. The reduction in amplitude was much less at the design speed of 8000 rpm. This prompted a serious consideration of bearing half-speed whirl. As a result, Unit No. 2 of the Mark I design was modified by reducing the weight of the turbine wheel in order to better distribute the bearing loads. Also a more sophisticated analysis of the bearing characteristics and rotor dynamics was made for the Mark II design.

The intermediate potassium Rankine cycle system was not specifically designed to obtain turbine pump performance data; however, since an electromagnetic pump was used for system startup, it was possible to operate the turbine pump in a closed loop, comprised of the condenser jet pump recirculation system, and obtain some non-cavitating pump performance on one flow resistance line. These data correlated very well with water test data and increased the confidence in predictions of the cavitation coefficient.

Unit No. 1 of the Mark I design operated for a total of 18 hr and was subjected to a total of six starts and stops. The bearing drag, as measured by the turbine nozzle differential pressure required for starting, became progressively higher with each startup. The difficulty stemmed from having the vapor throttling valve in the turbine exhaust line because this gave a high bearing load under starting conditions.

Unit No. 2 basically had two modifications, both of which have been mentioned previously, that would affect the operation of the unit. The turbine wheel was lightened to improve the loading conditions for the journal bearings and vanes were machined into

the back of the impeller to reduce the thrust bearing load. In addition, the system was modified to locate the vapor throttling valve in the turbine inlet line.

Since being placed in operation, Unit No. 2 has accumulated a total of over 1500 hr of operation with 14 starts and 13 stops. Table 3 shows typical operating conditions for the turbine pump over a range of system operating conditions. Notice that, for non-cavitating operation, the pump suction pressure is relatively high, while for cavitating operation the pump suction pressure is reduced.

Speed measurements of the potassium units are made with a high temperature electro-magnetic pickup made by Electro Products Laboratories and rated for 5000 hr of operation at 800°F. The interruption of its magnetic field is provided by small slugs of Hyperco 27 located in the impeller outside diameter. Another slug of Hyperco 27 is brazed into the turbine pump housing to enhance the magnetic coupling between the pickup and the rotating Hyperco slugs. The pickup is connected to an appropriate amplifier and counter. Its use has provided satisfactory speed measurements at about 700°F. However, after 1000 hr, there are indications that the generated voltage had decreased more than 50%.

Aeronutronic Mark II Turbine Pump

Extensive test experience with the seven ORNL turbine pumps and two of the Aeronutronic Mark I units made it clear that a new design should be prepared based on the Mark I design but with numerous modifications. A new contract was negotiated with Aeronutronic for the redesign work and the fabrication of the stainless steel and TZM molybdenum parts for five turbine pumps. We are procuring tungsten carbide parts in rough form from vendors and are having them finished to meet the tight dimensional tolerances and surface finish requirements that we have found to give satisfactory performance. These five units have been designed so that they can be used interchangeably in any one of the three potassium Rankine cycle systems of widely different sizes employed in our development program. This approach has the advantage that it cuts the cost for the engineering design, procurement and development, as well as the costs associated with the inventory of spare parts. In addition, considerable operating experience can be obtained with one design.

The Mark II turbine pump, shown in Fig. 10, is basically a modification of the Mark I unit designed to meet the needs of the large potassium system. The increased power is obtained by using a two-stage, partial admission, impulse-type turbine. Additional flow capacity is provided by using two diffusers from a concentric collector. The impeller is a Barske wheel, which is completely open; i.e., no front or back shroud, just four radial vanes. Operational and design criteria for the turbine pump is shown in Table 1.

As the design of the Mark II turbine pump progressed it became apparent that the new design could, by virtue of its improved turbine performance characteristics, be operated as a replacement for both the ORNL and the Aeronutronic Mark I turbine pump in their respective systems as well as in the large boiling potassium system for which it was designed. In the small potassium system it would require all the boiler vapor flow to obtain about one-half of the turbine pump design flow at design head and speed, but in the intermediate potassium system it could be operated at design conditions.

Many features in the Mark II design were the result of experience gained in designing, fabricating, maintaining and operating both the ORNL and the Mark I turbine pumps. Based on this experience, several precepts were established that were to serve as guidelines in the design of the Mark II turbine pump. Some of the more notable precepts and their backgrounds are as follows:

Access. We felt that since the turbine pump was one of the more likely components in the system to give trouble in the early stages of the development program, it should be designed so that the entire rotary assembly, i.e., turbine wheel, shaft, impeller, bearing housing and retaining flange, could be removed and a new one installed without cutting any of the lines to the unit. Removal of a rotary assembly from the intermediate potassium system was a considerable inconvenience, as noted previously, requiring several new weld joints and making it difficult to avoid contamination of the system.

Assembly and Disassembly. The bolts holding the rotary assembly in place should be located outside the outer diameter of the turbine wheel so that the entire rotary assembly could be removed from the turbine housing without removing the turbine wheel from the shaft.

Screw holes should be provided for jacking the rotary assembly out of the turbine housing should that be necessary.

All mating assembly surfaces should be selected to eliminate material combinations with bad galling characteristics. It was found in the Mark I unit that the clean TZM molybdenum turbine wheel bore sliding on the clean TZM molybdenum shaft was extremely susceptible to galling while being assembled and disassembled. This material combination in particular was to be avoided in the Mark II design. Wherever possible, sliding fits should consist of tungsten carbide against itself or against molybdenum.

To facilitate the disassembly of the rotary assembly, the shaft and nut screw threads should receive special treatment. The threads (National Coarse) should be ground and lapped and the roots and crests of the thread profile should be given a radius to reduce surface asperities and discontinuities that would cause high contact stresses and subsequent galling, especially under the combined effects of high temperature and potassium. All retaining nuts (TZM molybdenum) should be carburized and provided with slots to make it easier to cut or break them off should galling or sticking occur. The threads on the cap screws (304 stainless steel) holding the rotary assembly into the turbine housing (304 stainless steel) should be nitrided to reduce galling.

Operation. To avoid bearing loading conditions conducive to bearing failure, the center of gravity of the assembled rotating parts is located well inboard of the two journal bearings and the casings are designed so that the fluid dynamic forces on the turbine and impeller should be oriented to give satisfactory bearing load conditions over the full operating range.

Differential Thermal Expansion. To accommodate the differential thermal expansion of the tungsten carbide bearing sleeve and the type 304 stainless steel turbine housing, a set of conical surfaces should be used. The apices of the two cones have a common point of intersection on the centerline of the shaft so that the differential thermal expansion between the two parts results in slippage along the conical surfaces with no change in the contact pressures. The contact pressure on the conical faces is maintained by means of a preloaded spring washer that will provide a force of approximately 100 lb.

A test of a pair of preloaded conical surfaces (tungsten carbide versus nitrided stainless steel) in potassium has been initiated. An earlier test of TZM molybdenum versus stainless steel in potassium consisted of thermally cycling the test pieces from 1050°F to 1450°F through 42 cycles and from 200°F to 1450°F through 4 cycles. The damage to the surfaces was slight.

Bearings

The rotor of the Aeronutronic Mark II turbine pump is supported on two potassium lubricated bearings each having a diameter of 1.0 in., a length of 0.75 in. and a radial clearance of 0.0008 in. To provide some operating margin against bearing whirl the radial clearance was decreased from the 0.0017 in. used in the Mark I design. In addition, each bearing has two axial grooves located in the upper half. These are located 45° on either side of the vertical centerline to suppress half-speed whirl. This groove geometry was chosen because there is some question as to the exact direction of the bearing loads over the entire operating range of the turbine pump. The axial grooves do not extend over the entire length of the bearing in order to reduce the lubricant flow rate, particularly into the turbine cavity.

The loads on the journal bearings are made up of three components:

1. Radial resultant of the tangential impulsive forces acting on the turbine wheel.
2. Radial hydraulic thrust on the impeller.
3. Weight of the rotary element.

There is considerable control over the resultant bearing forces through the location of the turbine nozzle blocks and the two pump diffusers. In accordance with the precepts established for the Mark II design, the center of gravity of the rotary components has been located inboard of the two journal bearings by reducing the weight of the turbine wheel and increasing the span between bearings. Using the shaft made entirely of tungsten carbide also helps a little. The Mark II design has the turbine nozzle blocks and pump diffusers oriented to give equal journal bearing loads of about 3 lb at the design conditions.

An analysis of the bearing characteristics and rotor dynamics was made by Mechanical Technology, Inc.* This study indicates that over most of the operating range the journal bearings are operating in the laminar regime. It is only when the design condition of 8400 rpm is exceeded that the bearing will be in the turbulent regime. However, this is not of much importance since the main effect of turbulence is to increase the friction losses in the bearing. The minimum film thickness in the bearing was between 0.0001 in. and 0.0004 in., being about 0.0003 in. at the design point. As a result of the analysis it was decided to increase the bearing length from the 0.5 in. of the Mark I design to 0.75 in. in order to increase the minimum film thickness.

The completely open impeller was used to reduce its contribution to the thrust bearing load at all operating conditions. Thus, axial thrust is relatively independent of the turbine pump speed and primarily dependent on the differential pressure maintained between the turbine cavity and the pump suction. It is believed best to keep this differential pressure slightly biased so that the turbine end thrust bearing will be loaded to prevent an excessive flow of liquid potassium into the turbine cavity.

The thrust bearings are of a design similar to that used in the Mark I unit as is the flow path of the lubricating potassium. The base materials used in the Mark II units are the same as those used in the Mark I units except that the turbine pump housing is type 304 stainless steel.

Thermal Sleeve

To meet the requirements of the precepts set forth for the design of the Mark II unit, namely that the piping to the turbine pump would not have to be cut for removal of the rotary assembly of the unit, the turbine inlet line was brought through the body of the turbine pump housing adjacent to the bearing and pump portion of the unit. At the design conditions of the unit this meant that there would be a thermal gradient from the 1540°F vapor to the 1000°F liquid. A thermal sleeve was installed to provide a temperature gradient that would not result in excessive stresses.

*J. E. Lund, "Analysis of the Bearing Characteristics and the Rotor Dynamics of the Mark II Turbopump," MTL-65TR38, July 14, 1965.

Water Tests

Some test results have been obtained from operating a water mockup of the Mark II turbine pump while driving the turbine with nitrogen. The overall efficiency of the turbine pump versus pump flow is shown in Fig. 11. Head-flow test data at various speeds are shown in Fig. 12. Pressure data were also taken to determine the pressure distribution around the collector from which the resultant radial thrust on the impeller and its direction was calculated for use in the bearing analyses.

Status

At present the components of the Aeronutronic Mark II turbine pumps are being fabricated. A prototype, having the same basic configuration as the Mark II potassium unit, has been designed for testing the turbine pump with water. The radial clearance of the journal bearings has been increased to about twice that used in the Mark II potassium unit to provide equal Sommerfeld numbers and equal Taylor numbers for operation with water at 120°F and potassium at 1050°F. Induction type proximity probes are located adjacent to each journal bearing to measure bearing eccentricities, attitude angles, and the onset of instability. Another proximity probe is to be installed for measuring axial displacement of the rotor, in particular, for determining the pressure conditions for zero axial load on the thrust bearings. The bearing surfaces are chrome plate against bearing bronze. All the other components are stainless steel except for the turbine wheel which is a Mark II TZM molybdenum turbine wheel. Pressure taps are located in the collector of the pump to measure the pressure distribution around the impeller. This is being done to determine the effect of cavitation on the radial thrust on the impeller. A water-steam test stand has been designed for testing the prototype. The necessary design for installing a Mark II potassium unit in a small potassium system has been completed.

Conclusions

1. We have found that it is feasible to design potassium vapor turbine driven centrifugal pumps for use in potassium Rankine cycle systems and to obtain successful operation. A wide range of operating conditions has not been investigated.

2. The bearing material combination of tungsten carbide against itself (hard on hard) has proven to be operable in potassium up to 1000°F.
3. The performance characteristics of both the turbine and the pump (Aeronutronic Mark I turbine pump) have shown no signs of deterioration after 1000 hr of operation in potassium. During most of this time the pump has operated with cavitation coefficients varying from 0 to 33% but normally at about 15%.
4. Steam-water tests provided a good insight into most of the mechanical problems and provided adequate measures of the thermodynamic and hydraulic characteristics of the turbine pumps.

Bibliography

J. W. Lund, "Analysis of the Bearing Characteristics and the Rotor Dynamics of the Mark II Turbopump," MTI-65TR38, July 14, 1965.

R. E. MacPherson, "Potassium Rankine Cycle Operating Experience for the MPRE," Proceedings of the First Annual Rankine Cycle Space Power System Specialists Conference, AIAA, Cleveland, Ohio, October 26-28, 1965 (Confidential).

"Materials Compatibility Report," compiled by Bearing and Oil Seal Branch of the Bureau of Ships, USN.

Table 1. Turbine Pump Design and Operational Criteria

	ORNL	Aeronutronic	
		Mark I	Mark II
Speed, rpm	to 25,000	to 8,000	to 8,400
Turbine, inlet temp., °F	to 1,540	to 1,540	to 1,540
Turbine inlet pressure, psia	30	30	15
Turbine vapor flow, #/hr	130	280	180
Turbine exhaust temp., °F	1,050	1,050	1,140
Turbine exhaust pressure, psia	1.5	1.5	3.0
Pump flow, gpm	0.4	8.0	27.0
Pump suction temp., °F	1,050	1,050	1,050
Pump suction pressure, psia	1.5	1 to 4	1 to 4
Pump discharge pressure, psia	70	50	60

Table 2. ORNL Turbine Pump - Operating Experience on Potassium

Rotary Unit Number	Starts and Stops	Total Operating Time, Hours	Cause of Termination
1	53	90	Unable to attain full speed - Excessive nozzle to turbine wheel clearance and impeller inlet shroud rubbed housing.
2	10	92	Bearing sleeve broke - caused by tight mounting pins and stress concentration in eloxed holes in the WC bearing sleeve.
3	9	10	Low performance - impeller inlet shroud rubbed housing.
4	0	0	Particles clogged thrust faces.
5	3	1/2	Turbine end journal seized.
6	38	327	Impeller end journal seized.
7	15	20	Pump head rise unsatisfactory due to high resistance of filter in pump suction line.
8	1	1/2	Starting difficulty - System was operated 3 weeks on auxiliary pump before attempting to start turbine-pump - found shaft had broken at root of threads flush with out board end of turbine-wheel.
	129	540 hours	

Table 3. Performance of Aeronutronic Mark I Turbine Pump on Potassium

Run No.	Turbine Pump Speed rpm	Turbine				Pump				
		Inlet		Exhaust		Suct. Temp. °F	Suct. Press. psia	Head psi	Flow gpm	Cavitation %
		Temp. °F	Press. psia	Temp. °F	Press. psia					
A. Non-cavitating performance - Pump operating in closed loop with auxiliary pump supplying boiler feed.										
1	3360	1094	2.2	954	.7	652	1.6	10.3	.7	0
2	4860	1140	3.1	990	1.0	645	2.7	21.2	1.0	0
B. Cavitating performance - Turbine pump driving the system. Auxiliary pump off.										
3	4031	1095	1.6	975	~0	767	.1	13.3	2.0	9
4	5195	1144	2.9	1029	.5	854	~0	18.2	2.6	25
5	6116	1198	4.4	1096	1.4	915	.2	29.1	3.5	14
6	6611	1219	5.3	1108	1.8	935	.1	35.8	3.9	9

ORNL Turbine Pump
Fig. 1

ORNL Turbine Pump.
Fig. 2.

ORNL Turbine Pump - Pump
Head vs Pump Flow at Various
Speeds with Water.
Fig. 3.

ORNL Turbine Pump - Performance
Characteristics of ORNL Turbine
Pump on Steam and Water.
Fig. 4.

Aeronutronic Mark I
Turbine Pump.
Fig. 5

Aeronutronic Mark I Turbine Pump.
Fig. 6

Differential Pressure Versus Bearing Leakage During Water Test of Original Aeronutronic Mark I Impeller.

Fig. 7

- ORNL TEST WITH IMPELLER VANE-TO-HOUSING AXIAL CLEARANCE OF 0.057 in. TURBINE DRIVEN BY STEAM.
- AERONUTRONICS TEST OF PROTOTYPE PUMP WITH AXIAL CLEARANCE OF 0.026 in. TURBINE DRIVEN BY NITROGEN.

Aeronutronic Mark I Turbine Pump - Pump Head vs Pump Flow at Various Speeds.

Fig. 8

Aeronutronic Mark I Turbine
Pump- Typical Cavitation
Test Results.

Fig. 9

Aeronutronic Mark II Turbine Pump.
Fig. 10

Aeronutronic Mark II Pump-Overall Turbine Pump Efficiency vs Pump Flow.

Fig. 11

Aeronutronic Mark II Turbine Pump - Pump Head vs Pump Flow at Various Speeds with Water.

Fig. 12

BOILERS

Chairman: P. J. Berenson

AiResearch Manufacturing Company
Los Angeles, California

BLANK

285

DEVELOPMENT OF THE SNAP-8 BOILER*

by

A. H. Kreeger, J. N. Hodgson, and A. J. Sellers
SNAP-8 Division
Von Karman Center
Aerojet-General Corporation
Azusa, California

ABSTRACT

Development of the mercury boiler for the SNAP-8 system is one of the major tasks of the SNAP-8 program. The 1500 hours of testing accumulated on two of the first-generation boilers has permitted detailed evaluation of performance under various operating conditions. Initial tests showed that the boiler performed at only 40% of its thermal design capacity. Even after 600 hours of operation, the boiler could not deliver its rated superheat at design flow rate. Trace amounts of rubidium were added to the mercury. The boiler immediately responded to the additive, reaching its design thermal capacity.

A second unit of the same design reached rated thermal output in 26 hours without the use of an additive. This time-dependent conditioning process is the major problem in boiler development.

Scaled tests on a single-tube are reported to support a new boiler design which should minimize the conditioning problem. Increased liquid velocity is shown to condition a poorly performing boiler. These scaled tests also provided empirical correlations of heat transfer and pressure drop from which other designs may be extrapolated.

*The information appearing in this paper is based on work accomplished under National Aeronautics and Space Administration Contract NAS 5-417.

I. INTRODUCTION

SNAP-8 is a 35 kw, nuclear-electric, Rankine-cycle space power system. The electrical output is generated by a turbine-alternator assembly using mercury vapor as the working fluid. The vapor to drive the turbine is produced in the mercury boiler where the entering subcooled liquid is preheated, vaporized, and superheated in a single pass through the heat exchanger. A schematic diagram of the system is shown in Figure 1. The design and development of the once-through boiler is a major task of the SNAP-8 program.

The first generation boiler, shown in Figure 2, is a combination cross-counter flow, tube-in-shell heat exchanger. The mercury flows in four 60-ft-long tubes which are coiled on two double-lead helices. A "plug" to restrict flow is placed in the inlet to each of the four parallel flow passages giving a liquid velocity of 0.3 ft/sec. The plug in this restricted flow section is a solid rod spaced from the inside of the tube by a wire spring forming a spiral flow path for the mercury. This insert continues through the boiler for 10 ft. Downstream of the plug, the spiral flow is maintained by a twisted ribbon insert with a pitch of 7.2 in., which continues for the remainder of the boiler length. The swirl flow serves to separate the high density liquid from the vapor, making boiler operation insensitive to gravitational field and increasing heat transfer rates. The mercury coils are bounded by cylindrical shells which form an annular flow passage for the reactor coolant, NaK-78 (the eutectic mixture of sodium and potassium). The significant dimensions and materials of fabrication are shown in Table 1.

The boiler is the interface between the nuclear and nonnuclear systems and is subject to the most significant perturbation that occurs in operation; variation in the NaK inlet temperature as the nuclear reactor responds to its temperature control system. The boiler must continually maintain vapor superheat, an acceptable fluid inventory variation, and minimal outlet pressure fluctuations over a 50°F band of NaK inlet temperature. The other operating conditions are shown in Table 2.

This paper presents the performance data accumulated during the tests of two of the tube-in-shell boilers. The main development problem encountered is that consistent boiler performance on system startup has not been achieved. A time-dependent conditioning process

occurs wherein boiler performance increases with time, finally reaching the thermal performance predicted by a semiempirical boiling-heat-transfer model and earlier stabilized experimental results. Test results using trace amounts of rubidium added to the mercury to hasten the conditioning process are reported. Data generated with an experimental, single-tube boiler, are presented which formed the basis for the configuration of a second generation boiler designed to minimize the conditioning problem.

II. TUBE-IN-SHELL BOILER OPERATION

Two units of the tube-in-shell boiler have been tested. The first unit was placed in a facility designed specifically for component evaluation, while the second unit was tested in a more system-oriented loop. During the past year, 1400 hours of operation have been accumulated on the first unit, with a test time of 200 hours on the second boiler. Experience has been gained regarding boiler performance in general and boiler conditioning, both with and without additives. The boilers have operated under conditions of near-immediate conditioning, long-time conditioning, and conditioning through the addition of rubidium. The performance of the boiler as measured by heat-transfer effectiveness, pressure drop, and liquid entrainment in the vapor stream has been obtained under these various degrees of conditioning.

On initial startup of the first unit, thermal performance was poor. The total heat transfer was approximately 0.7×10^6 Btu/hr compared to the design thermal load of 1.75×10^6 Btu/hr. This poor performance was attributed to carbon contamination of the boiler when a pump failed on startup introducing oil into the mercury loop with the boiler at a temperature of 1300°F. Even after cleaning the boiler with an alkaline scale remover, the varnish-like residue from the "cracked" hydrocarbon may not have been removed. Additional evidence supporting this assumption was generated in capsule tests at NASA's Lewis Research Center, and with scaled boilers at Aerojet showing the deleterious effects of hydrocarbons on boiler performance.⁽¹⁾ A 600-hour run ensued during which the time-dependent conditioning process was observed. The thermal capacity of the boiler increased

Superscript numbers in parentheses refer to similarly numbered references listed following the text.

from 0.7×10^6 to 1.5×10^6 Btu/hr at fixed NaK and mercury conditions. This conditioning process is shown in Figure 3 with mercury vapor outlet enthalpy plotted as a function of time. Another measure of performance is the pressure drop across the boiler. Figure 4 shows the pressure-drop data obtained at the end of the conditioning run as compared with the semiempirical curve obtained with a stabilized or fully-conditioned experimental boiler. Even after 600 hours of operation, fully-conditioned performance had not been obtained.

The performance of the second unit installed in the system test loop is an interesting comparison to the first. Boiler performance as a function of time is shown in Figure 5. The boiler reached rated thermal capacity with over 200°F superheat after 26 hours of operation. Although performance was encouraging, this boiler still showed the effects of the time-dependent conditioning process with performance changing with time.

A. BOILER PERFORMANCE WITH ADDITIVES

In view of the boiler conditioning problem that existed with the first unit, it was decided that an additive would be used in the mercury loop in an attempt to improve heat transfer and obtain full-rated power output from the boiler. A study reported in Reference 2 indicated that an appropriate additive would be rubidium. Its low vapor pressure at boiling temperatures encourages it to circulate with the mercury. Arrangements were made to inject increments of a nominal 1% alloy of rubidium and mercury into the 300-lb mercury-loop inventory until full boiler heat transfer occurred.

The response of the system to a rubidium injection is very rapid. The reaction on the boiler is shown in Figure 6. The mercury-side functions plotted are outlet temperature, saturation temperature (NBS data), pressure drop, and flow rate. The pressure drop and outlet temperature are the boiler parameters most affected by the addition of rubidium.

It is seen that before injection of rubidium, the saturation temperature and outlet temperature coincide (no superheat). Upon injection of rubidium, the outlet temperature and pressure drop both increase. The response is seen to be quite rapid. The pressure drop reached equilibrium within one to two minutes. The outlet temperature required approximately five minutes to reach equilibrium, but most of the increase occurred

in the first one to two minutes. The overall effect of the addition of rubidium is a nearly instantaneous change in performance. Sufficient rubidium was injected to bring the loop concentration to 860 ppm. The actual concentration in the loop was not measured.

This boiler operated approximately 400 hours with rubidium used as an additive under a range of flow conditions and additive concentrations. Although actual loop concentrations were not measured, relative concentrations were determined by the quantity of alloy injected into the loop. Two groupings of data were obtained, one set with a "high" concentration covering the approximate range from 600-900 ppm, and the other set with a "low" concentration from 250-400 ppm. Two performance parameters shown to be sensitive to rubidium concentration are boiler pressure drop and liquid carryover (droplets of liquid entrained in the otherwise superheated vapor).

1. Boiler Pressure Drop

Boiler pressure drop as a function of mercury flow rate is shown in Figure 7. The upper of the three sets of data represents operation with high rubidium concentrations. This curve is considered a maximum in pressure drop and is the applicable relationship for design purposes. The center set of data represents operation with a low concentration of rubidium in the test loop. The lower curve is repeated from Figure 4 for comparison between conditioned and deconditioned operation. In general, a considerable span of pressure drop is represented by the three sets of data in Figure 7. However, these data do not represent uncertainties or inadequacies in the ability to predict the pressure-drop relationship for the boiler. Rather, they demonstrate the fact that conditioning affects the pressure drop of the boiler. Under suitable conditions, the boiler may be expected to perform according to the relationship of a fully conditioned boiler as represented by the upper set of data.

The data generated in these multitube boiler tests agree well with the semiempirical curve shown in Figure 7 once the boiler is conditioned.

2. Liquid Entrainment

Another important measure of boiler performance is the amount of liquid entrained in the vapor stream leaving the boiler. As droplets of mercury are boiled during

passage through the boiler, their size continually decreases. It is possible for a given droplet to not come in contact with suitable boundaries for a complete vaporization, and the minute droplet is simply carried along in the vapor stream. The quantity of liquid entrained is important in that it has considerable effect on the system cycle efficiency.

With a fully-conditioned boiler, a shorter length is required to reach a given fluid state than with a lesser degree of conditioning. Consequently, with higher rubidium concentrations, it follows that the superheat length of the boiler is longer and there is more opportunity for a complete drying and superheating of the mercury vapor. Whether or not complete drying of the mercury ever occurs is not known. It is likely that, even with a long superheat length, some small droplets are never vaporized and remain in liquid form. This is especially true in that the relationship between droplet vapor pressure and surface tension makes it more and more difficult for vaporization to occur as a droplet becomes smaller.

An analysis of data has been conducted to investigate changes in liquid entrainment. The analysis is based on a comparison between the vapor flow rate as measured by a sonic nozzle and the liquid flow rate as measured by a venturi flow meter. The assumption that the vapor meter is insensitive to liquid droplets implies that flowing quality can be determined from the ratio of vapor to liquid flow rates. The data on liquid entrainment have been plotted in Figure 8.

There is considerable scatter in the data, but the overall trend is seen to be a confirmation of the foregoing postulates. At low flow rates, the data of the runs essentially intermix and have an average value a little above unity. The shift from unity is the result of instrumentation inaccuracies which are assumed to be caused by a zero-shift in the pressure transducers. The important feature is that the data have essentially the same average value. At high flow rates, however, a definite divergence from unity is seen. The high flow-rate divergence, thus, is an evidence of a greater liquid entrainment.

Liquid carryover in future boilers should be reduced by changing from a twisted tape to a spiral wire turbulator downstream of the plug insert section.

Isothermal two-phase flow tests using nitrogen to simulate the-mercury vapor have shown that some liquid droplets can be trapped on the surface of the tape and move slowly to the boiler outlet, thus contributing to the liquid entrainment. Tests with a spring-wire turbulator showed no visual droplets in the vapor core, the droplets being effectively separated from the gas. Comparable results have been verbally reported by AiResearch personnel engaged in experiments with boiling Freon.

3. Side Effects

Rubidium is not in general use in SNAP-8 system operation because of potential problems associated with it. Oxides in the loop react with the rubidium resulting in an accumulation of solid particles of rubidium oxide. This problem is compounded with in-leakage during operation, or when the loop is opened to air for repairs. After these tests on the boiler, examination of the pump showed that rubidium oxides had indeed accumulated. Apparently no performance degradation occurred as a result of the buildup but, potentially, this is undesirable.

Other possible side effects are the changes in thermodynamic properties associated with rubidium-mercury solutions during evaporation and the buildup in rubidium concentration at the space seals of the SNAP-8 turbine-alternator and pump-motor assemblies.

4. Boiler Stability

Typically, forced-convection boiling systems are not completely stable. The instability in forced convection boiling systems manifests itself in variations in outlet pressure and flow. The causes for these instabilities have been classified by AiResearch⁽³⁾ and include:

Flow regime transition

Negative derivative of pressure drop with respect to flow rate

Compressibility

Parallel channel coupling.

The data obtained in tests with the boiler included transient measurements of outlet and inlet pressure, flow rates, and temperature to evaluate the stability of the boiler system. The measurements indicated that the outlet pressure varied

approximately sinusoidally with a period of 4 seconds per cycle and an amplitude of 9 psi measured from peak-to-peak. The results for each data point were tabulated and correlated with boiler pressure level and mercury flow rate, as shown in Figure 9.

At rated flow of 11,200 lb/hr, the spread in data indicates that the outlet pressure variation will be from 1.85 to 2.70% of the rated boiler outlet pressure of 270 psia.

Other forced convection systems have shown a sensitivity in pressure stability to liquid preheat. Data in this test series, with the degree of subcooling varying from 575 to 875°F, showed no effect on outlet pressure stability.

From the data generated, it is not possible to classify the form of instability present in the tube-in-shell boiler. However, both single-tube and multitube boilers of this design have shown comparable forms of instability tending to eliminate parallel channel coupling as a dominant cause. Each of the boiler tubes was orificed at its inlet to give a liquid to two-phase pressure drop ratio of 0.5 at rated flow rate of 11,200 lb/hr. This restrictor probably damped out much of the fluctuation, and reduced the probability of a negative derivative of pressure drop with flow rate. Additional tests on single-tube units are presently in progress, and may shed additional light on the cause of the oscillatory nature of the pressure.

The spread in the data of outlet pressure fluctuation brackets the expected value of $\pm 2.5\%$ at rated flow rate and outlet pressure. The tube-in-shell boiler is considered adequate as far as pressure fluctuations are concerned.

B. OPERATION WITHOUT ADDITIVES

The last 400 hours of operation of the first boiler was without the use of additives. The boiler and liquid mercury legs of the system were cleaned of rubidium and a new batch of mercury was loaded into the system. On startup, fully conditioned performance was evident. The pressure drop, liquid entrainment and outlet temperature and pressure measurements showed performance comparable to the runs with the highest rubidium concentrations. On shutdown, evaluation of the dump tank showed less than 50 ppm of rubidium in solution with the mercury. The operation of a boiler for an extended period

of time in the conditioned state apparently resulted in a boiler the performance of which was no longer dependent on rubidium concentration.

III. DEVELOPMENT OF TUBE-IN-TUBE BOILER

The difficulties encountered in boiler conditioning and the lack of diagnostic information in the tube-in-shell configuration of the boiler led to the design and development of a second-generation boiler. Sealed, single-tube boiler tests were performed to generate test data in support of this design effort covering mercury-side heat-transfer and pressure-drop characteristics, as well as boiler conditioning. Boiler performance was studied by means of external shell-side thermocouples reading the local NaK temperature along the boiler length. The slope of the temperature profile gives the heat flux and vapor quality locations. A heat balance is used to determine the local quality and is correlated with internal mercury-side pressure measurements in the plug insert region of the boiler.

The details of the single-tube tests are reported in Reference 4 with the following significant conclusions:

- A. The introduction of a flow restrictor at the liquid-preheat and low-quality regions of the boiler bringing the inlet liquid velocity to approximately 6 ft/sec, can possibly overcome difficulties associated with nonwetting of the tube wall surface. This inlet velocity also results in additional liquid phase pressure drop leading to an improvement in boiler stability.
- B. Two-phase pressure drop data are adequately correlated by a modified Martinelli parameter.
- C. Boiling heat-transfer rates obtained from a fully conditioned boiler are in fair agreement with proposed semiempirical correlations reported by TRW in the development of the SNAP-2 boiler.⁽⁵⁾

The boiler conditioning program is reported in detail in Reference 1. The initial results of these tests are best illustrated by the NaK profiles shown in Figure 10. Curve 1 shows the profile after 10 hours of operation and Curve 2 shows it after 670 hours of operation, in both cases with a 0.8 fps inlet mercury velocity. Note that very little

improvement is shown in the NaK profile although the mercury outlet conditions did improve, and finally reached superheated vapor. The plug geometry was then changed to give a liquid velocity of 6.0 fps. There was substantial improvement in the NaK temperature profile, and hence boiler performance, immediately upon startup. Curves 4 and 5, which were taken later in time, indicate that the profile was getting better and that the boiler was still conditioning. This, then, indicates that the boiler does not condition fully immediately, but the improvement is still very marked. The original 0.8 fps velocity plug replaced the 6 fps plug and Curve 6 was generated. This curve is representative of a conditioned boiler tube. From this it appears that, once a surface is conditioned by high velocity and additional contamination is avoided, the boiler operation is independent of inlet velocity and the plug insert design.

Two-phase pressure drop in the various regions of the boiler were expressed as:

$$\frac{\Delta P_{TP}}{\Delta P_g} = \phi = \frac{\Delta P_{TP}}{f_v \frac{\Delta L}{D_e} \frac{(\bar{x} G)}{2g_c \rho_v}}$$

where ΔP_{TP} = two-phase pressure drop
 ΔP_g = pressure drop if gas phase were flowing alone
 f_v = vapor-phase friction factor
 D_e = equivalent diameter
 ΔL = incremental length
 $\bar{x}$ = average quality
 G = mass flow per unit area
 g_c = gravitational constant
 ρ_v = vapor-phase density

The product ϕf_v was correlated in terms of a modified Martinelli parameter λ , as suggested by TRW in their SNAP-2 boiler development⁽⁵⁾

$$\lambda = \left(\frac{\bar{x} G D_e}{\mu_v} \right)^{.8} \frac{\mu_v}{\mu_f} \frac{\rho_f}{\rho_v} \frac{\bar{x}}{1-\bar{x}}$$

where μ_v, μ_f = absolute viscosity of vapor and liquid, respectively
 ρ_f = liquid phase density

Figure 11 shows the data obtained in the 6 fps preheat and low vapor quality regions. These data were obtained on three variations of plug insert configuration with liquid phase velocities from 4 to 6.5 fps.

Figure 12 shows the data obtained in the intermediate quality region of the boiler tube. This region includes a restricted flow section with a wire spring wrapped plug inserted in the mercury containment tube, again creating a spiral flow path. A "least squares" fit of these data resulted in the following empirical correlation:

$$\phi f_v = 130 \lambda^{-0.67}$$

Figure 13 shows the data obtained in the region of the boiler where no restricted flow section is employed. A spiral wire is inserted to encourage phase separation. The data, while scattered, give an adequate correlation for design purposes.

Heat-transfer rates were determined at the initiation of boiling and averaged for each 20% increment of vapor quality. Figure 14 shows the heat flux obtained in the region of the liquid vapor interface. The curve is best fitted by an equation of the form $q'' = (\text{const}) (\Delta T_m)$ implying that single-phase forced-convection heat transfer is the dominant mode, and negligible local boiling is involved. Figure 15 shows the curves obtained for each of the five quality increments. A composite curve is shown in Figure 16 which indicates the peak heat flux occurring at a vapor quality of $x = 0.50$. This quality "burnout" of 50% was characteristic of all the data generated, independent of plug-insert configuration.

These data were applied in the design of the tube-in-tube boiler. The resulting profiles of the various boiler parameters are shown in Figure 17. The velocity profiles

are comparable to those employed in the scaled tests to minimize the difficulties with boiler conditioning.

IV. CONCLUSIONS

A. The extensive tests performed on the first-generation (tube-in-shell) boiler have shown that boiler conditioning is a major problem in boiler development.

B. When conditioned, either by operating time or with the use of rubidium, the boiler performed as designed thermally, but with liquid droplets entrained in the otherwise superheated vapor.

C. When conditioned, additives are not required to achieve satisfactory performance, if contamination is avoided.

D. Sealed tests on single-tube boilers have shown that adequate semiempirical correlations for forced convection mercury boilers are available to predict pressure drop and heat-transfer characteristics, and that inlet restrictor design can affect the time-dependent conditioning process.

REFERENCES

1. R. S. Carey, B. E. Farwell, and G. M. Thur, Effects of 0.1 Contamination on Heat-Transfer Performance of Forced-Convection Mercury Boilers, Paper Presented at AIAA First Rankine Cycle Space Systems Specialist Conference, Cleveland, Ohio, 26-28 October 1965.
2. L. T. Clark and M. F. Parkman, Effects of Additives on Wetting During Mercury Pool Boiling Heat Transfer, ASME 64-WA/HT-22.
3. Flow Stability in Multitube Forced Convection Systems, AiResearch Report L-9443, October 1964.
4. A. J. Sellers and M. K. Wong, Experimental Investigation of Forced Convection Once-Through Mercury Boiler Performance Characteristics, Aerojet-General Corp. Technical Memorandum TM 4933:65-8-311, 3 August 1965.
5. The SNAP-2 Power Conversion System Topical Report No. 17, Mercury Boiling Research, TRN Report No. ER-4833, October 1964.

TABLE 1

SNAP-8 TUBE-IN-SHELL BOILER DESIGN PARAMETERS

1. Design Parameters

NaK

Inlet Temperature - 1298°F

Inlet Pressure - 41 psia

Temperature Drop - 170°F

Mercury - Flow Rate - 11,500 lb/hr

Inlet Temperature - 515°F

Outlet Temperature - 1280°F

Inlet Pressure - 340 psia

Outlet Pressure - 270 psia

2. Size - Overall Length - 55 in.

Diameter - 24 in.

Tube Length - 60 ft

Boiler Weight - 850 lb

3. Inventories - NaK - 150 lb

Mercury - 20 lb

TABLE 2

SNAP-8 TUBE-IN-TUBE SIGNIFICANT DIMENSIONS & MATERIALS OF FABRICATION

Shell Material - 316 SS

Shell Dimension

OD (in.) = 24

ID (in.) = 17.6

Wall (in.) = 0.187

Length (in.) = 45 (including toroids)

Tubing Material - 9M (SSTU Type 9Cr1Mo)

Tubing Dimensions

OD (in.) = 0.902

Wall (in.) = 0.125

Length (ft) = 60

Plug Inlet Material - 1020 rod (solid) - 1010 wire 0.135-in. OD

Plug Inlet Dimensions

Length (ft) = 10

Plug (rod) OD (in.) = 0.602 - 0.599

Wire Pitch = 3 in. pitch for 1st 3 ft of plug length
 4 in. pitch for next 2 ft of plug length
 5 in. pitch for remainder of plug length

Ribbon Turbulator Material - 0.016 in. sheet steel type C1009

Ribbon Turbulator Dimensions

Length = 17-1/2 ft (average)

Pitch = 7.3 in.

Fig. 1 SNAP-8 Electric Generating System Schematic

Cutaway Drawing of Tube-in-Shell Boiler

Fig. 2 SNAP-8 Tube-in-Shell Boiler

Fig. 3 Tube-in-Shell Boiler 1st Unit Time-Dependent Performance

Fig. 4 Tube-in-Shell Boiler Conditioning Pressure Drop

Fig. 5 Tube-in-Shell Boiler 2nd Unit Performance as a Function of Time

Fig. 6 Tube-in-Shell Boiler Response to Rubidium Injection

Fig. 7 Tube-in-Shell Boiler Pressure Drop Variation

Fig. 8 Liquid-Mercury Entrainment - Tube-in-Shell Boiler

Fig. 9 Tube-in-Shell Boiler Mercury Outlet Pressure Fluctuations

LIQUID VELOCITY FT/SEC	CURVE NO.	RUN NO.	TIME, HRS	
.80	1	1	10	
	2	1	670	
6.00	3	2	4	
	4	2	20	
	5	3	1-1/2	RESTART
.80	6	4	1	RESTART

Fig. 10 Single-Tube Boiler NaK Temperature Profiles

Fig. 11 Two-Phase Flow Pressure-Drop Correlation in Plug-Insert Preheat Section

Fig. 12 Two-Phase Flow Pressure-Drop Correlation in Plug-Insert Quality Section

Fig. 13 Two-Phase Flow Pressure-Drop Correlation - Unplugged Tube

Fig. 14 Initial Boiling Heat Flux vs NaK-to-Mercury Temperature Difference

Fig. 15 Boiling Heat Flux with Variation in Mercury Quality

Fig. 16 Typical Boiling Heat Flux Profile

Fig. 17 Tube-in-Tube Boiler Parameter Profiles

EFFECTS OF OIL CONTAMINATION ON HEAT TRANSFER PERFORMANCE*
OF FORCED-CONVECTION MERCURY BOILERS

By

R. S. Carey
SNAP-8 Division
Von Karman Center
Aerojet-General Corporation
Azusa, California

B. E. Farwell
Aerojet-General Nucleonics
San Ramon, California

G. Thur
SNAP-8 Project Office
Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio

ABSTRACT

The major requirements associated with mercury once-through boilers for space powerplants are repeatable startup and reliable long-term heat-transfer performance without deleterious effects on other system components such as turbines, pumps, and condensers. During the past few years, there has been a number of instances where mercury boilers have been known to decondition either during startup or after some period of operation. The connotation is that any performance in the direction of providing poorer heat-transfer performance than expected from either design correlations or some minimum accepted performance is termed deconditioning, while any trend providing improved performance after initial thermal equilibrium is termed conditioning. This performance uncertainty during system startup and endurance operation can prevent adequate startup, cause turbine corrosion/erosion and reduce system power.

Recent tests were performed in Corrosion Loop 4 single-tube boiler at Aerojet-General Nucleonics, San Ramon, California to investigate the effect on boiler performance of the SNAP-8 lubricant-coolant, mix-4P3E, when mixed with the boiler mercury. This was

*The information appearing in this paper is based on work accomplished under National Aeronautics and Space Administration Contract NAS 5-417.

considered necessary since the SNAP-8 turbine-alternator assembly and the mercury pump-motor assembly use dynamic space seals that allow interdiffusion between the mercury and oil during startup and endurance operation. In addition, Dr. L. Rosenblum here at NASA-Lewis Laboratory demonstrated in unpublished work that very small amounts of oil contamination on the surface of a mercury containment capsule had dramatic effects in preventing mercury droplets from wetting the surface of the capsule. The experimental data from the CL-4 single-tube boiler tests were evaluated to determine the conditioning rates for various velocities in the boiler preheat region after the boiler was deconditioned by injection of mix-4P3E into the mercury loop.

I. INTRODUCTION

A. DESCRIPTION OF THE SNAP-8 SYSTEM

The SNAP-8 system is a nuclear-powered auxiliary space power system capable of generating 35 kilowatts of electrical power for up to 10,000 hours. The system, shown schematically in Figure 1, consists of a primary loop that cools the reactor and transfers heat to the mercury boiler. This loop contains liquid metal eutectic NaK as a heat-transfer medium. The working loop employs a Rankine mercury cycle. The mercury is heated and vaporized in the boiler, expanded through a turbine, cooled in the condenser, and returned to the boiler via a mercury pump. Power is extracted from an alternator driven by the mercury turbine. The third loop extracts heat from the mercury condenser and expels it through a system radiator. The heat-transfer medium in this system is also eutectic NaK. A fourth loop is employed as a lubricant-coolant loop. This loop utilizes a fluid designated as mix-4P3E as a lubricant for the bearings and a coolant for various mechanical components in the system. Mix-4P3E is a mixture of six isomeric bis-(phenoxy-phenyl) ethers, where the 4P and 3E designate 4 phenyl groups and 3 ether oxygens. This fluid was chosen for its combination of lubricant, heat transfer and radiation resistance properties.

B. REASONS FOR TESTING

The principal problem attacked in the tests discussed in this paper concerns the potential interrelationship of the lubricant-coolant loop with the mercury boiler heat exchanger. Difficulties have been encountered in achieving rated boiling heat-transfer fluxes in NaK heated once-through mercury boilers. These difficulties have been

encountered in boilers using 316 SS, Haynes 25, and 9Cr-1Mo alloys for tubing used for mercury containment. Regardless of the material used, varied and spasmodic performance has been experienced during boiler testing.

The major cause of unpredictable performance is believed to be nonwetting surface contamination of the mercury containment tube. Dr. L. Rosenblum here at NASA-Lewis Laboratory demonstrated in unpublished work that very small amounts of oil contamination on the surface of a mercury containment capsule had dramatic effects in preventing mercury droplets from wetting the surface of the capsule. In these experiments, surface wetting was defined as a contact angle of the droplet being greater than 90° of the surface of the tube wall. Qualitative tests conducted at Aerojet-General Nucleonics verified that degradation of heat transfer would result from oil contamination in a forced convection boiler.

The SNAP-8 system employs two space seals that separate the mix- $^{40}\text{P}^{32}\text{E}$ from the mercury. One seal is employed in the turbine-alternator assembly; the second seal is used in the mercury pump-motor assembly. It has been calculated that less than 40 grams of mix- $^{40}\text{P}^{32}\text{E}$ will diffuse through these seals and into the mercury system during steady-state SNAP-8 operation over the 10,000-hour lifetime. The 40 grams represents an additive "worst-condition" calculation, and leakage rates are expected to be a factor of ten lower, or about 4 grams. Possibility also exists for diffusion across these seals during system startup. The space seals are dynamic and do not function properly until the system achieves rated speed. During startup, lift-off seals are employed to prevent leakage until the dynamic seals are operating. If the lift-off seals do not function properly or become worn during operation, they may provide the source of larger mix- $^{40}\text{P}^{32}\text{E}$ contamination on restart. The purpose of these tests was to determine a capability of a conditioned boiler to digest mix- $^{40}\text{P}^{32}\text{E}$ without degradation of thermal performance, and to determine the ability of these boilers to recover original conditioning if such thermal performance was lost.

In that the goal of these tests was to intentionally decondition the boiler, the amount of mix- $^{40}\text{P}^{32}\text{E}$ injected was much larger than that corresponding to the SNAP-8

system during the steady-state operation. A factor of twenty based on flow rates and surface area comparison can be used for comparative values of oil contamination. The 12 grams injected would give a SNAP-8 mix-4P3E equivalent maximum for these tests of 240 grams.

II. EXPERIMENTAL PROCEDURES

A. DESCRIPTION OF CORROSION LOOPS

The corrosion loops (Figure 2) were designed to simulate the SNAP-8 dynamic cycle conditions for corrosion study. It is a three-loop system, consists of a heated primary NaK-circulation loop, coupled through a boiler to a simulated mercury Rankine-cycle loop, which in turn rejects its heat through the condenser to an air-cooled NaK-circulating loop. The following operating ranges were used for the boiler plug insert experiments:

NaK Primary Loop

Boiler inlet temperature	1330°F maximum
NaK flow rate	2050 lb/hr maximum
Heater power input	35 kw maximum

Mercury Loop

Mercury flow	500 lb/hr nominal
Boiler inlet temperature	500°F
Boiler inlet pressure	500 psia maximum
Boiler outlet pressure	100 psia minimum

1. Primary NaK Loop

The NaK primary loop employs a direct-resistance heater in which low-voltage electrical current is passed through the NaK-carrying tube and the NaK. The heater is coiled, with the two grounded leads on the loop side and an insulated low-voltage lead at the midpoint of the coil. Since the resistance of each leg is fixed, the power input is varied by controlling the voltage across the terminals. The voltage is regulated with a saturable core reactor-transformer. An electromagnetic pump maintains the NaK flow, which is measured by a magnetic flowmeter. A purification system is

employed to reduce and maintain the oxide level in the system at less than 25 ppm. Other major components in the loop include level indicators, an expansion tank, a dump tank, and a cover gas system.

2. NaK Condensing Loop

Major components of the NaK condensing loop consist of a conventional air-cooled finned-tube heat exchanger, a magnetic flowmeter, and an electromagnetic pump. A convective type of cold trap is employed to maintain a low oxide level in the loop.

3. Mercury Loop

The mercury loop uses two Chempumps,* Model CFHT-7 1/2-65 (one on a standby basis), for pumping the liquid mercury. A venturi flowmeter is used to measure mercury flow rate. Two semistandard valves are used for control purposes and for imposing a resistance between the pump and boiler. An adjustable choked nozzle is located downstream of the boiler outlet for the regulation of boiler outlet pressure. The adjustable choked nozzle is a convergent-divergent nozzle in which the throat area may be varied with a movable pintle. The mercury vapor is then passed through a desuperheater and a turbine-blade mockup section before entering the condenser. The desuperheater and the blade mockup are not necessary for the boiler experiment, but were originally designed for turbine simulation during a corrosion study. The NaK-cooled mercury condenser is a counter-flow heat exchanger consisting of three tapered condensing tubes with a straight length for subcooling.

The rest of the mercury loop consists of conventional liquid metal components such as bellows-sealed valves, electrical resistance level probes and a cover-gas system.

B. BOILER DESCRIPTION

The mercury boiler (Figure 3) is a single-pass counter-flow tube-in-shell heat exchanger wound into helix form with a protruding inlet and outlet. The mercury flows in the tube, and the NaK through the annulus passage. An inlet plug is inserted

* Manufactured by Chempump Division of Fostoria Corp., Huntington Valley, Penna.

into the straight part of the protruding mercury inlet section to increase the liquid-mercury and low-quality velocity. A spiral wire vortex-generator is installed in the coiled mercury tube to promote heat transfer. Spacers are used to center the mercury tube in the annulus. The NaK inlet and outlet are located near the ends of the shell.

The boiler configuration and geometry are as follows:

Mean helical coil diameter (in.)	18
Helical shell pitch (in.)	1-3/8
Shell size (in.)	1-1/8 OD x 0.083 wall x 9.959 ID
Tube size (in.)	9/16 OD x 0.083 wall x 0.397 ID
Vortex wire diameter (in.)	0.049
Vortex wire pitch (in.)	1.5
Coil tube length (ft)	55
Straight tube length (ft)	5
Tube material	9Cr-1Mo
Shell material	316 SS

The boiler was installed in a metal tank assembly and insulated with about 6-1/2 in. of fused-alumina bubbles. Six 1/2-in.-thick layers of Fiberfrax insulation were affixed to the outside of the boiler supporting tank assembly.

C. BOILER TEST INSTRUMENTATION

1. Temperature Measurement

The NaK boiler inlet, outlet, and the boiler shell temperatures are measured by 1/16-in. diameter, stainless-steel sheathed, MgO-insulated thermocouples. These thermocouples have ungrounded junctions, and are attached to the 316 stainless steel tube that constitutes the outside shell of the boiler. The NaK boiler inlet temperature is measured by three thermocouples attached circumferentially at the tube 18 in. from the boiler. The boiler outlet temperature is measured by a thermocouple located about 9 in. from the boiler shell.

2. Flow Measurement

An ASME standard venturi flowmeter with a throat diameter of 0.100 in. is used for metering mercury flow. The differential pressure across the flowmeter is parallel to an electronic transducer and a pneumatic transmitter. The electrical output signal from the transducer is then amplified and indicated on a Motorola 4-in. strip chart recorder. The pneumatic transmitter output signal of 3 to 12 psig is transmitted directly to a 12-in. circular recorder-indicator.

The NaK flow is measured by a permanent-magnet flowmeter manufactured by MSA Research Corporation. The flowmeter dc output is amplified and indicated on a Motorola 4-in. strip chart recorder-indicator.

D. PREVIOUS PERFORMANCE OF CORROSION LOOP

Two corrosion loops of the type described above have been constructed and operated. They are designated as Corrosion Loops 3 and 4. A brief description of the performance of each boiler in these loops is given below.

1. Corrosion Loop 3

The boiler inlet plug installed in this boiler was as shown in Figure 3. The liquid velocity in the preheat region was 0.6 fps and, when this boiler was first put into operation, superheated mercury could not be achieved using any mode of operation. After 200 hours of operation with saturated conditions at the boiler outlet, superheated vapor was suddenly obtained although not at rated SNAP-8 conditions (1265°F, 265 psia). After an additional 200 hours of operating time (total 400 hours), rated conditions were obtained.

The Corrosion Loop 3 boiler was operated a total of 4400 hours. During the first 2000 hours of operation, the boiler performance continued to improve as indicated by the NaK shell temperature profiles shown in Figure 4.

The degrees of conditioning are well illustrated with this boiler. At the start it was totally deconditioned and saturated conditions were obtained at the boiler outlet. After the sudden attainment of superheat conditions (200 hours), there

was constant improvement to the point where the boiler was considered conditioned. It is postulated that the mercury vapor scoured or cleaned the surface permitting better heat transfer between the tube wall and mercury. The mechanism of this cleaning action has not been pinpointed.

2. Corrosion Loop 4

a. Description

The Corrosion Loop 4 boiler was a duplicate of Corrosion Loop 3 except for the inlet plug at the mercury inlet. This plug was modified so that the spiral wire pitch on the plug (Figure 3) was 0.75 in. instead of 1.5 in. This change was made to increase the liquid mercury inlet velocity (from 0.6 fps to 0.9 fps) to see if this would influence boiler performance. In addition, the mercury containment tube of the boiler was degreased just prior to installation in the loop.

b. Boiler Performance

When this boiler was first started, superheated mercury vapor was obtained but not at rated SNAP-8 conditions (1265°F, 265 psia). After 650 hours of operation, a slight improvement in the boiler performance was noted but it did not compare to the improvement shown in Corrosion Loop 3 boiler.

At this time, a further modification was made to the inlet plug so that a multipitch was wound with the spiral wire. The first 6 in. of the plug had no spiral wire, the next 18 in. was wound on a 0.125-in. pitch, and the last 36 inches was wound on a 0.75-in. pitch. Operation of the boiler with this inlet plug showed an immediate improvement in the boiler performance and continued improvement during the next 16 hours of operation. The NaK shell temperature profiles shown in Figure 5 illustrate this improvement.

The improvement in boiler performance using this multipitch inlet plug in Corrosion Loop 4 was greater than had been obtained in Corrosion Loop 3 even after 4400 hours of operation. The main difference between the two boilers was the increase in liquid mercury velocity in the preheat region from 0.9 fps to 6 fps.

Boiler inlet plug development was carried out as a separate test program.⁽¹⁾ The information reported in this study was generated to determine pressure drop and boiler stabilization data versus plug design. A plug with a liquid-mercury velocity in the preheat region of 4.5 fps was developed and tested. This inlet plug and the two inlet plugs described above were used in the oil contamination tests.

E. TEST RUNS WITH SNAP-8 LUBRICANT-COOLANT FLUID (MIX-4P3E) ADDITIONS

1. Description of Mix-4P3E Addition Apparatus

The additions of mix-4P3E to the mercury were made using a parallel manifold system at the boiler inlet. When the mix-4P3E was added to the mercury, the mercury stream was diverted through the manifold section that contains the mix-4P3E and it was carried into the boiler with the mercury stream.

2. Test Run with 0.9 fps Liquid Velocity Boiler Inlet Plug

The first test to determine the effect of mix-4P3E on boiler performance was done using a boiler inlet plug designed to give a liquid velocity in the boiler pre-heat region of 0.9 fps. Prior to the run, the boiler was considered conditioned.

After loop operation had stabilized, mix-4P3E was admitted to the mercury stream in amount to total one, three, six and twelve grams. After the one-gram mix-4P3E addition, the loop was shut down because of a cooling-water system failure. The loop was not shut down between the remaining additions until after the last addition was made.

There was no effect on the boiler performance when additions were made while the loop was operating; however, when the mercury boiling was restarted after the last addition, the boiler performance, as measured by the NaK temperature profile, was found to be degraded. From this series of tests, it was observed a restart of mercury boiling was more detrimental to boiler performance than continuous operation, when mix-4P3E is present in the loop.

The boiler was operated for approximately 72 hours after 12 grams of mix-4P3E had been added to the mercury stream and the boiling had been stopped and

Superscript numbers in parentheses refer to similarly numbered references listed following the test.

restarted. No conditioning of the boiler was noted during this time. A plot of the average boiling heat flux during this test run and a typical NaK temperature profile are shown in Figure 6.

3. Test Run with 4.5 fps Liquid Velocity Boiler Inlet Plug

The boiler inlet plug used in the run described above was changed to an inlet plug that gave a design liquid preheat velocity of 4.5 fps. Prior to injecting the mix-4P3E, the boiler was operated until conditioned performance was obtained (about 150 hours).

For this run, three grams of mix-4P3E were injected with the mercury at the start of mercury boiling. This amount of mix-4P3E injected at the startup caused the boiler performance to degrade immediately. The loop was allowed to operate for approximately 150 hours and there was continuous conditioning of the boiler during that time. Operation of the boiler was continuous except for a 2-hour shutdown caused by an electrical power failure. The conditioning rate was slow for the first 70 hours prior to shutdown. After the shutdown, the rate of conditioning increased rapidly, as shown in Figure 7, even though the performance at the start of the second period of conditioning was worse than before the shutdown.

4. Test Run with 6 fps Liquid-Velocity Boiler Inlet Plug

A boiler inlet plug was installed in the boiler that was designed to give a liquid preheat velocity of 6 fps. The boiler was considered to be conditioned prior to the mix-4P3E injections.

During the mix-4P3E injection phase of this run, three additions were made while the loop was operating so that the total amount of mix-4P3E in the loop was 3, 6, and 12 grams. After each addition, the mercury boiling was stopped and restarted.

For this test, no effect was noted on the boiler performance until 12 grams of mix-4P3E had been added, the mercury boiling stopped, and then restarted. During the conditioning run, after the last restart of the mercury boiling, the conditioning rate was rapid and the boiler was fully conditioned after 24 hours of operation, as shown in Figure 8.

5. Inspection of Boiler Inlet Plugs

After each of the test runs described above, the boiler inlet plug was removed from the loop and examined. A black deposit (analyzed to be 80% carbon) from the thermal decomposition of the mix-4P3E was found on all the plugs. There was no indication that the black deposit was removed in the 0.9 fps inlet plug. However, in the tight pitch sections of the 4.5 and 6 fps inlet plugs there were spotty areas free of deposit. It appeared as if the mercury had scoured the deposit from the plug exposing bare metal. Although the inside diameter of the tubing that contained the plug could not be examined, it is assumed that the black deposit from the breakdown of the 4P3E coated that surface also.

III. DISCUSSION

The effect of increased liquid velocity in the preheat region of the boiler was seen in these tests from two points. First, the mercury boiler with the higher liquid velocity inlet plug was able to tolerate more mix-4P3E before the heat transfer of the boiler was degraded and, second, once the boiler heat-transfer rate is degraded, the higher velocity inlet plug causes the boiler to recondition at a faster rate.

The mechanism for the degradation in heat-transfer performance of the boiler was not determined in these tests. Original test plans recognized the limitations of the test equipment in this regard. Effects, rather than mechanisms, were sought. However, some insight on the problem can be gained. The mix-4P3E additions tended to decrease the heat-transfer rate in the first few feet of the boiler. This performance is similar to that experienced when mix-4P3E has not been added, but the boiler operates in a "deconditioned" state. From the inspection of the inlet plugs and the appearance of the black deposit (analyzed to be 80% carbon) caused by the thermal decomposition of mix-4P3E, it can be postulated that surface contamination is responsible for the decrease in boiler performance.

Previous tests⁽²⁾ have shown that the heat fluxes are high for mercury boiling when the mercury wets the tube wall. Additives to promote wetting have been successful in improving heat transfer. Since wetting is a function of the surface energy relationships

between the mercury and the surface it contacts, the introduction of a film between the metal surface of the boiler tube and the mercury surface can change this surface-energy relationship.

When a mercury drop wets a surface, it tends to spread out over the surface and thereby gives more contact area for heat transfer to the drop.

Another mechanism by which a drop could be flattened against the tube wall is the centrifugal force created by the velocity generated in the tight pitch section. This action would also give more area for heat transfer to a mercury drop.

When the boiler tube surface is conditioned, the inlet-plug velocity is not critical since the wetting mechanism promotes adequate heat transfer. But when the wetting mechanism is destroyed by a surface film, such as the thermal decomposition products from mix-4P3E, the mechanical flattening effect on the mercury drops caused by the liquid velocity must be present to achieve the proper heat transfer rates. Experience indicated that proper heat transfer at the boiler entrance (in the plug region), promotes vapor generation that provides the impetus for remaining liquid-phase breakup. This early controlled boiling results in a boot-strapping effect that results in good performance throughout the rest of the boiler.

This is a generalized picture of the boiling process and there are many combinations of the two factors (wetting and centrifugal force) that cause the various degrees of boiler conditioning, as noted from the NaK temperature profiles.

Inspection of the boiler inlet plugs after loop operation with mix-4P3E revealed that surface contaminants, such as the mix-4P3E residue, are removed by the mercury flow, possibly because of the scouring action of the mercury drops and vapor and the subsequent corrosion attack of the metal under the surface films. Since the conditioning is a time-dependent process, the rate of surface film removal with a fixed liquid velocity may be the controlling factor in conditioning.

During the operation of the loop for these tests, it was observed that the mix-4P3E additions did not degrade the boiler performance while the mercury boiling was established and the boiler was operating at steady-state conditions. A decrease in boiler performance

was not noted after a mix-⁴P₃E addition until the mercury boiling was stopped and restarted, unless the mix-⁴P₃E was injected during boiler startup. The additions of mix-⁴P₃E in all cases caused an increase in the condensing pressure from the gases generated when the mix-⁴P₃E decomposed. A sample of the atmosphere in the condenser was analyzed and found to contain a large percentage of CO and H₂.

These observations indicate the mix-⁴P₃E decomposition products may not have coated areas of the boiler tube that were wetted when the boiling process was established. However, when the boiling was stopped and all the mercury was evaporated from the tube surface, the bare metal surface was contaminated by loose decomposition products or the back-diffusion of the gases from the condenser.

Once the boiler was conditioned during operation, there was no degradation of thermal performance.

IV. CONCLUSIONS

A. The initial boiler performance is dependent on the mercury-flow-passage wall surface characteristics and the thermal dynamic conditions imposed by the mercury-flow-passage geometry.

B. Once "conditioned," forced-convection mercury boilers have a good tolerance for mix-⁴P₃E additions; therefore, the SNAP-8 boiler should operate at steady-state conditions with no noticeable effects from mix-⁴P₃E contamination from space seal diffusion.

C. During conditioning, surface contaminants are apparently removed which exposes the mercury to bare metal surfaces. This reduces the liquid-solid energy relationships and allows better heat transfer.

D. When the mercury boiler tube surfaces were contaminated with the thermal decomposition products of mix-⁴P₃E, the higher liquid velocities in the boiler preheat region exhibited shorter reconditioning periods when compared to lower velocities.

ACKNOWLEDGMENT

The authors wish to acknowledge the assistance of Mr. M. Wong and Mr. G. Redfern for the data reduction, and Mr. Jack Ralph for his work in operation of the loop during these tests.

REFERENCES

1. A. J. Sellers, G. Thur, and M. K. Wong, Experimental Investigation of Forced Convection Once-Through Mercury Boiler Performance Characteristics, Paper Presented at Argonne National Laboratories, 27-30 September 1965.
2. A. H. Kreeger, A. J. Sellers, and J. N. Hodgson, Development of the SNAP-8 Boiler, Paper Presented at AIAA First Rankine Cycle Space Systems Specialist Conference, Cleveland, Ohio, 26-28 October 1965.

Fig. 1 SNAP-8 Electric Generating System Schematic

Fig. 2 Corrosion Loop Layout

Fig. 3 Mercury Boiler

Fig. 4 Mercury Boiler Temperature Profiles - Corrosion Loop 3

Fig. 5 NaK Temperature Profiles - Corrosion Loop 4 Boiler

TYPICAL NAK TEMPERATURE PROFILE AFTER 12 GRAMS OF MIX-4P3E ADDED
(PROFILE DID NOT CHANGE DURING RECONDITIONING PERIOD)

AVERAGE BOILING HEAT FLUX DURING 72 HR RECONDITIONING PERIOD

Fig. 6 Test Run With 0.9 fps Liquid-Velocity Boiler Inlet Plug

Fig. 7 Test Run With 4.5 fps Liquid-Velocity Boiler Inlet Plug

Fig. 8 Test Run With 6 fps Liquid-Velocity Boiler Inlet Plug

MULTIPLE-TUBE POTASSIUM BOILER PERFORMANCE ✓

J. A. BOPPART, AiResearch Manufacturing Co., Phoenix, Arizona

K. O. PARKER and P. J. BERENSON, AiResearch Manufacturing Co.,
Los Angeles, California

ABSTRACT

Performance testing of a multiple-tube, sodium-heated potassium boiler has been completed as the latest step in a program to develop boilers for potassium Rankine-cycle space power systems. The test boiler contained 19 tubes, with inlet orifices for flow stability and twisted-tape inserts to improve two-phase heat transfer. Test results indicated stable operation over a wide temperature range and at output vapor qualities from 40 percent to 150°F superheat. Performance confirmed predictions at high flows and high exit quality, but was lower than expected at partial flow and low exit quality.

This paper describes the design of the test boiler and of the test facility, describes the boiler test, and presents and discusses the test results.

1. INTRODUCTION

Development of high-temperature potassium Rankine-cycle space power plants achieved a major advance recently with successful operation of a 19-tube, sodium-heated, 1600°F potassium boiler. This test was part of an effort begun in

1960 under the Air Force SPUR program, and continued under the AEC/Air Force SNAP 50/SPUR program. The objective of this program is to develop potassium saturated-Rankine-cycle space power plants for use with liquid-metal-cooled reactors, to provide electrical power output in the 300-1,000 kw range. In the boiler of such a power plant, heat is transferred from the liquid reactor coolant to the potassium, which is vaporized to 100 percent quality for delivery to the turbine.

Preliminary steps in the boiler development included studies of boiling stability and flow phenomena using Freon in quartz tubes (Ref. 1); measurement of potassium liquid heating and forced-convection boiling heat transfer coefficients in a single-tube potassium heat transfer test rig (Ref. 2); and analytical and experimental studies of mechanical design and stress aspects of a boiler design (Ref. 3). The test described herein is the first of a planned series of tests of prototype liquid-metal-heated potassium boilers, the purposes of which are to obtain overall performance and stability data and to prove mechanical design concepts and fabrication techniques. In this test, about 300 hours of high-temperature boiling operation were obtained, covering the following range of parameters:

Potassium (inside tubes)

Mass velocity 12 to 38 lb per sq ft sec

Inlet liquid temperature 900 to 1500°F

Exit saturation temperature 1450 to 1650°F

Exit quality 40 percent to 100 percent to
150°F superheat

Sodium (outside tubes)

Inlet temperature 1600 to 1850°F

Inlet-to-outlet temperature drop 20 to 150°F

Average Boiling Heat Flux

40,000 to 200,000 Btu per sq ft hr

The following sections describe the design of the boiler and of the boiler test facility, and discuss the test operation and test results.

2. BOILER DESIGN

A. General

The boiler design details are illustrated in Figure 1, and the boiler is shown installed in the test facility in Figure 2. The boiler is a shell-and-tube unit, with the potassium flowing through the 19 tubes and the sodium flowing on the shell side in a counterflow arrangement. Salient features of the design are:

1. The shell is curved into a 180-degree circular arc. This minimizes the thermal stresses induced by temperature differentials between the shell, which operates at sodium temperature, and the tubes, which operate close to the potassium saturation temperature.
2. Several inches of shell length are provided at each end between the potassium headers and the sodium plenums. These "thermal buffer regions" moderate shell thermal gradients and locate the region of maximum shell thermal gradient away from the shell welds.
3. The tubes (5/16-inch outside diameter, 0.030-inch wall thickness) are arranged in a triangular array, and supported at axial intervals

by slotted plates which permit some differential thermal expansion between tubes and shell.

4. Twisted-tape inserts are used in the tubes to centrifuge entrained droplets to the tube walls, improving heat transfer and avoiding liquid carryover into the discharge vapor. In the fabrication process, the tubes are drawn down on the tapes to provide a tight fit, and the tapes are diffusion bonded to the tube walls. The tapes extend several inches before the end of the tube, to permit liquid flowing on the tape to be centrifuged to the wall by the swirl persisting beyond the tape end.
5. Orifices are used at each tube inlet to provide a liquid pressure drop sufficient to prevent tube-to-tube two-phase flow instability. The orifices are 0.035 inch diameter, with an L/D of about 10. They are incorporated in plugs fitted tightly into the tube ends.

B. Heat Transfer

In order to determine the heat transfer area required by the full size SNAP 50/SPUR boiler, which established the tube length and diameter selected for the 19-tube boiler test, it was necessary to predict the values of liquid metal heat transfer coefficients for axial flow outside a tube bundle and both single and two-phase flow inside a tube with a twisted tape insert. A number of equally accurate correlation equations have been developed for predicting liquid metal heat transfer coefficients for axial flow outside a bundle of tubes in an equilateral triangle array. The one selected is the following (Reference 4):

$$\frac{hD_e}{k} = 7 + 3.8 \left(\frac{P}{D_e}\right)^{1.52} + 0.027 \left(\frac{P}{D_e}\right)^{0.27} \left[\frac{\psi G D_e C_p}{k} \right]^{0.8} \quad (1)$$

ψ is the average value of the ratio of the eddy diffusivity of heat transfer to that of momentum transfer; its value was predicted from the correlations presented in Reference 5.

The following equation was used to predict the liquid potassium heat transfer coefficients in the subcooled region near the entrance of each boiler tube:

$$\frac{hD_e}{k} = 3 + 0.025 \left[\frac{\psi G D_e C_p}{k} \right]^{0.8} \quad (2)$$

Equation (2) was obtained by modifying the standard correlation for a uniform wall temperature (Reference 6) so that the value of the heat transfer coefficient of tubes with twisted tape inserts relative to plain tubes is the same when the second term on the right is negligible at low Peclet numbers, and higher at high Peclet numbers where the decreased equivalent diameter ($D_e = 0.611D$) should produce an increase. The resultant velocity inside the pipe, which for the tape twist ratio of 3 in the 19-tube boiler is 13 percent greater than the axial velocity, is used in all potassium side correlation equations.

That portion of the tube where the two-phase flow exists was divided into two regions:

- (1) wet wall (annular flow) and
- (2) partially dry wall (mist flow).

The wet wall region heat transfer coefficient was predicted using the following equation.

$$\frac{hD_e}{k_\ell} = 3 + 0.025 \left[\frac{\psi G D_e C_p x \rho_\ell}{k_\ell \rho_v} \right]^{0.8} \quad (3)$$

Equation (3) was obtained by applying the reasoning, suggested in Reference 7, that annular flow be correlated by using liquid correlations modified by substituting vapor velocity in place of liquid velocity. This requires that the Peclet number in Equation (2) be multiplied by the quality

times the liquid-to-vapor density ratio given in Equation (3) which compares favorably with previously obtained potassium heat transfer coefficients. The increase in the wet wall region heat transfer coefficients that may be caused by nucleate boiling was neglected.

Data described in Reference 2 indicated that the dividing point between the wet and dry wall region occurs at approximately 90 percent quality in tubes with twisted tape inserts. The potassium heat transfer coefficient in the dry wall region was predicted by assuming that it decreased from the annular flow (wet wall) value at 90 percent quality to the dry vapor value at 100 percent quality along a straight line on a semi-log plot of h vs x . This is in reasonable agreement with the results of Reference 2. The potassium vapor heat transfer coefficient at 100 percent quality is predicted using standard single-phase heat transfer correlations.

The heat transfer equations presented above were used in the full-scale boiler optimization studies, and then to predict the performance of the 19-tube boiler whose testing is described below. The predictions show that the potassium side heat transfer resistance is negligible except at qualities greater than that at which the wall becomes partially dry.

C. Pressure Drop

The pressure drop inside the tubes is predicted using standard correlations. The effect of the twisted tape is included by using resultant velocities and path lengths, in addition to the equivalent diameter. The two-phase pressure drop was calculated using the Martinelli correlation (Reference 8) increased by 30 percent because of the results reported in Reference 2. The momentum pressure drop was calculated assuming the liquid volume fraction and velocity were negligible.

D. Flow Stability

Provision of adequate flow stability is essential to meeting the design objective of dry vapor output with no liquid carry-over. The design approach is to place at the entrance of each boiler tube an orifice which has a large enough pressure drop to provide stable, uniform flow. To determine the pressure drop required, the tests described in Reference 1 were performed in which the dimensionless parameters which influence the flow stability problem were simulated using Freon 113. The results of these tests show that, for the range of entrance subcooling, density ratio and quality of interest here, the ratio of the orifice to the two-phase pressure drop required to provide stable flow is 0.2 to 1.6. Based upon these results, the orifices in the 19-tube boiler were designed to have a pressure drop equal to the two-phase pressure drop.

3. TEST FACILITY

A. Design

The test facility, which was designed and built for boiler testing of the type described herein, is illustrated schematically in Figure 3. The facility contains two dynamic liquid metal loops; a sodium loop and a potassium loop.

The primary heat source is a d-c resistance heater in which the resistance elements are two sections of the sodium pipe loop. Heat inputs up to about 400 kw (thermal) are achieved by passing up to 7,000 amps d-c through each section. The sodium carries the heat to the test boiler, where it is transferred to the potassium. The sodium is returned to the heated section through an electromagnetic flowmeter by a helical-induction electromagnetic pump. A sump tank is provided for sodium storage when the loop is not operating, and for emergency dump. An expansion tank accommodates liquid thermal expansion. An OCI (oxide control and indication) system provides for sodium oxide detection and removal.

Potassium is supplied to the boiler by a helical-induction electromagnetic pump through a flow meter, a flow control valve, and an auxiliary heater. The valve is used to add restriction in the potassium loop if needed

for loop flow stability. The auxiliary heater is included so the liquid inlet temperature to the boiler can be varied while operating the pump and flowmeter at a constant, relatively low temperature. The potassium is vaporized in the boiler, and the vapor discharged to the air-cooled condenser, where it is condensed, sub-cooled, and returned to the pump. An expansion tank and sump tank are provided, along with a zirconium-foil-packed hot trap for oxide removal.

The physical arrangement of the facility is illustrated in Figure 4. Using the philosophy that the test item (boiler) should be the most likely failure point, the boiler is located at the highest point of both sodium and potassium loops. Thus, if a leak occurs in the boiler, it can be quickly stopped by initiating dump of the sodium and/or potassium into the sumps. Loop thermal stresses are minimized by supporting the pipe and most components on constant-load hangers permitting free lateral motion.

B. Instrumentation

The primary boiler data consists of:

Sodium side

Flow

Inlet and outlet temperatures

Shell temperatures

Potassium side

Flow

Inlet and outlet temperatures

Inlet and outlet pressures

Flows were measured with electromagnetic flowmeters which were calibrated analytically and by actual weigh-tank-type liquid-metal flow checks.

Temperatures were measured by sheathed Geminol N and P thermocouples which had been calibrated with the use of melt-point standards and checked for lack of drift at operating temperatures. Pressures were measured by both Wiancko (modified Bourdon tube) and Taber (strain-gauge) transducers.

C. Operation

Each loop was initially evacuated and filled until a liquid level was indicated in the expansion tanks. Boiling was initiated by regulating the boiler discharge potassium pressure to the desired saturation pressure, and then raising the boiler sodium inlet temperature from below to above the desired saturation temperature. Inception of boiling was indicated by displacement of liquid into the expansion tank.

Each test condition was established to provide pre-selected values of potassium flow, WK; inlet temperature,

TINK; exit saturation temperature, TSAT; exit quality, XOUT; and sodium temperature drop, ΔTR . The required sodium flow WR was computed, and the sodium pump was adjusted to give this flow. The potassium pump was set to provide the desired WK, and the auxiliary heater and condenser controls were adjusted to give the desired TINK. The potassium boiler discharge pressure POUTK was adjusted to the desired exit saturation pressure. The electrical heater power input was adjusted to give the desired value of ΔTR . Data was recorded after loop parameters had been at steady-state values for at least one hour. One complete set of data points was run during loop shakedown and operator training, and then a second complete set of data points was run.

Superheat was attained by increasing heat input until the potassium exit temperature TOUTK rose above the value of TSATK corresponding to the fixed POUTK. Operation of the loop was entirely manual, with the exception of the boiler potassium discharge pressure, which was controlled by regulating the argon cover pressure on the expansion tank.

4. TEST RESULTS

A. Heat Transfer

Overall Boiling Heat Transfer Coefficients, UBO

One of the test objectives was to measure overall heat-transfer performance as a function of potassium flow, exit quality, and saturation temperature level. Figure 5 presents preliminary data of this type. In Figure 5, overall average boiler heat-transfer coefficient UBO is plotted as a function of boiler exit quality XOUT for potassium mass velocities GK of 12, 25, and 38 lb per sq ft sec (flow rates WK of 5, 10, and 15 lb per min) and for exit saturation temperature levels of 1450, 1520, and 1600°F. The potassium inlet liquid temperature was about 100°F below the exit saturation temperature in most cases. The basic test data which was obtained after the initial training and shakedown period is shown as solid circles, and in addition the data from preliminary loop operation are shown as broken circles.

The data presented was obtained as follows:

Exit Quality XOUT

$$XOUT = \frac{WR(HINR-HOUTR) - WK(HSATL-HINK) - QL}{WK(HSATV-HSATL)}$$

where: WR = sodium flowrate

WK = potassium flowrate

QL = boiler heat loss to surroundings
 HINR = sodium enthalpy at boiler inlet
 HOUTR = sodium enthalpy at boiler exit
 HSATL = enthalpy of saturated potassium
 liquid at the boiler exit saturation pressure.
 HINK = potassium liquid enthalpy at
 boiler inlet
 HSATV = enthalpy of saturated potassium
 vapor at boiler discharge pressure.

Overall Boiling Coefficient, UBO

$$UBO = \frac{WR(HINR-HOUTR) - WK(HSATL-HINK) - QL}{AB \cdot LMTD}$$

where, in addition to parameters defined above:

AB = boiling heat-transfer area, taken as
 total heat-transfer area minus area
 required for liquid heating, the latter
 calculated from test data and from correlations
 determined by previous single-tube testing.

LMTD = boiling-section log mean temperature
 difference

$$LMTD = \frac{(TINR-TOUTK) - (TBR-TSATK)}{\ln \frac{TINR-TOUTK}{TBR-TSATK}}$$

TINR = sodium inlet temperature

TOUTK = potassium outlet temperature

TBR = sodium temperature at transition
between liquid heating and boiling
section

TSATK = potassium saturation temperature
at start of boiling

Superimposed on the 1600°F data are predicted values based on previously-obtained single-tube potassium boiling test data.

Local Boiling-Heat-Transfer Coefficients, UBL

Measurement of local values of UBL as functions of potassium flow, temperature, and local quality XL was not among the original objectives of the test, as this data is considered to be provided with greater accuracy from single-tube tests. However, boiler-shell axial temperature distributions were measured, and these provide insight into the relationship between UBL and XL. Typical shell temperature data is presented in Figure 6. In each case, the corresponding estimated two-phase potassium temperature distribution is also plotted.

There is considerable scatter in the shell thermocouple readings, due in part to their location at different circumferential positions and in part to their

location in different axial relationships to the tube support plates inside the shell. However, certain trends are notable:

1. The sodium temperature profiles indicated for the cases in which $WK = 15$ pounds per minute ($GK = 38$ lb per sq ft sec) are all concave-upward. This is the type of profile expected when local UBL is generally constant throughout the boiler.
2. The profiles in which $WK = 10$ pounds per minute ($GK = 25$ lb per sq ft sec) are not of this type, and are reasonably well characterized by straight lines. This curve shape is consistent with local values of UBL which decrease as the local potassium quality increases (from right to left on the temperature profile curves).
3. The profiles in which $WK = 5$ pounds per minute ($GK = 12$ lb per sq ft sec) are consistently of a third type, convex-upward. This shape would result from local values of UBL decreasing with increasing local quality XL, but at a more rapid rate than in cases in which $WK = 10$ pounds per minute.

By selecting a curve to characterize the sodium temperature profile for each set of data, local overall heat-transfer coefficient UBL and local quality XL can be calculated at each point along the boiler length. The plots of UBL versus XL resulting from such analysis of the data of Figure 6 are of the type shown in Figure 7. The characteristics indicated for WK = 15 agrees closely with that predicted from the single-tube test data (3), which was taken at similar and higher values of potassium flow per tube. The characteristics for WK = 10 and WK = 5 were not expected on the basis of single-tube data; however, these flows are lower than those covered by the single-tube tests.

Sodium-Side Liquid Heat Transfer

A check was made on the effect of sodium flow parameters on overall heat transfer, by holding total heat input, potassium flow, saturation temperature, and exit quality constant and varying sodium flow over a wide range. Figure 8 presents the results of these tests in dimensionless form and indicates that the variation of overall heat transfer with sodium flow agreed closely with that predicted.

B. Pressure Drop

Overall potassium pressure drop was measured, and two-phase pressure drop was computed by subtracting the inlet orifice pressure drop. Two-phase pressure drop data is presented in Figure 9, along with predicted characteristics based on the previous single-tube tests.

C. Flow Stability

Oscillograph recordings of potassium flow were made to evaluate the effect of boiler inlet subcooling on the magnitude of flow oscillations. Tests were made with no flow restriction in the loop outside the boiler, and with a relatively high loop flow restriction achieved by throttling with the loop flow-control valve between the pump and the boiler. These results are shown in Figures 10 and 11. In Figure 10 it is seen that the magnitude of flow oscillations decreases as subcooling increases, substantiating the results of Freon tests (1). In Figure 11, with a high degree of loop flow restriction, there is little or no effect of subcooling. It is important to note that flow is stable in all cases; the maximum flow oscillation is only ± 3 percent. It is interesting to note that this flow stability was achieved with a boiler inlet orifice pressure drop that was only about 30 percent of the boiler two-phase pressure drop.

D. Flow Distribution

It was observed during the tests that the liquid potassium pressure drop across the boiler was only a fraction of the inlet orifice ΔP expected on the basis of orifice design and orifice water tests. After completion of potassium testing, the individual orifice/tube flow/ ΔP characteristics were investigated in water tests to determine the reason for this discrepancy. It was determined that substantial bypassing of the orifice plugs had occurred; that is, liquid had flowed from the potassium inlet plenum into the individual tubes through gaps between the orifice plugs and the tube walls, as well as through the orifices. The bypass and orifice flow characteristics of each of the 19 tubes is shown in Figure 12. As shown, nonuniformity in bypassing resulted in relatively high flow in three tubes, with the highest-flow tube passing almost twice as much flow as the lowest-flow tube. This bypassing occurred despite the facts that the orifice plugs were sized for tight fit into the tubes, the tube inlet welds were ground flat to provide a seal against the orifice plug cap, and care was taken in the boiler design to assure that the header plate, tube inlet, and orifice plug would operate at uniform temperature, precluding differential thermal expansion effects. It appears that the diametral clearances between plugs and tubes was larger than expected, and that distortions from tack welding on the plug caps made the seal against the tube inlet weld ineffective.

5. DISCUSSION

The results of this initial multitube potassium boiler test provide encouraging tentative answers in a number of areas:

1. Operation of the boiler was stable at all times, in the sense that the output flow and pressure did not fluctuate or vary appreciably with time.
2. Initiation of boiling was easy, being consistently achieved smoothly and with only a few degrees of liquid superheat.
3. Substantial degrees of exit vapor superheat were readily obtained.
4. Heat transfer performance agreed closely with predicted values in the operating range of maximum interest; that is, 100-plus percent exit quality at high temperature, high flow, and high average heat flux.

The fact that these positive results were obtained is especially encouraging in view of the tube-to-tube flow maldistribution and the reduced stabilizing effect of orificing caused by orifice plug bypassing.

The heat transfer data also raises some questions for future investigation. Overall heat transfer coefficients

decreased sharply at low exit quality, instead of rising slightly as predicted. To compound this anomaly, conflicting indications on the trend of overall heat transfer coefficients vs local quality were obtained from the heat transfer data and from shell temperature profile analyses. The shell profile data suggests overall coefficients which generally decrease with increasing local quality, while the overall heat transfer measurements suggest coefficients increasing with increasing local quality. The extent to which the potassium flow maldistribution may contribute to this ambiguity is not known. The authors are not able to explain these apparent contradictions; it appears that the results of further tests will be needed to shed further light.

Post-test disassembly and inspection of the boiler is now under way. All preliminary indications are that the boiler design was satisfactory in all material and fabrication aspects. The tape-to-tube fabrication was successful in preventing tape wear or damage. Fabrication of a similar second boiler, in this case with twelve 3/8-inch-diameter tubes, incorporating minor design improvements and more precise shell temperature measurement provisions, is now under way. Testing of this boiler, anticipated in the next few months, will provide opportunity for further investigation of the phenomena discussed herein.

References:

1. "Flow Stability in Multitube Forced-Convection Vaporizers," Air Force Aero Propulsion Laboratory Report APL TDR 64-117, October, 1964.
2. "Forced-Convection Vaporizations of Potassium in a Single Tube," Air Force Aero Propulsion Laboratory Report APL TDR 64-115, November, 1964.
3. Quarterly Technical Progress Report for Period Ending June 30, 1965, SNAP 50/SPUR Contract AF33(615)-2289, BPSN: 5(6399-675A) 63409124 [and Contract AF33(615)-1551], AiResearch Report APS-5152-R3, July 18, 1965.
4. A. J. Friedland and C. F. Bonilla, "Analytical Study of Heat Transfer Rates for Parallel Flow of Liquid Metals Through Tube Bundles, Part II," AIChE Journal, vol. 7, no. 1, 1961, pp. 107-112.
5. O. E. Dwyer, "Eddy Transport in Liquid-Metal Heat Transfer," AIChE Journal, vol. 9, 1963, pp. 261-268.
6. R. N. Lyon, "Liquid Metal Heat Transfer Coefficients," AIChE Trans. 47, pp 75-79, 1951.
7. P. Sachs and R. A. K. Long, "A Correlation for Heat Transfer in Stratified Two-Phase Flow with Vaporization," Int. J. of Heat and Mass Transfer, vol. 2, pp 222-230, 1961.
8. R. W. Lockhart and R. C. Martinelli, "Proposed Correlation of Data for Isothermal Two-Phase, Two-Component Flow in Pipes," Chemical Engineering Progress, vol. 45, No. 1, January 1949, pp 39-46.

FIGURE 1. MULTITUBE POTASSIUM BOILER DESIGN

FIGURE 2. BOILER TEST INSTALLATION

FIGURE 3. BOILER TEST LOOP SCHEMATIC DIAGRAM

FIGURE 4. BOILER TEST LOOP ARRANGEMENT

FIGURE 5
 19-TUBE POTASSIUM
 BOILER HEAT TRANSFER
 TEST RESULTS

FIGURE 6 TYPICAL SHELL TEMPERATURE PROFILES

FIGURE 7. TRENDS IN LOCAL HEAT-TRANSFER COEFFICIENTS

FIGURE 8. BOILER SODIUM-SIDE HEAT-TRANSFER CHARACTERISTICS

- NOTES:
1. PREDICTION BASED ON MARTINELLI CORRELATION MODIFIED PER AFL TOR 64-115
 2. TSAT = 1600°F

FIGURE 9. TWO-PHASE POTASSIUM PRESSURE DROP

FIGURE 10. FLOW OSCILLATIONS AT VARIOUS SUBCOOLING WITH MINIMUM LOOP FLOW RESTRICTION (FLOW CONTROL VALVE OPEN)

FIGURE 11. FLOW OSCILLATIONS AT VARIOUS SUBCOOLING WITH MAXIMUM LOOP FLOW RESTRICTION (FLOW CONTROL VALVE THROTTLING)

FIGURE 12. POST-TEST WATER CALIBRATION OF 19-TUBE BOILER ORIFICES

HIGH PERFORMANCE "ONCE-THROUGH" BOILING
OF POTASSIUM IN SINGLE TUBES

J.R. Peterson

Space Power and Propulsion Section
Missile and Space Division
General Electric Company
Cincinnati, Ohio 45215

Presented at AIAA 1st Rankine
Cycle Space Power System Conference
October 26-28, 1965
NASA Lewis Research Center
Cleveland, Ohio

ABSTRACT

Boiling potassium heat transfer results obtained in a two-fluid Haynes-25 alloy facility are presented. The single-tube test section used is a shell and tube heat exchanger, approximately eight feet long, in which boiling potassium in the tube is heated by sodium flowing in the annulus. Boiler tubes of 0.92-inch and 0.67-inch inside diameter were tested, both with and without instrumented vortex generator inserts. Boiling heat transfer and pressure loss data were obtained from 10% quality to over 200°F of vapor superheat at the test section exit at boiler exit temperatures from 1500 to 1750°F at average heat fluxes up to about 200,000 Btu/hr-ft².

The heat transfer data presented include "overall" performance, "average" heat transfer coefficients, critical heat flux determinations, and local transition boiling heat transfer coefficients.

INTRODUCTION

In recent years, an experimental program has been conducted by the General Electric Company for the National Aeronautics and Space Administration under Contract NAS 3-2528 to investigate the problems of forced convection vaporization of potassium in boilers for application to Rankine cycle space power systems. The "once-through" boiling process, wherein a subcooled liquid is transformed into a saturated or superheated vapor in one continuous pass through a heated tube, was studied. This type of boiler is attractive for space power application because it removes the need for a boiler recirculation loop, resulting in increased reliability and reduced weight.

The heat transfer and pressure loss data from the 300 KW Facility are of interest in relation to "once-through" boiler studies. The 300 KW Facility is a two-loop system which simulates the two-fluid conditions anticipated in a space power boiler. Data from this facility, presented in this paper, indicate the feasibility of the "once-through" boiling process for potassium and provide a basis for the initial thermal design of Rankine cycle boilers for space power application.

TEST FACILITY

The experimental results presented were obtained in the two-fluid liquid metal test facility shown in Figure 1. This facility is constructed of Haynes-25 alloy, which permits continuous test operation to a maximum potassium boiler exit temperature of 1750°F. Liquid sodium flows in the primary

loop of the facility, accepting heat from a gas fired furnace and giving the heat to boiling potassium in the test section. A reversible helical induction pump is used in this loop and the sodium flow rate is indicated by an electromagnetic flowmeter. The liquid level in the primary loop is indicated by a "J" type probe in the standpipe located at a point above the boiler and is controlled by inert gas pressurization of the sodium dump tank.

Potassium flowing in the secondary loop is vaporized by boiling in the sodium-heated test section and is condensed in vertical and horizontal air-cooled single tube condensers. The vapor-liquid interface is maintained in a head tank fitted with level probes, positioned immediately downstream of the horizontal condenser. The liquid potassium flows through an electromagnetic pump and flowmeter and finally a throttling valve upon its return to the test section from the head tank.

Control of oxygen content in both loops is accomplished by zirconium gettering grids located in the dump tanks. Oxygen levels have been held to less than 50 ppm by heating the alkali metals in their dump tanks before and between periods of testing.

TEST SECTION

Figure 2 shows a disassembled view of the boiling test section employed in the experiments. It is a shell and tube heat exchanger, potassium flowing inside the tube being heated by sodium flowing in the annulus. The boiler tube is maintained concentric within the shell by means of centering pins located at three axial positions. The relative expansion between the boiler tube and the shell is accommodated by a bellows located upstream of the heated section of the tube near the potassium inlet.

Figure 3 is a schematic drawing of the test section and of one of the helical inserts used, showing the instrumentation employed. Thermocouple wells, each containing three calibrated platinum-platinum rhodium thermocouples, are located at the inlet and outlet of both sodium and potassium streams. Absolute Taylor pressure transducers are positioned at the inlet and just upstream of the boiler tube exit to determine the two-phase pressure loss. Thermocouples which provide information regarding the axial variation of sodium temperature are positioned on the outside shell of the test section in eleven rings of five thermocouples each.

In cases where an insert is used, the local potassium temperature as a function of axial position is determined from thermocouples positioned inside the centerbody of the helix.

Table I shows the combinations of insert and tube diameter investigated. The potassium was in vertical upflow in all cases. The sodium flow was countercurrent to the potassium in all tests except Case III, in which both countercurrent and cocurrent sodium flows were employed.

Table I
Boiling Test Geometries

<u>Case</u>	<u>Boiler Tube Length</u>	<u>Boiler Tube I.D.</u>	<u>Helical Insert P/D</u>
I	91.5"	0.92"	2
II	91.5"	0.92"	6
III	91.5"	0.67"	No Insert
IV	91.5"	0.67"	6

TEST PROCEDURE

The test procedure consisted of changing the potassium exit quality while the potassium mass velocity and boiler exit temperature were held constant. The potassium exit quality was varied from approximately 10% up to the maximum attainable by increasing the sodium temperature up to 1850°F, the temperature limitation of the Haynes-25 material. At a constant potassium mass velocity, the power and thus the average heat flux were increased with the result that the exit quality also increased. Two or three potassium mass velocities at nominal potassium exit temperatures of 1700°F and 1550°F were investigated in this manner. The sodium flow rate was held constant as the various potassium parameters were changed. The potassium subcooling at the test section inlet was about 300°F for most of the data reported.

Recording of a large number of instrument readings was required to define a single test. This data acquisition was accomplished by the use of a high speed digital recorder. Complete readout on paper tape of the 150 data points required was obtained in approximately one minute.

EXPERIMENTAL RESULTS

Overall Performance

Typical overall results for once-through boiling with exit superheat are shown in Figure 4*. Two cases for the

*The word "overall" refers to a measurement in which the heat transfer effects of sodium heating fluid, boiler tube wall and boiling potassium are combined.

0.92-inch I.D. boiler tube are shown which have similar values of sodium inlet temperature, potassium exit pressure and mass velocity but helical inserts of different twist ratio (P/D). Considerably more exit superheat is obtained with the tighter twist ratio. The payment for this improvement in performance is in terms of an increased boiling pressure drop; for example, the measured potassium pressure drop across the boiler tube for the $P/D = 2$ helix insert is 10 psi compared to 4 psi with the $P/D = 6$ helix insert. The improvement of the overall performance obtained with helix inserts when compared to the no insert performance is impressive in view of the fact that no exit superheat could be obtained in tests at similar conditions without an insert, an exit quality of approximately 90% being the maximum obtained.

In Figure 5, the static potassium pressure measured just upstream of the boiler exit is plotted versus the measured exit temperature. The data obtained for boiling conditions with calculated exit qualities between 10% and 90% are shown, as well as the data obtained with superheated vapor exit conditions. The data with exit qualities less than 100% fall on the saturation curve within the expected accuracy of the Taylor pressure transducers, which is approximately 1 psi. The largest amount of exit superheat was obtained with the insert of tightest twist ratio ($P/D = 2$), as is also shown in Figure 4.

The overall results are useful in that they indicate the feasibility of once-through potassium boilers. It is unlikely, however, that a space power potassium boiler would be operated under conditions identical to the test conditions. It is therefore necessary to separate the thermal resistance of the primary fluid and of the boiler tube wall from the overall thermal per-

formance, so that the effects due to the potassium alone may be evaluated.

Average Heat Transfer and Pressure Loss Results

The first step taken in the treatment of the overall performance data was the computation of "average" boiling potassium heat transfer coefficients.* The details of the computational procedures employed are given in Reference 1. In brief, the liquid or subcooled heating length, was subtracted from the total boiler length and an overall heat transfer coefficient was computed for the length of the tube in boiling by employing a logarithmic average of the overall sodium-to-potassium temperature differences at the location of initiation of boiling and at the potassium exit. The subcooled potassium length was about 10% of the total boiler tube length. The curves of Figure 6 are typical in this respect. The average boiling potassium heat transfer coefficient was calculated from the overall boiling heat transfer coefficient by subtracting the sodium and boiler tube wall thermal resistances from the overall thermal resistance. Lyon's equation (Reference 2) for an annulus was used to compute the sodium resistance, and the tube wall resistance was calculated from its thermal conductivity (Reference 8) and thickness.

Values of the average boiling heat transfer coefficient obtained with the 3/4-inch nominal diameter boiler tube containing no insert are shown in Figures 6 and 7 for both cocurrent

*The word "average" refers to the performance of the potassium fluid only, averaged over the tube length in boiling.

and countercurrent flow of the sodium heating fluid with respect to the boiling potassium fluid. Figure 6 shows the effect of mass velocity whereas Figure 7 indicates the effect of potassium exit temperature. The results obtained with the 3/4-inch nominal diameter boiler tube containing a helix insert with $P/D = 6$ are shown in Figure 8 with both mass velocity and boiler exit temperature as parameters. In Figure 9, the non-insert heat transfer results for both cocurrent and counter-current sodium flow are compared with those obtained with inserts.

The following observations can be made from the heat transfer results presented in Figures 6 through 9.

1. The average boiling heat transfer coefficient, both with and without insert, increases with increasing potassium boiler exit temperature over the range covered. This effect is less pronounced with inserts at qualities near 100%, apparently disappearing or possibly reversing for superheated exit conditions.

2. At qualities up to approximately 60%, the average heat transfer coefficient without insert increases both with increasing quality and increasing mass velocity (Figures 6 and 7).

The increase of the heat transfer coefficient with mass velocity and quality is absent or considerably less significant for the data with insert (Figure 8).

3. All of the results presented show that above a certain quality, the heat transfer coefficient decreases markedly with

further increase in quality. Figure 6 indicates that the quality at the maximum heat transfer coefficient increases with decreasing mass velocity for both cocurrent and counter-current flow of the sodium heating fluid for the tube without insert.

4. Figure 9 shows that the quality at the maximum heat transfer coefficient point is higher with cocurrent flow conditions than with countercurrent flow at the same mass velocity.

5. The results presented for the 3/4-inch nominal diameter tube (Figure 9) show the helical insert to be effective in that exit superheat was obtained with the insert whereas at similar conditions without insert a maximum potassium exit quality of only 90% was attained. The penalty paid for this performance improvement is a higher boiling pressure loss with insert than without insert, which requires an increase in the pumping power required for the two-phase loop. For given values of sodium inlet and exit temperatures and a fixed potassium exit pressure, this increased pressure loss also reduces the overall sodium-to-potassium temperature difference, which tends to decrease the overall performance. At small insert twist ratios, the sodium and potassium temperatures approach each other near the point of boiling initiation, resulting in longer boiler tube lengths being required for a specified heat transfer rate. There is an optimum insert twist ratio for any boiler design which results from the opposing effects of enhanced heat transfer performance and increased pressure loss.

The boiling pressure losses for all tests were determined from absolute Taylor slack-diaphragm pressure transducers positioned at each end of the boiler tube. Correction for liquid head was applied to the measurements. Only the pressure

losses in tests with inserts could be analyzed, since the pressure losses without inserts were of the same magnitude as the accuracy of the gages, about 1 psi. The frictional component of the boiling pressure loss was calculated by subtracting the momentum pressure loss. Using the square root of the ratio of liquid to vapor density for the slip ratio (Reference 9), the momentum component was found to be approximately 20% of the total two-phase pressure loss. The two-phase frictional pressure drop multiplier was computed as the ratio of the frictional component to the all-liquid pressure drop. The all-liquid pressure loss was calculated for the total mass flow rate. Friction factors for the computation of the all-liquid potassium pressure drop were obtained from pressure drop tests performed with water in the boiler tubes with inserts. The results of the water tests are shown in Figure 10. Figure 11 shows the calculated two-phase potassium frictional pressure drop multipliers as a function of exit quality for the boiling data taken with two different test geometries. These data are compared in Figure 11 to predictions using the "Martinelli" and "Homogeneous" models, which are derived and discussed in Reference 3. The "Martinelli" model was obtained (Reference 3) by adapting the Martinelli-Nelson curves (Reference 10) to potassium and the "Homogeneous" model was derived using quality-weighted densities and viscosities and a slip ratio of 1.0. Figure 11 is typical for all the pressure loss results in that the experimental data generally fall between the predictions of the two models.

Local Results

Considerable effort is being devoted to use the data from the 300 KW Facility to determine local boiling heat transfer

coefficients and values of the critical heat flux. This work is not complete; the available results, however, are presented.

Figure 12 is a suggested model of the behavior of the boiling fluid in a two-fluid once-through potassium boiler. Entering subcooled, the potassium is heated until sufficient liquid superheat is attained to initiate boiling. The point of net boiling initiation is the beginning of the nucleate boiling region.

The point of critical heat flux marks the end of the nucleate boiling region and the beginning of the transition region, which is presumed to be characterized by partial wetting of the heat transfer surface. The heat transfer performance in the transition region decreases with increasing quality since an increasing fraction of the heat transfer surface probably becomes dry. The transition region is terminated by the onset of film boiling, at which point the liquid droplets and liquid patches become insulated from the heat transfer surface by a vapor film. The film boiling region extends to the point at which bulk superheating commences, the superheat region being the final step in the once-through boiling process.

Recent results obtained by investigators at Geoscience (Reference 4) regarding the vaporization lifetimes of single potassium droplets on heated surfaces indicate that film boiling for individual droplets does not occur until the temperature difference between the heated surface and drop is greater than about 300°F. These results imply that the film boiling region as defined above may be absent in the controlled temperature two-fluid boiler used to obtain the

results presented in this paper. No experimental tube to potassium temperature differences as large as 300°F were attained in the boiling data reported.

Figure 13 shows temperature measurements as a function of boiler length for a boiling test run with an insert in which superheated vapor exit conditions were obtained. Figure 14 shows the calculated local heat fluxes, qualities and heat transfer coefficients as a function of boiler length for the same run. The change in slope of the sodium temperature profile indicated by the shell temperature measurements shown in Figure 13 is assumed to be the critical heat flux point, and the point at which the potassium temperature begins to rise near the tube exit is assumed to mark the beginning of the vapor superheat region. The amount of heat transferred is calculated from the potassium temperature rise in the superheat region. The point of boiling initiation and the amount of heat transferred in the subcooled liquid region are similarly obtained from the potassium temperature profile measured by the insert thermocouples.

The average heat flux in the nucleate boiling region is calculated from the axial shell temperature gradient in this region, which is assumed to be equal to the sodium axial temperature gradient. From the average heat flux and the length of the nucleate boiling region, the amount of heat transferred in this region is calculated. The heat transferred in the transition region is determined as the difference between the total heat input and that transferred in the other regions; the average heat flux in the transition region is then calculated from the measured length of the region.

The local quality and local sodium temperature at various points along the tube, including the critical heat flux point, are calculated by heat balance, employing the values of heat flux and length determined for the various regions. The local potassium temperature, obtained from the insert thermocouples, and the calculated local sodium temperature are utilized to calculate overall heat transfer coefficients for the various regions. From these coefficients, an average potassium heat transfer coefficient for each region is obtained by subtracting the thermal resistance of the sodium and boiler tube wall from the overall thermal resistance.

The average nucleate boiling heat transfer coefficients calculated in the above manner are large compared to the tube-wall and sodium coefficients and therefore represent a small fraction of the overall thermal resistance in the nucleate boiling region. This fact precludes their accurate determination in the two fluid facility utilized. The overall heat transfer coefficient in the nucleate boiling region, which includes the average nucleate boiling heat transfer coefficient and the effective coefficients for the tube wall and the sodium, can be employed, however, to calculate with good accuracy values of the local heat flux in the nucleate boiling region. In particular, the critical heat flux can be readily calculated, being the product of the overall nucleate boiling heat transfer coefficient and the sodium-to-potassium temperature difference at the critical heat flux point. A more detailed discussion of the calculational procedures is given in Reference 5.

Figure 15 shows a critical heat flux parameter as a function of quality obtained with the 3/4-inch nominal diameter boiler tube, both with and without insert. Experiments in pool boiling have indicated that the critical heat flux is proportional to

the fourth root of the local acceleration (Reference 6, 7); therefore the forced convection critical heat flux values obtained for potassium with inserts have been divided by the fourth root of $(1 + a_R)$ to give a critical heat flux parameter. The radial acceleration of the fluid in g's (a_R) developed by the insert is given by Equation (1) as a function of the insert and fluid constants. This equation was derived and discussed in Reference 7 and is as follows:

$$a_R = \left[\frac{x_c G_K \frac{\pi}{(P/D)}}{\rho_v \sqrt{\rho_l / \rho_v}} \right]^2 \frac{2}{g_o D_i} \quad (1)^*$$

Figure 15 shows that the critical heat flux parameter data obtained with the insert are in fair agreement with the values obtained without insert. For the insert data presented, the calculated radial acceleration varies from 65 to 90 g's. The correlating curve indicates a decrease in critical heat flux with increasing quality. The data shown in Figure 15 are presented in two plots, one with potassium temperature as parameter, and the second with potassium mass velocity as parameter. No definite groupings of the values with respect to either parameter is observed, indicating the effects of mass velocity and temperature to be less than the data scatter of about $\pm 30\%$ over the range covered.

A correlation of the transition boiling heat transfer coefficients was attempted, based upon the following reasoning:

1. The results from potassium droplet vaporization experiments by Geoscience (Reference 4) show that the vaporization

* A list of nomenclature is given on page 23

lifetimes of single droplets increase rapidly with increasing temperature difference between the heated surface and the droplet before the onset of film boiling. The results suggest that the wall-to-fluid temperature difference ΔT is an important parameter affecting the heat transfer coefficient in transition boiling.

2. The local vapor quality x is an important variable, because the amount of liquid which can be in contact with the heat transfer surface decreases with increasing quality.

3. The transition boiling heat transfer coefficient h_{TB} must reduce to the vapor phase heat transfer coefficient h_v at 100% quality.

4. The radial acceleration generated by a helical insert a_R , given by Equation (1), should increase the transition boiling heat transfer coefficient, because the centrifugal force generated tends to increase the fraction of liquid in contact with the heat transfer surface.

These four considerations resulted in the following equation:

$$\frac{\left(\frac{h_{TB}}{h_v} - 1\right)}{(1+a_R)^a} = A \left(\frac{1-x}{x}\right)^b \left(\frac{1}{\Delta T}\right)^c \quad (2)$$

The exponent c was determined by plotting the left side of Equation (2) versus $1/\Delta T$, using the non-insert data for which $a_R = 0$. The parameter $(1-x)/x$ did not vary significantly for the non-insert data used in the determination of c . The other two exponents, a and b , and the coefficient multiplier A were obtained by plotting all of the data, adjusting the constants until the best fit straight line was obtained. This best fit resulted in the following empirical equation, which is plotted in Figure 16 with the transition boiling heat transfer coefficient data:

$$\frac{\frac{h_{TB}}{h_v} - 1}{(1-a_R)^{1/5}} = \frac{2.55 \times 10^5 \left(\frac{1-x}{x}\right)^{0.7}}{(\Delta T)^2} \quad (3)$$

The Colburn equation (Reference 11) was used to calculate the vapor phase heat transfer coefficient h_v in Equation (2) for the plain tube without insert. For the data taken in tubes with helical inserts, h_v is calculated from a modified form of the Colburn equation, which is developed in Reference 8 using Greene's data for liquid water in helical flow (Reference 12). The average deviation of the experimental transition boiling heat transfer coefficients from Equation (3) is $\pm 23\%$.

As discussed previously, nucleate boiling heat transfer coefficients cannot be calculated with accuracy from data obtained in the two-fluid boiling tests, since the large heat transfer coefficients characteristic of nucleate boiling represent a small fraction of the overall thermal resistance of a two-fluid boiler. A controlled heat flux facility is

more appropriate for the determination of these large coefficients since there is no primary fluid thermal resistance to be accounted for under controlled heat flux conditions. Typical forced convection potassium nucleate boiling data obtained by Bond (Reference 5) are presented in Figure 17 to show the magnitude of the nucleate boiling heat transfer coefficient. The values plotted were obtained in a controlled heat flux facility at a saturation temperature of 1990°F in a 0.77-inch ID boiler tube containing no insert. It can be seen from Figure 17 that the nucleate boiling heat transfer coefficient is generally above 10,000 Btu/(hr-ft²-°F) and increases with increasing heat flux. There is very little effect of quality upon the heat transfer coefficient for the range of data shown.

DISCUSSION OF RESULTS

The quality beyond which the average heat transfer coefficients decrease with further increase in quality is associated with the critical heat flux and onset of transition boiling. As the boiler exit quality is increased beyond the quality at which the critical heat flux occurs, the low performance transition boiling region occupies an ever increasing fraction of the boiler tube length, causing the average heat transfer coefficient to decrease.

Figure 7 shows that the quality corresponding to the maximum average heat transfer coefficient increases as the mass velocity decreases. This is expected, since the experimental heat flux decreases as the mass velocity decreases. The critical heat flux results (Figure 15) indicate that higher qualities are obtained

before onset of the critical heat flux as the heat flux is reduced. For similar reasons, the maximum average heat transfer coefficient in cocurrent flow occurs at a higher quality than in countercurrent flow (Figure 9). Under countercurrent flow conditions, the local sodium-to-potassium temperature difference increases in the direction of potassium flow due to the drop in potassium temperature caused by two-phase pressure loss and due to the increase in local sodium temperature. Under cocurrent flow conditions, the sodium temperature decreases in the direction of potassium flow. The local heat flux, which is proportional to the local temperature difference, is therefore higher at the potassium exit under countercurrent conditions than under cocurrent conditions, which results in a lower predicted critical quality for the former case. This result indicates that operation with cocurrent sodium flow is advantageous for boilers exiting with less than 100% quality. For boilers with potassium leaving superheated, however, the increased temperature difference obtainable with counterflow in the very low performance vapor superheat region could counterbalance the lower quality advantage of cocurrent flow. Thus, the proper flow direction for a particular boiler must be determined from the specific operating conditions required.

APPLICATION TO BOILER DESIGN

The overall boiling data obtained in the 300 KW Facility can be employed for boiler designs by direct scale-up if the specified conditions duplicate the test conditions. The average potassium heat transfer coefficients and pressure drop multipliers can be used for design conditions other than the test conditions if reasonable judgment is employed. It is recommended that the radial acceleration supplied by the insert be employed to generalize the average heat transfer coefficients,

implying tighter twist ratios at higher temperatures due to the increase in vapor density with increasing temperature. The average heat transfer coefficients should not be applied at higher heat fluxes than those obtained experimentally, as poorer than predicted performance would result due to early onset of the critical heat flux. Care should also be exercised if the heat transfer area of a particular boiler design is to be increased over that calculated in order to obtain a "conservative" design. A decrease in heat transfer performance could conceivably result if the calculated tube length were arbitrarily increased, for example, as the increase in heat transfer area obtained could be at least partially offset by a reduction in primary to secondary fluid average temperature difference due to increased boiling pressure loss, for a fixed potassium exit pressure.

SUMMARY AND CONCLUDING REMARKS

The feasibility of high-performance once-through potassium boilers through the use of vortex-generator inserts has been experimentally demonstrated by single tube tests in a two-fluid boiling facility. Adequate forced convection potassium boiling heat transfer and pressure loss data have been obtained to permit the thermal design of once-through boilers. The usefulness of local data in the understanding of the once-through process and in the application of test results to design has been indicated and local values of the nucleate and transition boiling heat transfer coefficients and the critical heat flux have been presented. The results presented are regarded as a step towards the development of high performance potassium boilers for space power application.

ACKNOWLEDGMENTS

The results presented are part of the work performed under Contract NAS 3-2528 for the Lewis Research Center of the National Aeronautics and Space Administration.

Miss R.N. Weltmann is the NASA contract project manager, and F.E. Tippetts is the General Electric Company program manager.

BIBLIOGRAPHY

1. "Alkali Metal Boiling and Condensing Investigations", Quarterly Progress Report No. 9, Contract NAS 3-2528, SPPS-MSD, General Electric Company, NASA-CR-54215, October 20, 1964.
2. Lyon, R.N., "Liquid Metal Heat Transfer Coefficients", Trans. Am. Inst. Chem. Engrs., 47, 75-79, (1951).
3. "Alkali Metal Boiling and Condensing Investigations", Quarterly Progress Report No. 8, Contract NAS-3-2528, SPPS-MSD, General Electric Company, NASA-CR-54138, October 20, 1964.
4. Poppendiek, H.F., et al, Quarterly Technical Report on High Acceleration Field Heat Transfer for Auxiliary Space Nuclear Power Systems, AEC Contract No. AT(04-3)-409, September 1 to November 30, 1964.
5. "Alkali Metal Boiling and Condensing Investigations", Quarterly Progress Report No. 11, Contract NAS 3-2528, SPPS-MSD, General Electric Company, NASA-CR-54405, April 23, 1965.
6. Zuber, N., "Stability of Boiling Heat Transfer", Trans. ASME, April, 1958.
7. Costello, C.P., and Adams, J.M., Paper No. 30, "1961 International Heat Transfer Conference", Boulder, Colorado, August, 1961.
8. Haynes Stellite Corporation, "Haynes High Temperature Alloys", June, 1962.
9. "Alkali Metal Boiling and Condensing Investigations", Quarterly Progress Report No. 6, Contract NAS 3-2528, SPPS-MSD, General Electric Company, NASA CR-54037, April 20, 1964.
10. Martinelli, R.C. and Nelson, D.B. "Prediction of Pressure Drop During Forced-Circulation Boiling of Water", Trans. ASME, Vol-70, pg. 695, 1948.
11. Rohsenow, W.M. and Choi, H.Y., Heat, Mass and Momentum Transfer, Prentice-Hall, 1961.

BIBLIOGRAPHY (Cont'd)

12. Gambill, W.R. and Bundy, R.D., "An Evaluation of the Present Status of Swirl-Flow Heat Transfer", ORNL Report (central files number 61-4-61), April 1961.
13. "Alkali Metal Boiling and Condensing Investigations" Quarterly Progress Report No. 10, Contract NAS 3-2528, SPPS-MSD, General Electric Company, NASA-CR-54308, January 20, 1965.
14. Wesling, G.C., and McCarthy, M.E., "Thermodynamic Properties of Potassium Vapor from 1300°R - 2700°R - Using Naval Research Laboratory Experimental Data", GE Report R64SD3012, SPPS, MSD, General Electric Company, October 6, 1964.

NOMENCLATURE

a_R	Radial acceleration developed by insert at tube wall, g's
D_i	Inside diameter of boiler tube, feet
f_i	Friction factor based upon inside tube diameter, dimensionless
G_K	Potassium mass velocity, lb/(sec-ft ²)
g_o	Standard gravitational acceleration, $4.17 \times 10^8 \frac{ft}{hr^2}$
K	Potassium
L	Length, feet
Na	Sodium
N_{Re}	Reynolds Number, dimensionless
P	Pitch, axial distance for 360° revolution of insert, feet
P/D	Tube diameters per 360° revolution of insert, dimensionless
Q	Total heat transferred, KW
q''	Heat flux, Btu/(hr-ft ²)
$\overline{q_B''}$	Average boiling heat flux, Btu/(hr-ft ²)
q_c''	Critical heat flux, Btu/(hr-ft ²)
T_c	Potassium temperature at point of critical heat flux, °F
T_{KO}	Potassium outlet temperature, °F
V_a	Axial velocity, ft/sec
x	Quality, dimensionless
x_c	Local quality at point of critical heat flux, dimensionless
ΔP	Pressure loss, psi
ΔT	Wall-to-potassium temperature difference, °F
$\frac{\Delta P_{TPF}}{\Delta P_o}$	Integrated two-phase frictional pressure drop multiplier, dimensionless
μ	Viscosity, lb _m /hr-ft
ρ_ℓ	Liquid density, lb/ft ³
ρ_v	Vapor density, lb/ft ³

Figure 1. 300 KW Two-Fluid Heat Transfer Facility.

Figure 3. Instrumentation of 300 KW Facility Test Section.

Figure 4. Measurements and Overall Results for Once-Through Boiling With Inserts in 300 KW Facility.

Figure 5. Comparison of Saturation Properties With Measured Potassium Pressures and Temperatures in 300 KW Facility.

Figure 6. Average Boiling Potassium Heat Transfer Coefficients with No Insert, Effect of Mass Velocity.

Figure 7. Average Boiling Potassium Heat Transfer Coefficients with No Insert, Effect of Boiler Exit Temperature.

Figure 8. Average Boiling Potassium Heat Transfer Coefficients with Insert, Effect of Mass Velocity and Temperature.

Figure 9. Comparison of Average Boiling Potassium Heat Transfer Coefficients with and without Insert, with Countercurrent and Cocurrent Sodium Flow for the Non-insert Case.

Friction Factor,

$$f_i = \frac{\Delta P}{\left(\frac{L}{D_i}\right) \left(\frac{\rho V_a^2}{2g_c}\right)}$$

Figure 10. Single Phase Friction Factors For Water in Tubes With and Without Inserts.

Figure 11. Boiling Potassium Pressure Drop Data From 300 KW Facility.

Figure 12. Suggested Model of Once-Through Boiling.

Figure 13. Measured Fluid Temperature Profiles in 300 KW Test Section.

Figure 14. Local Results for Typical Once-Through Boiling Run With $P/D = 6$ Insert in 300 KW Facility.

Figure 15a. Effect of Mass Velocity

Figure 15b. Effect of Temperature

Figure 16.
Potassium Transition
Boiling Data and Empirical
Correlation
(Equation 3).

Figure 17. Potassium Nucleate Boiling Data Obtained Under Controlled
Heat Flux Conditions (Reference 5).

TURBOALTERNATOR

Chairman: D. L. Southam

TRW Inc.
Cleveland, Ohio

BLANK

1.5 kw_e ORGANIC RANKINE CYCLE
COMBINED ROTATING UNIT

by

Donald A. Malohn
Manager, Advanced Projects
Sundstrand Aviation-Denver
Denver, Colorado

ABSTRACT

The 1.5 kw organic Rankine cycle combined rotating unit (CRU) consists of a single stage partial admission impulse turbine, main alternator, speed control alternator, and working fluid pump on an integral shaft supported on hydrodynamic tilting pad bearings. An opposing pair of tilting pad thrust bearings are used to control the shaft axial location. The cycle working fluid is used as the bearing lubricant. The working fluid pump is of the pitot type. The organic working fluid used in this Rankine cycle is Dowtherm "A," a eutectic mixture of Biphenyl and Biphenyl Oxide. The alternators are of the radial gap homopolar type and produce 3-phase, 3200 cycle power. Power level of the CRU can be varied by changing the size and number of nozzles used. This unit was designed, fabricated, and has been tested in excess of 2,000 hours total cumulative test time. A single set of bearings installed in the CRU has operated in excess of 600 hours, cumulative running time, and has been subjected to in the excess of 100 start-stop cycles. Associated turbine performance testing has indicated good agreement between test results and analytical predictions.

INTRODUCTION

This paper describes the design and development status of the combined rotating unit used in the 1.5 kw organic Rankine cycle. The initial design studies on this system were completed under Bureau of Naval Weapons contract NOW-62-1037c. The development test phase was under contract NOW-64-0071-f.

The 1.5 kw system employs a Rankine cycle turbine directly coupled to an alternator and liquid pump. The cycle processes are consistent with those of conventional closed Rankine cycles. The working fluid in the liquid phase is pumped to the maximum cycle pressure, heated to saturated liquid conditions, boiled and expanded through the turbine. The low pressure turbine exhaust vapor is cooled to saturated vapor conditions, condensed, and subcooled. The subcooled liquid then re-enters the pump and repeats this path around the closed loop. The working fluid selected for this system is Dowtherm A, an organic substance.

Dowtherm A is an almost clear, slightly straw colored liquid at ambient temperatures and has a freezing point of 53.6 degrees F. A temperature entropy diagram with the cycle paths superimposed is shown in Figure 1. An important characteristic of this working fluid is the positive or reverse slope of the saturated vapor line. This particular physical property differentiates Dowtherm A from other working fluids such as water and liquid metals, which have negative sloped vapor lines.

A flow schematic of the 1.5 kw system is shown in Figure 2. The working fluid is pumped to the maximum cycle pressure (1-2) and is heated in the alternator cooling passages by heat from alternator losses (2-3). The high pressure liquid is further heated in the liquid side of the regenerator (3-4) and is then heated to saturation temperature and boiled in the heat receiver (4-5). The heat input to the boiler is obtained from solar heat concentrated in the heat receiver cavity by the mirror. The saturated vapor then expands through the turbine (5-6) and some of the waste heat is removed from the exhaust flow in the vapor side of the regenerator (6-7). The waste cycle heat is rejected to space by radiation as the working fluid is desuperheated, condensed, and subcooled in the radiator-condenser (7-1).

It is beyond the scope of this paper to report in detail on the complete system. This paper is limited to one portion of the 1.5 kw system, namely, the combined rotating unit.

COMBINED ROTATING UNIT (CRU)

The mechanical design of the 1.5 kw CRU is shown in Figure 3. The arrangement of the 1.5 kw CRU represents a straightforward integration of several system components into a single unit.

The critical speed of the 1.5 kw CRU was calculated using the Prohl method for flexible bearings. Bearing stiffness (spring rate) for this design can be adjusted to shift the location of the critical speed. Critical speed versus spring rate for this design is shown in Figure 4. The spring rate calculated for the 1.5 kw bearings is 27,600 lb/in. at design speed. The calculated first critical speed at the design spring rate is 15,100 rpm.

The weight breakdown for the CRU is shown in Table 1. The 1.5 kw CRU was designed and built to demonstrate a concept and no attempt to achieve a flight weight unit was made.

TABLE 1

<u>COMBINED ROTATING UNIT</u>	<u>WEIGHT</u>
Alternator Housing and Windings	32.00
Alternator Rotor	6.50
Turbine	2.25
Nozzle Housing	7.50
Pump Housing	1.75
Rear Bearing Housing	2.25
Pump and Bearings	<u>1.25</u>
	53.50

TURBINE

TURBINE ANALYSIS

Design charts for various types of turbines were generated in which curves of constant turbine efficiency and of principal geometric parameters are plotted, with specific speed (N_S) as abscissa and specific diameter (D_S) as ordinate.

An N_S - D_S diagram calculated for this application is shown in Figure 5. The optimum turbine geometry was calculated using this figure. To achieve optimum efficiency at various turbine mass flows requires different turbine designs. To permit maximum flexibility one turbine wheel was designed to operate from 0.01 to 0.05 lbs/sec. The diameter selected for the 1.5 kw CRU is 6.265 in.

The detailed design data were obtained using a digital computer and a partial admission turbine design program modified for the thermodynamic properties of Dowtherm A. The nozzle arrangement selected for this design consists of four nozzle housings set at 17 degrees, the number and size of the nozzle inserts is selected from the required power output.

TURBINE WHEEL DESIGN

The wheel design is based on the use of Inconel X material at a maximum temperature of 600 degrees F. Tertiary creep is not a problem at this temperature so that the 0.2 percent allowable yield stress is the limiting criterion for structural design.

The wheel and shaft are non-integral. This selection is based on mechanical and electrical considerations. The turbine wheel is held to the shaft by an interference fit. The radial hub displacement at design speed and temperature was calculated to be 0.0005 inches. The tip radial displacement is 0.009 inches. The wheel design used in the 1.5 kw CRU is conservatively below the 0.2 percent allowable yield stress of 84 ksi.

TURBINE BLADE FREQUENCY

A turbine blade passing through a jet of high velocity gas, as in partial admission turbines, vibrates at its natural frequency. For long life operation (10,000 hours) the turbine blades must be shrouded to increase the blade natural frequency to a point where the vibration amplitude produced with the available energy is not detrimental. The shroud used on the 1.5 kw turbine wheel consists of a Rene' 41 ring brazed to the Inconel X disc.

TURBINE WHEEL FABRICATION

Two Inconel X turbine wheel forging blanks were selected for use in the 1.5 kw program.

The heat treatment of the wheels was based on information from International Nickel and was selected to provide the best creep properties below 1100 degrees F. The various steps in the fabrication are shown in Figure 6.

TURBINE TEST RESULTS

Single nozzle (scaled) turbine tests were conducted by maintaining the turbine speed, inlet temperature and inlet pressure at constant values, and varying the condenser pressure. This technique was used to increase the number of data points available while running at design conditions. Three basic nozzle types were tested: axisymmetric, orifice, and conical. Two conical nozzles with different throat inlet ramp angles were included in the test program. The four nozzle configurations tested are shown in Figure 7. The axisymmetric nozzle (No. 1) was designed for an area ratio (A/A^*) of 32.40, which has a corresponding design pressure ratio (PR) of 190. Nozzle Nos. 2, 3, and 4 were designed for an area ratio of 19.49 with a corresponding design pressure ratio of 100.

Alternator performance from calibration tests was used to determine the power distribution for these tests. The alternator speed and bearing flows were held constant through-

out the test to eliminate any variation in bearing losses.

Because Nozzle No. 1 is designed with a smaller throat area and consequently passes a lower mass flow, direct comparison of nozzle performance cannot be made without normalizing the test results. The test results were normalized by calculating turbine efficiency based on actual turbine exhaust pressure. Turbine efficiency for the various nozzle configurations tested is shown in Figure 7.

ALTERNATOR

ALTERNATOR DESIGN

The alternator is a homopolar inductor type, rated at 1.75 kw, 120/208 volts, 3-phase, 3200 cps, directly coupled to and driven by the turbine at 24,000 rpm. The alternator is rated for continuous operation at AIEE Class H (180 degrees C) temperatures and in an environment of Dowtherm A liquid or vapor or both, at 0.5 psia pressure. At full load and under normal operating conditions efficiency is 85 percent, including all control power. The alternator is cooled by liquid Dowtherm A flowing through passages integral with the alternator stator housing. The alternator is designed to operate for 10,000 hours in a space environment, unattended and with no maintenance.

The alternator assembly incorporates a second homopolar inductor alternator section (rated 0.75 kw, 0 to 120/208 volts, 3-phase, 3200 cps, 24,000 rpm) mounted on the same shaft as the main machine. This second alternator transfers its output power, infinitely variable from 0 to 0.75 kw to a fixed load. The amount of power transferred is controlled by varying the excitation of the second alternator and is always such that it exactly compensates for any change in load on the main alternator. In this way a constant shaft load, and hence a constant shaft speed, is maintained on the turbine driving the two alternators. This method of speed control requires only a very small, simple, electronic control package.

Selection of the homopolar inductor alternator is based upon its inherently rugged and reliable rotor construction which does not require the use of composite construction techniques. No provision for rotating windings, bars, cages, or magnets need be made in the rotor for excitation of the machine because of the homopolar inductor machine, magnetic circuit arrangement. Brushes, slip rings, and other means of supply electrical power to the rotor are eliminated. This permits construction of a rotor of very high mechanical strength with an overall improvement in reliability.

ALTERNATOR CONSTRUCTION

Alternator materials were selected and optimized for electrical, mechanical, and thermal properties. The rotor is fabricated from AISI 4130 steel alloy. The magnetic flux-carrying stator yoke is fabricated from MT 1015 steel alloy. These materials combine good mechanical properties with reasonable magnetic qualities and compatibility with Dowtherm A. The alternator stator laminations are of Monimax material, 0.006 inch thick.

The excitation winding of the homopolar inductor is a stationary concentric-wound coil in the stator. The relative location of the stator laminations and the field coils are shown in Figure 8.

ALTERNATOR CALIBRATION

Electrical checkout and calibration were performed on an air-bearing dynamometer. After alternator calibration, the alternator was used as a dynamometer to measure turbine performance. The overall main alternator performance characteristics as a function of output power are shown in Figure 9.

During calibration of the alternator, it was found that electromagnetic capabilities predicted for the machine were not being obtained. The basic rated load design capacities were satisfactory at lower temperatures, but rated load and overload capacities at elevated temperatures were marginal. Test results indicated this was due to a lower than expected alternator magnetic capacity. Static leakage and magnetic material tests were made and analytical work reviewed and compared to these test results. In all cases the test results were found to be in accordance with analytical predictions, and the analytical work reviews showed no errors in original design assumptions or development. Tests did indicate, however, that the magnetic behavior of the alternator stator lamination material in the machine was lower than characteristic for the material. This pointed to the possibility of improper material heat treatment, which is rather critical for this material.

Inasmuch as improper heat treatment of material was believed to be the reason for marginal machine performance, it was decided to conduct additional machine performance tests with stator lamination material known to be properly heat treated. A test stator assembly, consisting of a stator housing, field coil, main alternator stacks, and armature winding was made using Armco Trancor T laminations. This test stator was of the same configuration and dimensions as the final CRU assembly, but did not incorporate detail items needed for the CRU (i. e., alternator control windings, stator cooling provisions, exhaust ducts, stator elements for dissipator). Tests were then conducted using a 16 pole solid rotor, and the existing dynamometer and adaptor equipment.

Results of these tests are shown in Figure 10. As expected, considerable improvement in machine output capability was evident when these results were compared with earlier test results with the Monimax laminations. The comparison indicates that the previous marginal machine performance with Monimax material was due to incorrect heat treatment of the Monimax material.

PITOT PUMP

PUMP SELECTION

A pitot pump was selected for this unit because of its inherent characteristics. It is insensitive to cavitation, requires no high pressure seals and can separate insolubles. This type of pump is a rotating cylinder (impeller) with internal vanes. The fluid is admitted at the center of the impeller where impeller rotation imparts a head to the fluid. The fluid is then extracted from the impeller by a pitot tube. A schematic of the pump is shown in Figure 11.

PITOT PUMP DESIGN

The pitot pump design was extrapolated from previous test data and is based on two empirical coefficients, flow factor (ϕ) and head coefficient (ψ). A flow factor of 0.3 with a corresponding head coefficient equal to 0.6 was selected for the small mass flow requirements of the 1.5 kw system. The calculated efficiency for the pitot pump is 26 percent at the design conditions.

PITOT PUMP TESTS

The pitot pump did not function as expected during preliminary tests, because of the pump inlet configuration. To rectify this problem, a new inlet plate was fabricated and installed. This modification was also incorporated into the CRU hardware. After the initial pump checkout program was completed, additional testing was undertaken in order to evaluate pump performance as a function of variations in pump geometry.

The primary pump performance characteristic to be investigated was pump efficiency. However, as the test program progressed it was found that net changes in pump efficiency were not discernible because of the effects of bearing power and pump leakage. Figure 12 shows the effect of probe strut configuration. As a result of this series of tests, it

was concluded that except for configuration 1 (rectangular probe cross section) all configurations tested yield the same performance.

BEARINGS

Bearing loads are very light under zero-g conditions, but ground test conditions require that the bearings be able to carry the dead weight of the rotating mass of the CRU. To allow freedom of orientation of the CRU shaft axis from horizontal to vertical, the thrust bearings are designed to carry the entire nine-pound shaft weight, and the radial bearings are designed to nominally carry 4.5 pounds each. No pump thrust allowance was made as the shaft thrust of a pitot pump is negligible.

JOURNAL BEARING

Tilting pad journal bearings were selected mainly because of their inherent dynamic stability, a particularly important consideration for high speed bearings and the light shaft loads under zero-g operating conditions. This bearing is self-aligning. The low viscosity of the lubricant and the small sizing of bearing elements result in thin fluid films. High reliability can be expected from this type of bearing because the adverse effects of assembly tolerances, thermal distortion, and dimensional instability are minimized by the self-aligning characteristics of the bearings. A journal set consists of three pads equally spaced around the journal; each pad is equipped with a screw adjusted pivot pin. Since total bearing clearance is small, provisions to eliminate backlash of the pivot adjusting screws are incorporated in the design. This bearing type behaves well at or near a critical speed, and can be relied upon to pass through critical speeds without damage. It is easier to adjust shaft critical speeds away from design speed for ultimate zero-g operation, since the bearing spring rate range is considerably narrowed in this operating condition.

Estimates of liftoff speed for these bearings were made assuming the bearing surfaces to be perfectly smooth and located at the position defined by the root-mean-square height of the surface asperities. With a 4.5 pound load and a 4 rms finish, liftoff should occur at about 500 rpm. Liftoff speed is deferred to about 2,000 rpm when the surface finish is 8 rms.

THRUST BEARINGS

The two tilting pad bearing design was selected primarily for the simplicity of the gimbal design. Three or more pads require a much more complex leveling system involving a large number of pieces. This system was not desirable with the small bearings required for this application.

To provide adequate capacity within a small envelope and to take advantage of the large available space between pads, the pad breadth (dimension parallel to relative runner motion) was increased about 40 percent over the radial pad length.

BEARING MATERIALS

For both the journal and thrust tilting pads capable of multistart operation, sliding of the runner against the bearing is required for liftoff and let-down when no hydrodynamic wedge separates the surfaces. A low shear strength film against a hard, polished surface is favored for this duty. The bearing pads were fabricated from AISI 4130 steel alloy plated with silver and topped with a dry film lubricant.

BEARING TESTS

A complete dynamically similar bearing system, pictured in Figure 13, was tested as a separate entity for proof of design and for bearing system evaluation. The bearing test rig used hardware geometrically identical to the CRU hardware to insure proper bearing system loads and dynamic conditions.

The bearing component tests were run using water as a lubricant. The test rig was operated at various pressurizing gas densities to determine the bearing power absorption. Extrapolation of these test results to zero gas density, or zero pressure, yields 0.25 hp the true loss attributable to the bearings. This empirical loss is five times the calculated loss. As a result of this inconsistency the bearings were reanalyzed. The results of this modified analysis which included the effects of turbulence (Reynolds Number) are within 6.5 percent of the actual test results.

Several methods for starting the CRU were considered. One method required the CRU to start without liquid in the bearing cavities. To determine the number of starts permissible under these conditions, a start-stop test was conducted in which no lubricant was present in the bearing cavity except the dry film already applied to the bearings. A new set of bearings was installed and the test rig was accelerated to 24,000 rpm, power removed, and allowed to coast down. The coast down time for each start was recorded. The results of these tests are shown in Figure 14. This figure gives the rig deceleration time, from operating speed to stall, as a function of the number of starts. An interesting characteristic of these data is the increased deceleration time, or reduced bearing friction, at start number 3. This indicates that it would be advantageous to run-in an operational bearing before actual usage.

CRU TESTS

A single test loop was used for development testing, system integration testing, and endurance testing.

On initial build-up of the CRU, the rotor shaft retaining threads were found to be right-hand and therefore incorrect. The unit was assembled with the incorrect shaft thread to avoid a delay in the program. The effect on the unit was considered, and it appeared that if the pump unscrewed, it would simply cause the unit to stop. The unit did stop after 13.25 hours of preliminary checkout testing, and subsequent inspection showed that

the pump had unscrewed. During the initial testing, the second rotor was modified to correct the shaft thread. The unit was reassembled using a new set of bearings. Excessive loading started by the pump unscrewing had failed the thrust bearings.

At the completion of 149.4 hours, the unit was shut down to investigate a noise in the turbine wheel area. The vibration level at this time was 2 g's on the Z axis. The noise and vibration level was periodic in nature. An inspection revealed that the dowel pin used to retain the rear seal ring had come loose. The unit was reassembled with new seal rings in which the pin had been peened into the seal ring as a temporary fix. The seal design was also modified to use a larger pin welded in place. At 156.75 hours the unit was disassembled and new welded configuration seals were installed.

The unit continued to operate until the test was terminated at 173.9 hours. The test was intentionally terminated to rebuild the CRU with a shrouded turbine wheel. The initial design concept of the CRU had incorporated a shrouded wheel. However, in order to obtain loop checkout and operational experience, the original build was made with an unshrouded wheel due to the long lead time required for shrouding. Since the potential problem of blade vibration is long term, no effect was noted on the unshrouded wheel, even when inspected under 500 X magnification. The original turbine wheel was subsequently shrouded.

Except for miscellaneous test loop problems, the unit ran uneventfully to 1,000 hours. The unit was then disassembled for a 1,000 hour inspection of the shrouded turbine wheel. Inspection of the turbine wheel under high magnification with a super penetrant revealed no indications of cracking.

The CRU was reassembled with new bearings and seal rings; all other components were unchanged.

CRU was automatically shut down at 1,235.6 hours because of a facility power failure. The CRU was removed from the test loop and disassembled for inspection, a standard

procedure for this type of shutdown. Inspection revealed that the aft floating seal had seized, and pulled out the retaining pin. The load resulting from the seal seizure failed the front radial and thrust bearing. The CRU was reassembled with new bearings and a new aft seal; all other components were the same. The unit was restarted and the recorded vibration level was normal.

Up to this time all testing had been accomplished with a conical nozzle. In order to investigate the effect of nozzle geometry on turbine performance a fully profiled (axisymmetric) nozzle was manufactured as designed by the Shock Wave Cancellation Theory.

The full scale unit (Unit No. 2) was fabricated using an integral condenser, and all the life test hardware from the scaled unit. An exploded view of the full scale CRU hardware modifications are shown in Figure 15. The integral condenser on the full scale CRU was mounted above the unit to facilitate integration of the pitot pump into the complete system at a later date. On the first full scale CRU checkout run, it was discovered that the system was flow-limited and only two nozzles at a time could be used. A series of tests were run to determine the cause of the excessive pressure drop. The testing indicated that the pressure drop through the alternator coolant passages and through the liquid side of the regenerator were approximately equal, and were higher than anticipated. In order to reduce these pressure losses, the unit was further modified by installing a second regenerator in parallel with the one already installed. The modification was done in this manner because program time and funding did not permit fabrication of a new regenerator and a second regenerator core had been fabricated and was available. Although the total pressure drop would still not be at the design point, the effective regenerator pressure drop would be reduced significantly. Incorporating the CRU modifications for the second regenerator core are shown in Figure 16.

Initial open loop testing of the pitot pump operating in the CRU indicated that the pump performance was essentially as predicted from previous open loop component tests. The pitot pump was originally designed for scaled operation but direct integration proved unsuccessful. This was due to the increased bearing pumping losses when the condenser

was integrated into and relocated above the CRU. The additional pumping loss was approximately 200 watts, which is almost the net output in the scaled configuration. In order to use a jet pump to scavenge the bearing cavities and eliminate this loss, the pitot pump flow through would have to increase and provide jet pump primary flow. In doing this, the pitot pump would be driven so far off design as to negate its ability to provide the required output head. Attempts to redesign the pump and change its operating characteristics were unsuccessful. The outside diameter of the pump impeller could not be increased because of pump housing limitations. Attempts were made to increase the pitot tube diameter, but this reduced the pump inlet flow area. This, combined with increased pump through-flow created an inlet cavitation problem and the desired pump characteristics were not obtained.

In order to continue CRU testing the pitot pump was left in the system and augmented by the facility pump. The unit was therefore run in a semi-closed loop mode.

The accumulated running time for the various CRU components are shown in Table II.

TABLE II

<u>ITEM</u>	<u>SERIAL NUMBER</u>	<u>TOTAL HOURS*</u>
Bearing Support Pins	1	2, 102
Nozzle Housing	1	2, 102
Alternator Rotor	2	2, 089
Turbine	2	2, 029
Bearings	8	679**
Forward Seal	4	679
Rear Seal	2	2, 089
Alternator	1	1, 998
Regenerator	1	1, 998
Regenerator	2	236

* Accumulated as of 9/16/65

** 114 starts

CONCLUSIONS

The engineering and technological development tasks completed to date have clearly established the feasibility of this energy conversion concept. However, even though system feasibility has been experimentally demonstrated several areas have been identified which require additional development and design optimization. This, of course, is an expected situation in any development program.

Figure 1. Temperature-Entropy Diagram for Dowtherm Closed Rankine Cycle

Lube System Schematic

Figure 2. 1.5 kw System Schematic

Figure 3. CRU Cross Section

Figure 4. 1.5 kw CRU Bearing Spring Rate Versus Critical Speed

Figure 5. Diagram for Partial Admission - Axial Impulse Turbines

Figure 6. Turbine Wheel Fabrication Steps

Figure 7. Turbine Efficiency

Figure 8. Alternator Laminations

Figure 9. Load Performance Data

Figure 10. 1.5 kw Alternator (Test Machine) Performance

Figure 11. Pitot Pump Schematic

Figure 12. Pitot Pump Performance (Head Rise vs Flow) for Various Probe Strut Configurations ($a/\Delta = 5.84$, $L/\Delta = 5.84$)

Figure 13. Bearings Test Rotor and Bearing

Figure 14. Dry Start 1.5 kw Bearing Test

Figure 15. CRU No. 2 before Final Weldment
Installed Regenerator Core Shown in Right Foreground

Figure 16. Modified CRU (Front View)

MRPC (SNAP 2) TURBOALTERNATOR MODEL CRU V

By R. C. Biering, Project Engineer

TRW Inc.

and O. Decker, Sr. Project Engineer

TRW Inc.

ABSTRACT

The Mercury Rankine Power Conversion Program (MRPCP) formerly called SNAP 2 is being conducted at TRW Inc. to develop the technology required to design a turboalternator for space application. The turboalternator (CRU V) has provided much useful information for this objective, demonstrating to the extent possible the feasibility of a dynamic Mercury Rankine System operating in space. This report presents the CRU V design details and the results of the development program to date. This work is being performed by TRW Inc. under a subcontract from Atomics International as a part of the Atomic Energy Commission Contract AT(11-1)-GEN-8.

I. INTRODUCTION

Development of turboalternators for Mercury Rankine Systems was initiated at TRW Inc. in 1958 with the testing of the first SNAP I unit. Since that time, several different turboalternators have been developed at TRW Inc. for various applications; including SNAP I, SPUD Ballistic Test Flight, Mercury Rankine Power Conversion Program (MRPCP - formerly SNAP 2), SUNFLOWER, and SCAP. These units have a combined total of more than 32,000 hours of turboalternator operation. The turboalternator now being developed for the MRPC Program is called Combined Rotating Unit V (CRU V).

In January, 1962, design and development of the Combined Rotating Unit (CRU V) Turboalternator Package for the SNAP 2 flight system was initiated. A concept evaluation and study program was completed in July of 1962 and the basic CRU V concept was established. The necessary layout and detail drawings and supporting analyses were then completed. Fabrication was initiated in February of 1963 and the first CRU V was tested in September 1963. This report presents the important considerations and features of the CRU V design

and describes the results of the test program accomplished to date.

During the development of CRU V, some of the initial requirements were altered or deleted due to advancements in technology and/or changes in program objectives. These are as follows:

1. The requirement for a CRU shaft mounted magnetic type NaK pump was deleted in favor of a line mounted thermoelectric pump. This change resulted in improving system performance by allowing the power normally required by the shaft mounted NaK pump to be made available as electrical output power.
2. The program objectives were altered, deleting the flight test and directing activities towards the advancement of the technology required for Dynamic Mercury Rankine Systems. At this time the program name was changed from SNAP 2 to Mercury Rankine Power Conversion Program (MRPCP).
3. The requirement for startup with unpreheated CRU housings was deferred in favor of developing repeatable startup capability with a preheated CRU. The initial requirement for unpreheated startups was established in order to conserve system weight. Present booster rocket capability indicates that the small additional weight required for preheaters and batteries is no longer a problem. Thus, unpreheated startup development has been temporarily deferred until other development objectives have been completed.

The CRU V was designed to operate in the system indicated in Figure 1. System operation is initiated with preheat of the boiler by NaK flow circulation and preheat of the CRU housing by electrical heaters. At the completion of preheat, a valve on the discharge of the injection tank opens and mercury flow is established to the bearings and the boiler simultaneously. Bearing flow passes through the bearings, into the bearing drain lines, and ultimately reaches the condenser pressure regulator tank. Boiler flow is vaporized in the boiler and passes into the turbine causing the CRU shaft to spin up to design speed. Vapor flow leaving the turbine passes through the alternator stator cooling fins,

SYSTEM SCHEMATIC

FIGURE 1

out of the CRU housing, and into the condenser where it is condensed and remains until later in the startup cycle. When sufficient bearing drain inventory has entered the condenser pressure regulator tank, pump inlet pressure is established and the mercury pump primes. When injection tank inventory is exhausted, the mercury pump is coupled to the system supplying both bearing flow and boiler flow by drawing accumulated inventory from the condenser pressure regulator tank. When condenser pressure reaches design value, a valve at the discharge opens and allows inventory in the condenser to enter the condenser regulator tank and/or the mercury pump inlet. When inventory in the condenser stabilizes, start-up is completed.

The major specifications for the design of CRU V are indicated in Table I.

II. DESIGN DESCRIPTION AND COMPONENT STATUS

A. Design Description

A cross-section of the CRU V Turboalternator is shown in Figure 2. All of the CRU components including the alternator, the turbine, the mercury pump, and the bearings are contained within a single housing which consists of four major elements:

1. A lower one piece semi-cylindrical cradle-type bearing support structure which also supports the turbine housing and alternator stator.
2. An upper semi-cylindrical shell that is bolted and welded axially to the bearing cradle.
3. The turbine end cap which is bolted and welded circumferentially to the turbine end of the cradle-shell assembly.
4. A turbine exhaust bell that is welded circumferentially to the pump end of cradle-shell assembly.

All of the rotating components are mounted on a single shaft as shown and include the turbine wheels, alternator rotor, and pump impeller supported by two fluid film journal bearings and a double acting thrust bearing. Since the thermodynamic working fluid, mercury, is used as the lubricant for the bearings, only non-contacting, labyrinth type,

TABLE I
MAJOR CRU V SPECIFICATIONS

<u>Speed</u>	36,000 rpm
<u>Turbine</u>	
Inlet Pressure	115 psia
Inlet Temperature	1150°F
Discharge Pressure	9 psia
Discharge Temperature	628°F
Efficiency	55% minimum
Flow rate	20 lb/min.
<u>Alternator</u>	
Frequency	1800 cps
Voltage	63 ±9.5 volts, line-to-neutral 110 ±16.5 volts, line-to-line
Output Power	4.1 KW nominal
Power Factor	1.0 to 0.8 lagging
Phases	3-phase, 4 wire, "y" connected, external grounded neutral
Duty Cycle	Continuous
Efficiency at gross power output	87%
<u>Hg Pump</u>	
Inlet pressure to jet	5.9 psia minimum steady state
Maximum inlet temperature	443°F
Discharge pressure	525 psia ± 25 psia
Flow	50 lb/min. ± 5 lb/min.
Efficiency (with dynamic seal)	41%
<u>Bearings</u>	<u>Three-Pad</u>
Power consumption - alternator	210 watts
turbine	210
thrust	100
Flow rate - alternator	8-10 lb/min.
turbine	8-10 lb/min.
thrust	7-8 lb/min.

CRU V TURBOALTERNATOR ASSEMBLY

420

FIGURE 2

dynamic seals are necessary between components. The alternator bearing assembly which contains the static portions of the mercury pump, thrust bearing and alternator bearing is clamped to the cradle at one end and the turbine bearing assembly is clamped to the cradle at the other end. Lubricant leakage from the bearing supply lines is prevented by the use of static seals at the interface between the cradle and the bearing assemblies. The turbine housing is clamped in place next to the turbine bearing and the alternator stator is clamped in place next to the alternator bearing. Mercury vapor from the discharge of the turbine fills the inside of the CRU housing and removes losses in the form of heat from the alternator stator as it passes over cooling fins attached to the stator OD. The major CRU V design features are summarized in Table II.

B. Component Description

1. Bearings

The lubrication approach selected for the CRU V model and similar turboalternators for such systems as the earlier SNAP 1/SPUD, Sunflower and SCAP was dictated to a large degree by the requirements of the application. These requirements include:

- . Long life without maintenance
- . Ground and orbital operation
- . Low system weight to power ratio
- . Maximum reliability

As indicated earlier the approach selected was to use the system working fluid as the lubricant. In the mercury Rankine cycle this dictated the use of fluid film rather than rolling element bearings. Advantages of the selected approach included:

- . Elimination of the need for zero leakage seals
- . Elimination of the need for an accessory lube system
- . No contamination potential by the presence of a secondary fluid (lubricant)
- . Simplicity of concept (single rotating shaft rather than several rotating assemblies with duplicity of bearing and seal problems)
- . Improved reliability by eliminating seals and extra components associated with

an accessory lube system.

- Lower over-all weight
- Shorter shaft arrangements (with regard to critical speed considerations)

The major disadvantages of the selected approach can best be categorized by the associated problems. These include:

- The limited lubrication properties of liquid mercury
- Turbulent flow lubrication was required.
- The restrictions imposed by the application in terms of size, thermal gradients, operating modes and available flows and pressures
- Ground and space operation and attendant requirements on the bearings

These disadvantages imposed difficult development problems in the early days of mercury Rankine systems and led to the evaluation of the bearing types shown in Figure 3. However, improved understanding of the influence of turbulent flow in the clearance space on performance characteristics and careful attention to the integration of the bearings into the package (discussed elsewhere) minimized potential bearing problems.

The following bearing characteristics, in addition to the general CRU V requirements listed earlier, were used as selection criteria; stability, critical speed, load capacity, misalignment tolerance, mechanical and thermal distortion, injection start capability, power loss, tolerance to clearance change; damping, hydrostatic start-stop, life, freedom from cavitation-erosion damage, fluid film stiffness control, flow-pressure requirements, tolerance to particles, component test performance, CRU test experience, and ease of CRU integration.

As a result of careful and detailed analysis of plain, three sector, three pad and tilting pad (self-aligning) types, the three pad, orifice compensated journal bearing was selected for the CRU V.

The three-pad bearing shown in Figure 4 is essentially a plain journal bearing with three

TABLE II

MAJOR CRU V DESIGN FEATURES

- Thermal design requirements more independent of structural design requirements and hence more attainable in design.
- Capability of eliminating need for shaft disassembly during final CRU assembly process.
- One-piece, uniform-temperature bearing support structure.
- Material selection flexibility resulting from separation of structural and component functional elements.
- Relative structural independence of components which are related to the bearing structure rather than to each other.
- Fabrication advantages including:
 - 1. Minimum welding,
 - 2. Improved parts interchangeability and spares availability,
 - 3. Reduction in assembly process final (selective fits) machining.
- Relative ease of component design modifications.
- Minimized cutting and rewelding of external tubing for assembly and disassembly.

MERCURY LUBRICATED BEARING DEVELOPMENT

POCKET LAND THRUST BRG

SPIRAL GROOVE THRUST BRG

THREE SECTOR COMPENSATED BRG

PLAIN BEARING

TANGENTIAL INLET BRG

LOAD PAD BEARING

THREE SECTOR BEARING

THREE SEC BRG W/SUPPLY PADS

THREE PAD COMPENSATED BRG

TILTING PAD BEARING

TILTING PAD SHOES

TILTING PAD SHOE SHOWING PIVOTS

FIGURE 3

feed supply pads equally spaced around the circumference and three equally sized restricting flow orifices. These three orifices are called compensating orifices because they tend to equalize the flow and supply the highest pressure to the pad supporting the highest load. The supply pressure forces the lubricant through the bearing, and the journal rotation tends to cause lubricant to flow circumferentially. The bearing is defined as "hybrid" since its load capacity is derived from both hydrostatic and hydrodynamic lubrication. Total bearing flow is a function of the bearing clearance space and the supply pressure.

The selection of the three-pad orifice compensated bearing met the basic desire of a relatively simple journal bearing, free from half-frequency whirl instability, without compromising the attainment of the other primary CRU V requirements.

The double acting, spiral groove thrust bearing shown in Figure 5 was initially selected for the CRU V because by analysis and performance evaluation it satisfied the requirements and specific CRU V design ground rules. It also had a good history of satisfactory performance in previous CRU and component tests. The spiral groove thrust bearing is essentially a low flow capacity centrifugal pump. An outward pumping design was selected to utilize fluid inertia to achieve increased load capacity. Mercury is introduced under pressure at the eye (or inside diameter) of the bearing, and the shallow spiral grooves pump the fluid outward. Dams located at the end of the grooves result in a velocity-pressure exchange. The resultant pressure profile establishes the load carrying capacity. The bearing as integrated into the CRU is hybrid in nature in that mercury is introduced to the inlet (inside diameter) under pressure. Some measure of compensation is provided by an annular orifice formed by the outside diameter of the shaft and the inside diameter of the bearing. The dual face feature provides the bearing with load capacity in both directions, but the slope of the spiral grooves permits rotation in only one direction.

During reduced preheat startup testing of the CRU, it was determined that the imposed thrust loads during spinup were greater than the load capacity of the spiral groove thrust

FIGURE 4

SPIRAL GROOVE TEST THRUST BEARING

FIGURE 5

bearing. A hydrostatic thrust bearing utilizing the available high injection pressures, was designed to provide the necessary load margin to survive this startup mode. This bearing is purely hydrostatic and, therefore, its load capacity is independent of speed. Compensation is provided by supply orifices rather than by annular clearance space. The hydrostatic thrust bearing provides the required load capacity at low speeds without changing the existing space envelope, flows and pressures. The selected design is shown in Figure 6 from which it can be seen by the symmetry of the design that the bearing performance is independent of direction of rotation.

2. Turbine

The turbines required for space power machinery such as SNAP 1/SPUD, SNAP 2 (CRU V), Sunflower and SCAP differed from those of more standard applications in size and speed. Using mercury as the thermodynamic working fluid was pioneered in large public utility binary cycle power plants. Experience with these power plants, operating for 16 years without inspection or teardown of the large turbines provided assurance that proper material selection minimized blade erosion as a major problem. Since that time continued improvements in turbine blade materials has all but eliminated erosion as a potential problem in mercury Rankine systems.

The minimum size and weight requirements of space power machinery dictated the need for high rotative speeds. The resultant high speed, compact turbine represented the major difficulty in accomplishing a satisfactory, efficient design.

System and cycle analysis of the selected mercury Rankine concept established essential requirements such as power, speed, vapor flow, inlet and exit pressures and temperatures, and the available space envelope for the turbine assembly in the CRU. In addition the turbine had to meet the over-all requirements of long life without maintenance (no performance degradation below an acceptable level); ground and orbital operation; low system weight to power ratio; and maximum reliability.

Originally little design information existed on high speed compact mercury turbines. The

procedure was to scale down from large size steam turbines even though the existing technology was not very applicable because of the very small percentage admissions and low aspect ratios required. Losses associated with partial admission, high Mach numbers and condensation were some of the major unknowns in terms of mercury performance. Although these losses are inherent in high speed, compact mercury turbines they were better understood and more predictable in CRU V because of experience gained in earlier turboalternators.

The CRU V turbine was designed to meet the following specifications:

. Supply temperature	1150°F
. Supply pressure	115 psia
. Discharge temperature	628°F
. Discharge pressure	9.0 psia
. Nominal weight flow	20 lb/min
. Required shaft power	5610 watts
. Design efficiency	0.55
. Design speed	36,000 rpm
. Over-all available energy	29.4 Btu/lb
. Fluid	Mercury

To meet these requirements, several configurations were considered. The high desired efficiency and the reaction inherent in radial turbines made radial configurations inapplicable. The axial-type turbines offered more flexibility in that one or more stages could be used to divide the load most efficiently, impulse operation was possible, and partial admission could be used without extremely detrimental effects. Single, two-stage and three-stage axial-flow impulse types were evaluated.

A two-stage unshrouded impulse design was selected for CRU V with a partial admission first-stage. The reduced average moisture, lower blade loading and more predictable performance were primary advantages.

The major design details are listed below:

HYDROSTATIC THRUST BEARING

FIGURE 6

CRU V SHAFT ASSEMBLY

FIGURE 7

First stage	30% admission
Second stage	Full (93.5%) admission
Blade height	
First stage wheel	0.228 inch
Second stage wheel	0.208 inch
Material	
First stage wheel	Nivco 10
Second stage wheel	15-7Mo (integral with shaft)

The following design considerations dictated the final configuration:

- Velocity ratios between 0.35 and 0.45 were selected for maximum efficiency.
- Work split was arranged so that the first-stage was fully subsonic (Mach number 0.6) whereas the second-stage was slight transonic (Mach number 1.1). This assured elimination of losses associated with separation in the blades when operating at or near sonic inlet velocities. The second-stage which does approximately 62% of the work was maintained slightly transonic by a small degree of reaction (approximately 7.8%).
- Number of blades and solidity were adjusted to aid fabrication.

Figures 7, 8, and 9 show CRU V shaft assembly with the first and second stage turbine wheels, the second-stage (split nozzle), and first-stage nozzle, respectively.

3. Pump

A pump required for space power machinery such as SNAP 1/SPUD, SNAP 2 (MRPCP), Sunflower and SCAP, differs from commercial boiler feed pumps primarily due to high speed and low suction pressure operation. Pumping mercury in itself does not represent a major problem. In fact, the high density of mercury permits high heads with small diameter centrifugal impellers. This characteristic helps achieve the goals of a minimum size and weight turboalternator. For the mercury Rankine system, i.e., the CRU V, the requirements of the pump are to supply the required boiler and bearing lubricant flows at the required

FIGURE 8

pressures, prime as required during the startup sequence, operate with low seal leakage, maintain low thrust loads, and absorb minimum shaft power. These requirements must be satisfied for the life of the system.

Specific CRU V pump requirements were:

Inlet temperature	443°F max.
Inlet pressure	6.0 ±0.1 psia
Flow	50 ±5 lb/min
Discharge pressure	525 ±25 psia
Speed	36,000 rpm
Efficiency	30-50%
Operating life	1 year (10,000 hours)
Thrust	Minimum
Leakage	Minimum
Startup	Prime at 36,000 rpm At design inlet and shut-offhead
Power consumption	250 watts

The selected concept of a single rotating shaft with the pump providing both boiler and bearing flow established the operating specifications such as speed, flow, pressure and allowable power levels. These requirements in turn imposed several difficult design and development problems. They include:

- The high rotative speed (N) and small net positive suction head (NPSH) resulted in a suction specific speed (S) which was beyond the acceptable limit for long life (cavitation free) performance.

$$S = \frac{N(Q)^{1/2}}{(NPSH)^{3/4}} \approx 30,000$$

- Similarly low volume (Q) and the high head rise requirements to provide adequate bearing flow resulted in a specific speed (N_s) which was too low for good efficiency.

$$N_s = \frac{N(Q)^{1/2}}{(H)^{3/4}} \approx 500$$

- The high parasitic losses inherent in a low specific speed centrifugal pump design resulted in low efficiencies.
- The miniature size of the pump which evolves from standard design practices required expensive close tolerance machining to obtain reproducibility of performance.

A pump combination consisting of a centrifugal impeller with a vaneless diffuser and a single jet inducer was selected after evaluating the capabilities and limitations of various types and combinations. The selected concept reduced the problems associated with potential cavitation and low efficiencies to an acceptable level. Specifically the incorporation of the jet inducer resulted in the following:

- Reduction in suction specific speed for the centrifugal pump from 30,000 to 8,000.
- Improvement in combined pump efficiency from 28% to 35% by increasing specific speed from approximately 500 to 1,000.
- A mercury pump capable of meeting all life and performance requirements.

Particular attention was given to provide good flow guidance in the impeller to minimize separation and cavitation. To meet the requirements of minimum seal leakage, reduced losses within the impeller and diffuser, and balanced axial thrust, dynamic seals in the form of impeller back vanes were incorporated. As a result the impeller efficiency was increased from 54% to 64%.

The jet inducer selected for CRU V, shown schematically in Figure 10, includes a driving nozzle and an inlet tube which are at right angles to each other. The flows from these two sources converge in the mixing tube region which is coaxial with the nozzle. After the mixing tube, a divergent passage is employed to recover the velocity head of the fluid. The fluid is then transported from the jet pump, which is external to the hermetically sealed CRU, to the centrifugal pump impeller.

JET PUMP DESIGN

434

FIGURE 10

The centrifugal pump design selected initially for CRU V and shown in Figure 11 has five backward curved vanes with a ten-degree inlet blade angle and a thirty-one degree discharge. A second impeller design (Figure 12) was developed to achieve improved performance. Increased solidity was accomplished by addition of a sixth vane. The blade discharge angle was also increased from 31 degrees to 45 degrees. The result was a higher discharge pressure and greater flow at the same speed (36,000 rpm) and inlet pressure (5.8 to 6.2 psia). This impeller is currently in use on all CRU V models. The backface of the impeller, shown in Figure 13, has eleven radial vanes. The impeller is mounted on the shaft through an internal lock thread.

The diffuser consists of a constant height diffusion passage of increasing diameter, directing the flow into a semicircular toroidal collector. The diffuser inlet employs a slight convergence about one diameter in front of the impeller and has an impeller facing which will offer 0.003 inch running clearance. The collector discharge consists of a hole drilled tangentially to the toroidal passage, followed by a diffusion passage similar to that shown on the jet pump.

4. Alternator

The minimum size and weight and maximum reliability requirements imposed by the system applied to the alternator design as well as to components discussed above. Since size and weight of rotating electrical equipment is inversely proportional to speed for a given power output, a high speed alternator was required to minimize system weight.

Alternator selection for the CRU V was determined by system temperature, the mercury vapor atmosphere, and nuclear radiation specifications. Temperature, atmosphere, and reliability considerations required that the machine selected for CRU V be an alternator without brushes or rotating windings. The CRU V alternator was designed to meet the following detailed specifications:

Operating temperature	750°F
Frequency	1800 cps

5 VANE MERCURY PUMP IMPELLER, CRU V

FIGURE 11

6 VANE MERCURY PUMP IMPELLER, CRU V

FIGURE 12

Voltage	63 $\pm$ 9.5 volts, line-to-neutral
	110 $\pm$ 16.5 volts, line to line
Output power	4.1 KW nominal
Power factor	1.0 to 0.8 lagging
Phases	Three-phase, four-wire, "Y" connected, externally grounded neutral
Duty cycle	Continuous
Efficiency at gross power output (min)	87%
Compatible with a 700°F mercury vapor environment.	

A six-pole permanent magnet alternator was selected for use in the CRU V because of the ability to meet the specification requirements and experience gained with this type on earlier CRU models.

This experience placed the permanent magnet alternator in an advanced stage of development. In this selection, reliability and development schedule (i.e., use of proven components wherever possible) were key considerations. The permanent magnet alternator offered the following advantages:

- Slip rings, brushes, rotating windings, external excitation, and a voltage regulator were eliminated.
- Symmetry and simplicity of the rotor construction permitted operation at high speed and contributes to ease of CRU fabrication and assembly.
- Efficiency was high and weight and volume were low.
- The three-phase winding provided maximum power output and efficiency for a given volume.

The alternator design was optimized to operate with a nominal radial magnetic gap between the rotor and the stator of 0.015 inch. With a 0.004 to 0.006 inch thick bore seal (see below), the nominal radial gap is 0.009 to 0.011 inch. Figure 14 shows the CRU V alternator cross-section.

FIGURE 13

CRU V ALTERNATOR STATOR PRESENT DESIGN

FIGURE 14

The CRU V stator is a sealed design wherein the silver windings are protected from mercury by a seal-welded frame and end-bell structure. The bore of the laminated stator core is coated with a thin-film of glass ceramic. The electrical leads are brought through a tubular extension and a hermetic feed-through ceramic header outside of the CRU. The tubular extension is sealed to the CRU housing with a welded bellows assembly.

Because of the brittle characteristics of the glass ceramic, the reliability of the bore seal is difficult to control or predict. Therefore, to insure winding protection, the stator interior (winding cavity) is kept evacuated below mercury saturation pressure. Evaluation tests and actual CRU test experience has shown that even at 700°F, leakage of mercury vapor through an evacuated stator winding cavity has little or no corrosive effect on the silver conductors.

The rotor design was dictated largely by considerations of high speed, high temperature operation, and the need to shield the Alnico magnet from the mercury atmosphere. A solid six-pole Alnico V permanent magnet is used in the alternator construction. The solid cylindrical magnet is encased peripherally within a high strength shroud ring (PHL5-7Mo) which also forms the shaft continuity between the turbine and alternator bearing ends of the shaft. Each end of the solid magnet is isolated from the magnetic shaft material by thin, nonmagnetic steel wafers to prevent flux short circuiting of the rotor poles. The use of steel wafers rather than air gap provides improved cooling of the rotor.

5. Materials

It was not possible to design the CRU V with one or two materials which are simply compatible with mercury because of the many other performance requirements. As an example, the materials required for the bearings, the magnetic core, and the turbine wheels, require properties not available in a single material. Therefore, it was necessary to select materials which satisfied a particular requirement but still maintained the necessary degree of compatibility with the surrounding materials. Figure 15 indicates the selected materials.

CRU V TURBOALTERNATOR MATERIALS SELECTION

FIGURE 15

The following list represents some of the properties considered when selecting a CRU V material:

Thermal expansion rate	Hardness
Thermal conductivity	Castability
Modulus of elasticity	Cost
Yield strength	Availability
Ductility	Thermal shock and thermal stress resistance
Weldability	Erosion resistance
Resistance to oxidation	Cavitation resistance
Resistance to NaK corrosion	Machineability
Resistance to mercury corrosion	Dimensional stability
Magnetic permeability	Creep rupture strength
Electrical resistivity	Creep strength

Table III summarizes the selection criteria for each of the CRU V components.

C. INTEGRATION ANALYSIS

1. Shaft Dynamics

The critical speed of the CRU V bearing-rotor system were heavily dependent on the following factors:

- . total mass and mass distribution
- . shaft flexibility
- . bearing type and stiffness
- . bearing span
- . bearing housing and mounting flexibility

System and component requirements mentioned previously combined to fix the total mass and mass distribution of the shaft assembly so that the critical speed mode shape was established within certain limits. The major variable which was available for controlling the cross-over or critical speed of the CRU V bearing-rotor system was the bearing stiffness.

Figure 16 shows the first and second mode critical speed of the CRU V rotor as a function of bearing stiffness. Influence of bearing stiffness is illustrated in this figure. For the first mode, at values of bearing stiffness up to 1×10^5 lb/in, the shaft behaves as a rigid member (i.e. its deflections are very small) and the natural resonance is principally a function of the stiffness of the fluid film in the bearing. For bearing stiffness from 10^5 to 5×10^5 lb/in, the shaft stiffness exerts a progressively stronger influence on the critical speed. Beyond values of 5×10^5 lb/in the critical speed is independent of bearing stiffness (i.e. the resonance is the generally accepted bending critical of a simply supported beam).

For the second mode the same evaluation shows that the middle region, where both the bearing and shaft stiffnesses play significant contributing roles, stretches over a larger region of bearing stiffness. Values greater than 10^6 lb/in are required before bearing stiffness ceases to influence the resonant frequency.

The behavior of the rotor system at a third mode is dependent on all four parameters,

TABLE III

CRU MATERIALS CRITERIA

PART NO.	PART NAME	MATERIAL	REQUIRED PROPERTIES									
			HIGH MAGNETIC PERMEABILITY	LOW MAGNETIC PERMEABILITY	EXPANSION COMPATIBILITY (PART NUMBERS)	THERMAL SHOCK RESISTANT	HIGH CREEP	STRENGTH	HARDNESS	WELDABILITY	NONMAGNETIC	HIGH RESIDUAL MAGNETISM
1	ALTERNATOR MAGNET	ALNICO V										X
2	ROTOR SHROUD RING	15-7 MO		X	1,3	X	X		X			X
3	SHAFT	15-7 MO			8,7, 24,5	X		X				X
4	BEARINGS	BG-42			3,5,12,11	X	X	X				X
5	CRADLE	17-4 PH			10, ,3,4,6					X		X
6	TURBINE HOUSING	17-4 PH & NIVCO 10			5,7,11	X				X		X
7	TURBINE INTERSTAGE	17-4 PH			6,3					X		X
8	FIRST WHEEL	NIVCO			3	X	X					X
9	SECOND WHEEL (INTEGRAL SHAFT)	15-7 MO					X					X
10	ALTERNATOR STATOR CAN	HAYNES 25			5					X	X	X
11	BEARING FACE COVERS	15-7 MO			4,6					X		X
12	ALTERNATOR BEARING & Hg PUMP HOUSING	17-4 PH			5,4,13,3							X
13	MERCURY PUMP IMPELLER	17-4 PH			12,3	X		X				X

CRU V (W/o NaK ROTOR)
CRITICAL SPEED vs BEARING STIFFNESS PLOT

FIGURE 16

i.e., mass, mass distribution, shaft stiffness and bearing location. At bearing stiffnesses below 10^4 lb/in this mode is asymptotic to the first "free-free" mode, i.e. the shaft vibrating without constraints. Since the range is above 50,000 rpm, it was of little importance.

The importance of being able to control the bearing stiffness is demonstrated in Figure 16. The desire was to select a bearing having a stiffness such that the cross-over point occurred at a low frequency and which at design speed provided sufficient separation with the critical speed and was compatible with all the other requirements (mainly load capacity, stability and flow-pressure characteristics). The three-pad bearing operates at stiffness values below 1×10^5 lb/in.

However, the significance and problems associated with critical speeds was minimized by the high damping characteristics of liquid mercury lubricated bearings. The high damping served to greatly reduce the resonant amplitude. As a result, it was possible to operate close, at or through a critical speed with a reasonably balanced rotor.

2. Thermal Map

The CRU V thermal design satisfied the following objectives:

- . minimum bearing heat load
- . minimum condensation within the "can" during steady-state operation
- . moderate thermal gradients
- . sufficient alternator cooling capacity

Figure 17 shows the predicted thermal map of the selected CRU V design. A review of the heat loads, condensation rates and cooling capacity shows how the above objectives were satisfied.

a. Bearing Alignment Structure

The bearing cradle is maintained relatively isothermal. When combined with the uniform circumferential temperature distribution between the cradle and the top of the "can", bearing alignment was assured.

445

b. Bearings

Heat loads into the bearings were controlled by minimizing heat flow areas, providing insulation at critical sections, and reducing vapor flow to the bearings. Vapor flow into the bearing drain area is limited by venting into the can and maintaining the ΔP across the bearing vapor seals to 3 psi. The total turbine bearing and alternator-thrust bearing heat loads were estimated around 600 and 760 watts, respectively, at design point. Observe that the heat load into the alternator and thrust bearing area includes the thrust bearing power loss and 150 watts of condensation on the housing. The vapor condensation was calculated at a rate of approximately 4 lb/hr and is due to the heat transfer from the vapor through the insulation into the cooler bearing assembly.

c. Alternator Cooling

The alternator presented one of the more difficult areas for thermal analysis. Yet to assure proper alternator operation, adequate removal of the alternator losses was essential to assure that certain limiting temperatures were not exceeded.

In the CRU V alternator, cooling was accomplished by passing turbine exhaust vapor across finned surfaces attached to the stator outside diameter and by dissipating some of the rotor losses to the alternator bearing drain flow. Approximately 470 watts must be removed to assure that the limiting temperature of 750°F, controlled by the bore seal, is not exceeded. Analysis showed that 80 watts would be dissipated to the alternator bearing drain flow. The calculated heat rejection of the selected CRU V fin design was approximately 390 watts. These calculations included only the convective cooling of the mercury vapor. CRU tests have shown that most of the cooling occurs due to vaporization of the moisture from the turbine exhaust. Since vaporization occurs at constant temperature, the cooling capacity of the stator fins was greatly increased.

d. Turbine Housing

The turbine housing was designed to provide nearly saturation temperatures at the piloting diameters. The vapor temperature of the high pressure labyrinth seal leakage was reduced from superheat to saturation temperature by heat transfer to the turbine bearing.

Since this heat transfer was only 30 watts, it did not represent a substantial increase in bearing heat load.

3. Bearing Alignment Verification

During the conceptual study and design phases, particular emphasis and attention was paid to minimizing potential bearing misalignment due to both manufacturing tolerances and thermal distortions.

As indicated in the section on the CRU V design, the primary purpose for the pedestal-canned concept was to provide a single continuous structural member and a relatively isothermal environment to assure minimum bearing misalignment. The importance of alignment was clearly demonstrated in early CRU models which had a history of failures attributed to misalignment caused by turbine and alternator bearing supports related through a series of pilot diameters and intermediate members which had to maintain bearing alignment in the presences of severe thermal gradients.

The current CRU V model permits a maximum total bearing misalignment of 0.00024 inch for fabrication tolerances. In addition, the difference in thermal environment between the alternator bearing and the turbine bearing causes a difference in centerline shift which is equivalent to an additional 0.000035 inch of misalignment over the bearing axial length. The resultant maximum possible misalignment was specified as 0.0003 inch to provide further margin. This factor was then considered in establishing the bearing type, size, and clearance, along with the other requirements.

The ability of the CRU to maintain bearing alignment during extreme conditions of mechanical loading and during severe thermal gradients has been demonstrated in numerous bench type tests. During these tests misalignment was always well within allowable limits.

III. CRU TEST RESULTS

Following the completion of the design, five CRU V's were fabricated and put into test. A test program was established and followed which enabled evaluation of CRU performance

with respect to fulfilling the MRP program requirements. Some of the significant results of this program are presented in the following sections.

A. TEST OBJECTIVES

The following were the major test objectives of the CRU V test program:

1. Design Verification

These tests were conducted to establish the fundamental soundness of the design concept and to verify the assumptions, calculations, and predictions utilized to complete the design. The tests were aimed at exposing design deficiencies and to provide for the debugging necessary in all new designs. Component and CRU performance was determined for a range of operating conditions varied approximately $\pm 10\%$ from design point.

2. Endurance

Endurance tests were conducted to demonstrate feasibility of endurance operation for 10,000 hours. Because time and funding did not allow conducting a 10,000 hour test, a 90-day demonstration was the initial objective. A demonstration test of 2500 hours is the present objective.

3. Injection Startup

Injection startup tests were conducted to insure compatibility of the CRU V design in a simulated system configured environment. The initial objective was to develop startup capability with no preheat of the CRU. This has since been modified to allow for pre-heat.

4. Shock and Vibration

Shock and vibration tests were conducted to verify capability of the CRU to withstand the shock and vibration levels anticipated during launch. These tests were with the CRU in a non-operating condition.

5. Margin Tests

Margin tests were conducted to verify CRU operating capability at conditions remote

from design point. These tests were conducted to verify the range of safe operating conditions for the CRU.

B. TEST PROGRAM AND RESULTS

Testing of CRU V was initiated in September of 1963. In the succeeding two years, 19 separate assemblies of the original five CRU V's have been tested. The test program is indicated in Figure 18 and shows the type of tests conducted on each CRU assembly as well as the accumulated hours of operation. During this period of testing the following general results have been obtained.

1. The design concept was verified and the ability to meet design point specifications was established.
2. Endurance feasibility was demonstrated.
3. Injection startup capability was demonstrated with a preheated CRU.
4. The ability to withstand specification level shock and vibration with no detrimental effects was demonstrated.
5. The ability to operate satisfactorily at conditions far removed from normal design point was demonstrated.

1. CRU Package Performance

a. Design Verification and Margin Tests

Verification of the CRU V design concept was obtained in December of 1963 following the completion of a 200 hour parametric test on CRU V-1A. During this test the operating conditions of the CRU were varied approximately $\pm 10\%$ from the design point. Testing since that time has demonstrated the ability of the CRU to operate without detrimental effects over a much wider range of conditions. Figure 19 indicates the present range of operating conditions for which CRU V has demonstrated ability for steady-state operation. This figure indicates the design point conditions as well as the maximum and minimum conditions for which CRU V's have operated to date. The column marked maximum indicates the maximum operating point obtained for the single parameter. These operating points were not all obtained at the same time nor were they obtained on a

CRU V TEST HISTORY

FIGURE 18

CRU V OPERATING RANGE

	PARAMETER	DESIGN POINT	TEST RANGE TO DATE (STEADY STATE)	
			MAXIMUM	MINIMUM
SPEED, RPM	RPM	36,000 ± 360	40,000	20,000
TURBINE	INLET PRESSURE, PSIA	115 ± 5	142	—
	INLET TEMPERATURE, °F	1,150 ± 25	1,200	1,050
	DISCHARGE PRESSURE, PSIA	9.25 ± .25	13.0	7.4
	FLOW, PPM	20 ± .20	24.9	—
	EFFICIENCY, % @ DESIGN POINT	55	58	49
BEARING (JOURNAL)	FLOW, RPM	9.0 ± 1	10.0	2.5
	INLET TEMPERATURE, °F	450 MAX.	450	320
	INLET PRESSURE, PSIA	330 ± 30	490	130
	DRAIN PRESSURE, PSIA	6.1 ± .2	7.0	—
BEARING (THRUST)	FLOW, RPM	9 ± 1	10.0	5.5
	INLET TEMPERATURE, °F	450 MAX.	450	320
	INLET PRESSURE, PSIA	350	405	265
	DRAIN PRESSURE, PSIA	6.1 ± .2	7.0	—
ALTERNATOR	POWER, WATTS	4,100	5,600	—
	VOLTAGE, VOLTS	110 ± 16.5	130	110
	EFFICIENCY, % @ DESIGN POWER	87	91	88
PUMP	INLET PRESSURE, PSIA	6.1 ± 2	9.2	2.3
	INLET TEMPERATURE, °F	443 MAX.	460	320
	DISCHARGE PRESSURE, PSIA @ 45 PPM	525 ± 25	480	400
	FLOW, RPM	45 ± 5	66	—
CRU	MACHINE EFFICIENCY, % @ DESIGN PT.	43	44	37

FIGURE 19

single CRU. The same is true for the column marked minimum. It is important to note that CRU V without modification has demonstrated the ability to deliver 5.6 KW of power merely by increasing the vapor flow through the turbine to 24.9 lbs/min. The 5.6 KW does not represent the maximum power limit of the CRU but was established by a limit encountered in the test rig.

Some typical CRU test results are indicated in Figure 20 and Figure 21. Figure 20 indicates the CRU machine efficiency as a function of gross alternator output power. Data for this were obtained over a range of output powers from 1.5 to 5.6 KW. As can be seen from the curve, machine efficiency was 43 percent at the design power output of 4.1 KW. Figure 21 indicates the good agreement obtained between the predicted thermal map of the CRU and the temperatures obtained during test.

b. Endurance Testing

Highlights of CRU V endurance testing are indicated in Figure 22. As can be seen by the curve of accumulated run time as a function of calendar time, endurance operation has continued to grow from the start of testing in September of 1963 until over 14,700 hours of operating time have been accumulated to date. The longest run on a CRU V was 3,153 hours. Three CRU V's have completed the present endurance objectives of 2,500 hours. Four CRU's have operated for over 2,000 hours and five CRU V's have operated for over 1,000 hours. The longest continuous period of operation on a single CRU V was the 2,341.6 hours accumulated on CRU V-1A. Many of the major components of CRU V-1, -1A, and -1B accumulated over 5,738 hours of operation. These tests clearly establish the ability of CRU V to reach the present endurance specification of 2,500 hours and also demonstrate the feasibility of operation for 10,000 hours.

c. Startup Testing

As indicated by Figure 22 three CRU V's have undergone startup development tests. CRU V-1C has demonstrated repeatable startup capability by completing 25 injection startups with the CRU preheated to design temperature level (600°F). During these starts no detrimental effects were noted. The CRU had a total of 39 startups. CRU V-2C

CRU V MACHINE EFFICIENCY VS GROSS ALTERNATOR OUTPUT

FIGURE 20

CRU V THERMAL PERFORMANCE

CRU V TEST DATA..... °F
 PREDICTED VALUES.... (°F)

FIGURE 21

CRU V TEST HIGHLIGHTS

ENDURANCE DEVELOPMENT		FY
CRU V-1 A	2557.5 HOURS (2341.6 HRS. CONTINUOUS)	64
CRU V-3B	2049.0 HOURS	64
CRU V-5	860.7 HOURS (5.6 KW CAPABILITY DEMONSTRATED)	
CRU V-1, IA & IB	2ND STAGE NOZZLE, TURB. HSG, Hg PUMP IMPELLER, SHAFT (INCL. 2ND STAGE TURB. WHEEL), ALT. BRG.	64 & 65 5738.5 HRS
CRU V-1B	3153.4 HOURS	65
CRU V-5B	1277.4 HOURS	65
CRU V-3D	2500 HOURS	66
START-UP DEVELOPMENT		
CRU V-2C	400° F PREHEAT LEVEL - WITH NO DETRIMENTAL EFFECTS. AT 200° F PREHEAT LEVEL - WITH THRUST BRG. SCORING	64
CRU V-5A	400° F PREHEAT LEVEL - WITH NO DETRIMENTAL EFFECTS	65
CRU V-1C	25 PREHEATED INJECTION STARTS WITH NO DETRIMENTAL EFFECTS 39 TOTAL STARTS	66
SHOCK AND VIBRATION		FY
CRU V-4A	VIB. ONLY WITH NO ADVERSE EFFECTS	64
CRU V-2D	NO ADVERSE EFFECTS	65

TOTAL START-UPS = 109
TOTAL ACCUMULATIVE CRU V
OPERATING HOURS 14,727.1

FIGURE 22

and CRU V-5A demonstrated startup capability with the CRU's preheated to 400°F. Again no detrimental effects were noted. CRU V-2C completed one startup at 200°F preheat level. During this startup, thrust bearing scoring occurred due to high turbine thrust loads which developed during the spinup period. The spiral groove thrust bearing, being primarily hydrodynamic in load capacity, was not able to support this load at the reduced speed. To correct this problem a hydrostatic thrust bearing which has adequate load capacity at zero speed was developed and is now in use in all CRU V's. These tests have demonstrated the capability of the CRU to startup repeatedly and reliably with the CRU housing heated to 600°F. Thus, the present startup requirements have been fulfilled.

d. Shock and Vibration Testing

Shock and vibration testing was conducted in order to verify the ability of the CRU to withstand the loads encountered during a typical launch. During launch the CRU would not be operating; therefore, the vibration and shock tests were conducted without lubricant flow to the CRU bearings. CRU V-4A was the first unit to undergo these tests and was tested only for the vibration portion of the program. CRU V-2D received a 100 hour checkout test at design point, then was shock and vibration tested at specification levels, and finally received an additional 100 hour checkout test. Neither of these units experienced any adverse effects which could be related to shock or vibration. The maximum vibration level input to the CRU was 15 g's and the maximum shock load applied was 20 g's.

Thus, the ability of the CRU to withstand a typical launch environment has been demonstrated.

2. Component Performance

During the CRU V tests, sufficient instrumentation was incorporated to allow monitoring of the performance of each of the components. The data obtained from this instrumentation combined with the pre-test and post-test calibration of the component allowed for accurate evaluation of the component performance. A summary of the performance of

each of the components appears in the following paragraphs.

a. Turbine

Extensive mapping of the turbine performance has been conducted on several CRU V's. Some differences in turbine performance have been noted from one unit to another. These differences are due to variations in first-stage nozzle size. These variations are caused by manufacturing tolerances and can be reduced by closer control of the assembled dimensions. Variations in nozzle size have resulted in variations in turbine efficiency from 49 to 58%. Demonstrated design point turbine efficiency of a nominal CRU V is 54%.

The performance of a typical CRU V turbine is presented in Figure 23 and Figure 24.

The data presented is from CRU V-4A which has a first-stage nozzle area approximately equal to the design nozzle area. Figure 23 indicates the variation in turbine efficiency, gross electrical output power, and shaft power as a function of shaft speed. The characteristics are typical of what is expected at design supply pressure for a nominal unit. Figure 23 also indicates the variation in turbine efficiency and gross electrical power as a function of turbine inlet temperature. Figure 24 presents the turbine shaft power output and gross electrical output as a function of turbine supply pressure and also indicates turbine efficiency and gross electrical output power as a function of turbine discharge pressure. A summary of these curves indicates the following:

1. CRU V-4A turbine efficiency at design point was 54%.
2. Electrical output power varies approximately 50 watts per psi of turbine supply pressure.
3. Electrical output power varies approximately 160 watts per psi of turbine discharge pressure.
4. Electrical output power varies approximately 130 watts per 100°F of turbine inlet temperature.
5. Gross electrical output power varies approximately 90 watts per thousand rpm over the range of 31,000 to 40,000 rpm.
6. Turbine efficiency is nearly constant over the ranges of speed, inlet

FIGURE 23

FIGURE 24

temperature, and discharge pressure indicated in the figure.

Endurance testing has resulted in insignificant erosion to the CRU V turbine nozzles or wheels. Figure 25 is a photograph of the CRU V-1B turbine second-stage wheel which had accumulated 5,738 hours of operation. As can be seen from this figure the edges of the blades are still sharp and show no signs of erosion. The turbine indicates excellent potential capability for 10,000 hours of operation.

FIGURE 25

b. Alternator Performance

Electrical - The alternator forms a part of the electrical subsystem indicated in Figure 26. As can be seen from this figure, power passes from the alternator, through the series capacitors, and into the useful load. Any excess power not required by the useful load is absorbed by the speed control and the parasitic load bank. In this manner the speed control maintains the shaft speed constant even though variations occur in useful load, input energy to the turbine, or shaft losses. The purpose of the capacitors in the system is to prevent demagnetization of the alternator rotor in the event of a short circuit in the electrical subsystem.

Typical alternator performance characteristics are indicated in Figure 27. These data were obtained from alternator component tests, and verification in a CRU has been obtained within the range of CRU operating conditions. As can be seen from this figure, the alternator is capable of supplying 9 KW, however, the maximum power drawn from the alternator in a CRU test was 5.6 KW. As can be seen from the curve of load voltage versus gross output power, alternator voltage regulation is approximately $\pm 15\%$ at 4.1 KW, with 40 microfarads of series capacitance per phase. Voltage regulation is reduced with higher values of capacitance; however, the short circuit stability of the alternator is also reduced. Normal CRU V operation utilizes 40 microfarads capacitance in order to provide the maximum in short circuit protection. As can be seen from the curve of alternator efficiency versus gross power output, CRU efficiency at 4.1 KW is 89 to 91% with the 40 microfarads capacitance. Harmonic distortion of the alternator output is less than 7%. The maximum number of operating hours accumulated on an alternator stator in a CRU V is 4,757.9 hours. This time was accumulated in CRU V-3, -3B, -3C and -3D.

Bore Seal - The present method of preventing mercury damage to the silver windings in the stator is to provide a ceramic coating at the ID of the stator and then to evacuate the internal section of the stator to a pressure below the mercury saturation pressure for the coolest part of the stator. In this manner mercury vapor which might enter the stator through minute cracks in the ceramic passes through the stator and is condensed

ELECTRICAL SYSTEM SCHEMATIC

FIGURE 26

CRU V ALTERNATOR VOLTAGE REGULATION CHARACTERISTICS

TYPICAL ALTERNATOR EFFICIENCY CHARACTERISTICS

FIGURE 27

in an external condensing trap. Thus, mercury is not allowed to condense or collect in the stator where it might attack the silver windings or cause a short circuit between the windings. This technique has been shown to be feasible in both component and CRU tests. As shown in the insert in Figure 28, the stator wire resistance change at low stator evacuation pressure is less than 1.5% for 2500 hours. Normal stator operation in test is with pressures of less than 2 Torr. Thus, it can be seen that the mercury vapor should have little effect on the alternator electrical losses and efficiency. Several CRU V's have been operated with the stator evacuated and a total of 4,583 hours has been accumulated. The longest time accumulated on a single evacuated stator was 2500 hours on CRU V-3D. In all of these series of tests, there have been no malfunctions caused by mercury in the stator. Figure 28 shows also the CRU V test experience for post-test bore seal leak rates. As can be seen from this curve the nominal leak rate would result in approximately 10^{-3} pounds of mercury leaving the stator in one year. Included in this test experience are several units which utilized a glass bore coating which has been made obsolete by the ceramic coating. Finally, included in this test experience are several alternator stators in which the internal portion of the stator was pressurized with argon above atmospheric pressure level.

In only one alternator stator mercury condensation caused a short circuit resulting in a demagnetized rotor. This was during the reduced preheat startup testing of CRU V-2C prior to internal evacuation of the stator. Disassembly of the stator revealed that mercury had condensed in the alternator connector which is the coolest portion of the stator. Thus, the present technique of evacuating the stator below saturation pressure prevents such condensation. Recent startup testing has demonstrated the ability of the stator to complete a large number of injection-type startups. On CRU V-1C 39 total starts were completed including 25 injection type startups with the CRU preheated to 600°F.

Thus, the ability of the alternator to meet the present requirements of preheated startup and 2500 hours of operation has been demonstrated. The alternator appears to offer no restrictions for 10,000 hours operation.

BORE SEAL LEAKAGE & EVACUATED STATOR TEST EXPERIENCE

FIGURE 28

TYPICAL PERFORMANCE Hg PUMP

FIGURE 29

c. Mercury Pump Performance

CRU V has utilized two different mercury pump configurations. The initial pump has a five vane impeller designed for 45 lb/min, and a maximum capacity of about 50 lb/min. The present pump contains a six vane impeller providing 60 lb/min at a slightly higher discharge pressure. Typical performance of both pumps is indicated in Figure 29. While the six vane design has a flow capacity of 60 lbs/min, normal operating region is approximately 45 lbs/min. As indicated in Figure 29, the six vane design requires only 30 watts more shaft power than the five vane design at 45 lbs/min, thus only a small penalty was paid to obtain the additional performance margin. Figure 30 indicates the performance of the mercury pump as a function of suction pressure. As can be seen from the curve, reducing inlet pressure from 6 to 4 psia results in only 20 psia decrease in output pressure. These pumps have operated for short duration with inlet pressures as low as 3.5 psia.

The maximum accumulated operating time on a five vane impeller was 5,738 hours which was obtained in CRU V-1, -1A and -1B. Figure 31 is a photograph of this impeller after test. While some erosion of the inlet part of the pump impeller blades is evident, pump performance at design inlet pressure was not affected. The maximum accumulated time on a six vaned impeller was 1277 hours which was obtained in CRU V-5B. Figure 32 is a photograph of this impeller after test and shows insignificant cavitation damage. Thus, mercury pump performance has demonstrated 2500 hour capability and indicates good potential for 10,000 hour capability.

d. Bearings

Journal Bearings - The three-pad mercury lubricated journal bearings in CRU V are a hybrid bearing type. That is, a portion of the bearing's load capacity is supplied by hydrostatic pressures and thus is independent of speed, and a portion of the load capacity is developed hydrodynamically as a result of speed. Thus, the shaft is floated in a film of mercury during startup and shutdown by the hydrostatic pressures and still develops much higher load capacity at design speed where it is needed to support the higher loads. Because of this characteristic, the bearings have demonstrated repeated startup and

CRU V MERCURY PUMP CAVITATING PERFORMANCE

FIGURE 30

IMPELLER FRONT FACE, CRU V-1B
AFTER 5738.5 HOURS

FIGURE 31

IMPELLER FRONT FACE, CRU V-5B
AFTER 1277.4 HOURS

FIGURE 32

shutdown capability without wear and have demonstrated adequate margin above applied loads at design speed. Figure 33 presents the bearing load margins as well as a curve of Nominal Bearing Strength vs Minimum Film Thickness, hm , and Eccentricity Ratio. At the indicated minimum film of .0002 inch which has been established as the smallest safe operating film, bearing reliability is .958.

The 2500 hour endurance capability of the three-pad bearing has been demonstrated in three CRU V's and over 2000 hours was demonstrated in a fourth CRU V. The maximum time accumulated on a single CRU V bearing was 5738 hours on the CRU V-1, -1A and -1B alternator bearing. In general, CRU V alternator bearings have shown minor effects of endurance operation while the turbine bearings have suffered cavitation erosion damage. While there are several possible causes for this, one significant difference between the alternator and the turbine bearings is the assembled clearance. The turbine bearing clearance has been adjusted in CRU V-3D and CRU V-1C to be consistent with the alternator bearing. This change proved beneficial in CRU V-3D as the turbine bearing suffered little or no cavitation damage after 2500 hours of operation. CRU V-1C has not yet been disassembled. While the results are encouraging, further tests are required to provide verification. In any event, the three pad journal bearings have demonstrated capability for 2500 hours of operation and indicate feasibility for 10,000 hours.

Thrust Bearing - Initial CRU V's utilized a hydrodynamic (spiral groove type) thrust bearing which depended upon speed to develop load capacity. During the reduced preheat startup tests of CRU V-2C it was determined that at preheat levels lower than 400°F, the thrust forces during spinup exceeded the bearing load capacity and bearing scoring resulted. To eliminate this condition, the hydrostatic thrust bearing was developed which did not require speed to develop load capacity. The high thrust forces in the CRU were caused primarily by choking of the vapor flow across the second-stage turbine wheel. At the low housing temperatures during a reduced preheat startup, the saturation pressure of mercury vapor is extremely low and thus the housing pressure is extremely low. As mercury vapor flow leaves the second-stage nozzle and passes through the second-stage wheel, it tries to expand to the low housing pressure. The volume of the vapor increases,

producing a pressure loss across the second-stage wheel which results in choking and produces the high thrust forces. As the housing temperature increases due to condensation of the mercury vapor, the housing pressure increases, the flow through the wheel unchokes, and the thrust forces reduce to the steady-state values. The maximum thrust force anticipated during a startup with the CRU housing initially at room temperature is 38 lbs, while the hydrostatic thrust bearing has a load capacity of 45 lbs. Steady-state thrust forces are approximately 3 lbs. Thus, it can be seen that the hydrostatic thrust bearing has adequate load capacity for both conditions. The hydrostatic thrust bearing has been utilized in CRU V's that have accumulated 4,802.9 hours of operation. One hydrostatic thrust bearing operated for 2500 hours in CRU V-3D and another thrust bearing accumulated 39 startups in CRU V-1C including 25 injection type system startups with the CRU preheated to 600°F. Neither of these bearings exhibited any performance degradation during test. Disassembly of CRU V-3D indicated the thrust bearing was in excellent condition. Disassembly of CRU V-1C has not been completed yet, therefore, post-test inspection results are not available. However, the hydrostatic thrust bearing has demonstrated repeatable preheated injection type system startup capability, has demonstrated at least 2500 hours endurance capability, and appears to offer no restriction for 10,000 hours operation.

BEARING LOAD MARGINS

FIGURE 33

BEARING LUBRICANT STATIC SEALS

ACCUMULATED TEST TIME ON INTERIM DESIGN-14,100 HOURS

FIGURE 34

3. Problems and Solutions

During the development of CRU V a normal amount of development problems were encountered. Most of these problems had ready solutions requiring only a simple change in dimension, clearance, or configuration. A few of the problems required more extensive analyses and more difficult solutions. These problems are discussed below.

a. Lube Line Seals

Initial CRU V's utilized a metallic chevron type seal to seal the interface between the bearing assemblies and the lube supply line. It was found in CRU testing that the seal suffered fractures which allowed leakage of the lube flow into the CRU housing. Subsequent investigations revealed that the chevron seal material which was 17-4PH was in a highly notch-sensitive condition. Also, it was revealed that closer control was required of the seal tolerances and the groove alignment. A number of bench-type tests were conducted which indicated the allowable dimensional variation of the seal and seal cavity which would still permit acceptable sealing capability. These tests were by necessity short-term type. Subsequent tests in CRU's indicated a short-term sealing capability similar to that discovered during the bench-type test. However, long-term operation appeared to present a limit since even the smallest leak tended to cause erosion of the seal which resulted in increased leakage with time. The interim solution to this problem has been to utilize silicon rubber "O" rings in place of the chevron seals. These "O" rings are entirely satisfactory for present operation. However, in flight-type operation some restriction would result. With the "O" rings it is necessary to maintain the temperature below 550° for long-term operation. In the present CRU this is accomplished by maintaining bearing lube flow at 350° during CRU preheat and following CRU shutdown till the housing has cooled sufficiently. During normal steady-state operation no problem is encountered as the temperature limits of the "O" ring are not exceeded.

The ultimate solution to the problem is in attaching permanent lube supply lines directly to the bearing assembly. A design for accomplishing this is being pursued now. The essential features of the design are indicated in Figure 34. The lube supply line will be

shrink fitted or welded to the bearing assembly and expansion joints will be incorporated to provide for the differential expansion between the lube supply line and the CRU housing. In this manner high loads resulting from thermal differential expansion between the cradle and the tubes will not be transmitted into the bearing assembly.

b. First-Stage Turbine Wheel

Three CRU V's suffered radial fractures of the first-stage turbine wheel. The fractures occurred after the following periods of testing and did not appear to be time related: one occurred after 2,557 hours; the second after 438 hours; and the third after 176 hours. Investigations indicated that the fractures occurred during shutdown of the CRU when the shrink fitted turbine wheel cooled more rapidly than the shaft, thus increasing the stresses in the wheel but not beyond the yield point. A study of the wheel material, Nivco 10, indicated that this material in the composition utilized was highly notch-sensitive; a fact discovered by the material manufacturer after the fabrication of CRU V. A dual solution for this problem was incorporated. The turbine wheel to shaft shrink fit assembly was replaced with a clearance fit and pinned turbine wheel assembly which reduced the wheel stresses. A locking washer retained the parts in position. The notch sensitivity of the Nivco 10 material was reduced by increasing the zirconium content of the material. No turbine wheel fractures have occurred in almost 8300 hours of accumulated operation with the new design. The original design and improved design are indicated in Figure 35.

c. Bearing Cavitation Erosion

Several CRU V's have encountered cavitation erosion damage to the journal bearings. Damage has occurred in the turbine bearing whereas the alternator bearing is relatively unaffected. The amount of damage does not directly correlate to the number of hours of operation of the CRU. The damage to the three-pad turbine bearings has not terminated a CRU test and does not appear to present a limit for 2500 hour operation. However, it is thought that some improvement is required for 10,000 hours of operation. An alternate bearing configuration, a compensated three-sector bearing, was evaluated in several CRU V's but no improvement in cavitation erosion resistance was found. In addition, the

FIRST STAGE TURBINE WHEEL

ORIGINAL DESIGN

PROBLEM

RADIAL FRACTURE OF TURBINE WHEEL CAUSED BY HIGH STRESS LEVEL IN SHRINK FIT MOUNTING OF WHEEL.

IMPROVED DESIGN

SOLUTION

WHEEL MOUNTED ON SHAFT WITH 3 HOLLOW PINS AND LOCKED IN PLACE WITH LOCK WASHER.

ACCUMULATED TEST TIME ON IMPROVED DESIGN - 8300 HOURS

FIGURE 35

three-sector bearing had lower load capacity. Thus, it was abandoned in favor of continued development of the three-pad bearing. Analysis of this phenomena indicates several factors such as lubricant temperature, bearing clearance, shaft loading and bearing supply pressure could contribute to the problem. An adjustment in turbine bearing clearance which makes the clearance consistent with the alternator bearing clearance has been made in recent CRU V's; however, this adjustment has not been fully evaluated. A component test program is in process to determine the effects of each of these variables on bearing cavitation. It is anticipated that sufficient information will be obtained from this program by the end of this fiscal year to provide a solution to the problem if the adjustment in clearance already made in recent CRU V's does not provide the entire solution.

d. Alternator Bore Seal

In the present alternator stator design the stator laminations are assembled into an axial stack and compressive loaded to force the stack to act as a solid member. The ID or bore of the lamination is then coated with a glass ceramic to seal any spaces which might exist between the laminations so that mercury vapor will not enter the stator. It has been shown in component tests that if the lamination stack acts as a solid member, the glass ceramic is capable of providing an effective seal. However, in CRU V tests some leakage through the bore occurs in almost every case. While this has presented no problem in CRU operation, it is undesirable from a reliability standpoint. Present CRU test results with the stator internal section evacuated indicate that this concept is acceptable for the present startup and endurance requirements. Nevertheless, in order to improve system reliability, a more nearly perfect bore seal is being sought. A new design alternator stator is being evaluated at this time. The new stator provides better control of the compressive loading on the alternator stack and reduces the thermal gradients which occur within the stator.

e. Mercury Corrosion Products

The effect of mercury corrosion products has been seen primarily in the CRU journal bearings where deposition of the products in the bearing compensating orifices and annular

clearances has caused reduction in bearing flows. The effect has been more pronounced in new test rigs or in test rigs with new boilers and/or condensers. In test rigs that have operated for several hundred hours only slight deposition has occurred on the CRU bearings. The bearings have demonstrated a tolerance to deposition even during the initial periods of operation in a new test rig. In some cases, shutdown of the test was necessary so that the system mercury could be replaced with clean mercury. Then, the test was restarted and usually no additional deposition effects were noted. In some cases with continued operation, the corrosion product deposition was removed from the bearings. This experience correlates with corrosion tests conducted at TRW which indicated that the rate of boiler corrosion product generation was high with a new boiler but decreased rapidly after a few hundred hours of boiler operation and then stabilized at a low level after 800 hours of operation.

While corrosion product deposition in the bearings has not prevented completion of any of the major CRU test objectives, nor caused a test termination, a better understanding of this phenomena is desired. To accomplish this, a program has been initiated to determine the effectiveness of separators and filters in removing these corrosion products from the mercury. One design separator and filter has been in use during recent CRU tests and will be evaluated shortly.

IV. SUMMARY AND RECOMMENDATIONS

A. PRESENT ACCOMPLISHMENTS

In the two year test program conducted on CRU V to date, many significant achievements have been demonstrated, such as

1. Over 14,700 hours total operating time have been accumulated on all CRU V's.
2. Four different CRU V's accumulated over 2,000 hours each.
3. Several of the major components of one CRU V accumulated 5700 hours of operation.
4. Peak output power of 5.6 KW has been demonstrated.
5. Machine efficiency exceeds 43%.
6. Repeated fully preheated injection type startups have been demonstrated.
7. Several injection startups were accomplished at preheat levels as low as 400°F. One CRU started at 200°F.
8. Tolerance to shock and vibration environment has been confirmed.
9. Operation has extended over an extreme range of off-design conditions.

In addition, in support to the CRU program many significant technological advancements were made in the component activities. Some of these are as follows:

1. Understanding and development of analytical techniques in the area of turbulent flow lubrication.
2. Development of erosion-free, compact high-speed turbines with efficiencies ranging from 55 to 60%.
3. Development of compact jet-centrifugal pump combinations operating at very high suction specific speeds.
4. Development of reliable, high-efficiency, high-temperature alternators.

These achievements have provided evidence that the current CRU V is capable of meeting the present 2500 hour objective. Also, the long-term tests have provided assurance and confidence that the CRU V concept can be developed to achieve the long-term objective of 1 year (10,000 hours).

B. PRESENT PLANS

The CRU V accomplishments to date have demonstrated the feasibility of the CRU operating through a typical cycle of a flight type system. However, the degree of reliability to which the CRU could complete this cycle has not been demonstrated. Primarily responsible for this lack of demonstration has been the variety and types of tests which were conducted and the design changes which were continuously incorporated into the CRU as a result of these tests. Since the objectives for different CRU's have varied from parametric to startup to shock and vibration to endurance, it is difficult to compare the results of a unit run on endurance testing to the results of a unit run on startup testing. There has not been an effort to test CRU's under the same type of test conditions to establish inherent reliability. Design changes were incorporated into the CRU as deficiencies were discovered and corrected. Thus, CRU's in the early part of the program have had slightly different design features than the present CRU V. Reliability of the CRU V design, although thought to be high cannot be accurately calculated. In order to establish reliability of the existing design a series of controlled tests are planned on two CRU's of the same configuration. In this manner, non-correlating or conflicting results will not be obtained from extrapolated test data obtained from CRU's of slightly different configurations. Neither will it be necessary to make a series of assumptions nor to exclude certain results to establish actual reliability. These tests will consist of six preheated injection startup and shutdown cycles and 2500 hours of operation.

In addition to the reliability evaluation of the present CRU V design, a parallel program of reliability improvement is being conducted. This program consists of component and CRU analyses and tests to determine the CRU or component areas which can be significantly improved and thereby move toward the achievement of long-term reliability. The reliability improvement program is necessary to discover and initiate corrective action on problems which could be encountered in longer periods of endurance operation or in further excursions into off-design operating conditions. The program will provide valuable information for identifying future CRU design requirements as well as in

improving the margin in the present design.

C. RECOMMENDED FUTURE ACTIVITY

The one requirement of the Space Dynamic Mercury Rankine System that has not as yet been demonstrated is operation in zero gravity. During CRU V testing the tests were conducted in a manner in which gravity effects were minimized. However, it was not possible to eliminate these effects on all components simultaneously.

Testing of CRU V's to date has demonstrated the potential of this unit to successfully complete a flight test. Such a test would be a benefit not only to the CRU V turbo-alternator, but would benefit all Dynamic Mercury Rankine Systems by establishing the feasibility of system operation in space.

It is recommended that an orbital flight test of CRU V be conducted as soon as possible.

DESIGN AND DEVELOPMENT OF A TURBINE-ALTERNATOR ASSEMBLY*
FOR THE SNAP-8 SYSTEM

By

M. G. Cherry and G. Oiye
Von Karman Center
Aerojet-General Corporation
Azusa, California

ABSTRACT

The need for a reliable turbine-alternator assembly arose when the potential methods of electrical power generation were considered for the SNAP-8 system. The turbine-alternator for this system was designed to produce 51.6 kw at the alternator terminals at an overall efficiency of 49.5% and to do so reliably and unattended for at least 10,000 hours. Superheated mercury vapor is the working fluid.

The turbine-alternator assembly (TAA) consists of two separate but integrated units: the turbine assembly, and the alternator assembly. Polyphenyl ether (mix-4P3E) is supplied to cool the alternator stator and lubricate the bearings. When two fluids such as mercury vapor and polyphenyl ether are in contact with the same shaft, there is a distinct possibility that they will intermix. To prevent this in the SNAP-8 turbine-alternator assembly, a low-leakage seal to space was designed. This seal, situated between the turbine and support bearings, consists of a visco pump, a slinger pump, and a molecular pump in series.

The complete turbine alternator assembly has been designed within the "state of the art" to ensure high reliability. Where conflicts of reliability and performance occurred, unless the performance gains were substantial, reliability overruled.

Two units have been tested. The first unit was tested using nitrogen gas as the working fluid for the turbine. The purpose of these tests was primarily to check the functional operation of the unit at the design speed, and to obtain measurements of the bearing and slinger losses. The second unit has recently completed some 830 hr of operation at design and off-design conditions in mercury vapor.

*The information appearing in this paper is based on work conducted under National Aeronautics and Space Administration Contract NAS 5-417.

The basic problems to be resolved as testing continues will be stage performance, seal-to-space leakage, mechanical integrity of the basic design for 10,000 hr and the effects of liquid carryover on turbine performance. Basic design problems which have been evident from the development testing to date have been notch sensitivity of the material used for the nozzle diaphragms and retention method for the interstage and thrust balance labyrinth seals. A 1,000 hr endurance run is planned for the next unit which is to be tested later this year.

I. INTRODUCTION

The SNAP-8 turbine-alternator assembly (TAA), Figure 1, converts the thermal energy of the superheated mercury vapor working fluid into electrical energy for use within the SNAP-8 system. Two major assemblies comprise the TAA: an alternator assembly (AA) and turbine assembly (TA). These two assemblies are bolted together with the turbine rotor shaft directly connected to the alternator rotor shaft with a splined quill shaft.

The location of the TAA within the SNAP-8 system is shown in Figure 2.

The TAA is designed to produce 51.6 kw of usable power measured at the alternator terminals at an overall efficiency of 49.4%. The system conditions to which the unit is designed are as follows:

SYSTEM DESIGN POINT CONDITIONS

	<u>Value</u>
Mercury vapor flow rate (lb/hr)	11,200
Speed (rpm)	12,000
Inlet pressure (psia)	265
Inlet temperature ($^{\circ}$ F)	1,250
Exit pressure (psia)	16.5
Exit temperature ($^{\circ}$ F)	685.0
Overall efficiency (%)	49.5
Output (kw)	51.6
Power factor	0.75
Current (amps)	223
Voltage (RMS, 3-ph)	120/208
Frequency (cps)	400

The TAA package is designed to operate unattended, continuously for 10,000 hours. Extremely high mechanical, metallurgical, and electrical reliability were required. In support of this reliability all components within the TAA were to be designed within the state-of-the-art, wherever possible.

II. COMPONENT STATUS

A. TURBINE ASSEMBLY (TA)

The turbine assembly (Figure 3) is an entirely separate unit from the alternator assembly to allow for individual development. It comprises the turbine, its shaft and support bearings, with associated slingers and the seal to space.

1. Turbine

The turbine design included considerations of all the conventional turbine losses for nozzle and rotor rows, including the effects associated with partial admission, interstage leakage and the effects of moisture. Thermodynamically, the turbine was designed according to the equilibrium characteristics of the Mollier diagram. No supersaturation or liquid carryover was assumed. Pertinent dimensions and design parameters are shown in Tables 1 and 2.

The cantilevered turbine assembly, Figure 3, is a four-stage, impulse-type turbine with the first and second stages operating with partial admission (38%) and the third and fourth stages with full admission. The nozzle areas of the two partial-admission stages are halved and spaced 180 degrees apart to avoid unbalanced torques on the bearings and to better distribute the heat input into the turbine case.

The turbine case and inlet manifold housing are suspended by four arms extending from the bearing housing. The concentric location of the turbine case under thermal and mechanical loads is maintained by the four "cold-frame" support arms that are part of the bearing housing. These arms are tangentially and axially stiff, but radially flexible to accommodate thermal growth. Any tendency for the case to be eccentric is well constrained.

Since the turbine assembly must operate for 10,000 hours at high temperatures, one of the design concepts was to provide large rotor axial and blade tip clearances

so that thermal distortions and minor creep effects could be accommodated. As a result, the turbine-blade tip clearances were set at a nominal 0.040 in. for all four stages with the loss of turbine performance from this action taken into account. Axial clearances between nozzles and wheels were set at a nominal 0.100 in. A larger axial clearance of 0.50 in. was used, however, between the second-stage wheel and the full-admission third-stage nozzle. This was done to minimize the transitional losses that occur in passing from a partial to a full admission stage.

Figure 4 shows the major constituents of the turbine assembly while Figure 5 shows a complete turbine-alternator assembly, lacking only the lubricant-coolant plumbing.

Preloaded angular-contact ball bearings are used with the overhung turbine and a space-seal system is interposed between the turbine exhaust end and the outboard turbine bearing.

To reduce ball-bearing axial loads, the thrust load due to unbalanced pressure forces on the shaft is counteracted by a balance labyrinth seal piston at the extreme overhung position.

2. Bearings, Slingers and Lubrication

The turbine assembly is supported by two angular-contact ball bearings, spring loaded in a back-to-back arrangement. The bearings are a 208 size (40 mm bore) angular-contact ball bearing (ABEC, Class 7) that contains the maximum compliment of balls. The ball retainer is a lightweight, one-piece outer-land-guided type made of iron silicon bronze. Rings and balls are made of triple vacuum melt consumable electrode (CEVM) M-50 tool steel. The bearing material selection was made primarily for the 10,000 hours life requirements at 99.5% reliability.

The bearings are lubricated by multiple-jet injection of an organic fluid (mix-4P3E) polyphenyl ether. Scavenging dynamic seals are used on both sides of each bearing to provide nonflooded bearing operation with the bearing cavity at the vapor pressure of the lubricant.

3. Seals to Space

To permit the use of the organic fluid for bearing lubrication, it is necessary to prevent mixing the fluid and mercury where they occupy adjacent regions on the same shaft. Inter-contamination of oil and mercury is avoided by venting a section of the rotating shaft to space and permitting a small controlled leakage of each fluid to the space vent cavity.

The seal combination developed for these requirements consists of a visco pump, slinger pump, and molecular pump in series located between the fourth-stage wheel and the adjacent bearings. The visco pump and dynamic slinger elements serve to develop stable liquid-vapor interfaces for both mercury and oil. The visco pump is surrounded by a (mix-4P3E) heat exchanger which assures an adequate mercury temperature at the interface. The molecular pump provides the barrier to restrict the leakage to space of mercury vapors in the molecular regime which evaporate from the interface.

The seal configuration is the result of a separate development program where extensive theoretical and experimental work was done. Testing during this program revealed that leakage rates of less than one pound per year can be expected. The design and development of this seal to space are well covered in References 1 and 2.

4. Startup Seal

The contact seals in the space vent cavity prevent the loss of liquid during the startup and shutdown phases of operation when the respective dynamic seals are generally inoperative. The contact seals are disengaged by means of an auxiliary lift-off device after startup and prior to shutdown during ground test operations.

5. Materials

Material selections (Table 3) were based on cavitation-erosion resistance to wet mercury vapor, creep data, and thermal expansion data. All areas subjected to mercury vapor, wet or dry, at high velocities are made from Haynes Stellite 6B: turbine wheels, nozzle diaphragm assemblies, nozzle inserts for the inlet manifold assembly labyrinth seals and the turbine bolt. The inlet manifold assembly is forged 316 stainless steel.

Material for the turbine case and bearing housing is cast Croloy 9M with the thin exhaust duct made from 410 stainless steel. Shaft material is 4340 steel. The ABEC 208 Class 7 ball bearings are made from triple CEVM M-50 steel, as noted previously.

B. ALTERNATOR ASSEMBLY (AA)

The General Electric Company, Erie, Pennsylvania, was selected as Aerojet's subcontractor to develop an alternator for the SNAP-8 system.

State-of-the-art methods were the criteria for selection of a design capable of high reliability, long life and good performance. The homopolar inductor design was selected primarily for its simplicity and inherent reliability.

The alternator assembly is a hermetically sealed 3-phase, 120/208 volt, 400 cps, radial air gap, homopolar inductor (Figures 1 and 6). It features a brushless solid rotor straddle mounted on angular-contact ball bearings.

Dynamic seals are provided to scavenge the rotor-stator cavity and outboard sides of the ball bearings.

The lubricating and cooling fluid is the same organic polyphenyl ether (mix-4P3E) used in the turbine. It is distributed to the bearings in a mist form and to the stator cooling jacket in liquid form. Heat from the rotor is removed by radiation to the surroundings and by conduction to the inboard slinger seals, where it is rejected to the lubricating oil.

The electrical insulation system is primarily a Du Pont aromatic polyimide (ML) system coupled with a General Electric Company epoxy resin compound "NOVALAC."

The terminals are hermetically sealed and feature universal orientation capability.

Four preprototype units and three prototype units have been built and tested with results shown in Figures 6 and 7.

III. DEVELOPMENT TEST PROGRAM

A. TEST PROGRESS

The development test program for the turbine-alternator assembly (TAA) included initial testing using gaseous nitrogen as turbine working fluid, then using mercury vapor to obtain performance data, and finally endurance testing.

Operation of a TAA using nitrogen provided information regarding the lubricant and coolant system, allowed measurement of bearing and slinger losses and provided verification of the mechanical integrity of the unit at speeds and power levels in excess of design requirements. In addition, the general handling characteristics of the TAA and its associated subsystems could be evaluated and adjustments made in a comparatively safe environment.

The first TAA assembled was tested at a nitrogen test facility. Measurements of bearing and dynamic seal losses are shown in Figure 8. At design conditions, the overall bearing and slinger losses for the TAA were 3.95 kw. Speeds up to 125% (15,000 rpm) of design speed, and power levels of 52.53 kw were obtained smoothly and no undue increase in vibration levels was experienced.

The bearing and dynamic-seal losses were calculated from the measured temperature rise and flow rate for each particular bearing and dynamic seal combination.

The values of bearing and slinger losses obtained (Figure 8) were confirmed by measurements made during separate bearing and seal development tests.

The second unit assembled was tested in the mercury test facility. The basic objectives of these tests, apart from functionally checking the operation of a SNAP-8 TAA at system design conditions, were to obtain performance data with regard to the turbine and the unit as a whole. In addition, data were to be obtained for correlation with a theoretical thermal map and also verification of the bearing and slinger losses, and vibrational characteristics obtained during the nitrogen tests.

During the initial portion of the testing period, the turbine was subjected to mercury vapor qualities estimated to be between 55% and 85%. Later tests produced the desired superheat conditions. The results shown in Figures 8 and 9 are for the latter conditions.

The alternator, the performance of which was known from tests made at General Electric, was essentially used as a dynamometer for obtaining turbine performance. The turbine output power is defined as the sum of the alternator input power and the other shaft loads, such as bearing and dynamic seal losses. Thus, the turbine aerothermodynamic efficiency is defined as;

$$\eta_T = \frac{\text{output power}}{\text{available power}}$$

$$= \frac{\left[\frac{(kw)_A}{\eta_{alt}} + (kw)_{B\&S} \right]}{\left[\dot{W} (h_o - h_{ex}) \times .000292 \right]}$$

where

.000292 = conversion constant Btu/hr to kw

$(kw)_A$ = alternator output power in kw measured at the alternator terminals

$(kw)_{B\&S}$ = turbine assembly bearing and dynamic seal losses

η_{alt} = alternator overall efficiency

$\dot{W}$ = mercury vapor flow rate, lb/hr

h_o = total inlet enthalpy, Btu/lb

h_{ex} = static exit enthalpy, Btu/lb

The measured turbine aerothermodynamic efficiency (54.7%) was low due to the comparatively large amount of liquid carryover (6%). Correcting the turbine performance to zero liquid carryover (the condition for which it was originally designed) yields an aerothermodynamic efficiency of 58.5% at the design condition. This value compares very well with the predicted 58.15% for this particular turbine. It is to be understood that the difference between the predicted result and the design nominal efficiency (60%) is due to a number of conditions that were incorporated in the interest of design conservatism and reliability.

The thrust and interstage seals were originally considered fixed and concentric. The free-floating attribute, which was finally incorporated, allowed the seals to become eccentric and thereby increase the seal leakages considerably. In addition, clearance spaces were built in over the nozzle vane tips to ensure that the nozzle vanes would not yield in compression during a transient condition occurring in a startup.

The measured TAA power output (Figure 10) at the design vapor flow rate, and while the turbine was subjected to the liquid carryover, was 50.5 kw at an overall efficiency of 46.9%, compared with the design value of 51.6 kw at 49.4%. The fact that this power output has been achieved in spite of lower-than-design turbine efficiency, is due to the bearing and slinger losses being less than the original design estimates, and the efficiency of this alternator being slightly higher than that of the alternator performance estimates. It is to be noted that operation without the liquid carryover penalty would indicate that we should expect a TAA power output of approximately 53 kw with an overall efficiency of 49.6% at the design conditions. The steady-state thermal map (Figure 11) indicated good agreement with the estimates.

After the performance tests were completed, the turbine-alternator assembly and the mercury vapor facility were used for tests of the SNAP-8 voltage regulator and speed control units. Following the electrical control tests, testing was continued in order to accumulate 1,000 hours of operation. At approximately 830 hours of accumulated run time, a major failure occurred in the turbine.

B. FAILURE ANALYSIS

Disassembly revealed three separate and essentially independent mechanical design problems. The first of the problems, which incidentally was the cause of the final failure, was that the thrust and first interstage labyrinth seal retaining rings came out of their containment grooves and were subsequently destroyed.

These retaining rings were still in place in the fourth stage and had partially "walked out" of the third-stage groove. In addition, the eight labyrinth seal anti-rotation pins had all sheared and were missing.

Disassembly also revealed the progressive cracking of the first-stage nozzle weld tips at the juncture with the insert body. Cracks initiated in the heat-affected zone on the Stellite 6B-to-316 SS weld. The cracks and the movements of the nozzle insert blocks increased the effective area of the nozzle and accounted for the change in performance that was observed.

Associated with the notch sensitivity of Stellite 6B, were cracks that were observed to have propagated from the sharp corners of the nozzle diaphragm locating slots and other areas of high stress on the second stage nozzle diaphragm. The radial cracks were old and probably occurred some time prior to the major failure.

Another problem area which became evident during the tear down was in the visco pump ring, which exhibited excessive erosion-cavitation damage. On the brighter side, evidence of erosion on the rotor blade leading edges was extremely faint in spite of long periods of operation with low-quality mercury vapor.

C. MECHANICAL DESIGN IMPROVEMENTS

Straightforward engineering solutions are available for all the observed failure modes. At the time of writing several alternative solutions for each of the failure modes are in the process of evaluation.

D. SUMMARY

1. Present Accomplishments

Six turbine alternator assemblies have been fabricated and built. Two of the units have been tested; one on nitrogen, the other using mercury.

The first model of a turbine alternator assembly for the SNAP-8 system has indicated a performance on mercury vapor, at system design conditions, very close to the original design requirements despite a turbine efficiency which is slightly below design. Should it be desirable, improvements are readily available to increase the turbine efficiency by minor modifications.

The turbine alternator assembly has demonstrated a high degree of ruggedness. It has digested eight stainless steel pins, parts of two labyrinth seal retaining rings, debris from the cracked nozzle insert welds and operated with cracks in the second stage nozzle diaphragm. During tear-down inspection, metal parts were found lodged between nozzle and rotor blades (but without breaking them) and evidence of old cracks were found. There was no problem with the alternator.

Most of the tests were performed with no abnormal incidents related to the bearings except during one test when the TAA oversped to 19,000 rpm. It is important to note that acceleration due to the overspeed condition occurred smoothly without any significant increase in the bearing vibration level. The overload, as a consequence of the turbine failure, did not Brinell the balls or race-ways nor was there an apparent damage of any kind. The adequate overload capacity, quality, precision, and reliability of the bearings more than likely saved the third and the fourth stages of the turbine, and the dynamic and static seals from complete destruction, and subsequent loss of evidence of the failure sequence. The alternator-assembly bearings were not removed from the unit for an examination; the alternator assembly will be inspected at the General Electric Co. The space seal worked satisfactorily. The Stellite 6B exhibited excellent erosion resistance and excellent thermal stability.

2. Remaining Problems

Specific areas which still require verification are bearing life, seal-to-space leakage, and turbine erosion to 10,000 hours and, of course, the integrity of the mechanical redesign to be incorporated in the turbine.

IV. PRESENT PLANS

Following selection and incorporation of the redesign features in the turbine, a unit will be run in mercury vapor and subjected to several starts and stops. The unit will be disassembled and inspected and, if no problem areas are evident, a second unit also incorporating these features will be installed in the complete SNAP-8 power conversion system for performance and 10,000 hour endurance tests of the complete system.

V. FUTURE PLANS

Increase in system parasitic loads has brought about increased requirements for the TAA. An output of 54.5 kw at the alternator terminals at a power factor of 0.625 will be needed in the future. Since it is expected that the redesign features to be incorporated in the turbine will provide some increase in the turbine efficiency, a modest increase in mercury vapor flow rate is all that should be required to satisfy this requirement.

Endurance tests to verify a 10,000-hour capability will be initiated as soon as possible in the complete SNAP-8 power conversion system. A continuing evaluation of design and performance to improve component efficiency will be conducted. Improvements that do not impair reliability will be incorporated when practical.

REFERENCES

1. R. L. Lessley and J. N. Hodgson, Low-Leakage Dynamic Seal to Space, ASME Paper 65-GTP-14.
2. J. N. Hodgson, Designing a Molecular Pump as a Seal-to-Space, ASME Paper 65-GPT-15.

TABLE 1

TURBINE PHYSICAL DESCRIPTION

	<u>First Stage</u>	<u>Second Stage</u>	<u>Third Stage</u>	<u>Fourth Stage</u>
<u>Nozzle</u>				
Type	Convergent	Convergent	Convergent	Convergent
Number	14	14	37	37
Height (in.)	0.24	0.33	0.25	0.47
Exit angle (degrees)	17.0	17.0	17.0	17.0
Chord (axial) (in.)	0.30	0.30	0.30	0.30
<u>Blade</u>				
Number	79	79	79	79
Height (in.)	0.323	0.464	0.372	0.656
Tip clearance (in.)	0.040	0.040	0.040	0.040
Chord (in.)	0.289	0.289	0.289	0.289
<u>General</u>				
Mean diameter (in.)	5.10	5.10	5.10	5.10
Seal diameter (in.)	2.08	1.40	1.40	1.40

TABLE 2

TURBINE AEROTHERMO DESIGN

	<u>First Stage</u>	<u>Second Stage</u>	<u>Third Stage</u>	<u>Fourth Stage</u>
<u>Nozzle</u>				
Pitch/chord ratio	0.752	0.748	0.745	0.745
Exit quality (%)	1.00	0.97	0.96	0.93
Exit Mach number	1.00	1.08	1.02	1.05
Nozzle velocity coefficient	0.943	0.953	0.940	0.943
<u>Blade</u>				
Pitch/chord ratio	0.7	0.7	0.7	0.7
Mean peripheral speed (fps)	267	267	267	267
Blade velocity coefficient	0.86	0.88	0.86	0.89
<u>General</u>				
Percentage admission	38	38	100	100
U/c_o	0.42	0.39	0.41	0.43
Stage efficiency (%)	59.7	61.0	60.3	61.5
Aerodynamic efficiency (%)	<hr/>		60.0	<hr/>
Overall efficiency (%)	<hr/>		58	<hr/>

TABLE 3

TURBINE MATERIALS SUMMARY

<u>Component</u>	<u>Material</u>	<u>Reason for Selection</u>
Turbine wheels and nozzles	Stellite 6B	1. Erosion resistance. 2. High temp. strength. 3. Hg corrosion resistance.
Inlet housing	316 SS	1. Same material as inlet pipe. 2. High temp. strength.
Turbine case	9Cr-1Mo	1. Hg corrosion resistance. 2. Oxidation resistance.
Bearing housing	9Cr-1Mo	1. Same material as turbine case. 2. Same thermal expansion as bearings. 3. Oxidation resistance.
Shaft	4340 Steel	1. Same thermal expansion as bearings. 2. Good strength and hardenability for spline.

Fig. 1 SNAP-8 Turbine-Alternator Assembly

Fig. 2 SNAP-8 Loop Schematic Showing Turbine-Alternator Assembly (Cross Hatched)

SYSTEM DESIGN DATA

TURBINE TYPE - 4 IMPULSE STAGES

MERCURY FLOW RATE - 11,200 LBS/HR

PRESSURE

INLET - 265 PSIA

OUTLET - 16.5 PSIA

TEMPERATURE

INLET - 1250°F

OUTLET - 685°F

EFFICIENCY - 60%

SPEED - 12,000 RPM

Fig. 3 SNAP-8 Turbine Assembly Showing Significant Parameters

Fig. 4 SNAP-8 Turbine-Assembly Components

Fig. 5 SNAP-8 Turbine-Alternator Assembly

SNAP-8 ALTERNATOR

PARAMETER	DESIGN POINT	TEST RESULTS
OUTPUT - KVA	80	80
POWER FACTOR	.75 - 1.0	.75 - 1.0
EFFICIENCY (OVERALL) - %	87	87.5
WINDING HOT SPOT TEMP - °C	200	190
WEIGHT - LB	330	441
EXCITATION VOLTAGE - VDC	50	43.5
EXCITATION CURRENT - AMPS	21.4	19.8
RUNNING TIME - HR (1 UNIT)	10,000	894
ACCUMULATIVE RUNNING (7 UNITS)		1324

OVERLOAD CAPABILITY

OUTPUT - KVA	87.5
POWER FACTOR	.625
EFFICIENCY - %	85
WINDING HOT SPOT TEMP - °C	194
EXCITATION VOLTAGE - VDC	51.1
EXCITATION CURRENT - AMPS	22.5

Fig. 6 SNAP-8 Alternator Design Point vs Test Results

Fig. 7 SNAP-8 Alternator Performance - Overall Efficiency vs Terminal Power

Fig. 8 SNAP-8 Turbine-Alternator Assembly Bearing and Slinger Losses

Fig. 9 SNAP-8 Turbine-
Aerothermodynamic Efficiency

Fig. 10 SNAP-8 Turbine-Alternator Assembly Performance

Fig. 11 SNAP-8 Turbine Assembly Steady-State Thermal Map

POTASSIUM TURBINE-GENERATOR DEVELOPMENT ✓

By J. H. Dannan and L. E. Chadbourne
AiResearch Manufacturing Company,
a Division of The Garrett Corporation,
Phoenix, Arizona

ABSTRACT

A description of the design of a 300-kw high-temperature potassium turbogenerator is given, including representative results obtained from tests of critical turbine and bearing components.

Of particularly significance are the bearing development tests leading to the selection of hydrodynamic pivoted-pad journal bearings selected for the initial development turbine.

The successful development of a high-temperature potassium turbogenerator is expected to provide a technology for the design and construction of high-temperature power plants satisfying special requirements for space and ground applications.

The development work conducted to date on the turbine, and bearing components of the SNAP 50/SPUR turbogenerator unit, under United States Air Force Contracts (Air Force Systems Command, Research and Technology Division, Aero Propulsion Laboratory) is summarized in this paper.

The long-range goal of the turbogenerator development is to provide a unit capable of operation at turbine temperatures in the 1800°F range. The reference design point for this unit, Figure 1, is a nominal generator output of 300 kw, turbine inlet and outlet temperatures of 1800°F and 1200°F, and a speed of 24,000 rpm. The generator output frequency is 3,200 cps. A generator coolant and bearing lubricant temperature of 600°F is utilized.

A cross section of the turbogenerator is shown on Figure 2. Saturated potassium enters the first half of the turbine, passes around the generator via the crossover ducts, and enters the last half of the turbine and discharges at approximately 86 percent quality. The selection of six stages resulted from a high efficiency requirement and from the allowable tip speed (1,000 ft/sec.) of the last stage.

A radial-gap homopolar generator is located between the turbine sections and is straddle mounted on two journal bearings.

Liquid potassium for generator coolant and bearing lubricant, is pumped into and along the shaft as shown in the figure. Part of this flow is tapped off for lubrication of the front journal bearing. The remaining flow travels through the shaft removing generator rotor heat and passes into the rear journal and thrust bearing cavity.

Hydrodynamic seals are used on the turbine rotor side of each bearing to preclude leakage of lubricant into turbine flow passages. Vapor pumping seals are incorporated on the generator rotor in order to maintain a pressure in the generator cavity that is less than the vapor pressure of potassium corresponding to the lowest temperature within the generator cavity (approximately 700°F or 0.05 psia). This pressure is required, otherwise the potassium vapor in the cavity may condense on the cooler surfaces within the cavity. Windage losses are also reduced by the low pressure.

For power plants operating in the 1800°F range, the external housing of the unit would be constructed of super alloys or refractory alloys. However, to avoid the difficulty of fabricating and testing components constructed of these alloys, the initial design uses conventional material (Austenitic stainless steels) for the external housing which will permit testing up to 1600°F. Materials selected for other components such as for the turbine rotors and stators, bearings, seals, and generator rotor and

stator can be identical with those to be used in the final refractory metal unit but in the first units will be conventional superalloys and cermets.

Bearings: As indicated earlier, the use of potassium as a lubricant eliminates the requirement for complicated seals and supporting equipment that would otherwise be required with a different lubricant. As potassium freezes at approximately 150°F, a supply temperature of at least 200°F is required. Generator design studies have shown that 600°F is a desirable temperature for the generator coolant, and as Figure 2 suggests, a generally simpler turbogenerator design evolves if the lubricant and coolant temperature are the same.

Loads imposed on the journal bearings vary from approximately 250 pounds (i.e., 125 pounds per bearing from static rotating group weight plus unbalanced rotor and electrical loads), for ground operation, down to very small loads for a space power plant.

Thrust loads are nominal as aerodynamic pressure force balancing of the rotor is used. However, in special cases, operation with the rotor axis oriented vertically is required thus imposing a load on the thrust bearing equal to the weight of the rotating group.

To satisfy the above requirements. rolling contact bearings are not considered since experience with this type of bearing running in NaK and sodium at speeds of a few thousand rpm and low temperatures discourages their use in the high speed turbogenerator being considered here. Gas-- or rather vapor bearings--appear attractive because of lower power loss but problems of controlling temperature distribution within the turbogenerator; of effecting a startup; marginal load capacity, etc., make these unattractive for initial development. Therefore, film bearings--hydrodynamic, hydrostatic, and hybrid--are considered as candidate types suitable for initial development.

Design and development problems arise in three areas in applying 600°F potassium lubricated film bearings to the turbogenerator. First, during startup, adequate operation under boundary lubricating conditions is required--a problem of bearing design, and particularly of material selection. Secondly, journal bearing stability, particularly for space power plants, is required over a range of loads varying from near zero to in excess of the static weight of the rotating group. Third, bearing designs and materials, must be satisfactory for operation over extended periods against cavitation erosion--a problem arising in lubricants operating near their vapor pressure--and corrosion of potassium.

Experience, dating back over a decade, with hydrodynamic fluid film bearings in NaK and sodium was available initially from tests at moderate temperatures and surface speeds and indicated the feasibility of using fluid film liquid metal bearings in the turbogenerator. Therefore, early in the present program, tests were conducted on a simple full cylindrical journal bearing, in order to obtain preliminary data on the characteristics of potassium as a lubricant at higher speeds. Representative test data with unidirectional loads, 600°F potassium lubricant temperature, and 24,000 rpm (100 ft/sec.), is shown in Figures 3 and 4. Measured journal eccentricity and attitude angle variations with load are shown along with theoretical values. The differences between these are considered acceptable considering the uncertainty of potassium film clearance measurements at this temperature. The bearing stator ($M_0 - 0.5 Ti$) and journal (WC) surfaces are shown in Figures 5 and 6 after a 300-hour test run. (Darkening of the journal occurred during disassembly and cleaning by inadvertent oxidation of the material.) No measurable wear of these surfaces occurred. The results indicated a reasonable load carrying capability of potassium lubricated bearings at this temperature.

The selection of materials suitable for bearing surfaces operating under conditions of boundary lubrication--as occurs during startup and

shutdown of the turbogenerator--is being conducted in a special friction and wear test machine in which selected materials are tested using a cylindrical rotor pressed upon by flat stators imulating "line contact" of the turbogenerator shaft and bearing during startup. Selected combinations of materials are tested for wear, galling, friction, etc., have, to date, been found generally most suited for the shaft and stator parts.

A background of additional data exists from boundary lubrication tests conducted in sodium and NaK. All of these data indicate that a measurable degree of boundary lubrication can be obtained in liquid metals with proper selection of materials such as the cermets mentioned above.

A second area of bearing development is that of journal bearing stability, a problem common to all high-speed rotating machinery. Stability problems are aggravated by operation in a zero-g environment. Because of the high cost and long time required to set up and test candidate bearings in potassium, preliminary stability tests are done in a special test rig consisting of rotors supported on test bearings lubricated by conventional fluids. Water, which has a viscosity at 240°F equal to that of potassium at 600°F, has been found to be an acceptable fluid for stability testing. Candidate bearings, which have acceptable stability characteristics in the water tests, are tested using potassium in the test apparatus shown in Figures 8 and 9. The test rig shown in Figures 8 and 9, consists of a potassium vapor turbine supported by one thrust and two journal bearings lubricated by liquid potassium.

Special instrumentation, which was developed to measure dynamic shaft motions in the liquid-potassium environment, is shown located adjacent to the journals in Figure 8.

Stability tests in all test rigs are run with rotor axes oriented both horizontally and vertically to evaluate the effect of static load of the rotor stability. In a vertical orientation, the effect of a zero-g environment is determined to some extent since the stabilizing effect of

the rotor weight is removed from the bearings.

A number of journal bearings are being evaluated for stability characteristics, such as floating ring, pivoted pad, hybrid, and lobed types. Each type has demonstrated stable characteristics in various applications. The floating ring bearing is attractive for the potassium turbogenerator since the velocity of the lubricant relative to metal parts is approximately one-half as large as for most journal bearings, which is expected to make this bearing more resistant to potential erosion. The two clearances in series make this bearing tolerant to thermal distortion and misalignments. Hybrid bearings are also attractive as some hydrostatic "lift-off" of the rotor is provided by the supply pressure, thus reducing the boundary lubrication problem during startup. Pivoted pad journals are attractive as self-alignment capability is inherent in these types. Representative results obtained on several of these types regarding stability characteristics are of interest.

Figure 10 shows a floating ring bearing which was tested using a light oil at two temperatures. Representative operation of this bearing, showing motion of the shaft and frequencies comprising this motion and ring speed is shown in Figures 11 and 12. In the test at 100°F, Figure 11, the shaft motion is seen to be composed of three frequencies--one at shaft speed, one at one-half ring speed, and one at one-half the sum of ring plus shaft speed. These frequencies indicate, respectively, a synchronous whirl, a whirl generated in the outer film, and a whirl of the inner film. The shaft pattern shown, which consists of two inside loops, agrees with these frequencies since it is characteristic of a one-third frequency whirl component. These patterns are in accord with theory which indicates that whirl in both films may occur. The test at 200°F, Figure 12, shows no whirl at one-half ring speed. The reduced viscosity apparently weakened the outer film enough to permit stability in that film, and excursions of the shaft--over several orbits--are smaller. Modifications to this bearing

made in an attempt to reduce the strength of the films in order to obtain stable operation, have not been wholly satisfactory to date, particularly with the rotor axis oriented vertically in oil and water.

Figures 13 and 14 show a hybrid bearing and representative motions for a speed of 20,000 rpm at two supply pressures. The sensitivity of the stability of this bearing to supply pressure is clearly evident in Figure 14, in that a supply pressure increase of 5 psi completely suppressed the half-frequency whirl component. Several configurations of the bearing tested to date have been unsuitable since supply pressures required to obtain stable operation at full speed are higher than considered acceptable for operation in available test systems. The whirl patterns shown in Figure 14 are essentially the same with horizontal or vertical orientation of the rotor axis.

Figure 15 shows a 3-pad pivoted-pad journal, with operating characteristics shown on Figure 16, with the shaft oriented vertically. The good stability features are noted by the absence of whirl components, other than a synchronous whirl due to nonconcentricity between the bearing journal and measuring sleeve. Patterns are similar with the rotor oriented horizontally. This bearing was selected for testing in potassium using the test rig of Figure 8. Figure 16 also shows the shaft patterns at the end of a 200-hour test in potassium at 600°F. These results indicate that characteristics in potassium are quite similar to those obtained with water. Startup and shutdown of the turbine in potassium proved to be easy, without requiring excessive turbine power to initiate roll-over.

Turbine: The principal design requirements are good aerodynamic efficiency and long operating life using saturated potassium vapor at the inlet of the turbine. Potassium properties resemble those of water in that an isentropic expansion from saturated conditions enters into a regime of progressively lower quality. Therefore, the aerodynamic performance is

expected to be affected in a manner analogous to that of steam turbines due to "wetness" of the vapor flow. Also, as occurs in some steam turbines the wet vapor flow introduces a potential for erosion of blading due to impingement of condensed liquid droplets.

Aerodynamic performance estimates of a potassium turbine are made utilizing data from steam and gas turbine experience. Thermodynamic properties of potassium with sufficient accuracy for establishing preliminary aerodynamic designs and performance estimates are currently available. Other data and theory indicates that the expansion process of potassium vapor in the 1200°F/2000°F range will resemble that of steam at low temperature. It is expected, therefore, that potassium vapor will act analogously to steam in a turbine, and that designs and performance estimates based on characteristics of the latter will be accurate.

It has been shown that "wet" steam erosion is a function of blade speed and blade material. Available theories indicate that the lower density of liquid potassium, as compared to water, should allow the use of blade speeds of potassium turbines to be somewhat higher than those of steam. To check the prediction that water and potassium erosion are comparable, a simple erosion test facility was built. This apparatus is shown schematically in Figure 17, along with some representative test specimens constructed from several materials. The test rig consists of a rotor holding four specimens which rotates normal to the flow direction of two one-fourth inch diameter fluid jets. Water and potassium were both used in this rig and the weight loss of the specimens were compared. Figures 18 and 19 show the relative weight loss for several materials after testing in water and potassium up to approximately 10^8 impacts. The figures show that the molybdenum base alloys are more resistant to erosion in potassium than Stellite 6B (HS6B) which is widely used in steam turbines for its good resistance to erosion. However, the figures show that 1400°F potassium

provides approximately twice the weight loss compared to 200°F water. The effect of velocity on erosion is not precisely known but various investigators indicate that below a certain critical velocity no erosion takes place. Endurance testing of a turbine is the only accurate test for the erosion determinations, but the present design speed and material selection (the molybdenum alloy TZM) are believed to be satisfactory for extended running.

The rotational speed of 24,000 rpm was based on a compromise between generator characteristics (weight and size for 3,200 cps), and turbine design at various speeds, for the required constant power output. The six-stage design was selected since the tip speed of the last stage is satisfactorily low as only small gains in efficiency accrue from a greater number of stages.

At present, a test turbine, represented schematically by Figure 20, is being assembled and installed in the turbine test facility for testing in potassium. The purpose of this test is to verify the aerodynamic performance of one, two, or three stages of the turbine in potassium. A high-speed generator, using oil bearings, is being used as a dynamometer in this test, with potassium bearings supporting the turbine. Air testing of this unit has been completed and testing in potassium will allow correlation of air performance data with potassium data. The turbine test facility is shown in Figure 21.

The molybdenum alloy TZM ($\text{Mo} - 0.5 \text{ Ti} - 0.08 \text{ Zr}$) has been selected as the principal candidate material for construction of the turbine rotor. This selection of TZM over other potential materials was based on good creep resistance at 2000°F, good fabricability and reproducibility, and compatibility with potassium. Conventional alloys for gas turbine practice are not suitable because of either low strength or noncompatibility with potassium for operation over long periods at temperatures in the 1800°F

range (however, for short-time testing up to 1600°F, some of these are suitable).

A forging and manufacturing process development program has been conducted to optimize the heat treatment and forging processes for use in preparing this material for rotor discs and blade blanks. Figure 22 shows two rotor forging blanks of TZM which have been made from arc cast material. Figure 23 shows a partially machined experimental turbine wheel. Creep and fatigue data has been obtained on TZM at temperatures in the 1800°F range. Figure 24 shows the high-temperature creep and strength properties and estimated operating stress levels of the first- and last-stage rotors.

FIGURE 1. POTASSIUM TURBOGENERATOR

FIGURE 2. POTASSIUM TURBOGENERATOR

FIGURE 3. POTASSIUM BEARING
ATTITUDE ANGLE DATA

FIGURE 4. POTASSIUM BEARING LOAD - ECCENTRICITY DATA

FIGURE 5. POTASSIUM JOURNAL AFTER 300-HOUR TEST

FIGURE 6. POTASSIUM BEARING AFTER 300-HOUR TEST

FRICTION AND WEAR OF BEARING MATERIALS
IN POTASSIUM AT 500°F (50 IN. PER SEC; $O_2 < 50$ PPM)

Material (R - Rotor, S - Shoe)	T _S - Start Torque (in.-lb)	T _R - Average Running Torque (in.-lb)	Calculated C _f	Weight Change (mg) (+ gain) (- loss)	Remarks*
K94-R Mo - 1/2 Ti-S	18.3	13.1	0.266 0.190	+0.6 -4.0	Poor
K601-R Mo - 1/2 Ti-S	17.4	10.9	0.253 0.159	-1.1 -3.7	Poor
TZM-R K94-S	>26	Shutdown	0.380 --	-40.0 -9.2	Very poor
K94-R TZM-S	17.0	10.9	0.247 0.159	+1.0 -5.0	Poor
K601-R TZM-S	28.4	Specimen fractured	0.410 --	-- --	--
TZM-R K94-S	26.2	14.0	0.381 0.204	-59.0 -0.5	Poor
Mo - 1/2 Ti-R K94-S	24.0	17.4	0.350 0.253	-69.8 +0.2	Poor
K601-R TZM-S	28.4	Specimen fractured	0.410 --	-- --	--
TZM-R W-S	>35.0	Shutdown	0.510 --	-108.4 -14.0	Very poor
W-R TZM-S	22.7	17.4	0.333 0.253	-15.4 -0.5	Poor
K94-R K94-S	15.3	13.0	0.222 0.189	-0.2 -0.4	Excellent
K94-R W-S	26.2	19.6	0.381 0.285	-666.3 -288.6	Poor
W-R TZM-S	4.3	4.0	0.062 0.058	-5.5 -0.7	Poor
Mo - 0.5 Ti-R W-S	2.1	--	0.03 --	-50.0 +5.0	Poor
K96-R K96-S	8.3	1.75	0.121 0.025	-1.1 -0.1	Excellent
K11-R K11-S	4.36	2.18	0.063 0.052	-0.8 -0.3	Excellent
K94-R K94-S	4.36	4.36	0.063 0.063	+0.2 +0.2	Excellent
K701-R K701-S	2.18	3.9	0.032 0.057	-0.6 -0.2	Excellent
K701-R K11-S	3.06	2.62	0.045 0.039	-0.5 -0.2	Excellent
K11-R K701-S	4.36	3.92	0.063 0.057	-0.9 -0.1	Excellent

*Excellent: Negligible wear, no material transfer, polish both specimens.

Poor: Excessive wear, surface roughening, material transfer, high friction.

Very poor: Test not completed due to excessive wear, friction, material roughening, and transfer.

TZM (Mo - 0.5 Ti - 0.1 Zr-0.02C)
Mo - 0.5 Ti (Mo - 0.5 Ti - 0.02C)
W (Pure arc cast W)
K-11 (WC - 3 Co - 0.1 Ti C)
K-94 [WC - 12 Co - 2 (Ta, Cb)C]
K-96 [WC - 12 Co - 2 (Ta, Cb)C]
K-701 (WC - 9.5 Co - 0.1 TiC - 4.0 Cr₃C₂)
K-601 (84.5 W - 10 Ta - 5.5 C, no limit)

FIGURE 7. BEARING MATERIALS FRICTION AND WEAR TESTS

FIGURE 8. POTASSIUM DYNAMIC BEARING RIG

FIGURE 9. DYNAMIC BEARING RIG PARTS

FIGURE 10. FLOATING RIG BEARING

SHAFT MOTION

PANORAMIC FREQUENCY, CPS

SHAFT SPEED = 10,500 RPM

RING SPEED = 5,100 RPM

LUBRICANT = LIGHT OIL AT 100°F (4.3 CP)

SHAFT HORIZONTAL

FIGURE 11. FLOATING RING BEARING STABILITY DATA, 100°F

SHAFT SPEED = 8,700 RPM

RING SPEED = 4,800 RPM

LUBRICANT = LIGHT OIL AT 200°F (1.45 CP)

SHAFT HORIZONTAL

FIGURE 12. FLOATING RING BEARING STABILITY DATA, 200°F

FIGURE 13. HYBRID BEARING

SUPPLY PRESSURE = 80 PSIG

SHAFT MOTION

0 250 500
PANORAMIC FREQUENCY, CPS

SUPPLY PRESSURE = 75 PSIG

SHAFT MOTION

0 250 500
PANORAMIC FREQUENCY, CPS

SHAFT SPEED = 20,000 RPM

LUBRICANT = LIGHT OIL AT 160°F (2.2 CP)

SHAFT VERTICAL

FIGURE 14. HYBRID BEARING STABILITY DATA

FIGURE 15. PIVOTED PAD BEARING

FIGURE 16. PIVOTED PAD BEARING STABILITY DATA

FIGURE 17. POTASSIUM EROSION RIG AND SPECIMENS

WATER EROSION TEST RESULTS

FIGURE 18. WATER EROSION TEST RESULTS

1400°F POTASSIUM EROSION TEST RESULTS

FIGURE 19. 1400°F POTASSIUM EROSION TEST RESULTS

FIGURE 20. POTASSIUM TURBINE TEST RIG

FIGURE 21. POTASSIUM TURBOGENERATOR TEST FACILITY

FIGURE 22. T2M FORGING BLANKS

FIGURE 23. EXPERIMENTAL TURBINE MACHINING SPECIMEN

FIGURE 24. CREEP PROPERTIES OF TZM

PERFORMANCE TEST OF A TWO-STAGE, 200-HORSE-
POWER TURBINE IN WET POTASSIUM VAPOR

R.J. Rossbach, E. Schnetzer,
H.E. Nichols, S.E. Eckard

Space Power and Propulsion Section
Missile and Space Division
General Electric Company
Cincinnati, Ohio 45215

ABSTRACT

The General Electric Company has been engaged by NASA under Contract NAS 5-1143 to design, fabricate, and test a two-stage, 200-horsepower turbine for operation on potassium at inlet vapor temperature up to 1600°F. The two stages represent stages 3 and 4 in a 5-stage, 500-KW turbine for the generation of electrical power during long-time duration space missions. The turbine was fabricated from nickel, iron, and cobalt base alloys. One major objective was verification of a performance calculation method. Therefore, an overhung arrangement was selected with an oil-lubricated pivoted-pad bearing and a duplex ball-thrust bearing for turbine support. A combination argon buffer and potassium hydrodynamic seal was developed to separate the oil and potassium. The 316-type stainless steel facility having a 3000-KW gas-fired boiler has made it possible to test the turbine over a range of temperatures, speeds, and pressure ratios. The turbine blading was designed making liberal use of steam turbine technology. The off-design performance of the turbine was estimated using the characteristics of this blading.

Performance data was obtained at two inlet temperature levels, namely, 1550 and 1450°F and over total-to-static pressure-ratio and speed ranges of 1.4 to 4.4 and 15,400 to 20,000 rpm, respectively. The power output data obtained agreed within 5.8 KW (The measured power output at design speed and pressure ratio was 160 KW at 1550°F inlet temperature.) of the calculated performance, demonstrating ability to calculate performance within practical limits. Individual data points were repeatable within 6 KW, indicating consistency in setting data points with reasonable tolerances and consistency in all pertinent instrumentation readings.

INTRODUCTION

Potassium and other alkali metals are being used by a number of research and development groups as Rankine-cycle component two-phase working fluids because at moderate pressures this class of working fluids boil and condense at high enough temperatures to permit the design of space-power systems with heat rejection radiators of manageable size. The turbine is one of the critical components of the space-power system, and as such presents the designer with a number of challenging problems.

For a temperature-limited Rankine-cycle system and a given condenser temperature (or pressure), the cycle efficiency is known from elementary thermodynamics (1)* to be higher for a system in which the condition out of the boiler is nearly saturated rather than highly superheated. For a space power system these facts mean that for a given power output the radiator area will be larger for a cycle using superheated rather than nearly saturated vapor because the superheated cycle will have the higher flowrate and the latent heat per unit flowrate is the same for both cycles. As a result, Rankine-cycle space power systems utilize only moderate amounts of superheat, making it necessary for at least part of the turbine expansion to occur in the wet region. The operation of stages in the wet region causes losses in efficiency due to droplet drag and the occurrence of supersaturation. Turbine-blade erosion may result when part of the expansion of the turbine is in the wet region. The rate of erosion damage, if any, depends upon such things as working fluid, wheel speed, blade material, temperature level, etc. (2). Erosion limits the reliability of the turbine and adversely affects performance.

In order to take into account supersaturation and droplet drag on turbine design properly, test data must be available on representative turbine stages run in the wet region. However, before such turbine stages can be designed, equilibrium thermodynamic property data and the effect of initial vapor quality on supersaturation and reversion must be known along with the polytropic expansion exponents of the working fluid. The

* Numbers in parentheses indicate references listed at the end of this paper.

potassium-vapor properties used by the authors' company are based upon experimental data obtained at the Naval Research Laboratory and are programmed on the digital computer (3). The effect of wetness on supersaturation and the polytropic expansion exponents was determined by measuring flow rates and wall pressure distributions in a converging-diverging nozzle tested in potassium vapor in the temperature range of 1450 to 1580°F and over the inlet vapor quality range of 85 to 99 percent (4).

In order to establish performance levels attainable in potassium vapor turbines of moderate size (500 to 1000 KW) and to verify off-design performance calculation methods, under NASA Contract NAS 5-1143, the authors' company designed, fabricated and tested in potassium vapor a two-stage, 200-horsepower turbine. The turbine was operated for 56 hours in its final performance test at inlet temperatures of 1450 and 1550°F and over a speed range of 15,400 to 20,000 rpm. Seventy-two data points were obtained at six values of speed and six values of turbine pressure ratio. One third of the data points were taken up to two days later to determine repeatability.

Another major contract objective was to determine the integrity of the test turbine and determine the relative erosion resistance of candidate blading materials. This is presently being done in a 2000-hour endurance test. More than 600 hours of this test have been completed with the turbine still running. After 254 hours of testing, a visual inspection with borescopes showed no damage to the blading.

In this paper the important design features of the turbine and its test facility will be pointed out. The instrumentation used to obtain the turbine performance will be discussed. The methods for calculating off-design performance will be summarized. Finally, the experimental performance data obtained will be compared with performance calculated prior to the test of the turbine.

TEST EQUIPMENT

The test equipment consists of the two-stage test turbine, its performance instrumentation, and the turbine test facility.

TURBINE

The two-stage potassium test turbine represents stages three and four of a five-stage conceptual turbine design for a 500-KW space electric power system. The design point conditions of the third and fourth stages were as follows:

Inlet Temperature, °F	1600
Inlet Vapor Quality, percent	92
Flowrate, lb/sec	2.8
Rotative Speed, rpm	19,200
Total-to-Total Pressure Ratio	2.95

Stage three was chosen as a first stage of the test turbine because the temperature was low enough to permit the use of super alloys commonly used in jet engines, and compatible with the temperature limits imposed by fabricating the test facility from 316 stainless steel. In addition the third and fourth stage would be typical of the stages where moisture would be present. It was considered necessary to test more than one stage since an upstream turbine stage is about the only means of adequately providing moisture with the proper drop-size distribution.

In the design phase, the facility boiler was expected to produce potassium vapor having a quality in excess of 98 percent. Therefore, it was planned initially to spray liquid potassium into the turbine entrance. The spray device was designed to depress inlet vapor quality to as low as 85 percent, permitting the testing of the turbine at design inlet quality and below. However, a performance test carried out using the liquid spray gave strong indications that a very coarse spray was being produced rather than the fine droplets needed to simulate the expansion in a turbine in the wet region. As a result, the performance test was repeated without the use of the spray device.

The fluid design of the two-stage test turbine is fully described in Reference (5). Briefly, the following procedure was used. Steam turbine practice was followed whenever practical and computer programs which evolved from steam turbomachinery design calculations were utilized in determining velocity diagrams and blade shapes. The axisymmetric solution

for the flow path was first obtained using supersaturated expansions through each turbine stage with reversion between stages. This solution, which solves the complete radial equilibrium equation by numerical methods and takes into account flow-path slope and curvature as well as variations in loss coefficients span wise, provided the velocity diagrams for each row of blades. The design is characterized by having moderately high blade loading to minimize droplet drag losses and reduce the number of stages. Nearly impulse conditions were imposed at the hub and the span-wise velocity distributions were free vortex. Each nozzle was slightly unchoked. Catalog steam-turbine blade shapes were selected for several sections in each of the blade rows. The coordinates of these sections then were input to a cascade analysis program which solves in exact manner the potential flow in cascade, resulting in the surface velocity distribution around each of the blade sections chosen. Modifications were made to the blade shapes in regions where the velocity gradients were adverse or undesirable. Eventually the blade shapes evolved which were expected to produce a high-performance turbine. Shown in Figure 1 is a typical bucket section of the turbine and the calculated surface-velocity distribution.

Shown in Figure 2 is the estimated total-to-total efficiency of the turbine at the design point as a function of radial tip clearance. The top line represents the variation in performance with tip clearance assuming a frozen-equilibrium expansion. The blading loss coefficients utilized in this calculation take into account the effects of profile, secondary-flow and tip-clearance loss and were obtained from Reference (6). Blade thickness, solidity, turning angle and the amount of reaction affect the loss coefficients so obtained. The middle line accounts for the supersaturation loss and the bottom line accounts for the moisture loss, which is defined as the kinetic energy required to bring all the potassium liquid droplets entering the stage up to wheel speed.

Shown in Figure 3 is the variation of turbine total-to-static efficiency with total-to-static pressure ratio for various rotative speeds. These data were calculated prior to performance testing using a gas-turbine off-design program for the supersaturated expansion in each stage. A reversion calculation was introduced into the program after each stage to take into account supersaturation loss. The off-design performance calculation utilizes the

turbine design geometry and the loss coefficients from the design performance calculation described above. Provisions in the program increase the loss coefficients for off-design angles of attack on the blading. From the calculated quality before each stage the moisture loss is calculated separately taking into account the droplet drag on the buckets and the reduction in vapor flowrate.

Mechanical Design

In the mechanical design of the test turbine, it was decided to isolate the problem of potassium turbine performance from other complicated space-power-turbine problems. As a result, the turbine was designed for operation on oil-lubricated bearings. Buckets were dovetailed to each disc to provide an easy method for removing and replacing blades, and for test of multiple rotor blade materials. The requirement for obtaining detailed turbine performance data required making provisions for interstage as well as inlet and outlet thermocouples and pressure taps.

Shown in Figure 4 is cross-sectional drawing of the two-stage potassium test turbine. The potassium vapor enters from the left, flows around a bullet nose into an annulus, and then into the first-stage nozzle diaphragm. After leaving the second row of buckets, the potassium vapor flows through an outlet guide vane and into a scroll and discharges through the test floor. The present configuration, which is an overhung design, was chosen over a straddle mounted design because it eliminates a bearing and seal on the high-temperature, high-pressure end. Just aft of the last stage of the turbine is a pivoted-pad bearing lubricated with Mobil Socony DTE 797 oil. The thrust bearing consists of a preloaded duplex ball bearing shown at the right-hand side of the figure. Between the second stage and the pivoted-pad bearing there is a combination hydrodynamic and argon-buffer seal. The hydrodynamic seal, which consists of a rotating channel surrounding a stationary disc, is supplied with potassium liquid at 250°F. The buffer seal, consisting of two labyrinths with argon gas injected between them, is between the hydrodynamic seal and the bearing sump. Part of the argon gas flows out of the hydrodynamic seal with the potassium liquid and part of the argon flows across the second labyrinth into the bearing housing. The hydrodynamic and buffer seal

permits a pressure differential up to about 25 psi to be maintained between the second-stage exit and the pivoted-pad bearing. The turbine rotor, shown in Figure 5, is made up of two discs and a stub shaft connected by means of curvic couplings and held together by a single centrally-located tie bolt. This tie bolt is prestressed to a tension of 10,000 lbs during assembly of the turbine rotor.

As indicated earlier, dovetail connections are used between the blading and the rotor discs, permitting the insertion of rotor blades fabricated from various types of material in the endurance test so that a relative evaluation of erosion resistance can be obtained.

Because of the instrumentation internal to the turbine, which will be described below, it was decided to use a horizontally-split casing. In Figure 6 is shown the rotor assembled to the outlet scroll. One half of the horizontally-split casing is in place. A nozzle diaphragm, the tip and interstage seals can be seen in this view. Both the circumferential and horizontal flanges of the turbine are welded. These can be seen in Figure 7. In order to establish a welding procedure which provided no distortion of the casing within rather close tolerances, an additional casing was welded systematically and checked dimensionally before the actual casing of the turbine was assembled. The tip seals for each of the bucket rows and the interstage seal are made of unfilled abrasion-resistant 316 stainless steel material. Shown in Figure 8 is the installation of the potassium test turbine in the turbine test facility. The turbine is surrounded by an environmental chamber which provides a cover of argon gas around the turbine casing, and instrumentation leads coming from the casing. The major purpose served by the environmental chamber is to prevent air from entering the turbine at subatmospheric pressures in the event of a potassium leak.

A water brake for absorbing the energy output of the test turbine is mounted aft of the turbine. A steam turbine is connected to the test turbine shaft aft of the water brake to drive the test turbine to about 10,000 rpm at which speed the hydrodynamic seal can be established, maintaining the required pressure differential between the second-stage exit potassium vapor and the pivoted-pad bearing oil lubricant.

TEST FACILITY

Shown in Figure 9 is an isometric sketch of the turbine test facility. The natural circulation boiler is on the left. It has a natural gas-heated fire box containing twenty-four flat-flame burners. Heat is transferred to the 174 element tube bundle by radiation. The tubes are welded between a 20-inch outside diameter upper drum and an 8 5/8-inch outside diameter lower drum, each in excess of 25 feet in length. Four 4 1/2-inch outside diameter downcomers also connect the upper and lower drums.

The air-cooled condenser hangs from the test turbine dolly as can be seen in Figure 9. It consists of 84 finned-tubes arranged in two banks. The air enters the condenser fore and aft, so as to cancel tube-bundle air-drag forces, and discharges on the far side. The lower drum which contains about 200 lb of titanium hot trapping material is approximately 8-feet long and has an inside diameter of 20 inches. For the removal of argon gas from the potassium vapor, the condenser has a vertical packed column through which potassium liquid at 250°F circulates at the rate of 10 gpm. The argon gas is removed by means of a vacuum system.

The helical induction electromagnetic pump, shown on the floor in Figure 9 between the boiler and condenser, takes the condensate from the condenser and pumps it into the boiler. At 1400°F the rated head of the pump is 150 psi and the rated flow is 42 gpm. The pump is followed in the flow circuit by an electromagnetic flow meter which measures the amount of liquid leaving the condenser.

In addition to the main loop the turbine test facility has eight sub-loops. The argon extraction system removes argon which enters the facility from the seal or pressure instrumentation. The argon reclamation system precludes the necessity of venting the argon carried with the oil and seal potassium. The low-pressure lubricating oil system supplies oil to the turbine water brake and steam turbine. The high-pressure lubricating system supplies oil to the test turbine bearings. There is a water system for the water brake and a steam system for the steam turbine. A vacuum system is needed to evacuate the argon gas

from the loop before start up. Finally there is a subloop to supply the hydrodynamic seal with liquid potassium.

Shown in Figure 10 is an exterior view of the turbine test facility. The equipment in the foreground is the scrubber system which maintains a flow of cooling air through the test cell and is used to wash the potassium oxide from the cooling air in the event of potassium leaks in the test cell or boiler. The 3000-KW (thermal) boiler is shown at the right-hand side of the figure. The condenser cooling air discharge is also shown.

The turbine test facility is constructed of type 316 stainless steel which has a practical maximum material temperature of 1600°F. The facility provides potassium vapor at 1550°F at a vapor quality in excess of 98 percent. A condenser pressure of as low as 1.5 psia can be provided.

INSTRUMENTATION

Shown in Figure 11 is a drawing of the two-stage potassium turbine showing the location of instrumentation stations. The potassium vapor enters the turbine from the left, passes through the two turbine stages and discharges by means of a scroll downward into the condenser. Station 1 contains five thermocouples, five total head tubes, two static taps, a throttling calorimeter to measure inlet vapor quality and a slack diaphragm pressure gauge. Station 3 is in the inlet annulus to the turbine and at this station there are four total head tubes and four static taps, two each on the inner and outer walls, and a thermocouple. Behind each blade row (Stations 4, 5 and 6) there are two static taps, one each on the inner and the outer walls. At Station 4 there was also a thermocouple in the wheel space. At Station 7, which is the turbine exit, there are four total head tubes, two static taps, one on the inner and one on the outer wall and four thermocouples. At Station 8, which is downstream of the turbine scroll, there are five thermocouples, five total head tubes, two static taps, and a heating calorimeter. In addition at Stations 7 and 8 there are slack-type diaphragm-type pressure gauges. All thermocouples utilized to measure the turbine fluid dynamic performance are sheathed chromel-alumel.

The pressures measured ahead of, inside of and aft of the turbine are measured by specially developed argon efflux pressure measuring devices. In that system, a metered amount of argon gas flows through each sensor line to balance the potassium-vapor pressure and to act as a buffer between potassium vapor and a precision pressure transducer (Reference (7)).

Shown in Figure 12 is an installation drawing of the turbine showing torque meters. The water-brake torque meter measures the reaction between the turbine and water brake casing. The steam-turbine torque meter measures the reaction between casings of the test and steam turbines. The two torque meters are of the temperature-compensated, strain-gauge type.

The turbine flowrate is measured by instrumentation in the inlet annulus of the turbine. This instrumentation is comprised of several pairs of total pressure probes and wall static taps. This instrumentation was calibrated in an air test stand prior to the performance test.

The speed was measured by electronically counting electrical pulses produced by a six-toothed gear on the turbine shaft. This speed was read on continuous recorder charts and displayed on a dial for the operator. The speed used in data reduction was obtained from a small calibrated D.C. generator. The 126 bits of measured performance data were acquired by means of a digital data handling system. This device scanned the channels in 70 seconds and recorded the information on punched and printed tape. The printed tape was used in the test cell to determine whether certain critical test conditions were within established tolerances. The punched tape was processed on a digital computer and resulted in reduced data printed on multilith masters.

EXPERIMENTAL RESULTS

Because of the newness of some of the instrumentation utilized to measure the performance of the potassium test turbine, the first step in the evaluation of the test data was to make certain checks on the readings of the various instruments.

Comparison of Measured Pressures and Temperatures

At several stations both temperatures and pressures were measured and the comparison of these measurements was studied to determine the validity of the measurements recorded. Shown in Figure 13 is the variation of the static pressure at Station 1, upstream of the turbine, with the vapor pressure determined from the measured temperature. It can be seen from the data that there is a good correlation between the measured and the vapor pressures.

Shown in Figure 14 is the variation of static pressure with the vapor pressure at Station 3. The vapor pressure appears to be about 1 psi higher than the measured pressure. The discrepancy amounts to a 7.5 and 11°F temperature error at 1550 and 1450°F inlet temperatures, respectively. The temperature is suspected of contributing the bulk of the error because it reads substantially the same as at Station 1, but the flow area is smaller at Station 3 requiring a lower static pressure. In addition the calibration data of the pressure transducer and readout system showed a repeatability within 0.07 psi and a zero shift no larger than 0.1 psi.

Shown in Figure 15 is the variation of Station 7 static pressure with the vapor pressure. At the highest pressures shown the measured and vapor pressures correspond. At the lowest pressures shown the measured pressure is substantially above the vapor pressure. An explanation of this is given in Figure 16 where the ratio of measured to vapor pressure (degree of supersaturation) is plotted against the turbine total-to-static pressure ratio. For the two values of turbine inlet temperature the degrees of supersaturation agree at the same turbine pressure ratios, indicating that some supersaturation is being measured at Station 7.

Shown in Figure 17 is the variation of the static pressure at Station 8 with the vapor pressure at the same station. The correlation is good and indicates no measureable supersaturation. The amount of argon from the efflux pressure measuring devices carried along with the vapor gas stream is at least as large at Station 8 as it is at Station 7, precluding the possibility that the measured pressure at Station 7 is the sum of the partial pressures of argon and potassium while the temperature indicates the vapor pressure only.

Performance Data

Prior to carrying out the performance test it was necessary to calibrate the test turbine for the bearing and seal losses. This was necessary because the turbine blading torque rather than the shaft torque was desired for verification of fluid design techniques. First dead weight calibration tests were carried out on the water brake and steam turbine strain-gauge torque meters (See Figure 12). Next the water brake was driven by the steam turbine at several torque levels without the coupling to the test turbine in place. Adjustments of hose connections to the steam turbine and water brake were made until the two torque meters read the same torque within a tolerance of 5 in.-lb. The coupling between the test turbine and water brake was then installed. After rotation was begun and the hydrodynamic seal had been actuated by admitting potassium liquid, the tare test for establishing the torque absorbed by the bearing and seal was carried out at several values of rotative speed and argon pressure level in the test turbine. Measurements of rotative speed, water brake and steam turbine torque and argon pressure were made for each condition set. The net measured torque (steam turbine minus water brake) is the bearing and seal torque plus the unwanted test turbine blade windage.

Since the blade windage is a function of argon density in the test turbine, by measuring the constant-speed torque at levels of subatmospheric pressure, an extrapolation to zero pressure (giving zero windage) results in the net bearing and seal torque. The results of the test to determine the bearing and seal torque are shown in Figure 18 where the bearing and seal torque is plotted against the rotative speed. In the evaluation of the performance test data the net bearing and seal torque shown by the solid line was added to the difference between water-brake and steam-turbine torque to result in the net blading torque.

The performance test was carried out at inlet temperatures of 1450 and 1550°F. For each temperature data was taken at six speeds from 15,400 to 20,000 rpm and six turbine total-to-static pressure ratios from 1.4 to 4.4. Seventy-two data points were taken, taking two digital-data-handling-system scans for each. If neither of the two scans was within the specified tolerances the condition was immediately reset and scanned until acceptable. In

addition, one-third of the data points were reset after taking other data points to check on repeatability.

Shown in Figure 19 is a comparison between the experimentally-determined turbine power and the calculated values for inlet vapor temperatures and a rotative speed of 1450 and 1550°F and 19,300 rpm, respectively. The experimental data are a little lower at low pressure ratios than the calculated data. The standard deviation from the calculated values of all of the power data (including second scans and repeated points) at both temperatures is 5.8 KW. At pressure ratios of approximately 1.4 and 3.3 repeated runs were made. The repeatability is seen to be within 6 KW.

Shown in Figure 20 is a comparison between the experimentally-determined turbine flow rate and the calculated values for inlet vapor temperatures and rotative speed of 1450 and 1550°F and 19,300 rpm, respectively. These data were obtained using the calibrated turbine inlet annulus as the flow measuring device. The data are in close agreement with the calculated values. The standard deviation from the calculated values of all of the flow rate data for both temperatures is 0.044 pps. The repeatability of the data at pressure ratios of 1.4 and 3.3 is within 0.1 pps.

Shown in Figure 21 is a comparison between the experimentally determined total-to-static efficiency and the calculated values for inlet vapor temperatures and rotative speed of 1450 and 1550°F and 19,300 rpm, respectively. The experimental data agrees closely with the calculated data at low values of pressure ratio in spite of the steep slope. In the high range of pressure ratios the experimental data lies above the predicted values. The standard deviation from the calculated values of all the efficiency data for both temperatures is 4.3 percentage points. Repeatability at pressure ratios of 1.4 and 3.3 is within 4 percentage points.

CONCLUDING REMARKS

Under NASA Contract NAS 5-1143 two stages of a five-stage space power turbine have been designed using steam turbine technology and potassium properties, built and performance

tested in potassium vapor at vapor temperatures up to 1550°F and over speed and total-to-static pressure ratio ranges of 15,400 to 20,000 rpm and 1.4 to 4.4, respectively. A special argon efflux pressure-measuring system was developed for the performance test and applied to the measurement of turbine flowrate in the vapor state. In 56 continuous hours of turbine operation 72 data points were obtained within specified tolerances and twenty-four points were reset after a time interval to check repeatability. The measured power output at design speed and pressure ratio was 160 KW at 1550°F inlet temperature. The experimental performance data agreed well with the calculated data demonstrating ability to calculate performance within practical limits. For example, the power output data agreed within 5.8 KW of the calculated performance. Individual data points were repeatable within 6 KW, indicating consistency in setting data points with reasonable tolerances and consistency in all pertinent instrumentation readings. Endurance testing has been initiated at a temperature and speed of 1500°F and 18,250 rpm, respectively, and a total-to-static pressure ratio of 3.47. Over 600 hours of endurance testing has been completed without degradation in the turbine performance at the established flow conditions.

REFERENCES

1. D.B. Mackay. Prentice-Hall, Inc. Englewood Cliffs, 1963, pp 83-98.
2. B. Wood: "Wetness in Steam Cycles" Proc Instn Mech Engrs, V 174, N 14, pp 491-534, 1960.
3. G.C. Wesling and M.E. McCarthy: "Thermodynamic Properties of Potassium Vapor From 1300 - 2700°R Using Naval Research Laboratory Experimental Data". Appendix A of Quarterly Progress Report No. 14 of Two-Stage Potassium Test Turbine, General Electric Company, November 8, 1964, NASA - CR - 54285.
4. R.J. Rossbach: "Critical Flow of Potassium Vapor Through Instrumented Converging-Diverging Nozzle". A.S.M.E. preprint No. 65-GTP-22, January 15, 1965.
5. R.J. Rossbach: "Space-Vehicle Rankine-Cycle Power Plant Potassium-Turbine Development". A.S.M.E. preprint 63-WA-326, October 7, 1963.
6. D.G. Ainley and G.C.R. Mathieson: "A Method of Performance Estimation For Axial-Flow Turbines". A.R.C. (British) R. and M. No. 2974, December, 1951.
7. G.C. Wesling, R.J. Rossbach and W.M. Shaffernocker: "Development of Efflux Measuring Device For Performance Evaluation of Potassium Vapor Turbine". Conference On Application of High Temperature Instrumentation To Liquid Metal Experiments, Argonne National Laboratory, September, 1965.

ACKNOWLEDGEMENT

The authors wish to thank the National Aeronautics and Space Administration, under whose contract, number NAS 5-1143, the work reported was carried out, for permission to present this paper.

Figure 1. Velocity Distribution Around Second Stage Bucket Hub Section.

Figure 2. Effect of Tip Clearance and Other Losses on Turbine Design Efficiency.

Figure 3. Calculated Turbine Efficiency Vs. Pressure Ratio. Turbine Inlet Temperature, 1550°F.

Figure 4. Potassium Test Turbine.

Figure 5. Two Stage Turbine Wheels and Shaft.

Figure 6. Turbine With Casing Removed.

Figure 7. Turbine After Welding Casing.

Figure 8. Potassium Turbine Test Installation.

Figure 9. 3000 KW Turbine Test Facility.

Figure 10. Turbine Test Facility.

Figure 11. Turbine Performance Instrumentation Stations.

Figure 12. Turbine Installation.

Figure 13. Comparison of Measured Pressure With Vapor Pressure at Station 1.

Figure 14. Comparison of Measured Pressure With Vapor Pressure at Station 3.

Figure 15. Comparison of Measured Pressure With Vapor Pressure at Station 7.

Figure 16. Ratio of Measured Pressure to Vapor Pressure at Station 7.

Figure 17. Comparison of Measured Pressure With Vapor Pressure at Station 8.

Figure 18. Comparison of Bearing and Seal Torque Data as a Function of Rotative Speed.

Figure 20. Comparison of Measured and Calculated Weight Flow Rate. Rotative Speed, 19,300 rpm.

Figure 19. Comparison of Measured and Calculated Turbine Power. Rotative Speed, 19,300 rpm.

Figure 21. Comparison of Measured and Calculated Total to Static Efficiency. Speed, 19,300 rpm.

RADIATORS AND CONDENSERS

Chairman: S. Lieblein

NASA Lewis Research Center
Cleveland, Ohio

BLANK

RADIATORS AND CONDENSORS

INTRODUCTION

S. Lieblein

NASA-Lewis Research Center
Cleveland, Ohio

In selecting the general subject matter and contents for this session, it was necessary first to look at the overall status and state-of-the-art of the heat rejection field for Rankine systems. To do this, it was convenient to examine the major design considerations involved in radiator development as summarized below.

- I. External Heat Transfer
 - A. Fin tube geometry
 - B. Surface coatings
- II. Internal Flow
 - A. Heat transfer coefficients
 - B. Pressure drop
 - C. Stability
- III. Meteoroid Protection
 - A. Hazard
 - B. Geometry
 - C. Materials
- IV. Structure and Fabrication
 - A. Materials
 - B. Geometry
 - C. Loads
 - D. Vehicle integration

The first item, external radiation heat transfer, which essentially sizes the radiator, is comparatively well in hand. Numerous steady-state heat transfer solutions are available for a wide range of radiator geometries. Tests have shown that this theory adequately

represents actual operating temperature variations. Similarly, endurance tests in vacuum have indicated that stable high-temperature surface coatings can be obtained to give total hemispherical emittance values between 0.85 to 0.90 for 10,000 hours. The major remaining unsolved area in external heat transfer appears to be in unsteady or transient effects associated with system start-up, orbital variations, and part-power operation.

The second item, internal flow in the two-phase condensing regime, has received considerable attention in the past year or two, and some significant test information has been obtained with respect to heat transfer coefficients, pressure drop, and flow stability. It appeared desirable therefore to bring this area up to date, and summarize the latest findings. As will be indicated by the papers presented in this session, it appears that radiator and condenser internal flow may shortly also be considered sufficiently well in hand.

Similarly, considerable advances have also been made in the field of meteoroid protection in the past year. A much better picture of the meteoroid hazard in space has been obtained from analysis of the available flight experiments and meteor data. In general, it appears that the hazard is less severe than believed earlier, and we feel a lot more confident about the little we do know about the meteoroid situation. Also, considerable interesting work has been done on hypervelocity impact into radiator materials and actual radiator tube configurations using a light gas gun. It was believed important, therefore, to present the latest results from these tests as will be discussed later in the session.

Finally, the fourth item of structure and fabrication, which includes consideration of materials, geometry, loads, and vehicle integration, is, by comparison, relatively new in the progression of developments. Although, it is true that system-type radiators have been built and operated recently, as will be indicated in some of the session papers today, the general status of full-scale integrated system radiators is far from a highly developed state-of-the-art for high-temperature systems. Structural studies are currently underway, and with the assurance of satisfactory performance in the previous three categories, considerable further progress should be in the offing. In fact, the subject of radiator fabrication and vehicle structure integration could well form the subject matter for a future conference session.

Accordingly, it was decided for this session to concentrate on the factors of internal flow and meteoroid protection with some attention to radiator construction when present on conjunction with the first two items. Also, it was felt that only actual test results with hardware configurations should be considered, since in essence, this is the real "proof of the pudding". Fortunately, some excellent experimental programs have been underway in these areas and we will be able to learn of the results of these programs in the papers presented herein.

The session is divided into three categories. The first, containing three papers, deals with mercury vapor systems. The first paper summarizes the results of tests of heat transfer coefficients and pressure drop during condensation in single tubes under conditions of both constant heat flux and convective cooling. These results were obtained from the Mercury Evaporating and Condensing Analysis (MECA) project conducted by the NASA-Lewis Research Center in support of the SNAP-8 program. The second paper considers multitube radiator condensers designed, fabricated, and tested at Atomics International for the SNAP-2 programs. Included in the paper are discussions of thermal and structural component testing, and prototype system mockup test experience with respect to steady-state and transient analysis and performance. The third paper discusses design, construction, and preliminary performance characteristics of the NaK-cooled multitube condenser for the SNAP-8 system being developed by Aerojet-General. Comparisons between design, measured, and single tube condensing coefficients are indicated.

The second group of papers contains a parallel presentation of results with potassium condensers operating at high temperatures. The first paper presents results of pressure drop and heat transfer coefficient tests in several sodium-cooled single tube configurations conducted by the General Electric Company. The second paper discusses the design and test experience obtained with several part and full-scale multitube radiative condensers built in conjunction with the Medium Power Reactor Experiment (MPRE) program at the AEC Oak Ridge National Laboratory. Preliminary results of a prototype NaK-cooled multitube potassium condenser and a multitube radiative condenser operated in a vacuum chamber at the NASA-Lewis Research Center are summarized in the third paper of this group. The final paper of the session, the sole item in the third category, presents recent results of hypervelocity impacts into various radiator armor materials and configurations.

The general conclusion or impression to be reached from the results of these papers is that multitube prototype radiators and condensers can be designed and operated to satisfactorily meet design requirements for both mercury and potassium vapor applications. It appears that condensing coefficients in both cases are sufficiently high so as to not constitute an important design factor, and also that internal flow and heat transfer are adequately understood and predictable for design purposes. Furthermore, we are rapidly learning how to effectively protect the vulnerable radiator elements against damage from impacting meteoroids.

AN EXPERIMENTAL STUDY OF THE CONDENSING CHARACTERISTICS OF MERCURY

VAPOR FLOWING IN SINGLE TUBES

by James A. Albers and David Namkoong, Jr.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio

ABSTRACT

An investigation was conducted to determine the condensing characteristics of mercury vapor flowing in horizontal single tubes. For a crossflow-nitrogen-cooled condenser, flow regime and pressure drop were investigated for nonwetted flow in constant diameter and tapered tubes, under 1 and 0 g, and for wetted flow in tapered tubes under 1 g. For a counterflow-NaK-cooled condenser, heat transfer, inlet pressure, and pressure drop were investigated for a tapered tube under 1 g.

Within the limits of the investigation and predicated on SNAP-8 requirements, the results obtained from the nitrogen-cooled condenser indicated no significant influence of gravity on nonwetted condensing pressure drop. Wetted 1-g condensing frictional pressure drop was higher than that of nonwetted only at the higher flow rates in a tapered tube. Data compared with the Lockhart-Martinelli and fog-flow correlations showed deviation from the former and data scatter on the latter.

For the conditions investigated, the NaK-cooled condenser indicated that the mercury condensing coefficients were higher than the NaK heat-transfer coefficient; therefore the NaK side was the more significant factor affecting the overall heat transfer coefficient. The inlet pressure of mercury, which is intimately related to the mercury inlet temperature and therefore to the heat-transfer mechanism of the condenser, was found to be sensitive to condensing length at shorter condensing lengths and insensitive at the longer condensing lengths.

INTRODUCTION

Of the many energy-conversion-system concepts proposed for the generation of electric power for various space flight missions, one of the more promising for the production of higher power levels is the liquid-metal Rankine-cycle turbogenerator system. The development of the SNAP Rankine reactor series characterizes this type of power system. A characteristic of the Rankine cycle system is that the working fluid in the turbogenerator loop undergoes phase changes during the heat-addition and heat-rejection portions of the cycle, resulting in the coexistence of liquid and vapor in continuously varying proportions between the all-liquid and saturated-vapor states. These variable-quality flow conditions represent a major problem in the design of the boiler and condenser components with regard to the prediction of heat-transfer and pressure-drop characteristics. Furthermore, the mixed phase flows are subject to the effects of differing gravity fields to the extent that data on heat transfer and pressure drop obtained in a 1-g environment may not be representative of those to be encountered during system operation in a weightless condition. An additional consideration is the significance of the wettability of the working fluid with regard to types of flow regimes and heat-transfer and pressure-drop characteristics. Finally, the condenser, which is located between the turbine exit and the pump inlet in the system, is restricted to stringent pressure-level and pressure-drop limitations. Because of these considerations, an experimental program was initiated at the NASA Lewis Research Center to determine the condensing characteristics of mercury vapor flowing in single tubes. Values of flow rate and condensing tube geometries were predicated on the SNAP-8 requirements.

This paper describes results obtained from two facilities reflecting two methods of transferring the heat from the mercury vapor condenser. One method simulates heat rejection by radiation, by providing a uniform temperature heat

sink. In this facility, flow regime and pressure drop were investigated for non-wetted flow in constant diameter, and tapered tubes, under 1- and 0-g environments, and for wetted flow in tapered tubes under 1 g. The other method utilizes a liquid metal coolant, NaK, in a counterflow heat exchanger system. In this latter facility, heat transfer, inlet pressure, and pressure drop were investigated in a tapered tube under 1 g. In all cases, the single condensing tubes were oriented horizontally. In addition to obtaining data applicable to the SNAP Rankine cycle, some of the data were analyzed for a more generalized comparison. As such, pressure drop data were compared with the analytical predictions of Koestel et al. and Lockhart-Martinelli in references 1 and 2, respectively.

SYMBOLS

A	Heat transfer area
D_i	Inner tube inside diameter
D_o	Inner tube outside diameter
D_m	Inner tube mean diameter
g_c	Conversion factor
h	Heat transfer coefficient
k	Thermal conductivity
L	Length
LMTD	Log mean temperature difference
l	Distance from condensing tube inlet
l_c	Condensing length
P	Pressure
Q	Heat transfer rate
Δr	Wall thickness
T	Temperature
U	Overall heat transfer coefficient

u	Velocity
x	Quality
μ	Viscosity
ρ	Density
σ	Surface tension
ϕ_g	Lockhart-Martinelli parameter, $\sqrt{(\Delta P/\Delta L)_{TPF}/(\Delta P/\Delta L)_g}$
X	Two phase flow modulus, $\sqrt{(\Delta P/\Delta L)_{liq}/(\Delta P/\Delta L)_g}$

Subscripts

g	Mercury vapor
l	Local
liq	Liquid
NaK out	NaK leaving heat exchanger
O	Inlet of condenser
s	Static
TPF	Two-phase frictional
1	One inch from inlet of condenser

APPARATUS FOR CROSSFLOW-NITROGEN-COOLED CONDENSER

A test apparatus was used in which a constant and uniform heat sink was provided along the condensing tube by gaseous nitrogen in crossflow from two manifolds on opposite sides of the tube. Nonwetted data for 1- and O-g comparison of flow regimes and pressure drop were obtained from the experimental system installed in the bomb bay of a converted Navy bomber (AJ-2). Near zero-gravity durations of about 10 to 15 seconds were obtained by flying the aircraft through a portion of a ballistic path (ref. 3). Data for wetted and nonwetted comparison of pressure drop were also obtained in a ground facility. In both cases, however, the experimental system was essentially the same. In the test system, the heat flux ranged from 30,000 to 92,000 Btu/(hr)(ft²) depending on the flow

rate and condensing length. Vapor temperature ranged from 800° to 1000° F depending on the vapor pressure and the degree of superheat that was required to minimize liquid carryover. The range of the other variables for the different conditions investigated are listed in table I.

Experimental System

Schematic drawings of a single-pass boiling and condensing system used for 1- and 0-g comparison test are presented in figures 1(a) and (b). The ground facility used for the wetting and nonwetting comparison test is virtually identical to the system shown in figure 1 except that a return line was provided between the receiver and the expulsion unit. In general, the mercury system consisted of an expulsion cylinder, a liquid-flow-measuring unit, a preheater, a high heat flux boiler, a main boiler, a vapor-flow-measuring venturi, a horizontal condensing tube, and a receiver for collecting the condensed mercury.

Definitions of flow characteristics for 1- and 0-g conditions were determined by high-speed photographs of nonwetting condensing in constant-diameter glass tubes of 0.27-, 0.40-, and 0.49-inch inside diameters. Pressure-drop measurements for 1 and 0 g were conducted on both constant diameter and uniformly tapered stainless steel tubes. The constant diameter tube was 87 inches long with an inside diameter of 0.31 inch and a wall thickness of 0.032 inch. The uniformly tapered tube was 84 inches long with an inside diameter of 0.40 inch at the inlet and 0.15 inch at the exit and a wall thickness of 0.025 inch. Pressure-drop measurements for wetted and nonwetted conditions were conducted on 4-foot-long uniformly tapered tubes with an inside diameter of 0.50 inch at the inlet and 0.20 inch at the exit. The nonwetting measurements were conducted with a 0.035-inch-wall stainless steel tube and the wetting measurements with 0.149-inch-wall copper tubes.

Instrumentation

Stainless steel inductance-type pressure transducers, capable of operating in a mercury environment up to 900° F, were used to measure condenser-tube pressure drop, venturi pressures, and venturi pressure drop. In the aircraft, each transducer in direct contact with mercury was mounted with the centerline of the diaphragm parallel to the lateral axis of the aircraft, to minimize the effects of acceleration and deceleration prior to and following the 0-g maneuver. Thermocouples throughout the system were constructed of I.S.A. standard-calibration K Chromel-Alumel wires.

All temperature, pressure, and acceleration data needed for analysis (for the gravity comparison tests) were recorded on two multichannel oscillographs. For the wetted and nonwetted comparison test on the ground, the pressure data were recorded on multichannel oscillographs and the temperatures on self-balancing potentiometer recorders.

Procedure

System operating procedure was identical for both ground and aircraft testing. The mercury loop was initially evacuated to approximately 60 microns of mercury, and the mercury heaters were brought to operating temperature. Startup mercury vapor flow was allowed to purge the system for approximately 5 minutes to remove noncondensables. The receiver pressure was then increased to a constant value (between 14 and 15 psia), and the cooling gas flow was adjusted to establish the mercury interface at the desired location. Relocation of the interface was accomplished by further adjustment of the cooling flow rate.

Determination of the interface location depended on the condensing tube material and the test facility. For the glass-tube tests, the interface was located visually. For the metal tubes in the aircraft (used in 1- and 0-g pressure drop comparison tests), the tube-wall temperatures were scanned to locate

the interface. The interface was identified by the sharp temperature drop in the liquid-mercury leg, which was readily apparent during a rapid scan of the wall temperatures. For the nonwetting and wetting comparison tests, the interface was observed with an X-ray image-intensifier system. The interface for the wetting case was defined as the location where the liquid filled the tube.

The 1- and 0-g comparison tests (of flow regime and pressure drop) were accomplished by recording the 1-g data points in the aircraft while in level flight and 0-g points immediately after without changing system operating conditions. Prior to each data run, a complete calibration of pressure and temperature instrumentation was carried out.

RESULTS FOR CROSSFLOW-NITROGEN-COOLED CONDENSER

Effect of Gravity on Nonwetting Condensing Characteristics

An indication of the differences in flow characteristics in 1- and 0-g environments is shown in figure 2. Under 1-g conditions, droplet runoff down the tube wall was observed, resulting in a liquid accumulation on the bottom of the tube, particularly in the low velocity region. Under 0-g conditions, the drops in the stream and on the wall were more uniformly distributed and had no tendency to "gravitate" toward any part of the tube surface. Whether this apparent difference in flow characteristics is translated into differences of pressure drop is the subject of the following discussion.

The effect of gravity on local static pressure drop is shown in figure 3. The local static pressure drop $(P_0 - P_t)_s$ was obtained from the difference between the measured inlet static pressure and the local static pressures along the condensing tube. For both the straight and the tapered tubes, the local static pressure drop increased over the first half of the condensing length due to the high friction losses that resulted primarily from high vapor velocities. In the last half of the condensing tube the pressure rise due to momentum change exceeded the pressure loss due to

friction, resulting in a decrease in local static pressure drop. The data presented for 1- and 0-g conditions were obtained at approximately the same tube inlet flow conditions. With the inlet conditions approximately the same, there was no discernible difference between the distributions of local static pressure drop for 1- and 0-g environments.

The effect of gravity on overall static pressure difference (from inlet to interface) is presented in figure 4. The overall static pressure difference $(P_O - P_{liq})_s$ was obtained by subtracting the average static pressure in the liquid leg from the inlet static pressure. Examination of the data indicates little difference between the 1- and 0-g conditions, although the majority of the 0-g data fall slightly above the 1-g data. Values of $(P_O - P_{liq})_s$ ranged from a pressure drop of 0.2 to 2.2 psi for the straight tube and a 0.9-psi pressure rise to a 0.1-psi drop for the tapered tube for the range of condensing lengths and flow rates investigated.

Comparison of Nonwetting Pressure Drop Data with Theory

The experimental pressure-drop data were compared with the two-phase pressure-drop predictions of Lockhart and Martinelli as discussed in reference 2. The two-phase frictional pressure drops between two pressure taps were determined by subtracting the pressure recovery due to the momentum decrease from the measured local static pressure difference. A velocity ratio of one $u_{liq} = u_g$ was assumed in determining the pressure recovery. This assumption is based on the analytical predictions of velocity profiles of liquid drops being entrained into the vapor stream (ref. 1).

Comparison of the nonwetting pressure drop data with the Lockhart-Martinelli correlation (fig. 5) shows that the correlation agrees with the data for the lower values of the parameter X (i.e., corresponding to qualities greater than 0.4). For the higher values of the parameter X (i.e., corresponding to

qualities less than 0.4), the frictional pressure gradient is greater than that predicted by Lockhart-Martinelli, and the deviation increases with an increase in the parameter X . This deviation may result partly from the fact that the flow regimes in dropwise condensing are significantly different from the two-component adiabatic flow model assumed by Lockhart-Martinelli. This deviation may also be because the roughness of the tube wall surface due to the condensed drops on the wall is greater than that obtained by Lockhart-Martinelli, resulting in a larger frictional pressure drop especially for condensation in the low-quality region of the tube.

Though they deviate from the predicted curves, the data points fall into a distinct grouping. Such a grouping suggests a faired data curve through the points such as that shown in figure 5. This curve also fits the faired condensing mercury data points of Koestel et al., as presented in reference 1 as shown in figure 6.

This same set of nonwetting pressure drop data was compared to the fog flow correlation of Koestel et al. (ref. 1). The correlation treats the liquid and vapor as a homogeneous flow and takes into account the buildup of drops on the inside tube wall. As shown in figure 7, the fog flow theory roughly predicts the trend of the nonwetting pressure-drop data over the entire quality range. However, considerable scatter exists.

Effect of Wetting on Pressure Drop

An investigation of pressure drop under wetted conditions, conducted entirely in the ground facility, included photographs of the X-ray image of the interface. The contrast between the interface under wetted and nonwetted conditions is shown in figure 8. Condensation under a nonwetted condition in a tapered tube resulted in a well-defined vertical interface, whereas under a wetted condition an elongated interface that gave rise to periodic waves

resulted. Any difference in pressure drop between wetting and nonwetting flow then may be attributed, at least in part, to the difference in flow regimes as indicated in the figure. Figure 9 indicates that there is no significant difference in the frictional pressure drop for wetted flow compared with the nonwetting condition for the low flow rate (0.028 lb/sec). Greater differences of frictional pressure drop occur in the last half of the condensing length for the mass flow rates of 0.038 and 0.049 pound per second. This may be due to the buildup of the liquid film on the bottom of the tube in wetting condensation, which tended to reduce the vapor-flow area and may have exposed a large area (between the film surface and vapor) of turbulence.

In general, a small, overall static pressure rise was obtained for both wetting and nonwetting conditions at the weight flows and condensing lengths considered as shown in figure 10. At a mass flow rate of 0.028 pound per second, the overall static pressure difference $(P_1 - P_{liq})_s$ was approximately equal for the wetted and nonwetted condensers. At mass flow rates of 0.038 and 0.049 pound per second, values of $(P_1 - P_{liq})_s$ for nonwetted flow exceeded that for wetted flow by 0.3- and 0.6-psi rise, respectively, for the range of condensing lengths considered.

Comparison of Wetting Pressure Drop Data with Theory

A comparison of the wetting pressure-drop data with the Lockhart-Martinelli correlation is presented in figure 11. The result was similar to that found for the nonwetted case in that the data indicated that the experimentally obtained pressure drops were greater than the predicted values at higher X values. Again, a fairly well defined curve could be drawn through the data points. The frictional pressure drops for the wetted condition were slightly greater than those for the nonwetted condition.

The wetting pressure drop data are compared with the fog-flow correlation in figure 12. Under wetting conditions, drops in the vapor stream are formed by the breakup of a thin film that forms on the condensing heat transfer surface. The fog-flow correlation agrees with the wetted data in the high quality region of the condensing tube (i.e., for modified Weber numbers greater than 10). The frictional pressure gradients are higher than predicted, however, for the low-quality region of the condensing tube (i.e., for modified Weber numbers less than 10). This may be due partly to the departure of the wetted flow regime in the low-quality region of the condensing tube from the basis of the fog flow correlation - that of liquid homogeneity within the vapor stream. Again, the data-point scatter is significant.

APPARATUS FOR COUNTERFLOW-NAK-COOLED CONDENSER

Experimental System

A single, tapered, mercury vapor condensing tube, cooled by NaK, was used to obtain heat transfer and pressure data under nonwetted conditions. Tests were conducted entirely in a ground-based facility. A schematic drawing of the mercury-NaK system is shown in figure 13. The range of variables for this experiment is listed in table I. The mercury loop is virtually identical to the nitrogen-cooled mercury vapor condenser system except that a desuperheater was provided between the mercury boiler and the vapor flow measuring venturi. The desuperheater was operated as a heat exchanger such that mercury vapor cooling was accomplished without condensation. By this procedure, high-quality mercury vapor at a saturated condition was provided at the condenser entrance.

The NaK loop is a closed, continuously circulating flow system, consisting of an electromagnetic (EM) pump and an EM flowmeter, an air cooler, and electrical heaters to control the NaK temperature into the heat exchanger. Other branches of the loop provide a means of filling, draining, and filtering the NaK loop.

The condensing tube geometry used was that of a tapered tube, the inner diameter of which was uniformly tapered from a nominal 0.45 inch to 0.20 inch over a length of 40 inches. The inner diameter remained constant at 0.20 inch to the condenser tube exit (12 in.). The annular passage, through which the coolant (NaK) flowed counter to the direction of mercury flow, was maintained at a constant height of 0.10 inch. The condensing tube wall was 0.035 inch, and the tube material was 9M.

Instrumentation

Pressures on the mercury side were measured by stainless steel inductance-type transducers in the high temperature (up to 900° F) sections of the loop and by strain-gage-type transducers for the lower temperatures. Pressure data were recorded on an automatic data-acquisition system and on a continuous trace, galvanometer-type recorder.

Thermocouples throughout the system were constructed of I.S.A. standard-calibration E, Chromel-Constantan wires. On the test section, thermocouples were attached to the outer tube at 1-inch intervals along the tube length; in addition, thermocouples were immersed in the NaK passage at 5-inch intervals. Temperature data used in the analysis were recorded on an automatic data acquisition system.

Procedure

With the NaK lines filled, power was applied to the EM pump to start circulation. The NaK flow rate was indicated by a digital readout from the EM flowmeter.

The procedure for the mercury loop was similar to that of the nitrogen-cooled condenser; that is, the mercury loop was evacuated to approximately 60 microns of mercury, heaters to vaporize mercury were brought up to temperature, and startup mercury vapor was allowed to purge the mercury system to remove

noncondensables. Operating conditions predicated on the SNAP-8 design requirements were reached by gradually increasing mercury flow rate, NaK flow rate, and back pressure, each variable coordinated with the others in a "bootstrapping" operation. With the mercury and NaK flow rates and the NaK inlet temperature at design levels (155 and 515 lb/hr and 500° F, respectively), the interface position was controlled by manipulation of the back pressure.

The interface was located by means of a temperature profile monitor, a visual indication of NaK temperature along the condenser length.

RESULTS OF COUNTERFLOW-NAK-COOLED CONDENSER

Heat Transfer Coefficients

Typical profiles of NaK and mercury temperatures along the condensing length are shown in figure 14. Both the outer-tube (shell) temperatures and the NaK-stream temperatures are shown. These profiles were based on constant values of saturated mercury and NaK flow rates and NaK inlet temperature. While these values were kept constant, the interface was positioned by controlling the condenser back pressure. The NaK temperature profile used in the heat transfer calculations was a curve that was faired through the stream temperature points with the shell data points serving as guides.

The overall heat transfer coefficient was calculated incrementally, for any one NaK-mercury temperature profile, by considering the fluid temperatures entering and leaving any one increment of heat transfer area. This was then plotted as $\int U dA$ on the ordinate and $\int dA$ on the abscissa. Figure 15 is a typical result. Except for a small section of the tube near the inlet, the values are such that a straight line could be drawn through the points. The slope of this line can then be considered as a mean overall heat transfer coefficient for the condenser section. For the range of condensing length considered (7 to 32 in.), the mean overall heat transfer coefficient ranged from 2500 to

3150 Btu/(hr)(ft²)(°F). The significance of these values of the overall heat transfer coefficient in determining the mercury condensing coefficient may be seen in figure 16. The range of the mean overall heat-transfer-coefficient experimental values are shown with the mercury condensing coefficient as the ordinate for three values of the NaK heat transfer coefficient. For the conditions covered in this series of tests, the heat transfer coefficient of NaK as calculated from an equation of Dwyer and Tu (ref. 4) was approximately 5000 Btu/(hr)(ft²)(°F). The values of 3000 and 8000 Btu/(hr)(ft²)(°F) are included as arbitrary curves. The magnitude of the mercury condensation coefficient is indicated by the region of overlap between the expected NaK heat-transfer-coefficient value and the experimental data band. The results indicate that the mercury heat transfer coefficient is greater than 10,000 Btu/(hr)(ft²)(°F) and larger than any expected value of the NaK heat transfer coefficient. The NaK heat transfer coefficient then is the more significant factor in affecting the overall heat transfer coefficient.

Inlet Pressure

Determining the characteristic of the mercury inlet pressure as a function of condensing length is important to the designer concerned with turbine performance and pump operation. Where the liquid mercury leg serves as an inventory for the system, the range and limits of condensing length operation must be considered. Finally, the characteristic of the pressure curve affects the type of control that might be considered for condenser pressure or inventory control.

Under saturated conditions, the inlet pressure of the condenser is related to the vapor temperature at the inlet by the pressure-temperature saturation relationship. The inlet temperature, in turn, is dictated by the heat transfer characteristics of the condenser. For a clearer insight into the determination of inlet pressure, then, it is necessary to discuss the factors contributing to

condensing heat transfer.

The basic heat transfer equation with regard to the condenser is:

$$Q = UA(LMTD)$$

Substituting the terms involved in the definition of LMTD and considering the heat transfer area as a function of the condensing length result in

$$Q = \frac{U f(l_c) \Delta T_{NaK}}{\ln \left(1 + \frac{T_{NaK}}{T_{Hg} - T_{NaK \text{ out}}} \right)}$$

With further manipulation the equation can be stated in terms of the NaK-mercury temperature difference at the condenser inlet:

$$T_{Hg} - T_{NaK \text{ out}} = \frac{\Delta T_{NaK}}{\left[e^{\exp \left(\frac{U f(l_c) \Delta T_{NaK}}{Q} \right)} - 1 \right]}$$

If the factors on the right hand side are known and $T_{NaK \text{ out}}$ is controlled or determinable, the equation can be solved for T_{Hg} . From the saturation curve the mercury inlet pressure can then be obtained.

In this investigation the mercury and NaK flow rates were constant. The temperature rise of NaK and the overall heat transfer rate, therefore, was constant ($\Delta T_{NaK} = 180^\circ$, $Q = 19,400$ Btu/hr) for the range of condensing lengths considered. From the previous discussion on heat transfer coefficients, U ranged from 2500 to 3150 Btu/(hr)(ft²)(°F). Substituting each of these values and the values of ΔT_{NaK} and Q into the equation results in the curves in figure 17. Each curve then represents an inverse logarithmic function of the NaK-mercury temperature difference with condensing length. On this same set of coordinates, the experimental values of the NaK-mercury temperature difference at the inlet have been plotted. The results indicate a close agreement of the data with the $U = 3150$ Btu/(hr)(ft²)(°F) curve for condensing lengths less

than 16 inches. The curves tend to converge at the longer condensing lengths.

An additional condition imposed on this system was that of maintaining a constant inlet NaK temperature. This, together with a constant NaK temperature rise, resulted in a constant NaK exit (condenser inlet) temperature. As a consequence, the shape of the NaK-mercury temperature-difference curve is virtually identical to that of the mercury inlet temperature. The mercury inlet pressure determined from the temperature curve, under saturation conditions, together with measured pressure data is shown as a curve in figure 18. The experimental data and the calculated curve, obtained independently, are in good agreement.

The pressure characteristic curve indicates a high sensitivity of the inlet pressure with the shorter condensing lengths. This would be disadvantageous if the condensing length were not controllable in this vicinity. This region would be of advantage, however, if an inventory control system were based on the response to change in the inlet pressure. At longer condensing lengths, the inlet pressure is relatively insensitive; therefore, a change in the condensing length, such as in inventory depletion, will have little effect on the inlet pressure. At the longer condensing lengths, however, pressure drop will increase, a fact which must be considered in the pump operation.

Pressure Drop

The results of the overall static-pressure-drop investigation are shown in figure 19. The data appeared to fall into two distinct groupings, approximately 0.4 psi apart. The discrepancy is still under investigation, but it is felt that the two groupings establish the limits of pressure drop within the set of conditions. In either case there was an overall pressure rise for practically all condensing lengths and a tendency toward a pressure drop was noted for the longer condensing lengths.

SUMMARY OF RESULTS

An experimental study of the condensing characteristics of mercury vapor flowing in horizontal single tubes yielded the following results.

Crossflow-Nitrogen-Cooled Condenser

1. Comparison of 1- and 0-g pressure drop for nonwetter condensing showed no significant difference for all flow rates investigated in constant diameter and tapered tubes.
2. For nonwetted condensing flow, there was an overall static-pressure drop in constant diameter tubes. For wetted and nonwetted flow in tapered tubes, there was an overall static-pressure rise.
3. There was no significant frictional pressure-drop difference between the wetted and nonwetted conditions at the lower flow rates, but there was a greater frictional pressure drop under a wetted condition at the higher flow rates.
4. The Lockhart-Martinelli correlation predicted the experimental values of the frictional pressure gradient accurately in the higher quality region of the condenser; however, the experimental values were higher than the predicted values of Lockhart-Martinelli in the lower-quality region. Though deviating from the predicted curves, the data points fell into a distinct grouping.
5. The experimental data compared with the predictions based on the fog-flow theory showed significant data scatter.

Counterflow-NaK-Cooled Condenser

1. The mean overall heat transfer coefficient, resulting from calculations of local heat transfer data that showed little variation over most of the condensing section, was found to range from 2500 to 3150 Btu/(hr)(ft²)(°F) for condensing lengths ranging from 7 to 32 inches.

2. For the conditions investigated, the mercury condensation coefficient was high (above 10,000 Btu/(hr)(ft²)(°F) for any expected value of NaK) and constituted a smaller thermal resistance compared with NaK.

3. For the operating conditions investigated, the inlet pressure was determined by the heat transfer characteristics of the condenser, which compared well with the experimental data. This resulted in an inlet-pressure curve that was sensitive to condensing length at the shorter condensing lengths and insensitive at the longer condensing lengths.

4. There was an overall pressure rise for most condensing lengths, with a trend toward a pressure drop at the longer condensing lengths.

TABLE I - RANGE OF VARIABLES FOR MERCURY CONDENSING EXPERIMENTS

Item	Crossflow-Nitrogen cooled					Counterflow-NaK cooled
	Gravity comparison tests			Wetted comparison tests		Nonwetting tests
Gravity level, g's	1 and 0	1 and 0	1 and 0	1	1	1
Wetted condition	Nonwetted	Nonwetted	Nonwetted	Nonwetted	Wetted	Nonwetted
Tube material	Pyrex	304 Stainless steel	316 Stainless steel	304 Stainless steel	Copper	9M
Tube diameter, in.	Constant 0.27, 0.40 0.49	Constant 0.31	Tapered 0.40 to 0.15	Tapered 0.50 to 0.20	Tapered 0.50 to 0.20	Tapered 0.45 to 0.20
Condensing length, in.	68*	45 to 72	45 to 72	18 to 45	18 to 43	7 to 42
Vapor mass flow rate, lb/sec	0.025 to 0.05	0.025 to 0.05	0.025 to 0.05	0.025 to 0.05	0.025 to 0.05	0.041 to 0.045
Vapor inlet quality	0.85 to 1.0	0.85 to 1.0	0.85 to 1.0	0.80 to 1.0	0.80 to 1.0	1.0
Vapor inlet pres- sure, psia	12 to 26	16 to 22	12 to 22	13 to 18	14 to 19	15 to 22
Vapor inlet veloc- ity, ft/sec	120 to 390	190 to 330	120 to 230	90 to 190	90 to 190	120 to 150
Vapor inlet Reynolds number $\times 10^{-4}$	2.0 to 5.5	3.0 to 5.0	2.0 to 4.0	1.5 to 4.0	1.5 to 4.0	3.0 to 4.0

*Single value

REFERENCES

1. Koestel, Alfred; Gutstein, Martin U.; and Wainwright, Robert T.: Study of Wetting and Nonwetting Mercury Condensing Pressure Drops. NASA TN D-2514, 1964.
2. Lockhart, R. W.; and Martinelli, R. C.: Proposed Correlation of Data for Isothermal Two-Phase, Two-Component Flow in Pipes. Chem. Eng. Prog., vol. 45, no. 1, Jan. 1949, pp. 39-48.
3. Albers, James A.; and Macosko, Robert P.: Experimental Pressure-Drop Investigation of Nonwetting, Condensing Flow of Mercury Vapor in a Constant-Diameter Tube in a 1-G and Zero-Gravity Environment. NASA TN D-2838, 1965.
4. Dwyer, O. E.; and Tu, P. S.: Unilateral Heat Transfer to Liquid Metals Flowing in Annuli. Nucl. Sci. Eng., vol. 15, no. 1, Jan. 1963, pp. 58-68.

(a) Schematic drawing of system.

(b) Schematic drawing of components.

Figure 1. - Experimental system and components for crossflow-nitrogen-cooled condenser.

Figure 2. - Flow configurations at interface location for 1- and 0-g mercury nonwetting condensation (3/8-in. o.d. glass tube).

Figure 3. - Effect of gravity on local static pressure drop. (Vapor mass flow rate, 0.038 lb per sec.)

Figure 4. - Effect of gravity on overall static pressure difference (vapor mass flow rate, 0.028 lb/sec).

Figure 5. - Comparison of nonwetting data with Lockhart-Martinelli correlation.

Figure 6. - Comparison of nonwetting data curve of this investigation with the nonwetting data curve of Koestel, et al.

Figure 7. - Comparison of nonwetting data with fog-flow correlation.

Figure 8. - Flow configurations at interface location for wetting and nonwetting mercury condensation in copper and stainless steel tubes, respectively.

Figure 9. - Effect of wetting on local static pressure drop (0.50-in. inlet i.d. with taper ratio 0.075 in./ft).

Figure 10. - Effect of wetting on overall static pressure difference (0.50-in. i.d. with taper ratio 0.075 in./ft).

Figure 11. - Comparison of wetting data with Lockhart-Martinelli correlation (tapered tube).

Figure 12. - Comparison of wetting data with fog-flow correlation (tapered tube).

Figure 13. - Experimental system and components for counterflow-NaK-cooled condenser.

Figure 14. - Temperature distribution of NaK and condensing mercury in counterflow.

Figure 15. - Typical plot of $\int U dA$ as a function of $\int dA$.

Figure 16. - Mean mercury heat transfer coefficient versus mean overall heat transfer coefficient.

Figure 17. - Condenser inlet temperature difference as a function of condensing length.

Figure 18. - Mercury inlet static pressure as a function of condensing length for NaK-cooled condenser.

Figure 19. - Overall pressure difference as a function of condensing length.

MERCURY RANKINE (SNAP2)
RADIATOR CONDENSER DEVELOPMENT EXPERIENCE*

by

V. H. HEISKALA
R. C. SMITH
E. A. ELLIOTT
R. T. LANCET

Atomics International
A Division of North American Aviation, Inc.
Canoga Park, California

ABSTRACT

Radiator condenser (RC) concepts which have been designed, fabricated, and tested at Atomics International for the Mercury Rankine Program (SNAP 2) are presented. These include conical steel fin-steel tube honeycomb, conical aluminum fin-steel tube, and flat copper fin-steel tube radiator condensers. The emissivity coating development program, which has been carried on concurrent to the RC development program, is outlined along with applicable coating test experience. Thermal and structural component testing and prototype system mockup test experience (PSM-1, PSM-3) are summarized.

Steady-state RC heat transfer, pressure drop, and stability characteristics are predicted and compared to experimental data resulting from system testing with a horizontal RC and from testing in a single tube condensing rig. One-dimensional and two-dimensional heat transfer models of the RC are shown and the relative accuracy of the two models is compared. The results calculated from these heat transfer models are shown to be in good agreement with experimental data. The pressure drop model, which is based on the fog flow theory of Koestel but uses the two-phase flow friction factor correlation of Baroczy and Sanders (Model 2), is recommended for mercury condensing pressure drop calculations. Results calculated with this model are compared to experimental data from testing both at Atomics International and other sources; the comparison is shown to be fair. Types of instabilities which can occur during RC operation are discussed along with criteria for determining the limits of stability. Results are discussed as a photographic study of instabilities encountered in a glass tube in a single tube condensing rig.

*Work performed under AEC Contract: AT(11-1)-GEN-8

Transient RC operation, including preheat, drain, and shutdown, is simulated with several analytical models. The preheat model is based on experimental observations of a moving, condensing interface in the RC tubes during the startup period; results calculated from this model are shown to compare favorably with experimental data from system testing. The drain and shutdown models couple RC heat transfer to system fluid dynamics in order to simulate system operation. The results calculated from these models are shown to be in fair agreement with experimental observations.

I. INTRODUCTION

In general, it may be said that radiator condenser (RC) designers will have to design a structure to dissipate some specified amount of heat for either a power system or an environmental control system. Typically, this heat will be expressed in terms of the temperature, pressure, quality, and flowrate of the incoming vapor and probably the required temperature and/or pressure of the outgoing liquid. Many different possible radiators could be designed to dissipate the required amount of heat under the specified conditions. It is the designer's task to choose the best one in terms of some optimization goal which must be fixed by the particular application. Minimum radiator weight is often the aim and sometimes minimum area at a reasonably low weight. Another choice, especially for a system powered by an expensive isotope, might be the minimum-cost point in a balance between fuel costs (efficiency) and launch costs (weight). All of these considerations must come into play when designing a radiator condenser for a specific application. For the SNAP 2 system, minimum system weight was the optimizing parameter. This paper presents, first of all, actual condenser design, fabrication, and testing experience gained at Atomics International (AI) under contract* to the Atomic Energy Commission (AEC); followed by sections dealing with the theoretical thermal, hydraulic, and transient analyses of radiator condensers, primarily those developed in conjunction with the SNAP 2 Program (now called the Mercury Rankine Program). In addition, a comparison of predicted radiator condenser performance with experimental data obtained from prototype Rankine system testing is presented.

*Work performed under AEC Contract AT(11-1)-GEN-8

II. RADIATOR CONDENSER DESIGN AND FABRICATION EXPERIENCE

A. RADIATOR CONDENSER REQUIREMENTS

1. SNAP 2 Requirements

Basically the RC serves two functions in the Mercury Rankine system. First, it provides containment and heat rejection capability for the mercury and, secondly, it provides basic structure for supporting the system components, i. e., reactor, shield, boiler, thermoelectric pump, combined rotating unit and other miscellaneous components.

Initial SNAP 2 design criteria required minimum weight for a RC that had a mercury flowrate of 19.8 lb/min, a maximum internal heat load (from radiation from internally mounted components) of 2.6 kw, total heat rejection load of 51.0 kw, a maximum pressure drop of 3.1 psia with inlet pressure and temperature of 9.0 psia and 625°F and outlet pressure and temperature of 5.9 psia and 371°F. The structural requirements included shock and vibration loads, aerodynamic loads, and inertia loads associated with an Atlas-Agena launch. Other design criteria included a one year life, armor to prevent puncture of the Hg circuit and geometrical limitations to maintain shielding protection from the reactor while achieving matchup to the boost vehicle. Operation in a zero and one g environment was also a requirement. There was no negative g condensing requirement. Figure 1 presents a schematic showing the general arrangement of the RC in the Mercury Rankine system.

2. Current MRP Requirements

Structural and configurational requirements for the present MRP are much less definitive due to the lack of specific flight applications. In order to provide as versatile a RC as possible, a modular or "building block" approach to RC design is being used; each module has the capability of being integrated into a structure and/or configuration that will satisfy particular performance criteria. Figure 2 presents an element typical of that planned for future use and shows how these elements can be integrated into a particular configuration. At present, development is continuing on the basic tube/fin element.

B. RADIATOR CONDENSER CONCEPTS

The development of the RC concepts followed two parallel paths consisting of a honeycomb RC and aluminum-steel development. The honeycomb design had the advantages of a large strength margin, and a large background in honeycomb technology. The aluminum-steel concept offered significant weight savings, lower

cost, and greater flexibility. The development of these RC's, leading to the final selection of the aluminum steel concept, is shown in Figure 3.

1. Conical Steel Honeycomb

One of the first RC concepts that was developed was the conical steel honeycomb type shown in Figure 4. PH 15-7 Mo was used throughout this design except for the fins which were O. F. H. C. (oxygen-free high conductivity) copper. The mercury circuit was comprised of 72 round, non-tapered PH 15-7 Mo tubes welded to a vapor manifold at the top and liquid manifold at the bottom or large end. The manifolds were also of PH 15-7 Mo material. Brazing was used throughout in joining tubes, face sheets, core and fins. Quarter panel construction was used, i. e., the RC was made up of four individual panels, each an individual braze unit. The quarter panels were joined into a RC cone at the 4 longitudinal joints by riveting after the liquid manifold segments had been welded. The vapor manifold was then introduced and manually heliarc welded in place. Two RC-6 units were made.

The second phase of the steel honeycomb concept emphasized improved braze quality, reduced number of tubes in the Hg circuit (higher RC pressure drop), tapering of tubes, and tapering of the copper fins. The first unit in this second phase, which is shown schematically in Figure 5, was used for structural testing. Fabrication of the second unit was not completed.

2. Conical Aluminum Fin/Steel Tube

The first aluminum fin/Haynes-25 tube design (Figure 6) used 1100 Al skin, Haynes-25 (L-605) tubes and gussets and 6061 Al frames. A second design, as shown in Figure 7, which represented an updated and optimized version of the first Al/H-25 RC was released for fabrication. The principal differences in the later design were: (1) thinner skin gauge — 0.040 in. and 0.030 in. vs 0.063 in., (2) less radiating surface — 120 ft² vs 134 ft², (3) more tubes — 32 vs 24, and (4) use of electron beam welded tubes.

The principal significance of this program, in addition to delivery of one RC-10 and fabrication of five RC-13 quarter panels, was the successful braze joining of aluminum to Haynes-25 with a ductile joint. A diffusion barrier of Ti plating on the tube provided the interface upon which the aluminum braze could be joined thus preventing the formation of brittle intermetallics of Al and Haynes-25.

3. Flat Copper Fin-Steel Tube

Two non-flight type RC's were designed in a flat configuration for use in ground system tests to more closely simulate a zero g condition. The first design utilized

a single sheet of copper for the fin and had 44 tubes of round, nontapered configuration brazed to the single fin plate. The tubes and associated manifolds at either end were of PH 15-7 Mo. This design (RCH-1) experienced considerable buckling and warping in the course of testing; however, the Hg circuit remained intact and the RC performance was satisfactory in excess of 1,000 hr. A refined version of this RC (RCH-2) shown schematically in Figure 8, uses tapered RC-11 tubes and individual double fins for each tube (fins on both sides of each tube). RCH-2 has been incorporated into a simulated zero g system (PSM-3) which is currently being tested.

C. EMISSIVITY COATING DEVELOPMENT

The special requirements of a system operating in a space environment creates unique problems in the development and qualification of radiation control coatings. A special coating system, designated AI-93, has been developed and tested for the outer surface of the radiator condenser. This coating which is a composite, consists of a sub-coat of chromium-cobalt nickel spinel bonded with aluminum phosphate, and a top coat of stannic oxide bonded with aluminum phosphate. The application temperature range of the coatings is 450 to 600°F in either an air or argon atmosphere. The coating exhibits a thermal emittance of 0.91 (400 to 700°F) and a solar absorptance of 0.35. Current experience at AI has shown the coating can be applied to at least three metallic substrates.

The stability of the coating in a nuclear and simulated space environment has been demonstrated as well as its capability to withstand the severe conditions of shock, vibration and aerodynamic heating which would be imposed upon it during the SNAP 2 system launch. The coating's stability has also been demonstrated in over 1,000 hr of actual space flight and an actual system launch while being used on the SNAP 10A flight. In addition, the coating has been tested in excess of 6,000 hr in FS-3, the ground test equivalent of the SNAP 10A flight item.

In component tests, no emittance degradation occurred in 8,000 hr coating stability tests in a vacuum. The coating has been exposed to nuclear irradiation (3×10^{18} n/cm², 1.9×10^{10} R) at 600°F in a vacuum with emittance remaining constant throughout the test. The coating system has also been subjected to electron bombardment without any detrimental effects. During this latter test, the coated sample was maintained at 600°F and received an integrated electron dose of 3×10^{18} , 1.25 Mev energy. Ultraviolet stability tests of 3,000 hr duration are complete with the following results:

1,000 hr — solar absorptance = 0.39

3,000 hr — solar absorptance = 0.46

This increase in solar absorptance could have a significant effect on RC heat rejection capacity and, thus, turbine exhaust pressure for a long duration mission; the influence of this effect on system power degradation would have to be taken into account when evaluating the performance capability of a real system.

D. CURRENT RC DEVELOPMENT STATUS

The current direction on the RC development program is to pursue the development of the Al-steel concept along the lines of a "building block" or modular concept (See Figure 2). This concept, i. e., individual tube-fin elements, will be developed to a point where the structural and thermal performance of the module is documented and proven. The approach permits a lower development cost and greater versatility, simply because any number of modules can be joined into the desired configuration and sizes to fit a wide variety of mission applications. The present effort on the aluminum fin-alloy tube concept is directed at evaluating and improving the braze bonded joint between the aluminum and steel and developing an alternate method of joining the materials. The braze joint requires a titanium plate to act as a diffusion barrier between the aluminum and tube alloy to prevent the formation of brittle Al-alloy intermetallics. The alternate approach is the use of diffusion bonding of the Al-alloy with a nickel plate on the tube to act as a diffusion barrier. The latter technique is currently being developed and optimized for time, temperature and pressure. Also, methods of stabilizing the tube to prevent buckling or crushing are being evaluated.

The braze bond has been extensively tested for time and temperature effects and found satisfactory within the limits tested, i. e., 2100 hr and 750°F. The quality of the joint, as delivered, was not as high as desired and techniques have been developed to improve this quality. Detailed analytical evaluation, however, has shown that the braze qualities obtained, more than adequately meet structural and thermal requirements.

A second problem area with the braze bond joint concerns the use of the titanium plate on the tube. For applications using a reactor heat source, the presence of hydrogen in the system and its permeation through the system results in the Ti plate acting as a getter. This ultimately results in the formation of titanium hydride which in turn is considerably more brittle than the initial braze joint. Tests are currently in progress to establish hydrogen permeation rates and titanium hydriding rates and the quantitative evaluation of the embrittling effects on the joint. This problem would not exist with an isotope system.

III. RADIATOR TESTING

A. COMPONENT TESTING

Figure 9 presents a summary of the types of tests which have been conducted on the various RC's which have been built.

The experience gained to date would allow the design of flight components with a high degree of confidence. The work which is continuing is primarily aimed at improved produceability, reliability, cost, adaptability, etc.

On a subcomponent level, the testing has been primarily directed at evaluating bond quality per se as well as the effect of thermal cycles and extended time in thermal environment on bond quality. Other testing on a component level has covered the evaluation of brazing of bimetallic joints and the effect of the presence of hydrogen in the system on the aluminum-steel joint with a titanium diffusion barrier.

Preliminary results of the hydrogen diffusion rates show no change in the rate of diffusion through the Haynes material as a result of the titanium plating because the titanium is found to act as a hydrogen sink. However, since the titanium plate does act as a getter, it prevents any hydrogen from escaping to the vacuum side of the test specimen. Also, impact tests on the joint show a significant increase in brittleness in the titanium plate layer in the braze joint. Results of the diffusion bonding effort show very good bond quality with the nickel plate on samples up to three feet in length. A second plating system using copper and nickel plating layers also shows very good bond quality on a three foot long sample.

B. SYSTEM TESTING

Two Prototype System Mockup (PSM) performance test systems evolved in conjunction with the SNAP 2/MRP program. These are PSM-1, which was a non-nuclear, flight configured system test and PSM-3, which was a non-nuclear system test, arranged to provide a zero gravity simulation.

1. PSM-1 (Flight Configured)

The PSM-1 system employed RC-6, a conical steel honeycomb, steel tube, copper fin RC. Major test results are shown along with a photograph of the system in Figure 10.

2. PSM-3 (Zero Gravity Simulation)

The early stages of PSM-3 system testing used RCH-1, 44 nontapered steel tubes brazed to a single horizontal flat copper fin; current PSM-3 system testing employs RCH-2, 40 tapered steel tubes, each tube brazed on each side to tapered copper fins. Major test results are summarized in Figure 11; some temperature and pressure drop data for the RC during the system startup phase are compared to analytical predictions in a later section.

IV. STEADY STATE ANALYSES

A. THERMAL PERFORMANCE

A comparison between the experimental data and analytical predictions for RC-6 is shown in Figure 12. The one-dimensional analysis^{*} predicts a slightly longer condensing length and a lower mercury exit temperature due to the omission of axial conduction along the fin. However even the considerably more complex two-dimensional analysis^{*} appears to predict too long a condensing length. The condensing length prediction was considerably improved when allowance was made for internal radiation from the hot portion of the RC to the cooler subcooler. However, the subcooler exit temperature prediction was only slightly improved. Because of the very low heat capacity of mercury liquid (approximately 0.0324 Btu/lb-°F), the exit temperature is very sensitive to system tolerances and experimental inaccuracies. The exit temperature discrepancy of 34°F represents only about 0.84% of the total heat rejected.

Because of the uncertainties associated with the surface emissivities, mercury flowrate and quality, braze quality, copper fin thickness, tube pressure drop, heat transfer coefficients, nonuniform sink temperatures and the analytical model, the analytical predictions are only expected to be good within $\pm 50^\circ\text{F}$ in mercury exit temperature. In the SNAP 2 system design, allowance was made for a possible $\pm 68^\circ\text{F}$ variation in RC exit temperature; of this, $\pm 20^\circ\text{F}$ is due to sun-shade variations and $\pm 48^\circ\text{F}$ is due to the above applicable uncertainties. Allowing for this variation imposes no particular problem, simply a small weight penalty. The discrepancy in the condensing region is larger than usual and is probably due to the combined effects of non-condensibles and a discrepancy between the predicted and actual pressure drop.

Figure 13 shows a similar set of curves for RC-10 with similar results. The difference between the two- and one-dimensional analysis here is felt to be due more to the inclusion of internal radiation in the two dimensional analysis rather than the difference between two-dimensional and one-dimensional approaches. The excellent agreement in exit temperature (in the two-dimensional analysis) is somewhat fortuitous since, as noted previously, the expected accuracy is only about $\pm 50^\circ\text{F}$. It should be noted that the agreement in the condensing region is

^{*}See Appendix B for details of this analysis.

considerably improved over the RC-6 results. Figure 14 shows a comparison between four tubes in the flat RCH-2 indicating significant differences in performance. Tubes 1 and 40 are end tubes and have a considerably larger view of the vacuum tank wall resulting in larger flows to these tubes and, consequently, higher exit temperatures than the internal tubes. The differences between tubes 20 and 10 are probably due to an uneven flow distribution since they have very similar thermal environment. While the maximum variation in exit temperature from these RCH-2 tubes is about 100°F, the measured exit temperatures of the conical radiators, when tested as separate components, differed by less than 20°F. The close agreement of the latter is partially due to the uniform thermal conditions and partially due to the strong static head effect which forces the subcooler lengths to be almost identical. In summary, it appears that the two dimensional analysis employed here is adequate for predicting the performance of the radiator condenser. Furthermore, a one dimensional analysis appears adequate for the less stringent requirements of system studies and, with further modification to include internal radiation interchange, it may be adequate for detailed component design.

B. INSTABILITY

The Mercury Rankine Program specifications are for zero g and one g operation where the gravitational forces are in the direction of the condensing flow. No attempt was made to design for large negative g forces. Therefore, the instability limits, discussed below, for g forces which are opposed to the condensing flow (negative g forces) represent design margins for this particular design rather than limits of design capability.

Condenser instability is very undesirable since in its worst forms it can have catastrophic effects on system performance. The primary effects of these instabilities are to cause large pressure fluctuations or even to cause large slugs of vapor to enter the effluent liquid stream. Either of these effects could cause large fluctuations in system performance or premature system shutdown due to pump or bearing failure. The major forms of instability that have been examined analytically and experimentally are discussed in the following sections.

1. Types of Instabilities

Four types of instability have been examined in the literature and under the Mercury Rankine Program. These consist of interface instability, runback instability, zero g instability, and liquid leg instability. The first two are single tube phenomena while the last two are peculiar to multiple parallel tube operations.

a. Interface Instability

Interface instability, the inability to maintain a relatively stable meniscus, does not appear to be a critical parameter in itself in a condensing system which has restoring forces in the form of vapor drag and/or impinging droplets. The primary influence appears to be in promoting nonwetting runback instability. The nature of the instability and the experimental determination of the critical Bond number are discussed in greater detail in References 1, 2 and 3.

b. Zero G Instability

This instability may occur in a system of parallel condensing tubes and results in the liquid leg of a perturbed tube spilling into the vapor manifold. The resultant effect could be a significant pressure fluctuation but no vapor should be entrained in the effluent liquid stream. The existence of this form of instability is primarily a function of the condensing length, the entrance pressure loss, and the degree of condensing vapor momentum recovery. This form of instability can only occur if there is a negative pressure drop; that is, the pressure rise due to vapor momentum recovery must exceed the frictional losses.

c. Liquid Leg Instability

This is the most severe form of instability as it may result in liquid spilling into the vapor manifold and vapor spilling into the liquid manifold. If the vapor in the liquid manifold does not condense prior to reaching the pump, it may deprime the pump shutting down the system. If the vapor does condense, it may raise the temperature of the liquid enough to reduce the net positive suction head to the pump below the critical value to maintain adequate pump performance; this will result in parallel tube instability when the acceleration forces opposing condensing exceed a critical value. The critical value is primarily a function of the condensing length, flowrate, and pressure drop.

d. Runback Instability

Runback instability has been noticed in wetting system condensing against gravitational forces. Runback occurs when the vapor drag is no longer capable to support the weight of the liquid in the annular film and the liquid bridges the tube trapping vapor behind it and shortening the condensing length. The net effect can be large pressure fluctuations which are detrimental to system performance.

In non-wetting mercury systems a runback type phenomenon can occur in tubes with unstable interfaces with similar consequences in terms of large pressure fluctuations. In this case the interface breaks down and runs upstream until the pressure and shear forces become large enough to slam the liquid back up to the interface.

2. Predicted Stability Limits and Experimental Observations

a. Interface Instability

For mercury the critical tube diameter above which the interface is unstable is given approximately by Reference 4

$$D_{cr} = 1.84 \sqrt{\frac{\sigma}{g/g_c (\rho_L - \rho_V)}}$$

In this equation the constant actually varies slightly depending upon tube material and surface finish.

In all the MRP applications considered to date, the interface in the condensing tubes is stable either because in space the acceleration level is essentially zero and hence the tube diameter is below the critical diameter and because the direction of any applied acceleration is from the vapor to the liquid, therefore inherently stable. As an example of the negative g design margin consider a local acceleration of 0.05 g's, the critical diameter would be about 4.5 times the value on earth. From the previous consideration, the critical value would be from 0.56 to 0.75 in. The increase in critical diameter was verified in Reference 2, where tubes of 0.625 in. diameter and 0.5 in. diameter, filled with water and Hg respectively, were found to have stable interfaces in a near zero-g environment. Since typical 10-ft-long MRP tube sizes run from 0.25 to 0.35 in. at the vapor inlet, to 0.125 to 0.20 in. at the liquid outlet, the tubes are well below the critical diameter. It was found that in experiments with rectangular tubes,⁽⁵⁾ the critical dimension is along the gravitational force vector. With the tube in a horizontal position, the horizontal dimension could be above critical but the vertical dimension governed the stability of the interface. On the other hand, if the interface is horizontal, both dimensions must be below the critical size to maintain a stable interface. Reference 5 found the critical dimension to be 0.18 in. for room temperature Hg in the glass ducts.

b. Zero G Interface Stability

To simplify the pressure-drop equation for the purpose of examining the factors influencing condensing stability at zero g, incompressible flow is assumed. To account for the fact that condensation occurs and the stream velocity varies along the condensing tube, it is assumed that:

- 1) Area = constant, not varying with distance

$$2) U_{(x)} = U_o^1 (1 - x/L_c)$$

$$3) U_o^1 = U_o (L_c/L_{oc})$$

While this represents a very special case, the major conclusion can be shown to be essentially independent of these particular assumptions.

Based on these assumptions, the resulting pressure drop equation is:

$$P = P_0 + [2 - y - K] \frac{\rho U_o^2 L_c^2}{2g_c L_{oc}^2} - \frac{4\rho U_o^2 L_c^2}{2g_c D L_{oc}^2} \int_0^1 \phi^2 f L \left(1 - \frac{x}{L_c}\right)^2 d\left(\frac{x}{L_c}\right) \quad (1)$$

which gives for the interface pressure, assuming an average ϕ^2 ($\bar{\phi}^2$) and an average f ($\bar{f}$),

$$P_i = P_0 + \left[(2 - y - K) \frac{L_c^2}{L_{oc}^2} - \frac{4L_c^3 \bar{\phi}^2 \bar{f}}{3 D L_{oc}^2} \right] \frac{\rho U_o^2}{2g_c} \quad (2)$$

Equation 2 shows that the condenser will be completely stable as long as $(2 - y - K) < 0$, or if the condensing length is greater than the critical condensing length when $(2 - y - K) > 0$. By differentiating Equation 2 and setting it equal to zero, the critical length is found to be

$$L_{cr} = \frac{(2 - y - K) D}{2 \bar{\phi}^2 \bar{f}} \quad (3)$$

Figure 15 is a plot of Equation 2, showing the interface pressure on the vapor side vs condensing length for equilibrium conditions at each condensing length. From Figure 15, the criterion for stability in zero g when one tube out of many is perturbed is seen to be:

$$\frac{dP_i}{dL_c} < 0$$

Figure 15 shows that it is possible, for small momentum losses during condensing, to have an interface static pressure which is greater than the incoming total pressure, hence the interface might move toward the inlet. To match the pressure drop data of Reference 6, it was empirically determined that the condensate must

give up an average of 28.5% ($y = 0.57$) of the mean stream momentum flux. For an acceleration and entrance loss factor (K) of 1.5, this would require the static interface pressure to be less than the inlet total pressure by seven percent of the inlet dynamic head, thus eliminating the possibility of this type of instability. However, more refined analytical models predict that this factor (y) is much closer to 10%, permitting the interface pressure to be greater than the inlet total pressure.

If the latter case turns out to be true, it will be necessary to maintain the condensing length greater than that given by Equation 3. For the MRP reference design, for $y = 0.1$, Equation 3 gives:

$$L_c \approx \left(\frac{2 - 0.1 - 1.5}{0.02} \right) D = 20D$$

During steady-state operation, the MRP condensing length is about 110 in. During startup, however, the condensing length decreases until the RC is heated and the condensing pressure reaches the steady-state condition. The minimum condensing length during startup of MRP condensers is about 20 in., or an $L/D \approx 65$, and no zero-g condensing instabilities are expected. This has been verified by flat horizontal RC's, and is true even when allowing for the errors resulting from the simplifications introduced during the derivation of Equation 3.

c. Liquid Leg Instability

Liquid leg instability occurs in a multiple tube condenser whenever the pressure gradient in the liquid exceeds the pressure gradient in the vapor, or:

$$\left| \left(\frac{dP}{dL_c} \right)_{\text{liquid}} \right| > \left| \left(\frac{dP}{dL_c} \right)_{\text{vapor}} \right|$$

The liquid pressure gradient is given by:

$$\left(\frac{dP}{dL_c} \right)_{\text{liquid}} = \rho n = 0.471 n \text{ psi/in.}$$

where ρ for mercury has been evaluated at 450°F.

The vapor pressure gradient was calculated using pressure drop Model 1 discussed in Section IV-C and the conditions given in Table 1. The variation with interface location is given in Figure 16. The actual location of the interface is established at the point where the vapor side interface pressure, as determined by the thermal and fluid dynamic behavior, is equal to the liquid side pressure, as determined by the acceleration and the liquid head between the pressure regulator and the liquid vapor interface. Figure 17 shows the resultant vapor side interface pressure and liquid side interface pressure as a function of condensing length. Cross-plotting the intersection of the liquid and vapor curves in Figure 17 gives the limits for stable operation. When the pressure regulator is located at the zero g interface level (108 inch condensing length in this case) the interface will remain at that level (for low acceleration) and the critical dP/dx from Figure 16 is 0.085 giving a critical acceleration of -0.18 g's. With the pressure regulator located 6 in. below the zero g condensing level the average interface location at the critical g level can be shown to be approximately 106.5 in. giving a critical acceleration of 0.16 g's. It can therefore be concluded that this particular radiator condenser design is stable against liquid leg instabilities for negative acceleration below 0.16 g's under the conditions given in Table 1.

TABLE 1
CONDITIONS FOR STABILITY ANALYSIS

Tube height	0.270 in.
Bottom tube width	0.125 in.
Top tube width	0.218 in.
Tube length	120 in.
Dog leg length	1.9 in.
Dog leg area change	0.0156 sq. in.
Dog leg entrance loss coefficient	0.5
Momentum loss factor (γ)	0.57
Flowrate (40 tubes)	19.8 lb/min

d. Runback Instability

Runback under wetting conditions was analyzed by Koestel (Reference 7). He assumed that runback occurred at the point where the wall shear went to zero. He found that the distance to the point of runback could be represented by the relation:

$$\frac{\left(\frac{x}{L_c}\right)^{1/3}}{1 - \frac{x}{L_c}} = \frac{204.5 \rho_V \bar{U}_o^2 \left(1 + \frac{2d \text{Re}_o}{64 L_c}\right)^{2/3}}{\rho_L^{1/3} \mu_V^{2/3} n^{2/3} \text{Re}_o^{4/3}} \quad (4)$$

From the experimental data, Koestel estimated that $L_b \approx 0.8 L_c$ would minimize the effects of runback slugging in tubes of diameters ≤ 0.5 in. This reduces Equation 4 to:

$$U_o = 308 (\rho_L \rho_V)^{1/2} \frac{d^2 n}{\mu_V} \quad (5)$$

where U_o is the required inlet vapor velocity to maintain a stable flow condition.

In the case of the MRP, if wetting of the condenser tubes occurs, the above criterion predicts stable operation (small pressure fluctuations) for a negative acceleration up to 0.078 g's. Nonwetting runback is not expected in the MRP applications since in all cases the tube is below the critical value for an unstable interface.

3. Experimental Observations

Experimental investigations of simulated zero g condenser stability have been carried out in a special mercury condensing rig and in the horizontal multiple tube radiator in PSM-3. Testing in the mercury condensing rig to date has been limited to single tube condensing. The rig is capable of condensing uphill or downhill at angles from +20° to -12° in a 10-ft tube. The boiler is capable of handling up to 20 lbs/minute. The multiple tube radiator in PSM-3 is capable of being tilted, while operating, through an angle of +2.7° to -0.9°. Figure 18 presents a summary of the type and quality of data obtained during the SNAP 2 and subsequent Mercury Rankine Program.

During the single tube tests it was observed that the critical (capillary) diameter as predicted for Hg does indeed have the main stabilizing influence upon surface stability at the liquid-vapor interface. Films taken indicated that the 0.125 in. diameter tube held a stable interface under all orientations tested; that even when large disturbances were created upon that interface by incoming droplets, the disturbances were rapidly damped.

Studies of the liquid-vapor interface in the above-critical-diameter tube also showed the expected phenomena. In a horizontal or "downhill" condensing

position, there were no forces that could cause a droplet (thrown from the interface by the surface waves) to move upstream. The interface therefore held an overall equilibrium position, although the interfacial tension was not sufficient to immediately damp all perturbations. However, in an "uphill" condensing orientation, droplets which separated from the interface were moved upstream by gravitational forces. Since the droplet was then moving against a flux of condensing vapor, its motion was eventually reversed and it was given a large momentum toward the interface.

Large amounts of liquid were released in the phenomenon being described. There appeared to be no actual separation of the interface into droplets, but the interface became elongated. This could be due to the time limitations, since it is known that the equilibrium configuration is a group of spherical droplets, but it is also known that the time to reach equilibrium is quite large. The upstream end of the interface, which had been exposed to the nonequilibrium conditions for the longest time, did actually form a group of spherical droplets. Some of these droplets moved in an oscillating pattern along the tube axis, moving upstream under gravity until the vapor momentum flux reversed the motion, then moving toward the interface until gravitational forces reversed that motion. This phenomenon is peculiar to non-wetted runback. Somewhat downstream of these oscillating droplets, a continuous but quite elongated interface exists. This interface distorts as liquid runs into its upstream end. As the upstream end grows in size (and mass), it moves farther upstream and encounters more vapor flux. The shear and momentum forces of the vapor deform this interface until it bridges the tube, at which time it receives a large impulse from the vapor, driving the entire liquid runback downstream into the stationary liquid leg. When the runback is slammed into the liquid leg, a large pressure pulse is transmitted into the liquid. Pulses as great in magnitude as ± 2.5 psia were observed. (The minus readings may be due to instrumentation oscillations rather than actual pressure fluctuations.)

Besides the fact that the 0.125 in. ID tube and 0.25 in. ID tube exhibited the predicted behavior typical of below and above critical diameter tubes, no analytical support of the previous observations has been obtained; however, the multiple-tube radiator has been run in a horizontal position and no instabilities have occurred at or near design flows to date.

Since non-wetting runback occurs only in above critical diameter tubes in a negative g environment, the Mercury Rankine Program does not suffer from this instability. In fact, so far, no form of condensing instability has been found to exist within the MRP system specifications.

C. PRESSURE DROP

The accuracy of the pressure drop prediction influences the RC area and consequently the weight since one of the tolerances which must be allowed for is the uncertainty in the condensing pressure. In addition, the pressure drop analysis is used in all the stability evaluations, and errors in the pressure drop analysis will be reflected in errors in the predicted margin for stable operations.

Three condensing pressure drop models have been investigated. The first two use the improved Lockhart-Martinelli correlation of Baroczy;⁽⁸⁾ the third uses the Fog-Flow correlation of Koestel. Model 1 has been used with moderate success in the past by AI to predict the total pressure drop for design purposes. Experimental data has shown that it does not predict the pressure profile along the tube very well; however, the profile is not as important for design purposes as the total pressure drop. Model 2 is a fog-flow model that uses the Baroczy correlation and predicts closely the same pressure drop as Model 1 and the pressure profile matches the data much better. The recently introduced Fog-Flow Model of Koestel has not been completely evaluated as yet to predict pressure drop. This model shows some merit mainly due to its simplicity. Model 1 and the Fog-Flow Model will only be discussed briefly below; Model 2 will be discussed in more detail.

1. Analysis

a. Improved Lockhard-Martinelli Correlation of Baroczy and Sanders

To date, the most accepted method for predicting pressure drop in isothermal two-phase flow in horizontal pipes is based on the Martinelli method.^(9, 10) This method relates the two-phase pressure drop (ΔP_{TP}) to the pressure drop that would occur if the vapor phase were flowing alone by

$$\Delta P_{TP} = \phi^2 \Delta P \text{ host fluid}$$

The coefficient ϕ^2 is correlated to the Martinelli parameter which is the ratio of the pressure drop of the liquid flowing alone to the pressure drop of the vapor flowing alone.

This correlation was found to be improved with the introduction of another parameter, the vapor Reynold's number.⁽⁸⁾ This improved correlation is used in both Models 1 and 2.

b. Condensing Pressure Drop Model No. 1 ^(8, 11)

For Model 1 the following assumptions are made:

1) The control volume includes both the liquid and vapor flow; however, the momentum of the liquid is assumed to be negligible, so that only the vapor is considered in the momentum terms.

2) The liquid volume fraction is very small so that the vapor flow area is taken to be the cross-sectional area of the tube.

3) No body forces are present since space operation with no acceleration is assumed.

4) The velocity of the liquid is assumed to be negligible.

The analysis resulting from these assumptions is presented completely in Reference 12. The results predicted from this model are compared to experimental data in Figure 19.

c. Condensing Pressure Drop Model No. 2

When the momentum of both the liquid and vapor are considered, the pressure drop equation can be written:

$$-A dP - \tau_w dA_w = \frac{1}{g_c} \left[w dU_V + w_L dU_L \right] + (U_V + U'_V) dw + (U_L - U'_L) dw_L \quad (6)$$

Assuming fog flow $U_L = U_V$ and zero initial entrainment velocity $U' = 0$, Equation 6 becomes

$$-A dP - \tau_w dA_w = \frac{1}{g_c} \left(w_t dU - U_y dw \right) \quad (7)$$

or in terms of gradients (see Reference 12)

$$\frac{dP}{dx} \left(\frac{w_t^2 \psi}{\rho A^2 g_c} \frac{d\rho}{dP} - 1 \right) = \frac{1}{g_c} \left(\frac{\phi^2 f}{D} \frac{w_t^2 \psi^2}{2 \rho A^2} + \frac{w_t^2}{\rho A^2} \frac{d\psi}{dx} - \frac{w_t^2 \psi}{\rho A^3} \frac{dA}{dx} - \frac{w_t^2 \psi}{\rho A^2} y \frac{d\psi}{dx} \right) \quad (8)$$

There is a difference in the momentum terms in Models 1 and 2. The first right hand term of Equation 8 includes the total weight flow, whereas in Model 1 only the vapor flow was included. Since the vapor flow decreases along the length of the tube, less momentum is recovered in Model 1 for the same velocity gradient. Figure 20 is a normalized plot of the momentum terms for the two models vs vapor quality. For simplicity, incompressible flow is assumed. Although the total momentum to be recovered is the same for both models (same area under curves), greater momentum recovery occurs at first in Model 1, thus reducing the slope of the pressure profile. Towards the end of the condenser, little momentum recovery is left. In Model 2, the momentum recovery is spread evenly over the length of the tube. The initial pressure gradient is steeper than in Model 1, but toward the end it is less steep. In most cases, the momentum recovery toward the end is larger than the frictional gradient so that the pressure gradient becomes distinctly positive, with the minimum pressure located approximately two-thirds of the condensing length.

In incompressible flow, either model would suffice for the total pressure drop since ϕ^2 , U , etc., would vary similarly along the tube and the momentum recovery is the same. But in compressible flow, the density drops more rapidly in Model 2 since the pressure gradient is steeper. The local vapor flow rates, however, are similar, so that higher velocities occur in Model 2, hence an unrecoverable steepening in the pressure gradient occurs. The overall effect is to increase the pressure drop over that predicted by Model 1. The increase is more noticeable when the ratio of the pressure drop to the pressure level is large.

d. Fog-Flow Condensing Pressure Drop Model

A second means for predicting the frictional pressure gradient in condensing flow of a nonwetting fluid was proposed by A. Koestel.⁽¹³⁾ Due to the simplicity of the model, an analytic value of the two-phase factor can be found:

$$\phi^2 = \left(\frac{D}{D_c}\right)^{4.75} \frac{1}{\psi^{0.75}} \quad (9)$$

where

$$\frac{D}{D_c} = f \left(W_{eT} \right) \quad (10)$$

$$\text{the tube Weber number} = \frac{\rho U^2 \delta_{cr}}{2 \bar{U} g_c} \quad .$$

The calculation of D/D_c involves a prediction of the drop size variation along the tube and is covered in detail in Reference 13.

Close to the interface (roughly the last 20% of the condensing length), the predicted diameter of the entrained droplet becomes very large, eventually exceeding the diameter of the tube and approaching infinity at the interface where the velocity is zero. This large droplet size is due to the fact that the total lift force of the vapor on the liquid droplet is dependent upon the droplet surface area. Furthermore, the test correlation for the droplet liftoff model was a very large flat plate where the droplet size did not significantly change the flow pattern as it does in a tube. In this region, ϕ^2 rapidly approaches infinity due to the reduction in the core diameter (D_c), and then becomes incalculable due to the negative value for the core diameter ($D_c = D - 2\delta_{cr}$). As a temporary remedy for this problem, limits have been set on the droplet diameter so that this model can be currently used to predict pressure drop. For the predictions presented in this report, ($\delta_{cr} \leq 0.1 D$) has been used.

Furthermore, the Reynolds numbers for this region are generally less than 2000, indicating that viscous flow exists. A laminar fog-flow, two-phase factor can also be obtained by the same method. In the development, the quality term vanishes and ϕ^2 becomes a function of (D/D_c):

$$\phi^2 = \left(\frac{D}{D_c} \right)^4 \quad (11)$$

But still, one is plagued with the problem of the diameter ratios and the questionable assumption that the mixture viscosity is equal to the vapor viscosity. The assumptions for the fog-flow model are the same as for Model 2 and the resulting momentum equation is identical, except that ϕ^2 is calculated using Equation 11.

2. Comparison of Models

A comparison of the three models (with y equal to zero) with experimental pressure drops taken from Reference 6, is shown in Figure 19. The effect of the difference in the momentum terms of Models 1 and 2 is clearly seen. The pressure gradient for Model 1 is not as steep due to the high initial momentum recovery, and the profile is fairly flat towards the end of the tube where the frictional gradient and momentum recovery are small. The opposite is true of Model 2; the initial gradient is steep, but the recovery toward the end is large.

The Fog-Flow pressure profile shows the same trends as does Model 2, since their equations are similar except for the evaluation of the two phase factor ϕ^2 .

Note also that the experimental data shows the same trends as Model 2 and the Fog-Flow Model; that is, a steep initial profile, a minimum point, and a considerable rise in pressure. The difference between Model 2 and the Fog-Flow Model is attributed to the large values of ϕ^2 in the latter. Note the gradients near the interface are steeper for Model 2 than the experimental data and less steep for the Fog-Flow Model. This is attributed to the lower ϕ^2 factor in the improved Lockhart-Martinelli correlation, and the extremely high ϕ^2 factors in the Koestel correlation. The ϕ^2 inferred from the data appears to be somewhere between the two. Some of the discrepancies between the models and the data may be due to flow measurement error. At the present time it is felt that Model 2 best describes the condensing pressure variation for both wetting and non-wetting fluids and is being used by AI in the design of RC's.

3. Comparison to Experimental Data From RCH-2

Current Mercury Rankine Program system testing (PSM-3, Phase 1-F) uses RCH-2, a flat, horizontal radiator condenser. In order to predict pressure drops to compare with data from this system, it is necessary to have an estimate of the momentum recovery factor, y . Previously, Model No. 1 was used to simulate condensing data taken from Reference 6. The y factor was allowed to vary in that calculation until a match could be made between experimental data and predicted results. The resulting y factor is shown as a function of pressure (the data were grouped at several pressures) in Figure 21. This figure shows that y is strongly pressure-dependent, decreasing in value with increasing pressure. The condensing pressure for RCH-2 is usually in the neighborhood of 8 psia so that y can be taken as 0.57 in this instance.

Model No. 2 was used to predict RCH-2 pressure drops for a variety of experimental data points taken from system testing. The pressure drops were calculated both for $y = 0.57$ and $y = 0$ (complete momentum recovery). Unfortunately, system pressure instrumentation was not accurate enough to provide meaningful pressure drop data; however, pressure drops were calculated from temperature measurements taken in the vapor manifold and at the end of the condensing region assuming saturated conditions. Since the computer program of Model No. 2 does not, as yet, include entrance effects, these were estimated by hand and added to the results of the code, which pertain only to the two-phase frictional pressure drop in the condensing region of the RC. The inlet pressure losses include: (1) an entrance loss from the vapor manifold into the curved dog-leg separating the vapor manifold from each tube in the RC (see Figure 8); (2) an acceleration loss; (3) pressure loss due to bends in the dog-leg; and (4) the frictional loss in the dog-leg. The first

three of these losses were taken as $1/2$, 1, and $1/2$ velocity heads, respectively. The fourth was estimated from the Fanning equation as 0.3 velocity heads.

The vapor quality can only be calculated at a point which is prior to the turbine of the combined rotating unit in the PSM-3 test. In order to provide an idea of the quality of the vapor entering the RC, the change in quality of the vapor as it passed through the turbine was estimated. For typical conditions, this change was found to equal 1.4%; this factor was applied (as a constant subtraction factor) in all cases to find the quality at the entrance to the RC for the computer runs.

The experimental and predicted pressure drops for 25 data points are summarized in Table 2 and in Figure 22. Figure 22 shows that, in almost all cases, the predicted drop is greater than the measured drop. Results calculated using a y factor equal to 0 apparently show a better correlation with the data than those calculated using $y = 0.57$. This suggests that, during the condensing process, the momentum recovered by the vapor stream may be quite large.

TABLE 2
RCH-2 PRESSURE DROP SUMMARY
PSM-3 PHASE 1-F

Run No.	Data Point No.	Tube No.	Vapor Manifold Temp (°F)	Vapor Manifold Press (psia)*	Con- densing Length (in.)	Flow (ppm)	Pressure Drop (psia)	Predicted Pressure Drop	
								$y = 0.0$	$y = 0.57$
309	1	10	619	8.2	95.1	0.49	1.9	1.9	2.2
309	5	10	618	8.1	89.1	0.49	1.5	1.7	2.0
307	4	20	616	8.0	83.1	0.48	1.6	1.4	1.6
307	23	10	620	8.3	83.1	0.48	1.6	1.3	1.5
312	6	10	623	8.6	89.1	0.47	1.7	1.5	1.7
312	6	20	618	8.1	87.9	0.47	1.5	1.6	1.8
313	9	10	623	8.6	89.0	0.49	1.4	1.6	1.9
314	3	10	612	7.6	95.1	0.49	1.9	2.1	2.4
314	3	20	613	7.7	91.5	0.49	2.1	1.9	2.2
319	24	10	628	9.1	89.0	0.49	1.4	1.6	1.9
309	5	20	618	8.1	87.8	0.49	1.1	1.6	1.8

*From temperature using saturation relation.

TABLE 2
RCH-2 PRESSURE DROP SUMMARY
PSM-3 PHASE 1-F (continued)

Run No.	Data Point No.	Tube No.	Vapor Mani-fold Temp (°F)	Vapor Mani-fold Press (psia)*	Con-densing Length (in.)	Flow (ppm)	Pres-sure Drop (psia)	Predicted Pressure Drop	
								y = 0.0	y = 0.57
319	20	10	624	8.7	88.9	0.49	1.2	1.5	1.8
319	20	20	623	8.6	83.0	0.49	1.0	1.4	1.6
319	38	10	620	8.3	76.8	0.43	0.7	1.0	1.2
319	38	20	622	8.5	77.3	0.43	0.6	1.0	1.2
316	15	10	618	8.1	88.9	0.49	1.3	1.7	1.9
316	15	20	618	8.1	82.9	0.49	1.3	1.5	1.7
317	9	10	621	8.4	89.0	0.49	1.4	1.7	2.0
317	9	20	622	8.5	85.9	0.49	1.7	1.6	1.8
316	3	10	606	7.1	95.0	0.49	1.7	2.2	2.6
316	3	20	606	7.1	95.0	0.49	2.2	2.2	2.6
311	1	20	602	6.8	95.0	0.48	1.6	2.2	2.6
320	1	10	612	7.6	95.0	0.52	2.1	2.3	2.6
320	1	20	611	7.5	97.7	0.52	2.1	2.5	2.9
321	4	20	609	7.3	97.9	0.50	2.1	2.4	2.7

*From temperature using saturation relation

V. TRANSIENT ANALYSES

There are three modes of transient RC operation that have been analyzed by AI; RC preheat, drain, and shutdown (See Figure 23). Mathematical models have been developed for these operational phases and, although the models are extremely simple, the predicted results compare quite favorably, on the whole, with experimental results.

A. PREHEAT ANALYSIS

1. Experimental Condensing Behavior

A full-scale mercury Rankine startup system test (Prototype System Mockup 3) led to considerable insight with regard to transient condensation. The PSM-3 system employs a multitube flat radiator on the same plane with the pump to eliminate gross gravity effects. Figure 24 is a reactor mercury system startup schematic. Of primary concern is heating the radiator-condenser from ambient temperature to that required to maintain the prime on the condensate pump. Originally, it was assumed that during startup, vapor from the turbine would flow into the condenser and condense evenly over the entire length of the condenser. This would result in a gradual temperature increase of the entire condensing section resulting in the predicted temperature profiles shown in Figure 25. However, when vapor entered the PSM-3 condenser the temperature profiles which were observed experimentally were radically different as shown in Figure 26 on the bottom right.

The reason for the large discrepancy became apparent after an evaluation of the experimental data. The initial analytical model did not include fluid dynamics, and it assumed that the pressure (and therefore saturation temperature) was constant along the condensing tube. Since the condensation rate is a function of $(T_s - T_w)$, saturation minus wall temperature, this model gave rise to a uniform heat flux throughout the length of tube. In the actual case, however, because of the limited flow area at the inlet to the tubes and the low saturation pressure in the condenser, the flow becomes choked at the tube inlet. The flow to the RC is fixed by a boiler inlet flow regulator, and a significant pressure (2 psi) is required at the tube inlet to allow passage of vapor into the tubes at sonic velocity. The condensing heat flux is a very strong function of $(T_s - T_w)$, and, at the saturation temperature equivalent to a pressure of 2 psia ($\sim 500^\circ\text{F}$), it is sufficient to condense all the vapor within the first foot of the tube. Therefore, condensation proceeds down the tubes until the condensed liquid reaches the closed-off end of the condenser and then proceeds at a more or less uniform rate, as illustrated in Figure 26.

This was further demonstrated by startup testing in the transparent tubes. The results of these experiments were used to formulate the transient models discussed below.

2. Analytical Model

The condensing mechanism during startup was explained above; i. e., a moving condensing interface pushing a growing liquid slug to the bottom of the tube. In addition, RC tube temperature profiles observed in testing showed that there existed an almost constant condensing temperature in the major portion of the condensing region. This is equivalent to saying that there is negligible pressure drop in the vapor for the major portion of the condensing region, or alternately, that all of the pressure drop (and condensation) during the preheat phase occurs over a small interval in the vicinity of the vapor-liquid interface. This can be attributed to the large condensing flux required to heat the newly exposed cold tube.

Utilizing the phenomena noted above: (1) constant temperature in the vapor section, and (2) a moving vapor-liquid interface, a one-dimensional condensing heat transfer model can be formulated as

$$\begin{aligned}
 & \left[\begin{array}{l} \text{rate of heat} \\ \text{gained by} \\ \text{total heat} \\ \text{capacity} \\ \text{associated} \\ \text{with condens-} \\ \text{ing region} \end{array} \right] = \left[\begin{array}{l} \text{condensing} \\ \text{heat load} \\ \text{in tubes in} \\ \text{condensing} \\ \text{region} \end{array} \right] - \left[\begin{array}{l} \text{heat radiated} \\ \text{from total} \\ \text{area associ-} \\ \text{ated with con-} \\ \text{densing region} \end{array} \right] + \left[\begin{array}{l} \text{heat gained as} \\ \text{liquid slug} \\ \text{moves and} \\ \text{exposes more} \\ \text{tube to condens-} \\ \text{ing region} \end{array} \right] \\
 & \frac{d}{dt} \left[MC (T_c - T_r) \right] = wh_{fg} - \sigma \mathcal{F} \eta A_R (T_c^4 - T_s^4) + (T_o - T_r) \frac{d(MC)}{dt} \quad (12)
 \end{aligned}$$

When vapor first starts into the RC, the pressure in the tubes is near zero. This results in choked flow at the entrance to the tubes which in turn sets the initial condensing temperature. The assumption is made that the vapor continues to condense at a temperature corresponding to the tube entrance choking pressure until the condensed liquid slug reaches the bottom of the tube. While this is not quite true, the deviation from a constant temperature, as shown by test, is small. Thus, in Equation 12, T_c is a constant. Since T_c is a constant and MC is only a function of x, Equation 12 can be rewritten

$$(T_c - T_o) \frac{d(MC)}{dx} \frac{dx}{dt} = wh_{fg} - \sigma \mathcal{F} \eta A_R (T_c^4 - T_s^4) \quad (13)$$

and the equation can be integrated to provide a relation between x and t . Note that x refers to the forward edge of the condensing region; that is, to the back side of the slug that is being pushed ahead of the condensing region. Knowing the flow rate of vapor into the tubes, the position of the forward edge of the slug as a function of x and t can be calculated from a knowledge of the system geometry. This analysis holds until the slug reaches the bottom of the tube.

Once the liquid slug hits the bottom of the tube, Equation 13 is no longer valid. Instead of a large pressure drop in the interface region which is continually moving and exposing more tube, the vapor is now condensing simultaneously in all sections of the available condensing length and uniformly heats up all of the heat capacity associated with this length, which in turn raises the condensing pressure. The applicable heat transfer equation during this process is

$$\left[\begin{array}{l} \text{rate of heat} \\ \text{gained by total} \\ \text{heat capacity} \\ \text{associated with} \\ \text{condensing} \\ \text{region} \end{array} \right] = \left[\begin{array}{l} \text{condensing heat} \\ \text{load in tubes} \\ \text{in condensing} \\ \text{region} \end{array} \right] - \left[\begin{array}{l} \text{heat radiated} \\ \text{from total} \\ \text{area associ-} \\ \text{ated with} \\ \text{condensing} \\ \text{region} \end{array} \right] - \left[\begin{array}{l} \text{heat lost} \\ \text{as liquid} \\ \text{slug grows} \\ \text{and removes} \\ \text{hot tube} \\ \text{from condens-} \\ \text{ing region} \end{array} \right]$$

$$\frac{d}{dt} \left[MC (T - T_r) \right] = wh_{fg} - \sigma \mathcal{F} \eta A_R (T^4 - T_s^4) - (T - T_r) \frac{d(MC)}{dt}$$

which reduces to

$$MC \frac{dT}{dt} = wh_{fg} - \sigma \mathcal{F} \eta A_R (T^4 - T_s^4) \quad (14)$$

This equation is integrated numerically to find T . As before, the position of the liquid-vapor interface can be found from geometrical considerations. This analysis continues until the desired preheat temperature is reached. The model is discussed in detail in Reference 14.

3. Comparison to Experimental Data

The above analysis was programmed for use on an IBM 7094 computer. Figures 27 and 28 show preheat transients predicted by this code in relation to experimental transients found from system testing (PSM-3). Whereas the temperature transient profiles only roughly approximate the experimental data, the code has demonstrated the ability to predict the startup inventory required to preheat RC's to within about 5%.

B. DRAINING ANALYSIS

1. Analytical Model

Once the RC reaches a pressure high enough to sustain the prime on the condensate pump, the RC exit valve is opened which allows the liquid inventory in the RC to drain to its steady-state operating value. The flow distribution during RC drain is shown in Figure 23. The following equations apply during the draining transient.

1) Heat Transfer

$$MC \frac{dT}{dt} = wh_{fg} - \sigma F \eta A_R (T^4 - T_s^4) - (T - T_o) \frac{d(MC)}{dt} \quad (15)$$

2) Continuity

$$\rho A_T \frac{dH}{dt} = w_R \quad (16a)$$

$$\rho A \frac{dx}{dt} = w_1 - w \quad (16b)$$

$$w_1 - w_R = \text{constant} \quad (16c)$$

3) Momentum

$$P_1 - P_2 = K_1 w_1^2 + K_R w_R^2 + \frac{1}{g_c} \sum_i \frac{1}{A_i} \frac{d}{dt} (M_i V_i) - \frac{w^2}{g_c \rho A} + \frac{(w_1 - w_R)^2}{g_c \rho A_L} \quad (17)$$

The term $d/dt (M_i V_i)$ represents the time-rate-of-change of liquid momentum in the system. For the system shown in Figure 23

$$\sum_i \frac{1}{A_i} \frac{d}{dt} (M_i V_i) = \frac{L-x}{A} \frac{dw_1}{dt} - \frac{w_1}{A} \frac{dx}{dt} + \frac{\ell}{A_L} \frac{dw_1}{dt} + \frac{h}{A_L} \frac{dw_R}{dt} + \frac{H}{A_T} \frac{dw_R}{dt} + \frac{w_R}{A_T} \frac{dH}{dt} \quad (18)$$

where ℓ , h are line lengths.

4) Regulator Tank Pressure - Inventory Envelope

$$P_w = k_1 + k_2 I \quad (19a)$$

$$I = I_o - \int_0^t w_R dt \quad (19b)$$

where

$k_1, k_2 = \text{constants}$

$I = \text{regulator tank inventory at time } t$

$I_o = \text{regulator tank inventory at end of preheat}$

5) Saturation Pressure

$$P_1 = \exp \left(a - \frac{b}{T} \right) \quad (20)$$

where a, b are constants. Note also that

$$\frac{dw_1}{dt} = \frac{dw_R}{dt}$$

and

$$\frac{d(MC)}{dt} = \frac{d(MC)}{dx} \cdot \frac{dx}{dt} \quad ; \text{ i. e. , the heat capacity}$$

of the RC is a function of length only. With these additional facts, the above set of equations can be solved numerically. This analysis was programmed for use on an IBM 7094 computer.

2. Comparison to Experimental Data

The code mentioned above covers the period of events from the time when the RC reaches its final preheat pressure (which activates the exit valve starting drain of the RC) until the RC drains down to its steady state operating level.

The code computes the average RC pressure and saturated temperature, the flowrate out of the RC, the receiver pressure (regulator tank), the flowrate, into or out of the receiver, the subcooler height (inventory) in the RC, and the receiver inventory as a function of time after the start of RC drain. The steady state operating level of the RC is also calculated by the code and the draining process is stopped automatically by the code when the RC reaches that level.

Figure 29 shows predicted subcooler height as a function of draining time compared to the experimental subcooler height from system testing (PSM-3); the agreement is within the accuracy allowed for system design.

C. SHUTDOWN ANALYSIS

1. Analytical Model⁽¹⁵⁾

The flow distribution during shutdown is shown in Figure 23. Because of the large radiating area of the RC, the temperature (and pressure) drops quickly with the result that the inventory in the regulator tank empties into the RC. Since the pump discharge flow rate drops to zero at shutdown, the feed to the pump also enters the RC. The applicable equations during shutdown can be written as follows (x is subcooler height):

1) Heat Transfer

Filling:

$$MC \frac{dT}{dt} = wh_{fg} - \sigma \epsilon \eta A_R (T^4 - T_s^4) \quad (21a)$$

Draining:

$$MC \frac{dT}{dt} = wh_{fg} - \sigma \epsilon \eta A_R (T^4 - T_s^4) - (T - T_o) \frac{d(MC)}{dx} \frac{dx}{dt} \quad (21b)$$

2) Continuity

$$w + w_B + w_R = \rho A \frac{dx}{dt} \quad (22)$$

3) Momentum (Steady-State)

$$w_R = -w_B + \sqrt{\frac{|P_2 - P_1|}{k_1}} \text{ for } P_2 \geq P_1 \quad (23a)$$

$$w_R = -w_B - \sqrt{\frac{|P_2 - P_1|}{k_1}} \text{ for } P_2 < P_1 \quad (23b)$$

4) Saturation Pressure

$$P_1 = \exp\left(a - \frac{b}{T}\right) \quad (24)$$

where, a b are constants.

5) Regulator Tank Pressure – Inventory Envelope

$$P_2 = k_1 + k_2 I \quad (25a)$$

$$I = I_o - \int_0^t w_R dt \quad (25b)$$

The vapor flow rate into the RC during shutdown depends on the flow decay of the system components upstream of the RC, e. g. , boiler outlet pressure decay.

The system of equations shown above must be solved numerically. An iteration technique that has been used successfully in an IBM 7094 computer code is as follows:

- 1) Vapor flowrate is calculated from the appropriate input equation.
- 2) Liquid position, x, is calculated from Equation 22 by numerical integration.
- 3) Geometry – dependent terms are calculated (functions of x).

4) Average RC temperature is calculated from Equation 21a (or Equation 21b) by numerical integration.

5) Equilibrium RC pressure is calculated from Equation 24.

6) Flowrate out of regulator tank is calculated from Equation 23a or Equation 23b.

7) Regulator tank inventory is calculated from Equation 25b.

8) Regulator tank pressure is calculated from Equation 25a.

2. Comparison to Experimental Data

RC shutdown pressure transients predicted by the above analysis are compared to experimental pressure transients (PSM-3) in Figure 30. Run 107 shows good agreement between predicted and experimental results; in Run 124 the agreement is not as good. However, some of the disagreement in Run 124 can be attributed to unknowns in the data associated with that shutdown.

VI. CONCLUSIONS

Several general conclusions become apparent from reading this report:

1) RC fabrication technology has advanced to the point that a specific design can be manufactured without encountering any major development problems; future development should be slanted toward optimizing individual tube-fin element assemblies which can then be joined together in a modular fashion to provide the required RC configuration.

2) RC component testing in the Mercury Rankine Program has been sufficient to verify the adequacy of the methods used for structural and thermal design of RC's in this program.

3) Analyses of steady state system operation of an RC have been completed, including thermal, stability, and pressure drop analysis. Sufficient experimental thermal and pressure drop data have been taken to verify these analyses; however, the stability analysis has only been verified in a qualitative manner so far for the Mercury Rankine Program.

4) Analyses of transient system operation of an RC have been performed and verified by experimental data.

The design criteria presented in this report have been developed specifically for direct condensing mercury radiators; however, the principles involved can be applied to any other fluid system.

APPENDIX A: NOMENCLATURE

- A = tube cross-sectional area
 A_L = line cross-sectional area
 A_R = radiating area of total condensing region
 A_T = cross-sectional area of regulator tank
 C, C_P = specific heat of liquid Hg
 D = tube hydraulic diameter
 D_C = core diameter
 f = friction factor
 g_c = gravitational constant
 H = height of liquid in regulator tank
 h = line length into regulator tank
 h_{fg} = latent heat of condensation
 k = thermal conductivity of fin
 K = entrance loss coefficient
 K_i, K_R = friction loss coefficients
 L = tube length
 L_c = condensing length
 L_{oc} = reference condensing length
 ℓ_m = average fin length of element m
 M_r = radiation modulus = $\ell m^2 \sigma \epsilon T_{fb}^3 / kt$
 MC = total heat capacity of condensing region
 n = number of g 's acceleration
 P = pressure
 P_o = inlet total pressure
 P_{LM} = regulated liquid manifold pressure ($= P_2$)
 P_1 = pressure of vapor in RC

P_2 = pressure in regulator tank
 Q_s = heat flux from space
 Q_i = heat flux from components within RC
 R_{eo} = axial Reynolds number based on diameter
 r_o = distance from center of rotation to liquid manifold
 r_1 = distance from center of rotation to interface
 T = temperature
 T_{out} = temperature of liquid Hg leaving node m
 T_{in} = temperature of liquid Hg entering node m
 T_{fb} = average fin base temperature
 T_c = mean condensing temperature of vapor
 T_o = initial temperature of RC
 T_r = reference (datum) temperature
 T_s = ambient (space) temperature
 t = time; fin thickness
 U = average axial velocity
 U_o = inlet axial velocity
 w = vapor flowrate
 $\bar{w}$ = vapor flowrate per half-tube
 w_B = bearing drains flowrate
 w_1 = drain flowrate from RC
 w_R = flowrate into regulator tank
 We_T = tube Weber number
 x = axial distance
 y = momentum loss factor
 ΔA_m = area of fin element m
 $\mathcal{F}$ = effective emissivity of radiating surface
 ψ = vapor quality

η = fin effectiveness

η_m = fin effectiveness of element m, obtained from plot of

η_m vs M_r at $T_s = 0^\circ R$ (the fact that T_s is, in general, than $0^\circ R$ is accounted for in the Q_s term)

σ = Stefan-Boltzmann constant; surface tension

ϵ = emissivity

ρ = liquid density

ϕ^2 = ratio of two phase pressure gradient to single phase vapor pressure gradient

δ_{cr} = droplet diameter

τ_w = wall shear stress

ω = angular frequency of rotation

APPENDIX B: STEADY STATE – RADIATOR CONDENSER ANALYSIS

A. ONE-DIMENSIONAL

An essentially non-iterative, one dimensional model has been developed for use in the screening of proposed radiator-condenser designs and in preliminary optimization work. For convenience the analysis will be discussed in terms of condensing mercury in a conical radiator.

Assume that the top diameter of the cone, the cone angle, and the number of tubes are known, so that the width of the fin is known as a function of its length. As a first step, one marks off a fixed distance from the top of the fin, thus completing the definition of an element of area. Since the condensing temperature profile along the fin base is known, ΔQ_m , the heat rejected by the element of area m can be calculated by means of Equation 26:

$$\Delta Q_m = \eta_m \sigma \epsilon \Delta A_m T_{fb}^4 - \Delta A_m (Q_s + Q_i) \quad (26)$$

Since the net heat rejected is taken from the mercury vapor, a part of the vapor condenses, and the quality of the vapor leaving the element of area is calculated by means of Equation 27:

$$\psi_{m+1} = \psi_m - \frac{\Delta Q_m}{wh_{fg}} \quad (27)$$

The exact location of the interface, i. e. the point at which $\psi_{m+1} = 0$ is found by interpolation; thereafter, removal of further heat corresponds to a reduction in the temperature of the liquid mercury according to Equation 28:

$$T_{\text{out}} = T_{\text{in}} - \frac{\Delta Q_m}{wC_P} \quad (28)$$

Since the calculation of ΔQ_m requires a knowledge of the average fin-base temperature, which in turn is to be calculated in the subcooling region from the value of ΔQ_m , a short iterative technique is used in this region. The tube length at which the required mercury outlet temperature is reached is found by interpolation.

B. TWO-DIMENSIONAL MODEL AND COMPARISON TO ONE-DIMENSIONAL MODEL

1. Analysis

After the original screening and approximate optimization work has been completed, the final optimization and final radiator design can be carried out with an accurate two-dimensional model developed for this purpose. This model, like the other, employs a typical radiator element, again a half-tube with its attached fin. In this case, also, the fin is considered to be bonded to the outside of the tube. The analysis is outlined below.

During steady-state operation, for a differential portion of fin,

$$\sum (\text{heat gained by conduction}) = \text{net heat lost by radiation} \quad (29)$$

The left-hand side of this equation is obtained by summing the heat entering the fin element by conduction across each of the four boundaries: r , $r + \Delta r$, θ , $\theta + \Delta\theta$ (see Figure 31), and taking the limit of this sum as Δr and $\Delta\theta$ both approach zero. The right-hand side is obtained by taking the difference between the heat radiated by an element of area and the heat absorbed by that element from its environment, according to Equation 30:

$$Q_{\text{net}} = \sigma \epsilon \Delta A \left[T^4 - T_s^4 \right] - Q_{\text{ext}} \quad (30)$$

Note that, for convenience, provision is made for introducing the heat load from the environment in terms of either a space temperature or an external heat load. The final equations in differential form are presented as Equations 31 and 32.

Rectangular Fin:

$$\nabla^2 T = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} = \frac{\sigma \epsilon}{kt} [T^4 - T_s^4] - \frac{Q_{\text{ext}}}{kt} \quad (31)$$

Tapered Fin:

$$\begin{aligned} \nabla \cdot (t \nabla T) &= t(\theta) \left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \left(\frac{\partial^2 T}{\partial \theta^2} + \frac{1}{t(\theta)} \frac{\partial T}{\partial \theta} \cdot \frac{\partial T}{\partial \theta} \right) \right] \\ &= \frac{\sigma \epsilon}{k} (T^4 - T_s^4) - \frac{Q_{\text{ext}}}{k} \end{aligned} \quad (32)$$

The following change of variable is effected in Equations 31 and 32 (see Figure 31):

$$\begin{aligned} r &= y_o + y \\ \xi &= \theta / \theta_{\text{max}} \end{aligned}$$

which then become Equations 33 and 34 (note that $b_o = y_o \theta_{\text{max}}$):

Rectangular Fin:

$$\nabla^2 T = \frac{\partial^2 T}{\partial y^2} + \frac{1}{(y_o + y)} \frac{\partial T}{\partial y} + \frac{y_o^2}{b_o^2 (y_o + y)^2} \frac{\partial^2 T}{\partial \xi^2} \quad (33)$$

Tapered Fin:

$$\nabla \cdot (t \nabla T) = t(\theta) \left\{ \frac{\partial^2 T}{\partial y^2} + \frac{1}{(y_o + y)} \frac{\partial T}{\partial y} + \frac{y_o^2}{b_o^2 (y_o + y)^2} \left[\frac{\partial^2 T}{\partial \xi^2} + \frac{1}{t(\theta)} \frac{\partial t(\theta)}{\partial \xi} \cdot \frac{\partial T}{\partial \xi} \right] \right\} \quad (34)$$

The differential portion of Equations 33 and 34 are put into manageable form by means of the following finite-difference approximations, which are obtained from Taylor's series expansion of $T(y, \xi)$. The fin is divided into M nodes of width Δ_1 , in the r direction, and N nodes of width Δ_2 in the θ direction.

$$\frac{\partial T}{\partial y} = \frac{T(m+1, n) - T(m, n)}{\Delta_1} = \frac{T(m, n) - T(m-1, n)}{\Delta_1}$$

$$\frac{\partial T}{\partial \xi} = \frac{T(m, n+1) - T(m, n)}{\Delta_2} = \frac{T(m, n) - T(m, n-1)}{\Delta_2}$$

$$\frac{\partial t}{\partial \xi} = \frac{t(m, n+1) - t(m, n)}{\Delta_2} = \frac{t(m, n) - t(m, n-1)}{\Delta_2}$$

$$\frac{\partial^2 T}{\partial y^2} = \frac{T(m+1, n) - 2T(m, n) + T(m-1, n)}{\Delta_1^2}$$

$$\frac{\partial^2 T}{\partial \xi^2} = \frac{T(m, n+1) - 2T(m, n) + T(m, n-1)}{\Delta_2^2}$$

The radiation expression involving T^4 is linearized as follows:

$$T_k = T_{k-1} + \delta T$$

where k refers to the k^{th} iteration, δT = the change in T from the $(k-1)^{\text{th}}$ to the k^{th} iteration. Then,

$$T_k^4 - T_{k-1}^4 = [T_{k-1} + \delta T]^4 - T_{k-1}^4$$

Expanding and neglecting higher order terms

$$T_k^4 \approx T_{k-1}^4 + 4T_{k-1}^3 [T_k - T_{k-1}]$$

Then if $s = 4T_{k-1}^3$, Equations 31 and 32 become, respectively, Equations 35 and 36:

Rectangular Fin:

$$\nabla^2 T_k - \frac{\sigma \epsilon}{kt} s T_k = \frac{-\sigma \epsilon}{kt} \left[3T_{k-1}^4 + T_s^4 + \frac{Q_{\text{ext}}}{\sigma \epsilon} \right] \quad (35)$$

Tapered Fin:

$$\nabla \cdot (t \nabla T_k) - \frac{\sigma \epsilon}{k} s T_k = \frac{-\sigma \epsilon}{k} \left[3 T_{k-1}^4 + T_s^4 + \frac{Q_{ext}}{\sigma \epsilon} \right] \quad (36)$$

Equations 33 through 36 are substituted into Equations 31 and 32 to give linear expressions in a numerical form which relate the temperature at the node (m,n) to those at the adjacent nodes.

2. Boundary Conditions

The finite-difference equations were modified at each boundary to satisfy the following boundary conditions:

1) The fin base temperature is equal to the mercury temperature minus the temperature drop through the mercury film, tube wall, braze, and meteoroid armor.

2) An adiabatic midplane exists between tubes.

$$\left[\left(\frac{\partial T}{\partial \xi} \right)_{\theta_{max}} = 0 \right]$$

This assumption is consistent with the earlier one that all the fins are under identical thermal conditions, which is implied by the original selection of a typical RC element for analysis.

3) There is no vertical heat flow at the top rim (vapor manifold) of the fin.

4) The temperature along the bottom rim (liquid manifold) of the fin is equal to the fin base temperature at the bottom of the tube. This temperature is approximately that of the exit mercury which fills the liquid manifold, which in turn is in contact with the fin along its full length.

The temperature drop between the mercury in the tube and the base of the fin covering the tube is a function of the film coefficient of the mercury, the dimensions of the tube, and any braze or armor which may be interposed between the fluid and the fin. These dimensions and the temperature profile of the mercury are often functions of length. The effect of the tube wall as a fin in conducting heat, and the brazed junction between the fin and tube, must also be considered.

In a hollow cone with a lengthwise temperature gradient, some heat exchange takes place by radiation from the hotter portions to the cooler ones. To account for this, one must impose a varying surface heat load down the length of the fin,

which appears in the Q_{ext} term of Equation 30. In a typical mercury radiator for a 5-kwe system, this effect has been found to change the interface location by approximately two inches.

Under some operating conditions a layer of non-condensable gas (hydrogen) can accumulate in the radiator tubes between the end of the condensing region and the top of the liquid leg. The three regions in the tube (condensing vapor, noncondensibles, and liquid) are accounted for by the use of different values of the film heat-transfer coefficient in the calculation of the fin base temperature profile along the tube-side boundary.

REFERENCES

1. N. Zuber and M. Tribus, "Further Remarks on the Stability of Boiling Heat Transfer," AECU-3631, January 1958
2. "SNAP II Power Conversion Topical Report No. 13 Offbase Testing," TRW ER-4670
3. R. Kiraly and A. Koestel, "Condenser Development and Design Study," The SNAP 2 Power Conversion System Topical Report No. 8, TRW Report No. ER-4104, June 1960
4. R. Bellman and R.H. Pennington, "Effects of Surface Tension and Viscosity on Taylor Instability," Quar. Appl. Math 12, 1954, p 181
5. R. Forslund, "Interface Stability in Rectangular Tubes Within Gravity Field," NAA-SR-TDR-8960
6. R.G. Gido and A. Koestel, "Mercury Wetting and Non Wetting Condensing Research," Progress Report No. 3, TRW Report No. ER-5214, January 1963
7. A. Koestel, et al., "Space Radiator Study," ASD-TDR-61-697
8. C.J. Baroczy and V.D. Sanders, "Pressure Drop for Flowing Vapors Condensing in a Straight Horizontal Tube," NAA-SR-Memo-6333
9. Martinelli, Boelter, Taylor, Thonsen, and Morsing, "Isothermal Pressure Drop for Two Phase, Two Component Flow in Horizontal Pipe," TRANS ASME, VOL. 66 (1944) p 139
10. Lockhart and Martinelli, "Proposed Correlation of Data for Isothermal Two Phase, Two Component Flow in Pipes," Chemical Engineer Progress, Vol. 45 (1949) p 39
11. R.P. Forslund, "Pressure Drop Program for Condensing Mercury in the SNAP 2 Condenser," NAA-SR-9064 (September 17, 1963)

12. R. P. Forslund, "SNAP Mercury Rankine Program Mercury Condensing Experiments," NAA-SR-9801
13. A. Koestel, et al., "Fog-Flow Mercury Condensing Pressure Drop Correlation," Third Annual High Temp. Liquid Metal Heat Transfer, ORNL (September 1963)
14. M. P. Berhold and V. H. Heiskala, "Description of R/C Preheat Inventory Code," NAA-SR-TDR-10714
15. M. P. Berhold and V. H. Heiskala, "Description of R/C Shutdown Transient Code; Comparison with PSM-3 Shutdown Test Data," NAA-SR-TDR-11038

8-21-64

7636-0478

Figure 1. 3 to 5 kwe Hg Rankine System

TESTS ON AL-STEEL TUBE-FIN ELEMENTS (AS OF DECEMBER 1, 1964)

THERMAL SHOCKS	-10
THERMAL CYCLES	720
THERMAL ENDURANCE	> 5000 hr

BASIC CONFIGURATION
NO STRUCTURAL
REQUIREMENTS

NUMBER OF ELEMENTS
DETERMINED BY REQUIRED
HEAT REJECTION

1-6-65

DEVELOPMENT UNDERWAY

- 1) TUBE-FIN FABRICATION TECHNIQUES
- 2) FIN STABILIZATION
- 3) ESTABLISH ASSEMBLY PROCEDURES
- 4) CONTINUE ENVIRONMENTAL TESTING
- 5) TUBE GEOMETRY OPTIMIZATION

APPLICATIONS CONFIGURATION
TYPICAL METHODS OF
STRUCTURAL INTEGRATION

ELEMENTS JOINED
BY RIVETING

ELEMENTS JOINED
BY WELDING

7636-0050C

Figure 2. Typical Aluminum Fin - Steel Tube Element

Figure 3. Radiator Condenser Development Progress

Figure 4. Steel Honeycomb Radiator Condenser (RC-6)

Figure 5. Steel Honeycomb (RC-11)

Figure 6. Aluminum/Haynes-25 Radiator-Condenser (RC-10)

Figure 7. Aluminum/Haynes-25 Radiator-Condenser (RC-13)

Figure 8. Schematic of a Horizontal Radiator Condenser (RCH-2)

TYPE	UNIT	PURPOSE	STRUCTURAL TESTING					AERO-HEATING	THERMAL PERFORMANCE CYCLES Hr	
			SHOCK	VIBRA-TION	AERO-LOADS	INERTIA LOAD	COMBINA-TION			
HONEY-COMB	RC-6	FEASIBILITY + COMPONENT TEST	✓	✓	✓	✓	✓			
	RC-6	THERMAL PERFORMANCE							34	1202
	RC-11	STRUCTURAL TEST	✓	✓	✓	✓	✓		—	—
Al-STEEL	RC-10	FEASIBILITY + COMPONENT TEST	✓	✓	✓	✓	✓	✓	16	37
	RC-13	FLIGHT PROTOTYPE + SUBCOMPONENT TEST							720	2000
COPPER-STEEL TUBE-FIN ELEMENTS	RCH-1	SIMULATED ZERO GRAVITY GRAVITY	—	—	—	—	—	—	90	1233
	RCH-2	ADVANCED ZERO GRAVITY SYSTEM TEST	—	—	—	—	—	—	20	260

Figure 9. RC Test Summary

APPROXIMATELY 1000 HOURS (550 w/CRU) OF TEST OPERATION ACHIEVED
STABILITY OF COMPLETE INTEGRATED SYSTEM VERIFIED
PREDICTED ANALYTICAL THERMAL AND HYDRAULIC PERFORMANCE WAS VERIFIED
OPERATION OF A SPACE CONFIGURED SYSTEM WITHIN A VACUUM WAS DEMONSTRATED
ADEQUACY OF INJECTION STARTUP AND GROUND SHUTDOWN SHOWN
SYSTEM LEAKAGE REQUIRES IMPROVED DESIGN AND FABRICATION TECHNIQUES
BOILER CONDITIONING PROBLEM INDICATED
CRU RUGGEDNESS SHOWN THROUGH SEVERE OPERATIONAL CONDITIONS

9-14-64

7636-0366A

Figure 10. Major PSM-1 Test Results

SYSTEM STABILITY IN BOTH STARTUP AND STEADY-STATE OPERATION VERIFIED—WITHOUT THE AID OF GRAVITY

FLAT RC DEMONSTRATED STABILITY AND ADEQUATE FLOW DISTRIBUTION

STARTUP ANALYTICAL MODEL DEFINED

Hg STARTUP INVENTORY REQUIREMENT IS 10% LESS THAT PREDICTED

CRU AND RC COUPLED SATISFACTORILY

TE PUMP SIMULATOR PROVED FEASIBILITY OF OPERATION

APPROXIMATELY 700 HOURS OF TEST OPERATION AND 60 STARTUPS WITH AN INSTALLED CRU HAVE BEEN ACHIEVED THUS FAR

9-14-64

7636-0461B

Figure 11. Major PSM-3 Test Results

Figure 12. Steel Honeycomb Radiator (RC-6) Performance Run 6B

Figure 13. Aluminum Steel Radiator Condenser (RC-10) Performance Run 5-3B

Figure 14. Flat Radiator Condenser (RCH-2) Performance Run 319

Figure 15. Typical RC Pressure Variations

10-19-65

7636-01144A

Figure 16. Evaluation of Stability Limits

10-19-65

7636-01145A

Figure 17. Evaluation of Interface Location

	NEGATIVE "G" FORCES				ZERO "G" FORCES				POSITIVE "G" FORCES			
	WETTED		NON-WETTED		WETTED		NON-WETTED		WETTED		NON-WETTED	
	STEADY STATE	STARTUP	STEADY STATE	STARTUP	STEADY STATE	STARTUP	STEADY STATE	STARTUP	STEADY STATE	STARTUP	STEADY STATE	STARTUP
INSTABILITY												
a) INTERFACE	◆		●◆	●	◆○		●◆△	●△			●□	●□
b) LIQUID LEG			○	○			○△◇Φ	○△			○■	○■
c) MOMENTUM RECOVERY												
d) RUNBACK			●	●			●◆	●			●	●
e) FLOW REGIME			○●	○			○◇●	○			○●	○
HEAT TRANSFER												
a) MECHANISM							△	△			□	□
b) CONDENSING COEFF.			●	●			●○	●△			●	●■
c) NON-CONDENSIBLE			●	●			●△	●			●	●
HYDRAULIC												
a) PRESSURE DROP			●○	●			●△◆●	●△○			●■	●■
b) LIQUID FRACTION			○	○			○	○			○	○
c) FRICTION FACTOR												
d) SLIP RATIO												
e) FLUID BEHAVIOR AT STARTUP							▲				□	
BEHAVIOR OF NON-CONDENSIBLES												
a) ENTRAINMENT			●	●			●△	●			●□	○
b) DIFFUSION	▼		▼		▼		▼		▼		▼	

APPLICABLE TEST DATA	TYPE OF INFORMATION SUPPLIED	
	QUALITATIVE	QUANTITATIVE
SINGLE-TUBE TEST	○	●
PSM-3	△	▲
PSM-1	□	■
TRW ER-5214 (TUBE TESTS)	◇	◆
TRW ER-4670 (OFFBACE)	Φ	⊙
VACUUM FURNACE DIFFUSION TESTS	▽	▼

Figure 18. Summary of MRP Condensing Experience

10-19-65

7636-0276A

Figure 19. Comparison of Predicted Pressure Profiles to Experimental Data From Series A Runs of Reference 6

5-13-64

7636-0298

Figure 20. Comparison of Momentum Terms in Pressure Drop Models

5-13-64

7636-0296

Figure 21. Momentum Loss Factor (y) vs Pressure. Used to Correlate Model No. 1 Prediction with Series A Data of Reference 14.

Figure 22. Comparison of Measured and Predicted RCH-2 Pressure Drops

Figure 23. Startup System Schematics

Figure 24. SNAP2 Orbital Startup Schematic

Figure 25. Predicted Temperature Distribution in RC During Injection

Figure 26. Mercury Rankine Program SNAP 2 PSM-3 System Test

Figure 27. Experimental vs Predicted RC Preheat Temperatures

Figure 28. Experimental vs Predicted RC Preheat Pressures

Figure 29. Experimental vs Predicted RC Draining Transient

Figure 30. Experimental vs Predicted RC Shutdown Pressure

$$\Delta_1 = y_{m+1} - y_m = \left(\frac{L}{M}\right), \text{ WHERE } L = \text{TOTAL LENGTH OF FIN}$$

$$M = \text{NUMBER OF LENGTHWISE NODES}$$

$$\Delta_2 = \xi_{n+1} - \xi_n = \left(\frac{1}{N}\right), \text{ WHERE } N = \text{NUMBER OF CROSSWISE NODES}$$

Figure 31. Definition Sketch of Fin for Use in Deriving Basic Differential Equation

44

DESIGN AND DEVELOPMENT OF THE SNAP-8 MULTITUBE CONDENSER*

by

A. H. Kreeger and J. N. Hodgson
SNAP-8 Division
Von Karman Center
Aerojet-General Corporation
Azusa, California

ABSTRACT

The design, development, and testing of the condenser for this SNAP-8 Electrical Generating System marks one of the major accomplishments of the SNAP-8 Program.

The SNAP-8 system condenser is a counterflow tube-in-shell heat exchanger. It consists of 73 tapered tubes surrounded by a tapered shell. Mercury condenses inside the tubes. The shell-side fluid is a sodium-potassium eutectic mixture (NaK) which flows through the condenser as part of the heat-rejection system. The tube configuration is designed to permit operation with a stable liquid-vapor interface in a zero-gravity environment.

The condenser development incorporated detailed analytical investigations of heat-transfer modes. These analyses extended from literature surveys to configuration studies conducted at Aerojet. The condenser size and number of tubes is influenced by the mercury inlet velocity, condensing mercury heat-transfer coefficient, and the diameter and spacing of the tubes. The existing condenser is the result of this parametric analysis.

The SNAP-8 condenser has been tested under conditions identical to its space application with the exception of zero-g. The objectives of the test program were to investigate the relationships between flow rates, temperatures, condenser inventory, and condensing pressure. A concurrent NASA-conducted program using a SNAP-8 condenser tube (MECA Project) provides additional test data for comparison.

The condenser has been found to comply to the SNAP-8 system requirements.

*The information appearing in this paper is based on work accomplished under National Aeronautics and Space Administration Contract NAS 5-417.

I. INTRODUCTION

SNAP-8 is a 35 kw, nuclear-electric, Rankine-cycle space power system. The electrical output is generated by a turbine-alternator assembly using mercury vapor as the working fluid. The excess heat from the cycle is removed through a liquid cooled condenser; the condenser coolant then dissipating its heat through a space radiator. This indirect heat removal system was selected in preference to a direct condensing radiator to avoid the complexities of flow distribution in a large two-phase radiator under a variable gravitational field.

The SNAP-8 system condenser, shown in Figure 1, is a counterflow tube-in-shell heat exchanger with fixed tube sheets. The 73 tapered tubes contain the condensing mercury with the coolant contained in the tapered shell. The shell-side coolant is the eutectic mixture of sodium and potassium (NaK) which flows through the condenser as part of the heat-rejection system. Tapered tubes are used to maintain vapor velocity through the condenser length providing a continual movement of condensate to the liquid vapor interface by the vapor drag on the condensing droplets in very low gravity operation. The condenser design parameters are shown in Table 1.

The condenser development program goals are to confirm the adequacy of the design in response to the variations in loop parameters through its 10,000-hour life. Initial tests have been performed to verify the thermal performance of the condenser in a one-gravity field. The variation in performance resulting with changes in mercury inventory (interface position) have been determined. The data have been analyzed to give the overall conductance leading to a determination of average mercury film coefficient for a calculated value of NaK-side film coefficient.

II. DESIGN ANALYSIS

The purpose of the condenser in the SNAP-8 system is threefold:

- To provide a back pressure on the turbine commensurate with system requirements,
- To provide adequate subcooling to result in an acceptable net positive suction head (NPSH) for the mercury pump, and
- To provide inventory storage capability for mercury leakage from space seals during the 10,000-hour operation.

To satisfy these requirements, the two major problems involved in the design of the condenser were the lack of a generalized correlation for condensing mercury heat-transfer coefficients and the absence of a rational set of design criteria for application to zero-gravity condenser operation. Accordingly, a detailed literature review at the initiation of the design was required to establish the required information for design purposes.

Since other systems used condensing radiators, the requirement for accurate values of condensing coefficients is unique in the development of a condenser for space application. With a condensing radiator, the heat removal is limited by the resistance to heat transfer from the radiating surface. However, with a liquid-metal-cooled condenser, the resistance offered by the shell-side and the tube wall are of the same order as the expected mercury film resistance. The sensitivity of condenser performance is a function of the limiting resistance; hence, a relatively accurate measure of film coefficient is required to minimize weight without sacrificing performance.

A. MERCURY FILM COEFFICIENT

Data on mercury condensing coefficients were limited. During operation of mercury system-power plants, Misra (working at General Electric Company) reported data⁽¹⁾ giving heat-transfer coefficients varying from 2840 to 870 Btu/hr-ft²-°F neglecting water-side fouling. Misra and Bonilla⁽²⁾ reported data on condensation of mercury on short vertical carbon steel, stainless steel, and nickel surfaces, and found that condensing heat-transfer coefficients vary between 2000 and 5000 Btu/hr-ft²-°F as the mercury vapor pressure varied from 15 to 2 psia. Based on these data, an average mercury-side heat-transfer coefficient of 2000 Btu/hr-ft²-°F for the reference design was selected for use in the design analysis.

B. ZERO-GRAVITY OPERATION

The development of a rational set of design criteria for zero-gravity operation hinges on the prevention of slug flow in the condensing tube and the development of a stable liquid-vapor interface. These factors are discussed in many papers and reports^(3,4,5,6) from which the principles of the condenser design were selected.

Superscript numbers in parentheses refer to similarly numbered references listed following the text.

The stability criterion for a circular tube is given as:

$$B_o = \frac{\rho_l g D_t^2}{\sigma} < 15.3$$

where

B_o = bond number

ρ_l = liquid density

g = local acceleration of gravity

D_t = tube diameter

σ = surface tension

leading to the first design principle that the critical tube dimension at the interface location is $D_t \leq 0.24$ in.

As the mercury vapor begins to condense, droplets of liquid form on the walls of the condensing passage. These grow in size until the drag force due to the mercury vapor flowing over the droplet is equal to the restraining force due to the surface tension between it and the wall. When this occurs, the droplet breaks away and moves downstream. To prevent slug flow, the critical droplet height, d , must be less than half the local channel diameter, D_t . The critical diameter is a function of the Weber number which relates dynamic to surface forces:

$$\frac{d}{D_t} = \text{const } (We)^{-1} < 1/2$$

leading to the second design principle, the minimum vapor velocity needed to prevent slug flow:

$$v_g = \left(\frac{2 \sigma}{\rho_g D_t} \right)^{1/2} \sim 100 \text{ ft/sec}$$

at the condenser inlet. The requirement to maintain vapor velocity down the length of the condenser tube to prevent slug flow implied the need to taper the condenser tubes. Figure 2 shows the effect of tapering on vapor velocity where the calculated velocity profiles for a straight and tapered tube of equal surface area are shown. The tapered tube configuration was selected with the amount of taper being variable in the design analysis.

C. CONFIGURATION STUDIES

With the design principles established, an engineering analysis was programmed for an IBM 7094 digital computer. The computer program resulted in a number of condensers which theoretically perform as required. A reference condenser was selected from the results of the study.

Fifty-six condenser designs were considered in the computer study, based on the following configurations:

1	Tube Inside Diameter	
	At mercury inlet	0.40, 0.45, 0.50 in.
	At condenser exit	0.125, 0.20, 0.25 in.
	At subcooler exit	0.125, 0.20, 0.25 in.
2	Tube Wall Thickness	0.020, .035 in.
3	Pitch [*] /tube outside diameter ratio at mercury inlet	1.375, 1.355 in.
4	Tube material	Croloy 9M (9Cr-1Mo)

The analysis was based on maintaining a constant clearance between the tubes (clearance is the distance between tube walls). Therefore, the P/D_o ratio at the mercury exit end varied from case to case.

The results of the 56 computer cases are summarized in Table 2. Some of the results are plotted as follows:

Figure 3 - Tube clearance and number of tubes as a function of tube inside diameter for mercury inlet velocities of 150 and 200 fps.

Figure 4 - Mercury condenser length as a function of tube inside diameter for condensing mercury heat-transfer coefficients of 1,000, 2,000, and 3,000 Btu/hr-ft²-°F and velocities of 150 and 200 fps. It is seen that, for the reference condenser, the condensing length will vary from 62.4 to 30.7 in. for the values of the mercury heat transfer coefficient of 1,000 and 3,000 Btu/hr-ft²-°F, respectively, with a mercury inlet velocity of 150 fps.

* Pitch is defined as the distance between tube centers. The ratio of pitch to tube outside diameter will be denoted by P/D_o ratio.

Figure 5 - Subcooler length as a function of tube inside diameter for mercury inlet velocities of 150 and 200 fps.

Figure 6 - Condenser heat transfer area as a function of tube inside diameter for mercury heat transfer coefficients of 1,000, 2,000 and 3,000 Btu/hr-ft²-°F.

Figure 7 - Condenser unit conductance as a function of tube inside diameter and for the three values considered for the mercury heat-transfer coefficients. The figure shows that the conductance is a weak function of the tube inside diameter. However, it varies strongly with the mercury heat-transfer coefficient.

Figure 8 - Mercury pressure drop calculated by the method of reference⁽⁷⁾ as a function of tube inside diameter for the three mercury heat transfer coefficients used.

As can be seen in Table 2, the tube exit diameter has a negligible influence on the condensing heat transfer area and overall unit conductance (U_{cond}) in the condenser for a given tube inlet diameter, mercury condensing coefficient, and number of tubes. For example, in Cases 1, 5, and 8 for a tube inlet diameter of 0.45 in. and an exit diameter of 0.125, 0.20, and 0.25 in., the corresponding condensing heat transfer areas are 19.7, 19.8, and 19.9 ft². The overall unit conductance for these cases is 1169, 1160, and 1156 Btu/hr-ft²-°F, respectively.

The final configuration, Case 56, was selected as the reference design. It was considered a conservative design with reasonable assurance of performance capability for first-generation component and systems testing.

The final condenser geometry was modified by increasing its length to give a design overall conductance of 960 Btu/hr-ft²-°F.

III. MECHANICAL DESIGN

The two areas of concern in the mechanical design of the condenser were materials compatibility with the working fluids, NaK and mercury, and the configuration of the tube-to-header joints. The requirement for a 10,000-hour operational goal requires materials that will not corrode or erode in either liquid-metal environment. The material

selected for mercury containment was a low-alloy, ferritic-base steel, 9Cr-1Mo, which showed excellent mercury corrosion resistance at condenser temperatures⁽⁸⁾. NaK is not particularly corrosive at condenser temperatures with appropriate cold traps provided to minimize the oxide level; therefore, there was a large group of acceptable materials for use on the shell side. The material selected was CRES Type 410, stainless steel, which was selected based on its comparable coefficient of linear expansion to the tube-side material. The equal coefficient permitted the fixed tube-sheet design.

The tube header configuration was the main design problem as numerous tube joints are required. Two approaches used with success were a welded and back-brazed joint and a rolled-and-welded joint.

The welded and back-brazed joint utilized a trepanned header, automatic tungsten electrode, inert-gas (TIG) weld, and a subsequent back-braze at 1800°F in a hydrogen atmosphere. This joint is shown schematically in Figure 9. Condensers using this fabrication method have accumulated over 2000 hours of service without a failure.

The rolled and welded joint eliminated the back braze (which was a costly and time-consuming process) replacing it with a grooved header into which the tube was rolled prior to welding. These joints, as shown schematically in Figure 10, have not been in service, but have passed nondestructive tests including helium leak checks and proof tests.

The gamma-radiography technique used for testing of the thin-wall tube-to-tube-sheet welds was developed specifically for the inspection of the SNAP-8 condenser. The inspection technique is shown schematically in Figure 11 and has been reported in detail⁽⁹⁾. The method is superior to conventional x-radiography as there is no interference from adjacent tubes to cloud the interpretation of the radiograph. Porosity to 0.003 in. in diameter was detectable as were cracks in welds and circumferential weld undercutting.

The remainder of the mechanical design is considered straight-forward heat exchanger fabrication with the materials selected and the fabrication process for the tube headers developed.

IV. DATA ANALYSIS

A. SINGLE TUBE TESTS

The first data available on the performance of the SNAP-8 condenser were

obtained from the NASA-conducted MECA (Mercury Evaporating and Condensing Analysis) program where a single SNAP-8 condenser tube was tested, comparing its performance with a straight condenser tube. The results of these tests have been reported⁽¹⁰⁾ and are summarized below:

1. The local overall heat transfer coefficients obtained from the tapered tube varied from 1200 to 3700 Btu/hr-sq ft-°F. In general, the local overall heat-transfer coefficient increased with increasing NaK weight flow and with decreasing condensing lengths.
2. For a constant mercury weight flow, NaK weight flow, and NaK inlet temperature, the mercury inlet static pressure decreased rapidly as the condensing length was increased to approximately 25 in. then leveled off as condensing length was increased further. Depending upon the combination of test variables used, inlet static pressures varied from 8 to 24 psia.
3. The overall static pressure losses for the tapered tube increased as the condensing length increased. An actual static pressure rise between 0 to 0.5 psi existed for condensing lengths less than 17 in., while static pressure losses as large as 3 psi were obtained for a condensing length of 40 in.

The results of this program were encouraging as extrapolation of these data to the multitube condenser gave an overall coefficient of 1900 Btu/hr-ft²-°F compared with the final design requirement of 960 Btu/hr-ft²-°F. The reason for the large difference in the MECA value of 1900 Btu/hr-ft²-°F and the design requirement of 960 Btu/hr-ft²-°F lies in the value of the local mercury condensing film coefficient. A value of 2000 Btu/hr-ft²-°F was used for design purposes. Extrapolation of the MECA data, however, indicated an average condensing film coefficient of about 14,700 Btu/hr-ft²-°F. Consequently, the condensing film offers relatively no thermal resistance. The total resistance is governed mainly by the NaK film and wall resistances rather than being significantly affected by the mercury coefficient as had been assumed.

B. MULTITUBE CONDENSER TESTS

The initial tests of the multitube condenser have been performed. The data are presented in Table 3. The ranges of variables tested and the calculated parameters are listed below.

Range of Testing

Mercury flow rate	9500 - 12,200 lb/hr
Condensing pressure	4.2 - 9.3 psia
NaK flow rate	30,000 - 35,500 lb/hr
NaK inlet temperature	333 - 404°F
Mercury inlet quality	83.9 - 99.2%

Parameters Calculated

Condensing area, ft ²
Logarithmic mean temperature difference, °F
Thermal load, Btu/hr
Overall heat-transfer coefficient, Btu/hr ft ² -°F
Mercury film coefficient, Btu/hr-ft ² -°F
Condensing length, ft
Condensing area effectiveness

1. Condensing Pressure

Figure 12 presents the plotted data from Table 3. Condensing pressure is plotted vs condensing length. The curve represents a fit of the data by the method of least squares.

The significance of the test data is that a change in interface position (condensing length or area) does not affect the condensing pressure. Thus, changes in inventory, either from space seal leakage or boiler demands, will not change the back-pressure on the turbine. Overdesigning the condenser is a relatively inexpensive means of decreasing the sensitivity of the cycle to inventory or system perturbations.

2. Heat Transfer

The most important condenser parameters to be evaluated are the overall heat-transfer coefficient and the local mercury condensing coefficient. To obtain the overall coefficient experimentally requires two basic sets of data. The first is the usual heat exchanger defining data of flow rates and temperatures which are straightforward in acquiring. The second requirement is a method of defining the quantity of available heat-transfer area actually used in the condensing process, as distinguished from areas used for subcooling or removing superheat.

The method for evaluating condensing area is based upon the NaK temperature gradient that exists along the condenser length. The method depends upon the considerable variation in heat-transfer rate existing between the heat-transfer processes of condensing and subcooling or desuperheating. Because condensing heat-transfer rates are high, the NaK in passing through the condenser experiences its highest temperature gradient in the section doing the condensing. Thus, a plot of NaK temperature vs condenser length identifies the high-temperature-gradient portion of the condenser and hence the condensing length. This method has been satisfactory for determining condensing areas.

A typical temperature profile is shown in Figure 13. The entering NaK is represented by the right-hand side of the profile. The initial relatively flat slope moving from right to left represents the subcooling area of the condenser with relatively low heat-transfer rates. The steep gradient near the center of the profile represents the high heat-transfer rate area characteristic of condensing. The flat portion of the profile at the left represents an area of no heat transfer.

The overall heat-transfer coefficient is found from

$$U = \frac{Q_c}{A_c \Delta T_{LMTD}}$$

where Q_c is the heat transferred in the condensing process, A_c is the condensing area, and ΔT_{LMTD} is the logarithmic mean temperature difference. The heat transferred is found from the mercury flow rate and the vapor quality calculated from the performance of the mercury vapor desuperheater and the condensing area is found as explained above.

The Logarithmic mean temperature difference cannot be evaluated directly from the condenser inlet and outlet temperatures since the entire condenser is not under consideration, only the condensing portion. The unknown in evaluating the logarithmic mean temperature difference is the temperature of the NaK at the position in the condenser marking the interface between vapor and liquid mercury. This NaK interface temperature is found from

$$T_{\text{NaK, INT}} = \frac{\dot{W}_{\text{Hg}} C_{P\text{Hg}}}{\dot{W}_{\text{NaK}} C_{P\text{NaK}}} (T_{\text{Hg IN}} - T_{\text{Hg OUT}}) + T_{\text{NaK IN}}$$

where W refers to flow rate and C_P refers to specific heat.

The overall heat-transfer coefficient has been calculated using the above procedure. The values obtained are tabulated in Table 3. The tabulated values have been statistically averaged and give a value of 1485 Btu/hr-ft²-°F with a standard deviation of 323 Btu/hr-ft²-°F.

The experimentally determined average overall conductance of 1485 Btu/hr-ft²-°F is considerably lower than the 1900 Btu/hr-ft²-°F value obtained by extrapolation from the MECA data. This discrepancy has been attributed to a lower-than-calculated NaK film coefficient caused by nonideal, shell-side flow distribution. An error analysis has been conducted on the data and instrumentation to determine the magnitude of error that can be expected in calculated values of condenser performance. The possible errors in calculated values are:

Inlet mercury quality	7.4%
Condensing area	4.0%
NaK temperature at interface	6.0%
Logarithmic mean temperature difference	7.2%
Overall conductance	12.2%

The effect of differences in overall conductance on condensing pressure is shown in Figure 14. Three curves of condensing pressure vs condensing area are shown. Two of the curves represent the performance with the MECA and multitube conductances of 1900 Btu/hr-ft²-°F and 1485 Btu/hr-ft²-°F, respectively. The third curve

shows the performance with the minimum design requirement of 960 Btu/hr-ft²-°F.

The local mercury condensing heat-transfer coefficient is more difficult to evaluate. The local coefficient defines one of three resistances in series, which, collectively, define the overall coefficient. The other two resistances are the NaK film resistance and the tube wall resistance. The local mercury condensing coefficient is found by calculating the NaK and wall resistances and then algebraically evaluating the coefficient using the known overall conductance. The relationship is

$$h_{\text{Hg}} = \left[\frac{1}{U} - \frac{1}{h_{\text{NaK}}} - \frac{\frac{x_{\text{wall}}}{k_{\text{wall}}}}{\quad} \right]^{-1}$$

The form of the equation for the local mercury coefficient makes it evident that difficulty could easily be experienced in accurately evaluating the coefficient. If the experimental value of U is a little high, or if the calculated NaK film coefficient* is a little low, the mercury condensing coefficient can become infinite or negative.

The difficulty in determining the local mercury coefficient is encountered in analyzing the SNAP-8 data. As a result, the calculated values of the coefficient suffer from considerable scatter. A statistical analysis of the data in Table 3 gives a local mercury condensing coefficient of 3390 Btu/hr-ft²-°F, but because of the data scatter, the standard deviation is 1290 Btu/hr-ft²-°F.

The condenser is found to be much less sensitive to the mercury condensing coefficient than was originally expected. This decreased sensitivity is illustrated in Figure 15 which presents overall conductance vs mercury condensing coefficient for various values of NaK film coefficient. At the design overall conductance of 960 Btu/hr-ft²-°F, the slope is 0.263 as compared to a slope of 0.066 at the measured overall conductance of 1485 Btu/hr-ft²-°F. The ratio of the two slopes is 4:1. Thus, the condenser is considerably less sensitive to the mercury condensing coefficient than was originally anticipated. This fact will be advantageous in future design work.

* Calculated using the correlation of Dwyer (11).

V. CONCLUSIONS/RECOMMENDATIONS

The first-generation condenser for the SNAP-8 system has been shown to satisfy its thermal design requirements. The structural integrity of the heat exchanger has been satisfactory in the 2000 hours of tests.

Unverified performance parameters include zero-gravity operation and confirmation of pressure drop in a multitube condenser test. These goals will be accomplished in future testing.

REFERENCES

1. B. Misra, Heat Transfer Coefficients in Condensation of Metal Vapors, Ph.D. Dissertation, Dept. of Chem. Eng. Columbia University, New York, N. Y., 1955.
2. B. Misra and C. F. Bonilla, "Heat Transfer in Condensation of Metal Vapors; Mercury and Sodium up to Atmospheric Pressure," Chem. Eng. Prog. Symp. Series No. 18, Vol. 52, 7 (1956).
3. Seymour Lieblein, Ed., Government-Industry Conference on Mercury Condensing, NASA Technical Note D-1188, Feb. 62.
4. Richard G. Gido, Quarterly Progress Reports 1 and 2, NASA Wetting and Non-Wetting Mercury Condensing Research, Contract NAS 3-2159, Jan.-June 62.
5. Stephen S. Papell, An Instability Effect on Two Phase Heat Transfer for Subcooled Water Flowing Under Conditions of Zero Gravity, ARS Space Power Systems Conference, 25-28 Sept. 62, Paper #2548-62.
6. W. C. Reynolds, Hydrodynamic Considerations for the Design of Systems for Very Low Gravity Environments, TR LG-1, Stanford University, Sept. 1961.
7. L. Hays, Investigation of Condensers Applicable to Space Power Systems, Part I, Electro-Optical Systems Report 1588, Final, September 1962.
8. SNAP-8 Topical Materials Report for 1963, Vol. II, Aerojet-General Corp. Report 2822, March 1964.
9. H. Derow, "Inspection of Thin Wall Tube-to-Tube-Sheet Welds Using Gamma Radiography," Welding Engineer, October 1965.
10. R. W. Vernon, R. A. Lottig, and W. D. Kenny, Status Report - MECA Project, Covering Period Sept. 1963 through Dec. 1963, NASA.
11. O. E. Dwyer, Eddy Transport in Liquid-Metal Heat Transfer, Brookhaven National Laboratory Report BNL 6046, February 18, 1962.

TABLE 1

SNAP-8 CONDENSER DESIGN PARAMETERS

1. DESIGN PARAMETERS

NaK FLOW RATE - 35,000 LB/HR

INLET TEMPERATURE - 496°F

INLET PRESSURE - 52.9 PSIA

OUTLET TEMPERATURE - 696°F

OUTLET PRESSURE - 45.9 PSIA

MERCURY FLOW RATE - 11,500 LB/HR

INLET TEMPERATURE - 680°F

OUTLET TEMPERATURE - 505°F

INLET PRESSURE - 15.5 PSIA

OUTLET PRESSURE - 12.0 PSIA

2. SIZE - OVERALL LENGTH - 61.2 IN.

DIAMETER - 10.2 IN. MAXIMUM

TUBE LENGTH - 51.5 IN. MAXIMUM

CONDENSER WEIGHT - 120.2 LB WET

3. INVENTORIES - NaK - 32.6 LB

MERCURY - CAPABLE OF STORING 17.6 LB

OF Hg AND OPERATING SATISFACTORILY

DURING THE LOSS OF THE 10 LB DURING

10,000 HR OF OPERATION

TABLE 2

CONFIGURATION STUDIES

Case No.	Tube I.D. (Inlet) in.	Tube I.D. (Exit) in.	Mercury Inlet Veloc. ft/sec	Mercury Cond. Film Coeff. Btu Hr-°F-ft ²	No. of Tubes	Length of Condenser in.	Shell Diam. (Inlet) in.	Shell Diam. (Exit) in.
1	.45	.125	150	2000	68	43.0	6.41	3.32
2	.40	.125	150	2000	86	36.3	6.48	3.53
3	.5	.125	150	2000	55	52.0	6.36	3.15
4	.4	.20	150	2000	86	32.2	6.48	4.34
5	.45	.20	150	2000	68	38.6	6.41	4.03
6	.50	.20	150	2000	55	45.4	6.36	3.79
7	.4	.25	150	2000	86	30.0	6.48	4.87
8	.45	.25	150	2000	68	36.2	6.41	4.51
9	.50	.25	150	2000	55	42.7	6.36	4.22
10	.4	.125	200	2000	65	48.3	5.61	3.06
11	.45	.125	200	2000	51	58.4	5.55	2.87
12	.5	.125	200	2000	41	67.3	5.50	2.72
13	.4	.20	200	2000	65	42.9	5.61	3.76
14	.45	.20	200	2000	51	51.8	5.55	3.49
15	.5	.20	200	2000	41	61.3	5.50	3.28
16	.4	.25	200	2000	65	39.9	5.61	4.22
17	.45	.25	200	2000	51	48.2	5.55	3.90
18	.5	.25	200	2000	41	57.6	5.50	3.65
19	.4	.125	150	3000	86	28.5	6.48	3.53
20	.45	.125	150	3000	68	34.0	6.41	3.32
21	.5	.125	150	3000	55	41.8	6.36	3.15
22	.4	.20	150	3000	86	25.4	6.48	4.34
23	.45	.20	150	3000	68	30.7	6.41	4.03
24	.5	.20	150	3000	55	36.3	6.36	3.79
25	.4	.25	150	3000	86	23.7	6.48	4.87
26	.45	.25	150	3000	68	28.8	6.41	4.51
27	.5	.25	150	3000	55	34.3	6.36	4.22
28	.4	.125	200	3000	64	38.0	5.61	3.06
29	.45	.125	200	3000	51	46.4	5.55	2.87
30	.50	.125	200	3000	41	53.7	5.50	2.72

Inlet denotes mercury inlet (or coolant exit)

Exit denotes mercury exit (or coolant inlet)

TABLE 2 (cont.)

Case No.	Tube I.D. (Inlet) in.	Tube I.D. (Exit) in.	Mercury Inlet Veloc. ft/sec	Mercury Cond. Film Coeff. Btu	No. of Tubes	Length of Condenser in.	Shell Diam. (Inlet) in.	Shell Diam. (Exit) in.
				Hr-OF-ft ²				
31	.40	.20	200	3000	65	33.9	5.61	3.76
32	.45	.20	200	3000	51	41.3	5.55	3.49
33	.50	.20	200	3000	41	49.2	5.50	3.28
34	.40	.25	200	3000	65	31.6	5.61	4.22
35	.45	.25	200	3000	51	38.4	5.55	3.90
36	.50	.25	200	3000	41	46.3	5.50	3.65
37	.40	.125	150	1000	86	59.5	6.48	3.53
38	.45	.125	150	1000	68	69.9	6.41	3.32
39	.50	.125	150	1000	55	82.5	6.36	3.15
40	.4	.20	150	1000	86	52.6	6.48	4.34
41	.45	.20	150	1000	68	62.4	6.41	4.03
42	.50	.20	150	1000	55	72.7	6.36	3.79
43	.40	.25	150	1000	86	48.8	6.48	4.87
44	.45	.25	150	1000	68	58.2	6.41	4.51
45	.50	.25	150	1000	55	68.2	6.36	4.22
46	.4	.125	200	1000	65	79.4	5.61	3.06
47	.45	.125	200	1000	51	94.2	5.55	2.87
48	.50	.125	200	1000	41	108.0	5.50	2.72
49	.40	.20	200	1000	65	70.1	5.61	3.76
50	.45	.20	200	1000	51	83.5	5.55	3.49
51	.50	.20	200	1000	41	97.7	5.50	3.28
52	.4	.25	200	1000	65	65.0	5.61	4.22
53	.45	.25	200	1000	51	77.6	5.55	3.90
54	.50	.25	200	1000	41	91.5	5.50	3.65
55	.435	.185	150	2000	73	40.0	6.74	4.28
56	.435	.185	150	2000	73	41.4	7.05	4.58

Table 2 Sheet 2

TABLE 2 (cont.)

Case No.	Tube Clearance in.	Heat Transf. Area (Cond) ft ²	U (Cond) Btu Hr-°F-ft ²	Length of Subcooler in.	U (Subcooler) Btu Hr-°F-ft ²	Film Coeff. of Coolant		Pressure Drop Mercury psi	Drop Coolant psi
						Inlet Btu Hr-°F-ft ²	Exit Btu Hr-°F-ft ²		
1	.184	19.7	1169	3.2	1600	4005	4908	3.66	.148
2	.165	19.3	1192	2.50	1610	4453	5332	2.82	.121
3	.203	20.8	1105	4.34	1452	3640	3524	4.70	.185
4	.165	19.4	1185	2.24	1190	4453	4925	.33	.078
5	.184	19.8	1160	2.85	1178	4005	4527	0.81	.097
6	.203	20.3	1136	3.55	1170	3640	4198	1.31	.117
7	.165	19.5	1182	2.13	1014	4453	4736	-.27	.061
8	.184	19.9	1156	2.72	1007	4005	4353	.043	.076
9	.203	20.3	1131	3.38	1000	3640	4035	.33	.093
10	.165	19.3	1193	3.28	1638	4461	5339	6.67	.280
11	.184	20.0	1148	4.36	1558	4014	4271	8.72	.351
12	.203	20.2	1137	5.29	1585	3650	4290	10.42	.415
13	.165	19.4	1186	2.94	1205	4461	4932	1.24	.179
14	.184	20.0	1153	3.79	1183	4014	4311	2.31	.225
15	.203	20.5	1122	4.76	1163	3650	3841	3.45	.275
16	.165	19.5	1182	2.81	1027	4461	4743	-.08	.140
17	.184	19.9	1156	3.57	1022	4014	4361	.47	.176
18	.203	20.6	1119	4.51	999	3650	3739	1.27	.218
19	.165	15.2	1517	2.50	1610	4453	5332	1.95	.098
20	.184	15.6	1477	3.2	1595	4005	4908	2.63	.121
21	.203	16.7	1374	4.34	1452	3640	3524	3.53	.153
22	.165	15.3	1502	2.24	1187	4453	4925	0.00	.063
23	.184	15.8	1460	2.85	1178	4005	4527	.38	.079
24	.203	16.2	1420	3.55	1170	3640	4198	.80	.096
25	.165	15.4	1494	2.13	1014	4453	4736	-.48	.049
26	.184	15.9	1451	2.72	1007	4005	5353	-.25	.062
27	.203	16.3	1411	3.38	1000	3640	4035	.02	.076
28	.165	15.2	1518	3.28	1638	4461	5339	4.77	.226
29	.184	15.9	1443	4.36	1558	4014	4271	6.48	.287
30	.203	16.1	1425	5.29	1585	3650	4290	7.87	.342

TABLE 2 (cont.)

Case No.	Tube Clearance in.	Heat Transf. Area (Cond) ft ²	U (Cond) Btu Hr-°F-ft ²	Length of Subcooler in.	U (Subcooler) Btu Hr-°F-ft ²	Film Coeff. of Coolant		Pressure Drop Mercury psi	Drop Coolant psi
						Inlet	Exit		
						Btu Hr-°F-ft ²	Btu Hr-°F-ft ²		
31	.165	15.3	1502	2.94	1205	4461	4932	.51	.145
32	.184	15.9	1448	3.79	1183	4014	4311	1.38	.184
33	.203	16.5	1398	4.76	1163	3650	3841	2.33	.227
34	.165	15.4	1495	2.81	1027	4461	4743	-.53	.113
35	.184	15.8	1452	3.57	1022	4014	4361	-.03	.143
36	.203	16.5	1392	4.51	999	3650	3739	.58	.180
37	.165	31.7	726	2.50	1610	4453	5332	5.43	.191
38	.184	32.0	720	3.20	1595	4005	4908	6.73	.230
39	.203	33.1	696	4.34	1452	3640	3524	8.20	.279
40	.165	31.7	726	2.24	1187	4453	4925	1.34	.122
41	.184	32.0	718	2.85	1178	4005	4527	2.08	.150
42	.203	32.4	710	3.55	1170	3640	4198	2.85	.179
43	.165	31.7	726	2.13	1014	4453	4736	.28	.096
44	.184	32.1	718	2.72	1007	4005	4353	.74	.118
45	.203	32.5	709	3.38	1000	3640	4035	1.27	.142
46	.165	31.7	727	3.28	1638	4461	5339	12.38	.441
47	.184	32.3	711	4.36	1558	4014	4271	15.43	.541
48	.203	32.5	709	5.29	1585	3650	4290	18.07	.635
49	.165	31.7	727	2.94	1205	4461	4932	3.44	.281
50	.184	32.2	715	3.79	1183	4014	4311	5.08	.348
51	.203	32.7	704	4.76	1163	3650	3841	6.81	.420
52	.165	31.7	726	2.81	1027	4461	4743	1.12	.220
53	.184	32.1	718	3.57	1022	4014	4361	2.12	.273
54	.203	32.7	704	4.51	999	3650	3739	3.34	.333
55	.179	22.03	1044	2.88	1090	4113	4562	1.16	.09
56	.210	22.8	1010	2.96	1061	3489	4022	1.25	.054

Table 2 Sheet 4

TABLE 3

SNAP-8 CONDENSER TEST SUMMARY

Data Point	Date Time	$\dot{w}_{\text{Hg}}$ Hg Flow lb/hr	P_c Hg-inlet Pressure psia	$\dot{w}_N$ NaK Flow lb/hr	$T_{N\text{-in}}$ NaK Inlet Temp °F	ΔT_{NT} NaK Temp Rise °F	X_{in} Hg Inlet Quality %	ΔP_c Hg Pressure Drop psi	A_c Cond Area ft ²
1	7-15-65 2320	9,500	6.2	30,000	350	166	96.8	15.5	8.8
2	7-16-65 0020	10,500	7.6	30,000	381	185	98.1	15.6	10.0
3	0115	11,200	6.8	30,000	367	192	98.6	14.2	9.5
4	0150	11,750	7.1	30,000	372	202	98.6	14.9	12.5
5	0225	11,500	7.5	30,000	371	195	98.7	7.5	8.1
6	0310	11,600	7.4	30,000	371	200	98.6	18.5	8.4
7	0355	12,200	7.0	30,000	374	213	99.2	19.6	11.0
8	0440	12,200	6.4	33,300	382	194	98.5	14.5	7.5
9	0520	11,750	6.7	33,300	378	196	98.0	15.9	7.3
10	0600	11,750	6.1	32,500	379	192	98.3	9.2	8.5
11	0645	11,750	6.1	33,300	383	193	98.2	18.9	9.1
12	0735	11,200	4.2	33,300	374	178	98.3	17.4	11.6
13	7-20-65 0135	11,750	6.1	35,500	334	183	88.9	14.3	7.9
14	0230	11,750	8.8	34,500	404	185	90.4	14.5	10.2
15	0312	11,750	9.0	33,000	398	194	90.8	13.4	10.0
16	0402	11,750	9.3	30,500	383	218	91.5	14.7	7.8
17	0445	11,750	7.6	30,000	353	216	90.9	14.5	10.0
18	0525	11,750	6.9	32,500	355	196	90.0	13.7	10.6
19	0609	11,750	6.7	34,500	359	183	89.8	14.2	9.4
20	0706	9,500	5.1	32,500	333	150	86.0	14.8	4.5
21	0748	10,500	5.8	34,500	351	162	86.5	15.7	6.1
22	0850	11,200	6.3	35,000	358	169	87.0	14.8	9.6
23	0935	11,750	6.8	34,500	359	178	86.2	15.4	9.6
24	1015	11,750	6.1	35,000	360	178	87.1	7.1	8.3
25	1100	11,750	8.0	35,000	370	181	83.9	16.1	6.1
26	1145	12,200	7.5	35,000	379	192	87.7	7.7	9.0

Table 3 Sheet 1

TABLE 3 (cont.)

Data Point	IMTD Log Mean ΔT	Q_c Thermal Load (Condensing) Btu/hr	U_c Overall Heat Transfer Coefficient Btu/hr-ft ² -°F	h_{Hg} Calculated Mercury Film Coefficient Btu/hr-ft ² -°F	L_c Condensing Length in.	ϵ Condensing Region Effectiveness
1	136	1.17×10^6	978	1515	18.5	.73
2	110	1.30×10^6	1194	2105	18.3	.84
3	120	1.40×10^6	1233	2227	21.3	.84
4	113	1.47×10^6	1043	1678	19.8	.88
5	117	1.44×10^6	1515	3356	36.7	.86
6	120	1.45×10^6	1449	3049	12.1	.85
7	108	1.54×10^6	1290	2421	9.9	.90
8	118	1.40×10^6	1724	4608	20.7	.84
9	117	1.46×10^6	1731	4525	17.7	.84
10	115	1.46×10^6	1496	3268	31.9	.84
11	119	1.46×10^6	1345	2625	11.4	.83
12	125	1.40×10^6	963	1479	14.5	.80
13	106	1.10×10^6	1320	2525	21.1	.77
14	117	1.35×10^6	1130	1912	20.7	.80
15	95	1.35×10^6	1430	2967	23.0	.87
16	104	1.36×10^6	1690	4348	20.2	.87
17	100	1.36×10^6	1360	2681	20.7	.89
18	67	1.42×10^6	1885	5917	22.3	.96
19	77	1.34×10^6	1863	5714	21.3	.91
20	127	1.04×10^6	1818	5319	20.0	.70
21	101	1.15×10^6	1859	5682	18.1	.79
22	90	1.21×10^6	1434	2985	20.1	.85
23	83	1.28×10^6	1623	3937	28.7	.88
24	77	1.30×10^6	2040	7813	36.3	.90
25	70	1.25×10^6	2910	00	17.4	.91
26	88	1.36×10^6	1715	4525	35.8	.88

Cutaway Drawing of the SNAP-8 Condenser

Fig. 1 SNAP-8 Condenser

Fig. 2 Comparison of Velocity Profiles, Straight and Tapered Tubes

Fig. 3 Tube Clearance and Number of Tubes with Inside Diameter

Fig. 4 Mercury Condensing Length with Tube Diameter and Mercury Film Coefficient

Fig. 5 Subcooler Length vs Tube Inside Diameter

Fig. 6 Condenser Heat Transfer Area with Tube Diameter

BASED ON: $V = 150 \text{ FT/SEC}$
 $D = .20 \text{ INCHES}$

Fig. 7 Condenser Unit Conductance vs Tube Inside Diameter

Fig. 8 Mercury Pressure Drop vs Tube Inside Diameter

TUBE - TUBE HEADER WELDMENT -1 CONDENSER
TREPAN - WITH 4° TAPER FOR BACK BRAZE

Fig. 9 Welded and Back-Brazed Joint Schematic

Fig. 10 Rolled and Welded Joint Schematic

PICTORIAL PRESENTATION OF GAMMA RADIOGRAPHY SETUP FOR INSPECTION OF TUBE-TO-TUBE-SHEET JOINTS

Fig. 11 Gamma-radiograph Inspection Setup

Fig. 12 Condensing Pressure with Interface Position

Fig. 13 Typical Condenser NaK-Side Profile

Fig. 14 Variation in Condensing Pressure with Overall Heat Transfer Coefficient

Fig. 15 Overall Conductance vs Condensing Coefficient

THERMAL AND FLUID DYNAMIC PERFORMANCE OF POTASSIUM
DURING CONDENSATION INSIDE A TUBE

S.G. Sawochka

Space Power and Propulsion Section
Missile and Space Division
General Electric Company
Cincinnati, Ohio 45215

Presented at AIAA 1st Rankine
Cycle Space Power System Conference
October 26-28, 1965
NASA Lewis Research Center
Cleveland, Ohio

ABSTRACT

Forced convection heat transfer and pressure change data for the condensation of potassium inside a vertical tube are presented and discussed. These data, obtained at saturation temperatures up to 1400°F and heat fluxes up to $3(10^5)$ Btu/hr-ft², provide a reasonable basis for the thermal design of potassium space power system condensers. A comparison of analytical and measured fluid temperature profiles is given which indicates that estimation of local potassium temperatures from measured condenser inlet and outlet temperatures is of questionable accuracy for calculation of reliable condensing heat transfer coefficients. The response of the test condenser to a change in location of the liquid-vapor interface is discussed.

INTRODUCTION

Experimental and analytical work was performed to define the thermal and fluid dynamic performance of potassium during condensation for application to space power system condenser design. Examination of the available literature shows that data for condensation of low Prandtl number fluids are scarce (1)¹, and that experimental heat transfer coefficients have been lower than predicted analytically by the Nusselt model (2). To explain this discrepancy between experiment and analysis a number of analytical investigations have been completed (3 - 8) which modify Nusselt's model to include the effects of gas phase shear, film turbulence, and property variation due to the temperature gradient across the liquid film. However, none of the modifications described in the literature predict heat transfer coefficients in satisfactory agreement with experimental condensing heat transfer data of low Prandtl number fluids. Also, pressure change data for forced convection condensation of potassium have not been reported in the literature.

Forced convection condensing heat transfer and pressure change data for potassium, obtained during this program at saturation temperatures from 1100 to 1400°F, are presented. The response of the test condenser to a change in location of the liquid-vapor interface is also briefly discussed.

1 Underlined numbers in parenthesis designate references at end of paper.

EXPERIMENTAL APPARATUS

Facility - Figure 1 is a schematic of the facility used in this investigation (9). The facility, except for the test section, was constructed of type 316 stainless steel, selected for its high strength and compatibility with alkali metals at temperatures up to 1600°F.

Potassium vapor was generated in a pot boiler with immersion heaters capable of 50 KW heat input. The high quality (approximately 100%) vapor from the boiler passed through connecting piping and then entered the top of the vertical sodium cooled test condenser. During heat transfer tests total condensation was obtained in the test section, and liquid was returned to the boiler through a calorimetric flowmeter, electromagnetic flowmeter, helical induction pump, and liquid throttle valve.

The sodium loop, an all-liquid loop, was used to cool the test condenser. The cooling loop was comprised of an electromagnetic pump, a flowmeter, and an air cooler for control of the sodium temperature into the test section. A dump tank was provided for storing the sodium during shutdown.

Test Section - The five test sections used in these experiments were of a design similar to that shown in Figure 2 (10). Each had an active heat transfer length of 36-inches with potassium condensation occurring inside a thick walled nickel tube cooled by a countercurrent sodium

flow in the surrounding annulus. All tests were performed with the potassium in vertical downflow. The five test geometries are listed in Table 1.

Table 1
Condensing Test Geometries

<u>Number</u>	<u>Geometry</u>
1	5/8-inch ID tube without insert
2	5/8-inch ID tube with tapered pin insert
3	3/8-inch ID tube without insert
4	5/8-inch ID tube with helical insert of pitch to diameter ratio = 6 (Instrumented insert)
5	5/8-inch ID tube with one-quarter-inch OD tubular insert (Instrumented insert)

The two instrumented inserts and the non-instrumented tapered pin insert are shown in Figure 3.

In the last four test sections, ten 0.050-inch diameter axial thermocouple holes, five entering from each end of the test piece, were drilled by an electrical discharge process to a 9-inch depth in the thick wall of the nickel tube. Thermocouples in these holes provided tube wall radial temperature profiles at two axial locations from which local values of the inner wall temperature and heat flux were determined. The axial positions of the top and bottom measuring stations varied with test geometry from 6 to 8-inches and 28 to 30¹/₂-inches respectively, from the test section inlet. During the first

test series, with the 5/8-inch ID tube without insert, a different thermocouple hole manufacturing technique was used, as described in Reference 11.

Potassium inlet and outlet temperatures were measured during the tests with the 5/8-inch and 3/8-inch ID tubes without inserts and with the 5/8-inch ID tube with tapered pin insert. The helical and tubular inserts tested in the 5/8-inch ID tube were furnished with eleven internal thermocouples, three of which preceded the active condensing length and eight of which were evenly spaced along the active condensing length of the test condenser. With these thermocouples the axial temperature and pressure distributions were determined for the last two test sections. The pressure distribution was obtained from the measured temperature distribution assuming that thermal equilibrium existed between the liquid and vapor phases, and therefore that the local pressure could be obtained from the equilibrium pressure-temperature relation. Inlet and outlet sodium temperatures were measured using three circumferentially spaced thermocouples at each end of the test condenser.

Test Procedure - To perform a systematic series of condensing tests, the following test procedure was used:

1. The potassium loop was evacuated at 600°F to a pressure of approximately 25 microns, and the difference between liquid and vapor boiler temperatures was used to check for the partial pressure of inert gases in the system. If a substantial difference, greater than 10°F, between those temperatures was obtained, the loop was re-evacuated.

2. After establishing sodium flow, a boiler power was set, and the temperature of the sodium was adjusted using the air cooler and line heat to bring the potassium to the desired test temperature.

3. At a constant boiler power, the sodium temperature was varied to obtain test data for potassium inlet temperatures from 1100 to 1400°F.

4. When a series of tests was completed at a given boiler power, the boiler power was increased, and the adjustment of the potassium condensing temperature was repeated using the sodium temperature as a control.

This procedure gave a series of heat transfer and pressure drop data at incremental condenser loads and potassium temperatures for the flow rate and temperature ranges of the facility. In addition, before and after each test series, the thermocouples in the potassium and nickel tube wall were calibrated in place relative to one of the potassium thermocouples at temperatures of 1200 and 1300°F. On the basis of these calibrations each thermocouple was assigned a systematic temperature correction as a function of temperature. The sodium well thermocouples were also calibrated relative to one of the inlet sodium well thermocouples using a technique described in Reference 12.

Measurements of the potassium saturation temperature, heat flux, and inner wall temperature were required to calculate the condensing heat transfer coefficient. As noted earlier, the inner wall temperature and local heat flux were determined from the tube wall radial temperature profile. These values were

obtained by a least squares procedure which was used to fit the measured temperatures within the nickel tube wall with the integrated Fourier conduction equation. Top and bottom measuring station radial temperature profiles are shown in Figure 4 for the 5/8-inch ID tube with tubular insert for an average heat flux of $1.2(10^5)$ Btu/hr-ft² and an inlet vapor temperature of 1295°F. The local heat flux was calculated from the slope of the radial temperature profile.

For the first three test sections, the 5/8-inch and 3/8-inch ID tubes without inserts and the 5/8-inch ID tube with tapered pin insert, only inlet and outlet potassium temperatures were measured. Estimation of the local potassium temperature from the measured inlet and outlet values with sufficient accuracy to allow reliable condensing heat transfer coefficients for these three test sections to be calculated could not be accomplished. Therefore, the inserts of the last two test sections, the 5/8-inch ID tube with helical and tubular inserts, were instrumented to measure the local potassium temperature thereby allowing reliable condensing heat transfer coefficients to be calculated.

RESULTS

Potassium Axial Temperature Distribution - The measured potassium axial temperature distribution and related saturation pressures, corresponding to the condenser radial wall temperature distribution of Figure 4, are presented in Figure 5a. As can be seen, the potassium pressure initially decreases indicating the influence of the frictional pressure gradient compared to the momentum pressure gradient. After the potassium pressure

reaches a minimum, it then increases due to the increasing influence of the momentum pressure gradient toward the end of the condensing length. For this run, vapor to tube wall temperature differences of 4.8 and 10.5°F were calculated for local heat fluxes of $0.98(10^5)$ and $1.3(10^5)$ Btu/hr-ft², respectively. Presented in Figure 5b are the dimensionless temperature profile of Figure 5a and analytical dimensionless temperature profiles for linear axial quality variation with homogeneous flow (13) and with a slip ratio of $\sqrt{\rho_f/\rho_g}$, as suggested by Fauske (14). Neither prediction gives good agreement, evidencing the complexity of the problem due to effects such as heat flux distribution and variation of the slip ratio with length. Since agreement between the analytical and experimental temperature profiles could not be obtained for the tubular insert test section, treatment of the data for which local saturation temperatures were not measured was limited to determining that the condensing heat transfer coefficients were the same order of magnitude as those determined with the instrumented insert test sections.

The axial temperature distribution presented in Figure 5a is representative of those obtained with the tubular insert test geometry in the 1100°F to 1400°F temperature range at condenser heat loads up to approximately 20 KW. At higher heat loads, however, departure from this type of distribution occurred. In fact, a net pressure drop was obtained across the condensing length under some operating conditions.

Heat Transfer - The experimental condensing heat transfer coefficients obtained for the two instrumented insert test sections are presented in Figure 6 as a function of the local film

Reynolds number. The calculations of the heat transfer coefficients are based on measured saturation temperature, and no allowance has been made for radial pressure gradients present with the helical insert. Nusselt's laminar film (2) and Seban's turbulent film (3) predictions are shown for comparison for a temperature of 1300°F. Figure 6 shows that the condensing heat transfer coefficients are relatively independent of the test section geometry and are generally lower than predicted by the Nusselt and Seban models, even though these two models neglect the effect of a cocurrent vapor shear at the vapor-liquid interface.

Substantial scatter is evident in the data of Figure 6. To explain this scatter, a comprehensive treatment of the data for the 5/8-inch ID tube with tubular insert was made. This data was selected for treatment since: 1) the test section geometry approximated the falling film model with a cocurrent shear due to the vapor flow; 2) the local potassium temperature was measured, and 3) a large range of test conditions was covered. The data were plotted as a function of local heat flux, superficial mass velocity, and local vapor temperature in Figure 7. As can be seen in Figure 7, the condensing heat transfer coefficient appears to be relatively independent of heat flux, mass velocity, and temperature within the experimental ranges. Over these ranges the average heat transfer coefficient is approximately 15,000 Btu/hr-ft²-°F. Figure 7 indicates a possible decrease in the condensing heat transfer coefficient with saturation temperature below 1100°F. This decrease of the heat transfer coefficient with decreasing temperature suggests the presence of a vapor phase thermal resistance similar to that proposed by Schrage (15) and applied by Rohsenow and Sukhatme (16).

A vapor phase thermal resistance was determined for the data shown in Figure 7 by making the following two assumptions:

1. It was assumed that the experimental condensing thermal resistance is a sum of a liquid phase and a vapor phase thermal resistance, then*

$$\frac{1}{h_c} = \frac{1}{h_f} + \frac{1}{h_g} \quad (1)$$

2. To estimate the liquid phase thermal resistance, annular flow with no entrainment was assumed, and the film thickness prediction of Dukler (17) with $h_f = k/\delta$ was used.

The vapor phase thermal resistance was obtained by subtracting the liquid film thermal resistance from the experimental values of the condensing thermal resistance. This experimentally obtained vapor phase thermal resistance for potassium was shown (10, 18) to exhibit trends and magnitudes consistent with the kinetic theory of gases for a "condensation coefficient" of 0.2. The "condensation coefficient" is defined as the fraction of the vapor molecules impinging on the liquid-vapor interface which actually condense and is equal to one for ideal gases. This indicated nonideal kinetic theory behavior can explain the observed trend of decreasing heat transfer coefficient with decreasing temperature and also the low experimental values of the heat transfer coefficient compared to analytical predictions which treat the liquid film as the only thermal resistance during condensation.

Pressure Change - The pressure gradient associated with the condensing process can be expressed (10) as the sum of the momentum and friction pressure gradients:

* Symbols are defined in the Nomenclature at the end.

$$\frac{dP}{d\lambda} = \left(\frac{dP}{d\lambda}\right)_M + \left(\frac{dP}{d\lambda}\right)_{fr} \quad (2)$$

or

$$\frac{dP}{d\lambda} = \frac{G^2}{g_o} \left[\frac{d}{d\lambda} \left(\frac{1}{\hat{\rho}} \right) - \left(\frac{\phi_f}{2D \rho_f} \right) \right] \quad (3)$$

where

$$\frac{1}{\hat{\rho}} = \left[\frac{(1-x) K}{\rho_f} + \frac{x}{\rho_g} \right] \left[\frac{1}{K} + x - \frac{x}{K} \right] \quad (4)$$

in which K is the vapor to liquid velocity ratio. For homogeneous flow, $K = 1$, and Equation (4) reduces to

$$\frac{1}{\hat{\rho}} = \frac{1-x}{\rho_f} + \frac{x}{\rho_g} \quad (5)$$

Integration of Equation (3) from 100% to 0% quality gives:

$$\frac{P_O - P_I}{G^2 / \rho_f g_o} = \left(\frac{\rho_f}{\rho_g} - 1 \right) - \frac{fL}{2D} \phi_{1-0} \quad (6)$$

To obtain the dimensionless integrated friction pressure drop component of the pressure change, $(\frac{fL}{2D} \phi_{1-0})$, it was assumed that the quality variation with length was linear. The dimensionless momentum pressure recovery term, $(\frac{\rho_f}{\rho_g} - 1)$, is independent of the functional variation of quality with length.

To determine whether a net pressure recovery or loss occurs over a condensing length, it is necessary to estimate the following ratio:

$$\frac{\rho_f/\rho_g - 1}{\phi_{1-o} \frac{fL}{2D}}$$

If this ratio is greater than one, a net pressure recovery occurs. If this ratio is less than one, a net pressure drop occurs. This criterion is only applicable to the condensing length, does not consider inlet and exit losses, and can be applied only to condenser tubes of constant cross-sectional flow area, under conditions for which the two-phase friction pressure drop multiplier can be estimated.

Measured temperature distributions and corresponding saturation pressures are presented in Figure 8 for the 5/8-inch ID tube with tubular insert for different heat loads and a potassium inlet temperature of approximately 1150°F. As can be seen, at the beginning of the condensing length $dP/d\lambda$ is negative indicating that the friction pressure gradient is dominant in relation to the momentum pressure gradient. As the condensing process continues, the two-phase frictional pressure drop multiplier decreases due to the decreasing quality, and $dP/d\lambda$ eventually becomes positive, indicating a dominant contribution of the momentum pressure gradient in relation to the friction pressure gradient.

For the two runs presented in Figure 8, a net pressure recovery was indicated across the condensing length. It can be seen by comparison of the results for the two runs that increasing the mass velocity G (by increasing the boiler heat input), at approximately constant potassium inlet temperature, resulted in an increased pressure rise across the condenser. The two data

runs presented in Figure 8 are typical of the majority of the data obtained at condenser heat loads less than 20 KW in that $\Delta P/\Delta G$ across the test condenser was negative. There was no evidence of instability during these runs.

Two-Phase Multiplier - The applicability of the homogeneous model and the Martinelli model (13) for estimating the integrated two-phase friction pressure drop multiplier was investigated using the data for the 5/8-inch ID tube with tubular and helical inserts. A comparison of the experimental data to the predicted integrated pressure drop multipliers from the Martinelli and homogeneous models is presented in Figure 9. As suggested in Reference 10, the maximum helical velocity and maximum helical length were used to estimate the single-phase liquid pressure drop for the helical insert. Figure 9 indicates that the experimental two-phase multipliers are in reasonable agreement with the predictions of the Martinelli and homogeneous models, with somewhat better agreement obtained with the Martinelli prediction.

Liquid-Vapor Interface Test - During the heat transfer tests, the liquid-vapor interface was maintained about a foot downstream of the condensing length. After completion of the heat transfer tests, the stability of operation was investigated with the liquid-vapor interface in the active condensing section of the 5/8-inch ID tube with tubular insert. The initial conditions with the interface downstream of the condensing length were:

Boiler Power	25 KW
Average Heat Flux	120,000 Btu/hr-ft ²
Potassium Flow Rate	65 lb/hr
Test Section Load	20 KW

Subsequently, the liquid potassium flow rate from the condenser was gradually reduced at constant boiler power and sodium flow rate. In this manner the interface was raised from its normal operating position downstream of the test section into the test section. The results of this test are presented in Figure 10. Figure 10 indicates that after the liquid flow rate from the condenser had been reduced to 56 lb/hr from 65 lb/hr during a period of about one hour, the interface entered the heat transfer length of the condenser. This change in interface position is indicated by a drop in potassium temperature at the test section exit to approximately the sodium inlet temperature. About 30 seconds later the interface passed the insert thermocouple located 5-inches upstream of the test section exit. With a subsequent reduction in flow rate, the interface was established at a point located between 10 and 15-inches from the test section exit (21 to 26-inches from the test section inlet), as indicated by the insert thermocouples at these positions, and was held there for an hour. During this period the insert thermocouple located directly above the interface, 15-inches from the test section exit, indicated a temperature approximately equal to the potassium test section inlet temperature, while the insert thermocouple located just below the interface, 10-inches from the test section exit, indicated a temperature approximately equal to the sodium inlet temperature. The difference between the sodium and potassium inlet temperatures was approximately 140°F. Figure 10 indicates that the system adjusted to the new interface position gradually without any severe temperature excursions. After the interface

had been maintained at the location between 10 and 15-inches upstream of the test section exit for a period of an hour, it was then returned to its normal operating position in the surge tank downstream of the test section by an increase in potassium liquid flow rate from the condenser. The conclusions of this test are:

1. The interface could be maintained within the condensing length of the vertical condenser without the system undergoing measurable temperature excursions.

2. A temperature differential in the potassium approximately equal to the difference between the potassium saturation temperature and the sodium inlet temperature is established across the interface.

3. No unexpected system excursions were detected when the interface was maintained within the condensing length of the test section.

SUMMARY AND CONCLUDING REMARKS

Data on the thermal and hydraulic performance of potassium during condensation inside a 3/8-inch ID tube without insert and a 5/8-inch ID tube without insert and with tapered pin, helical and tubular inserts have been obtained. The most reliable of these data are those obtained in the 5/8-inch ID tube with a helical and a tubular insert since for these data the local potassium saturation temperature, which is required to determine the local condensing heat transfer coefficient, was measured directly by thermocouples in the inserts. Evaluation of these data suggests the following:

1. The condensing heat transfer coefficient is relatively high, typically in excess of $10,000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, in the 1100 to 1400°F temperature range at local heat fluxes from $3(10^4)$ to $3(10^5) \text{ Btu/hr-ft}^2$, and film Reynolds numbers approaching 10^4 . No trends of the condensing heat transfer coefficient with heat flux, superficial mass velocity or temperature were noted for the above ranges of parameters.

2. Direct measurement of the local potassium saturation temperature, which was obtained using instrumented inserts, was found necessary to accurately determine potassium condensing heat transfer coefficients.

3. The two-phase friction pressure drop multiplier for the $5/8$ -inch ID tube with helical and tubular insert could be estimated by either the Martinelli or homogeneous methods, although somewhat better agreement was obtained with the Martinelli method.

4. The vertical test condenser operated in downflow equally well with either a pressure rise or pressure drop between the condenser inlet and exit. No instability could be detected when an increase in pressure rise across the condensing length was obtained by increasing the flow rate.

5. The test condenser could be operated satisfactorily with the liquid-vapor interface within its active condensing length. In this case, a temperature difference equal to that between the potassium and sodium inlet temperatures was measured across the interface.

BIBLIOGRAPHY

1. "Condensation of Metal Vapors: Mercury and the Kinetic Theory of Condensation," Wilhelm, D.J., ANL 6948, October 1964.
2. Heat Transmission, McAdams, W.H., 3rd Edition, McGraw Hill, Inc., New York, New York, 1954.
3. "Remarks on Film Condensation with Turbulent Flow," Seban, R.A., ASME Trans., 1954, p. 299.
4. "Heat Transfer and Temperature Distribution in Laminar Film Condensation," Rohsenow, W.M., Trans. ASME, Vol. 78, pp. 1645 - 1648, November 1956.
5. "Effect of Vapor Velocity on Laminar and Turbulent Film Condensation," Rohsenow, W.M., Weber, J.H., and Ling, A.J., Trans. ASME, Vol. 78, pp. 1637-1643, November 1956
6. "The Two-Phase Boundary Layer in Laminar Film Condensation," Koh, J.C.Y., Sparrow, E.M., and Hartnett, J.P., Int. J. Heat Mass Trans., Vol. 2, pp. 69-82, 1961
7. "An Analytical Study of Laminar Film Condensation: Part 1 - Flat Plates, and Part 2 - Single and Multiple Horizontal Tubes," Chen, M.M., Trans. ASME, Series C, J. Heat Trans., Vol. 83, No. 1, pp. 48-54 and pp. 55-60, February 1961.
8. "Turbulent Film Condensation," Lee, Jon, AIChE Journal, pp. 540-544, July 19, 1964, Vol. 10, No. 4.
9. "Alkali Metals Boiling and Condensing Investigations," Quarterly Report 9, Contract NAS 3-2528, Space Power and Propulsion Section, MSD, General Electric Company, October 20, 1964, NASA CR 54215.
10. "Alkali Metals Boiling and Condensing Investigations," Quarterly Report No. 12, Contract NAS 3-2528, Space Power and Propulsion Section, MSD, General Electric Company, July 23, 1965, NAS CR 54739.
11. "Alkali Metals Boiling and Condensing Investigations," Quarterly Report No. 4, Contract NAS 3-2528, Space Power and Propulsion Section, MSD, General Electric Company, July 25, 1963.

BIBLIOGRAPHY (Cont'd)

12. "Alkali Metals Boiling and Condensing Investigations," Quarterly Report 6, Contract NAS 3-2528, Space Power and Propulsion Section, MSD, General Electric Company, April 20, 1964, NASA CR 54037.
13. "Alkali Metals Boiling and Condensing Investigations," Quarterly Report No. 8, Contract NAS 3-2528, Space Power and Propulsion Section, MSD, General Electric Company, July 20, 1964, NASA CR 54138.
14. "Contribution to the Theory of Two-Phase, One-Component Critical Flow", Fauske, H.F., ANL-6633, October 1962.
15. A Theoretical Study of Interphase Mass Transfer, Schrage, R.W., Columbia University Press, New York, 1953.
16. "Heat Transfer During Film Condensation of a Liquid Metal Vapor," Sukhatme, S.P., and Rohsenow, W.M., Technical Report No. 9167-27, Dept. of Mech. Eng., MIT, April 1964.
17. "Fluid Mechanics and Heat Transfer in Vertical Falling-Film Systems," Dukler, A.E., Chem. Eng. Prog. Symposium Series, No. 30, Vol. 56, 1960.
18. "Heat Transfer During the Condensation of Potassium," Sawochka, S.G., Ph.D. Thesis, Department of Chemical Engineering, University of Cincinnati, 1965.

ACKNOWLEDGMENTS

The work described herein was accomplished under the sponsorship of the National Aeronautics and Space Administration under Contract NAS 3-2528.

NOMENCLATURE

Latin Letter Symbols

<u>Symbols</u>	<u>Quantity</u>	<u>Unit</u>
A	Flow area	ft ²
D	Diameter	ft
f	Friction factor	dimensionless
G	Mass velocity	lb _m /ft ² -sec
g _o	Conversion factor	32.2 ft-lb _m /lb _f -sec ²
h	Heat transfer coefficient	Btu/hr-ft ² -°F
k	Liquid thermal conductivity	Btu/hr-ft -°F
K	Vapor to liquid velocity ratio	dimensionless
L	Total condensing length	ft
ℓ	length	ft
P	Pressure	lb _f /ft ²
p	pitch	ft
q	Rate of heat flow	Btu/hr
q"	heat flux	Btu/hr-ft ²
R	Radius	ft
T	Temperature	°F
x	Quality	dimensionless

Greek Letter Symbols

Γ	Condensate flowrate per unit circumference	lb _m /ft-hr
Δ	increment	dimensionless
δ	Liquid film thickness	ft
μ	Absolute viscosity	lb _m /ft-hr
ρ	Density	lb _m /ft ³
Ø	Local two-phase drop multiplier	dimensionless
Ø _{1-o}	Integrated two-phase pressure drop multiplier (x = 1.0 to x = 0)	dimensionless

Subscripts

avg.	Average
c	condensing
f	liquid
fr	friction
g	vapor
i	inside
I	Inlet
K	potassium
ℓ	local
M	Momentum
Na	Sodium
O	Outlet
sat	Saturation

Figure 1 50 KW Heat Transfer Test Facility

Figure 2 Test Section Schematic

Figure 3 Inserts Tested in $5/8$ -inch ID Condenser Tube

Figure 4 Nickel Tube Wall Radial Temperature Profiles for $5/8$ -inch ID Tube with Tubular Insert

Figure 5. Potassium Axial Temperature Distributions for 5/8-inch ID tube with 1/4-inch OD Insert

Figure 6 Condensing Heat Transfer Results

Figure 7. Variation of Potassium Condensing Heat Transfer Coefficients with Mass Velocity, Heat Flux and Saturation Temperature (50 KW Facility, 5/8-inch ID Tube with 1/4-inch Instrumented Tubular Insert)

Figure 8 Integrated Two-Phase Multipliers as a Function of Temperature

Figure 9. Variation of Potassium Axial Temperature Distributions and Corresponding Saturation Pressures with Superficial Mass Velocity (5/8-inch ID tube with 1/4-inch OD Instrumented Tubular Insert)

Segment 1

Segment 2

Initial Conditions $(q/A)_{avg} = 120,000 \text{ BTU/Hr.Ft}^2$
 $\dot{W}_K = 65 \text{ lb/hr}$
 Test Section $q = 20 \text{ KW}$

Constant Boiler Power of 25 KW

Figure 10. Results of Liquid Vapor Interface Test
 (50 KW Facility, 5/8-inch ID Tube with
 1/4-inch Instrumented Tubular Insert)

POTASSIUM CONDENSING TESTS OF HORIZONTAL MULTITUBE

CONVECTIVE AND RADIATIVE CONDENSERS OPERATING AT

VAPOR TEMPERATURES OF 1250° to 1500°F

by O. A. Gutierrez, N. J. Sekas, L. W. Acker, and D. B. Fenn

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio

ABSTRACT

Two types of horizontally oriented multitube vapor condensers, dissipating heat loads up to 150 KW thermal, were tested over a range of vapor temperatures from 1250° to 1500°F. One condenser was a 7-tube counterflow shell-and-tube type convective heat exchanger, condensing potassium vapor inside the tubes and using NaK 78 as the shell side coolant. The other unit was a radiative type condenser, consisting of 9 coplanar, parallel-finned tubes manifolded to common inlet and outlet headers. This unit was tested inside an oil-cooled chamber in a vacuum environment of 3×10^{-3} torr. The objective of the tests was to study the performance and operating problems associated with current state-of-the-art units at near realistic operating conditions. The tests were conducted in the Space Radiator and Condenser Facility at the NASA Lewis Research Center. This paper presents some of the results obtained from these tests as well as a description of the units tested. A brief description of the facility and its capabilities is also included. It appears from these tests that it is possible to operate either type of multitube condenser in a stable manner over a large range of operating conditions. The advisability of using cocurrent coolant flow at the vapor inlet end of convective condensers is indicated, and the importance of properly evaluating inlet pressure drops is also pointed out by the results.

INTRODUCTION

Rankine cycle power generation systems for use in space must depend on radiation to reject the waste heat. As shown in figure 1, the waste heat can be dissipated in a radiative condenser, wherein the working fluid is condensed in a radiator. An alternate method is to use an intermediate liquid-cooled convective condenser, with the heat being lost to space from an all-liquid radiator. Each of these schemes has relative advantages and disadvantages, so that both should be studied and evaluated.

Also of interest in the application of Rankine power cycles to space missions is the use of alkali liquid metals as working fluids. The advantages of such fluids are obvious: their high saturation temperatures and low vapor pressures allow large reductions in system size and weight when compared with more conventional working fluids.

As part of the Lewis Research Center overall investigation in this field, a program was instituted to determine the steady-state operating characteristics of liquid metal condensers operating at near-realistic conditions and to define problem areas in their design and performance. Both a convective and a radiative condenser of multitube design were tested in a horizontal attitude to minimize the effect of gravity on their performance. The convective condenser was of the shell-and-tube type, and the radiative condenser was a coplanar centrally finned unit. Potassium was used as the test fluid.

This paper presents some experimental results from both condensers, as well as a description of the facility used.

FACILITY DESCRIPTION

The facility used for the condensing tests is shown in figure 2. It consists of three heat-transfer systems: an all-liquid heating loop, a two-phase loop, and a coolant system, which is either an oil-cooled vacuum chamber

whenever heat transfer by radiation is under study or an all-liquid loop whenever the condenser is convectively cooled.

Liquid NaK 78 in the heater loop is pumped at rates up to 35,000 lb./hr. through an I²R electric heater, with a capacity of 200 KW and using the flowing liquid metal itself as the resistance load. The heated liquid then flows through the shell side of the shell-and-tube boiler and returns to the pump for recirculation. The pump is of the electromagnetic type, as are the pumps in the other two liquid metal loops, but it has the added characteristic that its controls are set to allow reversal of the flow direction, permitting operation of the boiler as a countercurrent or cocurrent flow unit.

Liquid potassium, the test fluid, is pumped through the tube side of the boiler, which consists of 43 parallel 0.375-in.-o.d. by 0.035-in.-wall tubes 14 ft. long, mounted inside a 4-in.-i.d. shell. Each tube has a 0.040-in. orifice at the liquid potassium inlet. The boiler is mounted in a horizontal plane, as is the rest of the two-phase loop. The boiler outlet pipe is wrapped with radiant heaters having a capacity of 5 KW to serve as a superheater. After leaving the superheating section the potassium vapor is directed through either a radiative or convective condenser by means of manually operated shutoff valves. In either arrangement the condensate is returned to the pump, which is capable of handling a maximum flow of 600 lb./hr. of potassium during two-phase operation. Flow control is accomplished by a combination of throttling and pump speed regulation.

When a convective condenser is being studied, subcooled NaK is pumped through the shell side of the condenser at rates up to 35,000 lb./hr. The coolant is directed through shutoff valves to either an all-liquid radiator located inside the vacuum chamber, as would be the case in a flight system, or to an air cooler whose capacity can be controlled by varying the air flow

rate. In both arrangements, the cooled NaK is returned to the pump for recirculation.

The vacuum chamber used as a heat sink on radiative studies is an 8-ft.-diam. by 18-ft.-long cylindrical tank equipped with mechanical vacuum pumps capable of maintaining a pressure of 10^{-3} torr. All surfaces of the tank exposed to the vacuum environment are made of stainless steel. The shell of the tank is oil cooled, and the temperature of the cooling oil is controlled by a system that includes both water-cooled and steam-heated exchangers, as well as means of varying the oil flow rate. The oil system is capable of handling the full-rated heat load of the facility. Every loop is instrumented to measure flow, temperatures, and pressures to supplement research instrumentation on the test condensers.

Operating temperatures in the systems are limited by the materials used. All the heating loop and the portion of the test loop from the boiler inlet to the superheater outlet are manufactured from a cobalt-base alloy, which allows operation up to 1850° F. The rest of the test loop and the cooling loop are fabricated from AISI 316 stainless steel, which limits operation on this section of the facility to 1600° F.

CONVECTIVE CONDENSER TESTS

Test Section

The shell-and-tube NaK cooled convective condenser is shown in figure 3. This condenser consisted of seven 1/2-in.-o.d. by 0.030-in.-wall tubes 6 ft. long, built of AISI 316 stainless steel. The tubes were set on an 11/16 in. triangular pitch inside a 2-in.-i.d. shell. Potassium vapor flowed inside the tubes and was completely condensed before leaving the unit by the NaK stream flowing on the shell side in the opposite direction. The tube entrances were tapered and rounded as much as the proximity of the tubes allowed. The shell side was

unbaffled to maintain longitudinal flow in the coolant stream. The condenser was positioned with the tubes in a horizontal attitude. The unit was well insulated and heat losses in the order of 1000 Btu./hr. were measured for the operating conditions of the tests.

Test Program

The test program consisted of changing each of four independent variables over as wide a range as possible while holding the others constant and taking steady-state data at each operating condition. These independent variables were the coolant inlet temperature, the vapor inlet temperature, and the flow rates of both the NaK coolant and potassium streams.

The range of variables covered during the tests were as follows:

- (1) Coolant inlet temperatures from 1160° to 1400° F
- (2) Potassium saturation temperatures from 1250° to 1500° F
- (3) Coolant flow rates from 6000 to 25,000 lb./hr.
- (4) Potassium flow rates from 200 to 500 lb./hr.

Instrumentation

Potassium vapor pressure at the condenser inlet was measured by a Taylor gage as well as by an unbonded strain gage pressure transducer. Header-to-header pressure drop in the potassium stream was measured directly by an unbonded strain gage differential pressure transducer. Both strain gage transducers were maintained at 180° F in an oven to prevent freezing of the potassium. The transducers were mounted at a lower level than the condenser, and liquid potassium was trapped in the lines connecting the transducers to the loop. Provisions were made on the piping to allow in-service calibration of the transducers against a pressure standard during operation.

Since heat losses were very small as indicated previously, inlet and outlet temperatures of test fluid and coolant were measured by Chromel-Alumel

thermocouples welded to the outer skin of the pipes. In addition, a well couple was located in the vapor inlet pipe, and a skin differential thermopile was used to read the coolant temperature change directly.

Besides inlet and outlet instrumentation, the condenser shell was equipped with skin couples located about 2 in. apart along its entire length to obtain a temperature profile of the coolant. This profile was used to determine condensing lengths for the different test conditions.

Temperatures were read on a multipoint self-balancing potentiometer capable of recording one point per second. The thermopile output was read on a manually balanced precision potentiometer. Output of strain gage transducers were recorded on a multichannel oscillograph.

Test Results

Satisfactory operation of this multitube horizontal convective condenser was obtained over a wide range of operating conditions. The results discussed in this paper will be limited to the effect of coolant inlet temperature on condensing 1400° F potassium entering the condenser at a quality of about 88 percent, with a coolant flow rate of 15,000 lb./hr.

Changing the temperature difference across the condensing surface when dissipating a constant heat load changes the amount of surface, or length, required for condensation. This fact is shown in figure 4. The coolant shell skin temperature is plotted against distance from the vapor inlet for four different values of coolant inlet temperatures. From these plots the point in the condenser where condensation ends can be located (small arrows). It should be pointed out that the curve for 1194° F coolant inlet temperature represents a mean condensing heat flux of approximately $450,000 \text{ BTU}/(\text{hr.})-(\text{ft.}^2)$, while the curve for 1315° F has a heat flux of about $47,000 \text{ BTU}/(\text{hr.})-(\text{ft.}^2)$, a decade smaller. This results from the fact that as the coolant inlet temperature becomes higher, the coolant outlet temperature approaches the

vapor inlet temperature, and a longer length of very low heat flux occurs.

The vapor inlet temperature of 1400° F shown in the figure represents the temperature at the inlet header before the vapor accelerates into the tube entrances. It is to be expected that after the acceleration the tube vapor temperature will be lower than the header vapor temperature.

The condensing lengths obtained from profiles like those in figure 4 were crossplotted against their corresponding coolant inlet temperatures to obtain the performance map shown in figure 5. As the coolant inlet temperature increases at constant potassium flow rate, the vapor length becomes more sensitive to the coolant temperature as indicated by the increasing slope of the curves in figure 5. For instance, at the lower potassium flow rate curve of figure 5 and a coolant inlet temperature of 1220° F, an increase of 10° F in the inlet temperature changes the condensing length about 2 in. However, the same temperature change at a coolant inlet temperature of 1310° F changes the condensing length about 25 in. Such a change as the last one could be detrimental to system performance by an incomplete condensation of the vapor within the condenser or by introducing large oscillations in the system.

In a countercurrent condenser design with a liquid metal coolant, the condensate temperature generally is equal essentially to the coolant inlet temperature. Thus, the coolant inlet temperature is determined by the desired degree of condensate subcooling. The preceding results point to a condensing length sensitivity consideration when designing for small amounts of subcooling.

The condenser pressure drop data are presented in figure 6. The overall pressure drop from header to header is shown, including entrance losses to the tubes. Pressure drop increased with condensing length for a given inlet vapor flow rate. It is significant that satisfactory operation of the

multitube condenser was obtained even when the pressure drop was essentially zero

Preliminary analysis of the data indicated that pressure drop from header to tube inlet was higher than the 10 percent of the velocity head loss anticipated for rounded entrances from gas flow considerations.

Discussion

Proper evaluation of the entrance effects can be very significant in the design of convective condensers because of the change in the vapor saturation temperature corresponding to the pressure drop. This significance is greater with liquid metal condensers for space applications than for earthbound condensers using conventional fluids because of the larger saturation temperature to vapor pressure gradients characteristic of the liquid metals and the close approach of the coolant temperature to the vapor temperature dictated by the weight optimization inherent in the design of any space power system.

The radiator is one of the heavier components of a Rankine cycle space power system, and its size depends almost exclusively on its operating temperature to the fourth power. Therefore, increasing the coolant outlet temperature (radiator inlet temperature) for a given coolant inlet temperature as dictated by subcooling requirements would reduce the size of the radiator and decrease the total weight of the heat rejection system, even though the convective condenser size may increase. When coolant flow is countercurrent to the vapor flow, the saturation temperature of the vapor at the tube inlet throat represents the upper limit for the coolant outlet temperature. If, however, the coolant flow is cocurrent, it can be possible for the coolant outlet temperature to be higher than the value of the vapor saturation temperature at the vapor inlet throat. This can only be possible if the vapor saturation temperature increases along the condensing length. This saturation temperature rise can be obtained if the heat flux rate is

sufficiently high to allow the momentum pressure recovery to be greater than the frictional pressure drop, thus creating a static pressure rise as condensation progresses.

Of course, optimization of the complete heat rejection system, not the condenser and radiator alone, will determine the final operating characteristics of any one application.

RADIATIVE CONDENSER TESTS

Test Section

The other multitube condenser tested was the radiative unit shown partially fabricated in figure 7. This unit consisted of nine centrally finned tubes 14 ft. long manifolded into common inlet and outlet headers, with all tubes set in the same plane. Material of construction was AISI 316 stainless steel. The tubes had a 0.500-in. i.d. and a 0.420-in. thick wall. The centrally located fins were 0.90 in. long and 0.080 in. thick. The tubes were welded tangentially to the bottom of the inlet header on a 3.25 in. pitch, with the tube entrances being rounded and smoothed. A gap of 1/8 in. was left between the edges of adjacent fins.

This tube crosssection was selected by a weight optimization design procedure (ref. 1) developed by Lewis Research Center engineers for two-phase radiators. This procedure took into account the power system size, as well as mission requirements and environmental hazards. This tube configuration was selected for a 300 KW (electric) Rankine cycle power system with a 90 percent probability of successfully completing a 500-day mission on the basis of meteoroid population information available in 1961. The nine-tube panel used as a test unit constitutes one tenth of the 90-tube radiator required for such an application. Knowledge about meteoroid hazards has recently increased greatly, and the same design program today would dictate much thinner tubes.

Surface Preparation

After fabrication, the outside surface of the radiative condenser was sandblasted and oxidized in order to provide a stable high emissivity surface. Oxidation was selected over other coatings because of its relative simplicity and durability. Oxidation was accomplished by inserting the whole panel in an open-air furnace at 1200° F for a period of 24 hrs. A resultant total hemispherical emissivity of 0.72 to 0.75 was measured on test coupons that had been attached to the radiator panel throughout the surface preparation procedure.

Test Unit Mounting

The radiative condenser was mounted inside the vacuum tank shown in figure 8. The inside surfaces of the tank were sandblasted and coated with black paint which gave the wall an emissivity of 0.91 to 0.93. The pressure in the vacuum tank was maintained near 3×10^{-3} torr. In this rarified atmosphere the amount of heat transferred by convection from the radiator to the walls was less than 1/2 percent of the total heat load.

Instrumentation

Figure 9 shows the radiative condenser mounted inside the vacuum tank in a horizontal attitude and completely instrumented prior to operation. This unit lent itself more readily than the convective condenser to the application of extensive instrumentation. Each of the nine tubes had thermocouples spot-welded to its outside surface every 8 in. along its entire length. These thermocouples provided temperature profiles for every tube. In addition to these longitudinal profiles, thermocouples were applied to obtain cross-sectional profiles of tube and fin temperatures at three locations along the radiator.

Immersion couples were applied to the inlet header, and wall couples at inlet and outlet pipes outside the vacuum chamber were included as well.

Cooling-oil temperature rise across the vacuum chamber was measured as well as vacuum tank surface temperatures, both on the vacuum side and on the room side of the chamber. The former were used for monitoring the inside temperature of the tank, the latter to calibrate and meter the heat losses from the vacuum tank to the surrounding atmosphere. A total of over 250 thermocouples were employed to instrument the radiator and vacuum chamber and were recorded at the rate of 40/sec. with the aid of an automatic digital data recorder system.

Pressure instrumentation was applied to the inlet and outlet headers of the radiator as well as to the tube entrance of three different tubes. This pressure instrumentation was of the unbonded strain gage type and the application was similar to the one used on the convective condenser tests.

Test Program

The only independent variables affecting the performance of the radiative condenser were the potassium flow rate and temperature level. The vacuum tank walls were maintained at temperatures below 300° F, and any temperature variations at this temperature level had negligible effect on the radiator performance. The test program consisted of taking steady-state data at potassium flow rates ranging from 200 to 600 lb./hr. at saturation temperatures of 1235°, 1350°, and 1420° F.

In addition to the temperature, pressure, and flow data obtained for each run, visual and photographic observation of the radiative condenser during operation was carried out by utilizing two viewing windows located in the end of the vacuum tank. Over the range of vapor temperatures studied, enough light was emitted by the radiator as the only source of illumination to allow black-and-white as well as color photographs to be taken.

Results

The radiator was operated successfully as a condenser for a continuous

span of 250 hr., during which time more than 80 steady-state data conditions were obtained. Figure 10 shows a typical temperature profile obtained for one of the tubes. It can be seen that the outside temperature of the tube remained fairly constant along the condensing length. Once condensation was completed, the wall temperature fell steeply, indicating the subcooled temperatures of the condensate. The intersection of the two dissimilar temperature patterns clearly indicated the position of the interface in the tube. Similar profiles were obtained for all tubes, and the addition of all the condensing lengths for each run gave the measure of the radiator area used for condensation.

The high sensitivity of condensate outlet temperature to length available for subcooling should be noted. In contrast to the convective condenser, where the coolant inlet temperature fixes the amount of subcooling, the radiator is dissipating its energy to a very low temperature sink and any miscalculating on the length available for subcooling will alter the condensate outlet temperature drastically.

Mean condensing heat fluxes varied from 23,000 to 49,000 BTU/(hr.)-(ft.²) depending on vapor temperature. A photograph of the radiator taken during the same run as figure 10 is shown in figure 11. The photograph was taken as a time exposure by using the emitted light from the radiator. The subcooling length of the radiator dropped in temperature so abruptly that it appears dark in the photograph. It can be noticed that the condensing length in all of the tubes was not the same. For this particular run there was a difference of approximately 12 in. between the shortest and longest condensing lengths. This difference varied from 0 to 24 in. over the range of conditions tested.

The results from the different runs are presented graphically in figure 12 as average condensing tube length against average inlet vapor flow rate

per tube. The vapor phase flow rate was obtained from the total flow rate and the inlet quality as calculated from heat balance considerations. Results are grouped according to inlet vapor temperatures. The solid lines shown in the figure are performance lines obtained from heat fluxes calculated from an analytical study (ref. 2) of the cross section of the same fin-tube configuration. This analysis assumed the inside surface of the tube to be at a constant temperature equal to the inlet header vapor temperature; that is, it ignores any pressure drops or film coefficients on the vapor side. It can be seen that there is good agreement between experimental and simplified analytical results at low values of flow rate and high temperature levels. However, the deviation between actual and ideal values increased at higher flow rates, especially at low inlet vapor temperatures.

A reason for this increase in actual over ideal condensing length with increase in flow and decrease in vapor temperature is the effect of pressure drop. As the pressure drop increases the vapor temperature in the tubes becomes lower than the header temperature if it is to remain in equilibrium. Therefore, the header vapor temperature no longer represents the vapor temperature within the tubes. This effect becomes more pronounced at the lower values of vapor temperature because of the increasing vapor temperature-to-vapor pressure gradient of potassium.

The pressure drop characteristics of the radiative condenser appear in figure 13 as header-to-header pressure drop against average inlet vapor flow rate per tube. Comparing figure 12 and 13 shows that the higher deviations of condensing length from the ideal values correspond to the higher pressure drops.

Discussion

Examination of the results from the radiative condenser tests indicated that pressure drop characteristics were again the most significant factor in the operation of the condenser from the designer's point of view. In the case of the radiative condenser, errors in overestimating as well as underestimating pressure drops could cause significant design deficiencies. If actual pressure drop is less than predicted the radiator would be too large because of the higher surface temperatures encountered during condensation. The excess surface will then provide condensate subcooling below design temperature which may degrade the boiler performance. If actual pressure drop is greater than the predicted pressure drop, the radiative condenser would be too small to handle the design heat load. This condition is similar to that experienced on the convective condensers, but it is aggravated by the fact that the radiative unit cannot be designed with excess surface because of the effects of excessive subcooling.

CONCLUDING REMARKS

The results of the work performed during this program can be summarized as follows:

1. Successful operation of multitube potassium horizontal condensers, both convectively and radiantly cooled, was obtained over a vapor temperature range of 1250° to 1500° F. No appreciable instabilities were encountered over the range of operating conditions even when the pressure drop across the unit was essentially zero.
2. Pressure drops during condensing and entrance effects had a marked bearing on liquid metal condenser performance because of the large saturation temperature-to-vapor pressure gradient of the liquid

metal. Close evaluation of the effect of pressure variations on condensing length is of great importance to the designer.

3. The importance of properly predicting the condensing length is much greater in the design of a radiative condenser than in the design of a convective condenser. Improper evaluation of the condensing length on a radiative condenser will affect the subcooling obtained because of the very low value of the sink temperature. The same discrepancy on a convective condenser can be compensated for by adding extra surface which will not affect the condensate subcooling which is fixed by the coolant temperature.

REFERENCES

1. Krebs, R. P.; Haller, H. C.; and Auer, B. M.: Analysis and Design Procedures for a Flat, Direct-Condensing, Central Finned-Tube Radiator. NASA TN D-2474, 1964.
2. Stockman, N.; Bittner, E. C.; and Sprague, E. L.: Comparison of the One Dimensional and Two Dimensional Heat Transfer in Central Finned Tube Radiators. Proposed NASA Technical Note.

(a) Radiative condenser.

(b) Convective condenser.

Figure 1. - Rankine power systems.

Figure 2. - Schematic test facility.

Figure 3. - Seven-tube NaK cooled condenser.

Figure 4. - Coolant shell temperature profiles. Seven-tube potassium condenser; potassium flow, 300 pounds per hour; coolant flow, 15 000 pounds per hour.

Figure 5. - Vapor lengths plotted against coolant inlet temperature. Seven-tube convective condenser.

Figure 6. - Pressure drop plotted against vapor length. Seven-tube convective condenser.

Figure 7. - Nine-tube radiative condenser.

Figure 8. - Facility vacuum tank.

C-74566

Figure 9. - Radiative condenser installation.

Figure 10. - Radiative condenser wall temperature profile. Vapor inlet temperature, 1426° F; potassium flow rate, 569 pounds per hour; vapor quality, 83 percent.

CS-36459

Figure 11. - Photograph of radiative condenser during operation at 1400° F vapor inlet temperature.

Figure 12. - Average condensing length plotted against average vapor flow per tube. Nine-tube radiative condenser.

Figure 13. - Overall pressure drop plotted against average vapor flow rate per tube
Nine-tube radiative condenser.

DESIGN AND DEVELOPMENT TESTS OF DIRECT-CONDENSING
POTASSIUM RADIATORS*

A. P. Fraas

Oak Ridge National Laboratory
Oak Ridge, Tennessee

ABSTRACT

Six tapered tube direct condensing radiators have been built and operated in the course of the boiling potassium reactor system development test program at ORNL, and a number of tests have been run with individual tubes. The bases for the design of these units and the more significant results of the test program are summarized, and some implications relative to future design work are discussed.

Introduction

The boiling potassium reactor system under development at ORNL presents a number of fluid flow stability and control problems that involve close coupling and possible interaction between the boiler and condenser. Because of this, the MPRE (Medium Power Reactor Experiment) development program has been based on the use of a direct condensing radiator with a set of proportions reasonably representative of those that have appeared suitable for a space vehicle.

Six radiators have been built and operated, two at temperatures up to 1100°F - all as elements of systems rather than in rigs built expressly to test radiators. No effort has been made to obtain good data on condensing heat transfer coefficients or surface emissivities; our attention has been directed primarily at problems of radiator design to obtain good flow distribution, flow stability, a sound structure, and good system stability and control. Particular attention has been given to possible interaction between the flows in tubes operating in parallel, and to the flows in multiple banks of tubes.

In arriving at the basic radiator design to be used in the test program, a wide variety of configuration was considered from the standpoints of free-liquid-surface control under 0-g conditions, scavenging, fluid flow distribution, part load stability and

* Research sponsored by the U. S. Atomic Energy Commission under contract with the Union Carbide Corporation.

control, differential thermal expansion, installation in the launch vehicle, the stresses and vibration induced by launch loads, and orbital startup. Analyses disclosed that there were too many boundary conditions to be met to permit a conventional optimization study - it appeared better to begin with a structure that would be well-suited to withstand launch loads and at the same time would satisfy the fluid flow requirements. The proportions were then modified to reduce the weight to something like a minimum; this led to the reference design for a launch package shown in Fig. 1. The efficacy of this approach is indicated by the relatively low specific weight obtained for the resulting launch package - only 7 lb/kw of electrical power output. This specific weight is for the complete radiator including meteoroid armor, ducts, and the structure for launching. The background for this selection was outlined in Ref. 1 with some discussion of the problems presented by radiation scattering from the radiator and its effects on the shield design, meteoroid protection, two-phase flow in tapered tubes, and the effects of load on performance.

Reference Design

The design precepts upon which the ORNL radiators have been based can be presented by referring to the radiator in the reference design for a launch package shown in Fig. 1. This radiator consists of parallel banks of axially finned, tapered tubes arranged on the perimeter of a cylinder with an Alzak aluminum reflector placed behind each of the tubes to reflect heat from their rear surfaces to space as shown in Fig. 2. The reflector can be contoured so that any ray emitted from the rear of the finned tube will be reflected out past the tips of the transverse fins to space. Aluminum with an Alzak finish appears to be well-suited to this application since its efficiency for specular reflection is about 85%. Its equilibrium temperature at design conditions will be about 600°F, which is satisfactory for anodized aluminum even in air. While this tube-and-reflector arrangement represents a major departure from the approaches used by most designers, it appears to be a sound approach and has many advantages. Perhaps the most important of these is that it keeps both the fin weight and the amount of area vulnerable to meteoroids to

moderate values.¹ By surrounding the unit with a thin aluminum windshield, which could be in the form of the shutters indicated in Fig. 2, and foaming plastic between the windshield, the radiator tubes, and the aluminum reflector, a strong, stiff, vibration-resistant structure would be provided to resist the dynamic loads and severe vibration of a launch. When the power plant is warmed up in orbit, the polyurethane foam will be volatilized into the high vacuum surrounding the spacecraft, and the windshield can be removed if it is a continuous sheet around the perimeter, or flaps such as those shown in Fig. 2 can be opened.

The design calculations for the reference design radiator are summarized in Table 1. As shown in Fig. 2, the portion of the tube vulnerable to meteoroids was made 0.10 in. thick to reduce the probability of meteoroid penetration to 1% in 10,000 hr, while the rear portion of the tube was made 0.030 in. thick to give a probability of about 0.1% that an imperfection in the metal wall will lead to a leak. This consideration stems from extensive endurance test experience at ORNL with high temperature heat exchangers.^{2,3} This shows that individual grains often grow to a size of about 0.010 in., and since there may be imperfections in individual grain boundaries, it is best to make the tube wall thickness at least twice that of the largest grains, i.e., at least 0.020 in. thick.⁴ The thickness was increased to 0.030 in. to increase the system integrity because the resulting weight penalty was only 10 lb. For weight estimating purposes this gave an average tube wall thickness of 0.065 in. To reduce the fin weights, the fin thickness was varied so that it would be directly proportional to the square of the distance from the outer tip. Copper was chosen as the fin material because it is ductile, its coefficient of thermal expansion is close to that of stainless steel, and it is easily brazed. Ideally, beryllium fins would give a weight savings of about 60 lb, but beryllium is brittle, its coefficient of thermal expansion is only two-thirds that of stainless steel, and it is difficult to braze.

Subsequent to the preparation of Fig. 1 it became evident that it would be better to increase the number of ducts radiating to each manifold ring to six, and this was assumed in preparing Table 1. The weight of the ducts and manifolds totals 280 lb, or

28% of the total radiator weight at launch. If allowance is made for the fact that these elements are also designed to serve as structural members to support the turbine-generator under launching conditions, (a 10 g vertical acceleration would induce a 10,000 psi stress in these parts) this weight does not appear excessive. The two sets of six radial ducts that extend from the central hub to the two manifold rings are 3.5 in. in diameter and have a wall thickness of 0.040 in. to resist buckling under ground test conditions where atmospheric pressure would tend to cause them to collapse. The circular manifolds also have a base wall thickness of 0.040 in. with an additional 0.060 in. of thickness on the outer portion to provide meteoroid protection. These circular ring manifolds taper from a diameter of 3.5 in. at the joints with the radial ducts to a diameter of 2 in. in the regions midway between the joints.

Part Load Operation

It happens that, as the power to be dissipated by the radiator is reduced, its temperature must of necessity drop. Ideally, the vapor pressure of the potassium in the radiator tubes would vary in such a way that the velocity of the vapor down the tubes would be constant irrespective of the power plant output, and the pressure would fall off in direct proportion to the power. If no control is exercised over the heat dissipation from the radiator to space, the potassium vapor pressure will fall off more rapidly than the heat load so that, as the power is reduced, the vapor velocity in the tubes will increase to sonic velocity. Thus, while the pressure ratio between the tube inlet and outlet is quite small at full power, at part-load, if no shutters are operated to reduce the heat dissipation, this pressure ratio will increase to the point where a substantial temperature drop will occur along the length of the tube as a consequence of a series of oblique shocks in the vapor flow down the tube.

The effects of load on the axial temperature distribution in the tubes of a direct condensing radiator were investigated both analytically and experimentally with the heat sink temperature as a parameter. The radiator temperature varied, of course, with the heat sink temperature, but, for the radiator proportions used, the axial temperature varied less than 30°F irrespective of the heat sink temperature over the load range from

100% power down to about 60% power. Some typical results of the analytical work are shown in Fig. 3 for loads of 50%, 30%, and 10% power. Note the good agreement in Fig. 4 between the analytical estimates and a set of experimental data for the same set of conditions.

It is essential that the pressure at the inlet to a pump be appreciably above the vapor pressure. In ground tests it is possible to design the piping so that there will be sufficient static head to suppress cavitation at the inlet to the condenser scavenging pump no matter how low the pressure at the condenser outlet. Under 0-g conditions, however, cavitation suppression head may be obtained only by subcooling the condensate. Consequently, some means of reducing the heat losses from the condenser must be employed at reduced power so that condensing will occur at a temperature and pressure sufficiently high to make it possible to obtain the required cavitation suppression head by subcooling.

Figure 3 shows that increasing the heat sink temperature is a good way to increase the condenser outlet temperature, hence the shutter arrangement of Fig. 2 affords a promising approach. Some notion of the effectiveness of shutters can be obtained by considering some envelope cases. For example, if shutters which are black on both sides are in a completely closed position, ideally they will cut the heat dissipation rate for a given tube temperature in half. If the shutters are black on the inside but have polished gold surfaces on the outside with an emissivity of 0.02, the heat dissipation at a given temperature would be reduced to only 2% with the shutters in a closed position. The shutters will be particularly effective in reducing heat losses if their surfaces are specularly reflective and they are mounted close to the tubes as in Fig. 2 so that most of the heat rays emitted from the radiator tubes will be reflected back to the tubes after undergoing only one or a few reflections.

In analyzing the heat losses as a function of shutter angle for the arrangement of Fig. 2, it was found that the view factor associated with specularly reflecting surfaces on the inside of the shutter led to a rapid change in heat dissipation for relatively small changes in shutter angle in the vicinity of angles of 25 to 35 deg. This effect is obviously undesirable because it gives a highly non-linear relation between the heat dissipation rate and shutter angle, a poor characteristic from the control standpoint. A single full-span shutter door was then considered, and this was found to give very much

more desirable characteristics as can be seen in Fig. 5. Thus it appears that the single full-span shutter door is definitely superior to the two half-span doors, and hence the full-span door arrangement was chosen for further design purposes. Analyses indicate that the radiator temperature need not be controlled within close limits; the control device simply needs to reduce the effective emissivity and/or view factor sufficiently to keep the condenser heat dissipation rate within a factor of two to five of the ideal value.

The design of the shutter operating mechanism presents a number of problems. Differential thermal expansion between the shutter blades, the reflector, and the tubes must be accommodated with a light weight installation without seriously interfering with alignment of the hinge points. One promising arrangement employs a flat sheet with a stiffening bead along its outer edge and its inner edge pivoted on a wire stretched between brackets on the vapor and condensate manifolds. This would be actuated by bimetallic thermostat elements mounted at either end.

One problem posed by this method of controlling the heat dissipation from the radiator stems from the effects of thermal convection when ground testing a unit in air. The initial test consisted of an electrically heated tube 4 ft long mounted vertically with a reflector and shutter. The stack effect with the shutter closed was so large that the shutter was much less effective than expected. These tests are being repeated with transverse baffles at the ends designed to prevent vertical air flow over the tube.

Design of Radiators for Test Rigs

In laying out the development test program for the MPRE, it became evident that a number of component test systems would be required to investigate major problem areas, and that condensers would be required for these systems. It was found that in many instances an adaptation of the tapered-tube elements of the radiator of Table 1 served to give about as inexpensive a condenser as could be obtained in any other way, and that use of a properly designed and instrumented condenser would yield invaluable information on the performance of this type of direct condensing radiator. Since the most important single question that arose in the course of the design studies was concerned with the control of the free liquid surfaces in the condenser under 0-g conditions, it was decided to build some of these units for the test rigs so that they would operate with

the tube axes nearly horizontal. Since it was important that the liquid drain from the tubes after the test, a series of condensers was built in the form of flat panels as in Fig. 6 and run with the tubes at a small angle from the horizontal to facilitate drainage. It is important to note that analytical work indicated that the drag forces acting to move liquid toward the outlets of these nearly horizontal tubes would be of the order of 100 times the gravitational forces under full power flow conditions. Units were designed so that under high power conditions the tube taper gives a nearly constant vapor velocity down the tube from the inlet to close to the outlet.⁵ Since difficulties were encountered in the procurement of the tapered tubes, smaller tubes were employed to take advantage of some existing tooling. The proportions used are given in Table 2.

Test Experience

It happens that steam at about 100°F is an excellent stand-in for potassium vapor at about 1000°F in that the densities, viscosities, and heats of vaporization of the two fluids are about the same. Thus to reduce costs and to permit visual observation of the flow through glass and plastic parts, the initial system tests were carried out with steam. The condensers built for tests with steam have included a bank of 12 tubes 7 ft long that tapered from a 0.60-in.-ID to 0.20-in.-ID. Using tubes of the same size, a 144-tube bank similar to that of Fig. 6 was also built and tested in a steam system. Both of these units were water-cooled, hence fins were not required and were not used. The pressure drop at full power output was so low (a few millimeters of mercury) in the tests with steam that it was not possible to get an accurate check on the calculated pressure drop because of small uncertainties associated with the pressure tap locations. However, within the limits of experimental error the test data were consistent with the calculated values.

12-Tube Radiator

The first tests with potassium were run with the radiator of Fig. 7 which consisted of a single bank of 12 tubes. Because of a fabrication error, the feed pump discharge rate in the initial tests was not sufficient under some conditions to scavenge the condenser and, as a consequence, the tubes tended to load up with liquid, giving an abrupt

break in the axial temperature distribution at the liquid-vapor interface close to the outlet. When this condition was corrected, the temperature distribution in the condenser was quite uniform at the higher temperatures. At condenser temperatures below approximately 800°F, however, the temperature pattern shown in Fig. 8 was obtained. Note that near the outlet the outer tubes ran at temperatures almost 400°F below the inner tubes in the bank. This condition was undesirable because the consequent differential thermal expansion forced thermal distortion (see Fig. 9). At first the poor temperature distribution was charged to the greater temperature losses from the outer tubes but, if this had been the case, it is hard to see why the effect should have extended so far in from the outer boundaries. On further examination it was decided that the effects stemmed from sonic velocities in the vapor manifold. The vapor pipe feeding the center of the tube bank gave higher pressures in the center tubes and hence higher temperatures. Shock losses in the vapor manifold led to much lower pressures — and hence temperatures — in the tubes farther from the center feed point. This condition was not encountered with steam because the cooling water supply did not permit operation at a low enough temperature to give sonic velocities in the vapor manifold. As a consequence of this experience, all subsequent radiators have been designed so that choking will occur at the inlet to the tubes rather than in the vapor manifold so that the flow distribution to the various tubes will be uniform. Thus, when shock losses do occur, they take place within the tubes and lead to an axial pressure and temperature gradient that is essentially the same for all of the tubes in the radiator.

144-Tube Radiator

The potassium vapor direct condensing radiator of Fig. 6 was designed with the proportions indicated in Table 2 to dissipate 500 kw of heat at 1000°F. This has been operated for over 2000 hr in a potassium system. Figure 10 shows this condenser after installation in the heat sink, which consisted of a double-walled box with a cooling air stream directed between the walls. This was designed so that about 90% of the heat emitted from the radiator would pass by thermal radiation to the inner wall of the box,

while the remaining 10% would be dissipated by thermal convection.

The radiator was scavenged by two jet pumps, one serving each side. Each jet pump was connected to six banks of tubes (12 tubes per bank) by six lines of equal length to equalize the radiant heat losses and thus the amount of subcooling between the condensate manifolds and the jet pumps. The principal features of the installation can be discerned in Figs. 6 and 10. These jet pumps require a cavitation suppression head of about 0.2 in. of potassium at 10% load, hence the radiator would not scavenge properly if the outlet temperature were allowed to drop below about 560°F under 0-g conditions at 10% load.

Perhaps the most significant data obtained in the course of the test program are a series of infrared photographs taken of the condenser with no illumination other than the infrared light emitted from the radiator surfaces. Figures 11 and 12 were taken looking up from below at the essentially horizontal radiator (see Fig. 10). These pictures, made while the feed pump was scavenging the radiator properly, indicate that uniform scavenging of the radiator was occurring, and that the radiator temperature distribution was surprisingly uniform. An indication of the sensitivity of this technique is given by the fact that the temperature differential between the roots and tips of the fins was only 25°F. Thermocouples in the field provided base points for evaluating the temperature from the local density of the image. The radiator surfaces had not been treated to give a high emissivity, hence light from the higher temperature vapor manifold at the center was reflected from local areas on the fins (which had warped and wrinkled somewhat during the brazing operation).

After examining Figs. 11 and 12 with their apparently remarkably uniform temperature distribution, it was decided that it would be well to investigate the effect on the infrared pictures of loading up the radiator with a substantial amount of liquid. This was done by partially closing a valve in the condensate return line from the condenser to the feed pump, and the photos of Figs. 13 and 14 were taken. The system was stabilized with the liquid level in the expansion tank at the boiler reduced by an amount consistent with the quantity of liquid that Figs. 13 and 14 indicate had collected in the radiator tubes.

Note the sharp drop in temperature near what must be the liquid-vapor interface in the radiator tubes. Note, too, the small scallop in the liquid interface positions across the tube banks, with less liquid holdup in the tubes near the center of any given bank of 12 tubes. Refinements in exposure time and technique should permit detection of temperature differences less than 10°F. These pictures indicate that infrared photos will be an extremely useful tool for determining the temperature distribution in the condenser and the location of the liquid-vapor interface between the condenser and the condenser scavenging jet pump.⁶

Several observations can be made from the infrared photographs and the thermocouple and pressure gauge data obtained. The operating pressure in the radiator as estimated from temperature measurements was about 0.15 psia. The pressure drop, as indicated by a potassium manometer,⁶ ran about 0.1 psi at full load - too little to measure accurately. The thermocouple data and the infrared pictures indicate that vapor bubbles were leaving the radiator tubes, passing through the condensate manifold into the outlet line, and perhaps - in some cases - reaching the jet pumps. The unit was designed so that it should scavenge properly with no help from gravitational forces, and the test data indicate that this objective has been achieved when the scavenge pump has adequate capacity and the radiator outlet temperature is kept above some minimum value that depends on the cavitation suppression head requirements of the scavenging pump. The 144 tubes have operated nicely in parallel with no sign of flow instabilities and no maldistribution in the flow sufficient to cause any apparent warping of the tubes in 2000 hr of operation under a wide variety of conditions. The latter indicates that proportioning the vapor manifold so that sonic velocities will be reached in the tube inlets before they will be reached in the manifold is an effective solution to the problem cited earlier in connection with Figs. 8 and 9.

Projected Tests

The next step in our program will be to run a larger system with a full-scale version of the radiator of Fig. 1 and Table 1. The appearance of the test radiator in the course of fabrication is shown in Fig. 15. To reduce fabrication costs the tubes for this test unit have been made with a uniform thickness of 0.050 in., and the fins have been made of stainless steel-clad copper sheet with a uniform thickness of 0.040 in. of which 0.030 in.

lies in the copper core. The surfaces will be treated with a high emissivity coating. Another cost-saving measure was the increase in the wall thickness of the ducts and manifolds to about 1/8 in. to facilitate fabrication in our shops.

Conclusions

The results of a series of tests with both steam and potassium vapor show that direct condensing tapered tube radiators can be designed and built so that they will operate with a stable, uniform vapor flow distribution across large tube banks over a wide range of operating conditions. The radiators tested have scavenged well and run with very little liquid holdup in the tubes even with the tubes essentially horizontal so that gravitational forces have had little influence on the liquid flow. The tests also show that, to obtain proper scavenging at part load, the heat losses at part load must be reduced substantially through the use of shutters or some similar device, and the vapor manifold must be designed so that choking will occur at the tube inlets rather than somewhere in the vapor manifold. Good correlation between the analytical work and the test data has been obtained both in tests with steam and in tests with potassium vapor.

REFERENCES

1. A. P. Fraas, "The MPRE - A Boiling Potassium Reactor System," Third Biennial Aerospace Power Systems Conference, AIAA, September 1-4, 1964.
2. R. E. MacPherson, J. C. Amos, and H. W. Savage, "Development Testing of Liquid Metal and Molten Salt Heat Exchangers," Nucl. Sci. Eng., 8(1), July 1960.
3. Arthur P. Fraas, "Reliability as a Criterion in Nuclear Space Powerplant Design," Air Transport and Space Meeting, New York, April 27-30, 1964.
4. A. P. Fraas, "Design Precepts for High-Temperature Heat Exchangers," Nucl. Sci. Eng., 8(1), July 1960.
5. R. B. Korsmeyer, "Condensing Flow in Finned, Tapered Tubes," USAEC Report ORNL-TM-534, Oak Ridge National Laboratory, May, 1963.
6. H. J. Metz, and M. M. Yarosh, "Experiences and Developments in Instrumentation for Liquid Metal Experiments," Fourth High-Temperature Liquid-Metal Heat Transfer Technology Conference, September 28-29, 1965, Argonne National Laboratory, Argonne, Illinois.

Table 1. Performance, Dimensional, and Weight Data for the Radiator of the Reference Design for a Launch Package

Item	Value
Power to be dissipated, kw	860
Power to be dissipated, Btu/hr	2.93×10^6
Ideal dissipation at 1040°F to 40°F sink, Btu/hr·ft ²	8560
Emissivity of treated surface	0.92
Fin efficiency	0.86
Reflector efficiency	0.87
Area required, ft ²	435
Height of cylinder for 10-ft diameter vehicle, ft	14.0
Tube length, ft	7.0
Vapor temperature, °R	1500
Vapor density, lb/ft ³	0.0035
Latent heat of vaporization, Btu/lb	887
Vapor quality, %	91.7
Potassium flow rate, lb/sec	1.0
Vapor volume flow rate, ft ³ /sec	262
Vapor velocity at tube inlet, ft/sec	400
Vapor flow passage area at tube inlets, ft ²	0.655
Number of tubes	96
Tube inlet flow passage area per tube, ft ²	0.0068
Tube inlet ID, in.	1.1
Liquid density at tube outlet, lb/ft ³	44.0
Tube outlet liquid flow rate, ft ³ /sec	0.0228
Tube outlet ID, in.	0.30
Tube outlet area (per tube), ft ²	0.0049
Tube outlet area (total), ft ²	0.060
Tube outlet velocity, ft/sec (for liquid)	0.38
Fin span, in.	3.2
Fin cross-section shape	$t = b(x/w)^2$
$0.408 w \sqrt{h/kb}$ for 90% fin efficiency	0.46
Equivalent heat-transfer coefficient, Btu/hr·ft ² ·°F	20.0
Mean fin height, in.	1.2
Fin root thickness, in.	0.092
Vulnerable surface area (total), ft ²	70
Weight of copper fins, lb	290
Weight of 0.065-in.-average wall thickness tapered stainless steel tubes, lb (including 70 lb armor)	120
Weight of 0.010-in.-thick aluminum reflector, lb	100
Weight of 3.5 in. diam radial vapor ducts 5.2 ft long, 0.040 in. thick, lb	95
Weight of ring manifolds, lb	75
Weight of armor for ring manifolds, lb	60
Weight of inlet duct and hub, lb	50
Weight of 0.010-in.-thick aluminum shutters with operators, lb	100
Weight of 1.5-lb/ft ³ polyurethane foam, lb	100
Weight of aluminum meteoroid bumper at top end, lb	10
Total launch weight, lb	1000
Net weight after vaporizing polyurethane, lb	900

Table 2. Design Data for the 144-Tube Radiator for the Intermediate Potassium System*

Design power output	500 kw
Operating temperature, °F	1040
Black-body heat flux, Btu/hr·ft ²	8600
Fin material	SS-clad copper
Emissivity	0.75
Fin thickness, in. (total)	0.010 (0.006 in. Cu)
Heat transfer coefficient for thermal convection, Btu/hr·ft ² ·°F	1
Heat flux for thermal convection, Btu/hr·ft ² ·°F	700
Fin height, in.	0.60
Fin efficiency, %	76
Tube OD (mean), in.	0.57
Tube OD at manifold, in.	0.80
Tube OD at exit, in.	0.30
Tube wall thickness, in.	0.050
Fraction of surface in fins	0.636
Average fin heat flux, Btu/hr	4700
Average heat flux, Btu/hr·ft ²	5500
Surface area, ft ²	298
Average surface, ft ² /ft of tube length	0.295
Total length of tubing, ft	1008
Number of tubes	144
Tube length, ft	7
Manifold diameter at base, in.	8
Manifold diameter at tip, in.	2

*The radiator is shown in Figs. 6 and 10.

Fig. 1. Reference Design for a Launch Package Based on the MPRE System.

Fig. 2. Section Through a Finned Tube Fitted With Aluminum Reflector and Shutters for Installation in a Cylindrical Drum-shaped Radiator. Reference Design.

Fig. 4 Comparison of Experimental Data with the Analytical Estimated Axial Temperature Distribution in the Radiator of Fig. 6 at 17% Load for Three Heat Sink Temperatures.

Fig. 5. Effects of Shutter Opening Angle on the Heat Dissipation Rate at a Constant Radiator Tube Temperature for Both the Two Half-Span Shutters of Fig. 2 and a Single Full-Span Shutter.

Fig. 6. Direct Condenser with 144 Tapered Tubes in 12 Banks for Use in a Potassium System of Intermediate Size. The unit is shown in the shop while it was tilted 45° to inspect the installation of the jet pump.

Fig. 7. Direct Condenser with a Single Bank of 12 Tapered Tubes for Use in a Small Potassium System. The fin warpage noticeable in the photograph resulted from too rapid cooling in the fin brazing cycle. The inlet to the vapor manifold is at the center on the rear side.

Fig. 8. Temperature Distribution in the Condenser of Fig. 7 when Operating at a Reduced Load with Potassium Condensing at about 750°F.

Fig. 9. View of the Potassium Condenser of Fig. 7 Showing the Column Buckling Near the Small Ends of the Tubes (in the upper part of the photo) that Occurred as a Consequence of the Non-Uniform Transverse Temperature Distribution Shown in Fig. 8.

Fig. 10. Direct Condenser with 144 Tapered Tubes Installed in a Potassium System of Intermediate Size.

Fig. 11. Photo made with polaroid infrared film looking up from below at the radiator of Fig. 10 running at about 850°F with the scavenging pump cavitating. There was little liquid in the radiator except in the form of films on the tube walls.

Fig. 12. Closeup Infrared Photo of Portions of the Condensate Manifolds for Two Banks of Tubes Made at the Same Time as Fig. 11. The density of the image indicates that vapor bubbles must be moving through the condensate manifold and supplying heat by condensation to keep the temperature in that region within a few degrees of the temperature in the tubes.

Fig. 13. Infrared Photo of the Radiator at Essentially the Same Power and Temperature Conditions as for Fig. 12, Except that the Valve in the Line Leaving the Scavenging Pump Was Partially Closed so that the Scavenging Pump Stopped Cavitating, Liquid Inventory Distribution in the System Shifted and the Outer 8 in. or so of the Radiator Tubes Operated Full of Liquid.

Fig. 14. Closeup Infrared Photo of the Outboard Ends of the Tubes and the Condensate Manifolds Made at the Same Time as Fig. 13 to Show the Steep Temperature Gradient in the Vicinity of the Liquid Surface when the Radiator Tubes Operate Partially Loaded with Liquid.

Fig. 15. A 10 ft diameter Direct Condensing Radiator Similar to That of Table 1 Being Fabricated in the ORNL Welding Shop.

RESULTS OF HYPERVELOCITY IMPACTS
INTO SPACE RADIATOR MATERIALS

by Nestor Clough and James H. Diedrich

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio

ABSTRACT

The results of an experimental hypervelocity impact program designed to evaluate various materials as armor for waste-heat radiator applications are discussed. Total damage, such as dimple, spall, and perforation, as well as simple cratering, was investigated in both flat plate and tubular targets. The experimental program was conducted on the ballistics range facilities of the General Motors Defense Research Laboratories under NASA contract. Projectiles of approximately 0.016 and 0.040 gram were accelerated to velocities of nominally 25,000 feet per second and impacted against various materials and configurations at room and elevated temperatures. The materials investigated include various aluminum alloys, stainless steel, columbium 1-percent zirconium, molybdenum, vanadium, tantalum, Inconel, A-286, L-605, graphite, and various types of beryllium. Significant differences in the ability of the various materials to resist dimple, spall, and perforation were observed in both flat plate and tubular configurations. The unique behavior of materials subjected to high-velocity impacts is clearly demonstrated.

INTRODUCTION

The existence of a hazard to space vehicle components from the impact of meteoroids has been recognized as an important factor in the design of such components. In particular, space radiators contain fluid circuits that will have to be protected against critical damage from meteoroid impact. On a space radiator the thickness of the armor required to protect against critical damage by meteoroid impacts is determined by the vulnerable surface area, the time of exposure to the meteoroid hazard, and the damage characteristics of the armor material (ref. 1). Consequently, the radiator weight for a given application can vary widely depending on the severity of the meteoroid hazard and the material used for the armor. Recent information on the meteoroid hazard is given in references 2 and 3, and an example of the effects of different armor materials on radiator weight is given in reference 4.

The general cratering damage associated with hypervelocity impact into various materials is generally known (e.g., refs. 5 to 7). However, little information exists describing the specific damage likely to be incurred by the vulnerable components of space vehicles under operational conditions. The early published work on hypervelocity impact tests in various metals correlates the resulting crater depths with the target density and a strength property such as the modulus of elasticity or hardness (refs. 6 and 8). Examples of radiator analyses using a correlation based on material modulus of elasticity to predict the meteoroid crater depths in the armor are given in references 4, 9, and 10. In these analyses and in general radiator practice, and additional amount

of armor thickness is added to the crater depth to prevent perforation or critical damage of the armor wall. Because of a lack of detail data, all the armor materials are assumed to have the same material correlating coefficients and damage adjustments.

Recent results given in references 11 and 12 characterize the various types of damage that exist in tubes subjected to high velocity impacts. In addition to complete perforation, the inner surface of a tube could be made to dimple and spall with armor thicknesses significantly greater than the crater depth. The results of references 11 and 12 also indicate that the material coefficient for correlating simple penetration was unique to a particular material, and, therefore, it was assumed that the individual thicknesses to prevent dimple, spall, and perforation might also be peculiar to the material tested.

In order to assess the effects of individual material properties on the various correlating coefficients and critical damage in materials used in radiator weight calculations, the hypervelocity-impact-damage characteristics of 14 armor materials were investigated over a range of anticipated radiator-operating temperatures. The materials tested included several aluminum alloys, stainless steel, columbium 1-percent zirconium, molybdenum, vanadium, tantalum, Inconel, A-286, L-605, graphite, and several types of beryllium. The targets tested were in the forms of thick and thin flat plates, unlined tubes, and tubes with stainless steel liners.

The impact testing was performed on a ballistics range facility at the General Motors Defense Research Laboratories at Santa Barbara, Calif., as part of an overall research program on meteoroid protection concepts and

design data (NASA Contract Nos. NASw-468 and NAS3-2798). All impacts were carried out at a nominal velocity of 25,000 feet per second with either a 3/32 inch or a 1/8 inch diameter spherical pyrex projectile. In some cases 1/8 inch diameter spherical aluminum projectiles were used. A tabulation of the target configurations tested and conditions of impact is given in table I.

PLATE TARGETS

The impact data obtained from flat plate targets have been useful in determining damage information for use in radiator armor calculations (ref. 11). In plate targets, problems concerned with target fabrication are held to a minimum, and a relatively large target area can be used to ensure a reliable impact. Results concerning cratering materials coefficient and inner-surface damage obtained from impacts into flat plates will now be discussed.

Materials Coefficient

The estimation of the depth of penetration in a thick (semi-infinite) target due to spherical impacting particles has been made with various empirical equations. One such equation in which the penetration depth is correlated with the target-material modulus of elasticity and density is given from reference 8 as

$$\frac{P_{\infty}}{d} = \gamma \left(\frac{\rho_p}{\rho_t} \right)^{\phi} \frac{(V)^{2/3}}{\left(\frac{E_t g}{\rho_t} \right)^{1/3}} \quad (1)$$

where

P_{∞} penetration depth, in.

d projectile diameter, in.

γ materials coefficient

ρ_p projectile density, lb/ft³

ρ_t target density, lb/ft³

E_t Young's modulus, lb/ft²

V projectile velocity, ft/sec

g 32.2, ft/sec²

ϕ 1/2 or 2/3

Values of the materials coefficient γ have been reported to vary from around 1.5 to 2.5. In reference 1, an average value of 2.0 for $\phi = 1/2$ was proposed for use if no explicit value corresponding to the target material in question was available. A value of 2.28 was proposed in reference 8 as obtained from lead and copper targets for $\phi = 2/3$.

The various materials listed in table I were tested with the aim of defining the materials coefficient for each material and impact condition. In general, the materials were impacted in the form of thick flat plates at both room and elevated temperatures as given in table II. Equation (1) was solved for γ by using the measured values of P_{∞} obtained from the tests and the values of Young's modulus as given in the table. Table II also includes a description of the targets impacted, impacting projectile mass and velocity, measured penetration depth P_{∞} , and the various materials specific gravities used in the calculations for γ . The individual calculated values of γ presented are for a value of $\phi = 1/2$.

The average values of γ for the various materials and impact temperatures are summarized in table III. The values of γ are seen to vary considerably from a low of 1.13 for ATJ graphite to a high of 2.58 for 356-T51 cast aluminum. The variation of the values of γ calculated for room temperature impacts indicates the uniqueness of the materials coefficient for a particular armor material. Furthermore, the values of γ for each material are not constant with temperature. The value of γ for the refractory materials are seen to generally increase with temperature, whereas the other materials show

a slight decrease or no change with temperature. This variation of γ with temperature needs further amplification, however, since some of the values in table III are based on single impacts. These experimental values of γ for use in equation (1) can now form the basis for further design calculations and comparisons involving specific armor materials.

Inner Surface Damage

The major interest to the designer of a space radiator is the damage caused by an impacting particle to the internal flow passage rather than the depth of penetration of the armor. In reference 11, it was established that damage modes other than complete perforation of the radiator tube exist, which may be critical to the successful operation of the radiator. In particular, the inner surface of a tube could be made to deform and spall with armor thicknesses significantly greater than the crater depth. Therefore, it was necessary to study the effects of pertinent variables on inner-surface damage for a wider range of materials.

Test conditions.- The targets utilized in this series of tests consisted of flat plates of 2024-T6 aluminum, 316 stainless steel, columbium 1-percent zirconium, A-286, L-605, and Inconel 718. The experimental procedure involved impacting flat plate targets at room temperature with progressively less thicknesses until complete perforation of the targets was observed. The targets were then sectioned at the point of maximum crater depth and examined. Generally, each target configuration required 5 to 8 impacts to define the thicknesses at which the various modes of rear surface damage occurred.

The definition of the damage modes of interest, as they will be used herein, is shown in figure 1. Dimple is defined as any measurable permanent displacement of the surface below the crater without dislodgement of material, spall is a breaking away of any part of the surface of the target below the crater, and perforation is used in the conventional sense. In these tests,

the objective was to define the points of incipient rear-surface-damage. Visual observations of the sectioned targets was utilized to qualitatively classify the degree of damage and to establish the incipient conditions.

A tabulation of all the shots fired in conjunction with this phase of the program is given in table IV. The table includes a complete description of the targets impacted (i.e., material, armor thickness, dimensions), the conditions of impact (i.e., projectile mass and velocity), and a qualitative description of the rear surface damage sustained by the targets. Representative sectioned targets after impact are shown in Figure 2 for the columbium-alloy targets tested. The figure clearly depicts the transition from simple penetration to perforation with varying degrees of spall between. It is clear from the photograph that the damage incurred in a high-velocity impact of a thin section can be greatly extended beyond that of simple crater formation.

Damage thickness factors.- From examination of the sectioned targets impacted in this phase of the program, it was possible to closely estimate the target thickness corresponding to the onset of perforation, spall, and dimple. For each material, damage-thickness factors were defined for each damage mode as the value of the target thickness at threshold damage (i.e., either dimple, spall, or perforation) t^* , divided by the semi-infinite penetration depth P_∞ , as obtained from equation (1).

Damage thickness factors t^*/P_∞ were determined for six materials and the results are summarized in table V. The factors are seen to vary considerably for all three modes of damage among the materials tested. The results show that the damage thickness factor at perforation is not a constant value for all the materials, as has been previously assumed. (A value of $t^*/P_\infty = 1.5$ to prevent perforation has been widely used). Furthermore,

although the columbium alloy tested has a damage thickness factor at perforation consistent with the other high strength alloys tested, the values of t^*/P_{∞} for the columbium at spall and dimple thresholds are from 1.5 to approximately twice the corresponding damage factor values for the other materials tested.

It should be noted that the damage factors given in table V define incipient damage conditions and therefore will not be adequate to completely prevent the particular mode of damage. An increase in the design thickness above the values indicated by these factors will generally be necessary to prevent occurrence of the chosen damage.

TUBULAR TARGETS

Prior results reported in reference 11 indicated that the damage thickness factors determined using flat plates may not be identical to those in comparable tubular configurations. Consequently, tests of comparable impacts on lined and unlined tubes were conducted.

Unlined Tubes

In this study using unlined tubular targets of 316 stainless steel and 2024-T6 aluminum, the effects of the magnitude of tube inside diameter on inner surface dimple, spall, and perforation were investigated. The results are presented in table VI, which contains the shot identification, target dimensions and materials, conditions of impact (i.e., projectile mass, velocity, impact angle), a description of the inner surface damage, and the various calculations used later in determining the damage thickness factors. Representative sectioned targets after impact are shown in figure 3 which illustrates the three damage modes.

The threshold damage thickness factors for the tubular targets tested were determined in the same fashion as for the aforementioned flat plate targets. For this case, however, the normal component of the impact velocity was used in equation (1) for the calculation of P_{∞} . This procedure was shown to be valid in reference 11. Factors for incipient dimple, spall, and perforation were plotted against the inverse of the tube inside diameter as shown in figure 4. The variations given in figure 4 were determined visually from the sectioned targets and represent a best estimate of the points of incipient conditions. The damage thickness factors for the flat plates ($1/I.D = 0$) were obtained from the thin plate study discussed previously.

Figure 4 indicates that a slight reduction in required armor thickness at perforation, spall, or dimpling can be obtained with reduced tube inside diameter. The variations indicate that the tube-size effect is very small for 316 stainless steel tubes in all three modes of damage. For the aluminum alloy tested, the general trend of reduced damage thickness factors with reduced tube inside diameter is also apparent. It appears that the largest variation of t^*/P_{∞} with tube inside diameter is for the onset of spall in the aluminum tested. The trend of reduced damage factors with reduced tube inside diameters may be associated with the increased ability of a curved surface to tolerate a given shock loading without fracture. The curved inner surface of a tube under the point of impact has to go through a compressive loading condition as it is deformed before it can be stressed in tension. Therefore, if the free surface tensile strength is important in suppressing damage, reduced inside diameter tubes will have effectively higher free surface tensile strengths and will tend to suppress inner surface damage.

The observed variations of damage factors with tube radius indicate that, for more exact design information, the damage factors should be determined with the configuration and material in question. However, the tests

indicate that in all cases the flat plate values of the damage thickness factors are slightly conservative compared with the tube values and, therefore, can be applicable to general design usage.

Lined Tubes

Preliminary results of impacts into cast 356-T51 aluminum over L-605 liners, reported in reference 11, indicated that the liner could deform drastically without rupture or spall for the particular target-material combinations tested. A series of experiments was performed in order to further explore the characteristics of lined tubes and to quantitatively assess the advantages of lined tubes compared with unlined tubes. These experiments used 356-T51 cast aluminum armor over 316 stainless steel liners of various thicknesses and inside diameters as listed in table VII. As before, the aluminum armor thickness was successively reduced to define the threshold thickness for the three significant damage modes. In these tests, however, the critical damage was the damage that occurred on the inner surface of the liner.

For the liner-armor combinations impacted in this series of tests, it was found that for the 0.015 and 0.028 inch liner thicknesses the transition from liner dimpling to liner perforation was completed with no evidence of liner spall, and, in some cases, complete closure of the tube by the liner was observed without puncture of the liner. However, for the tubes with liner thickness of 0.065 inch spall occurred on the inner surface of the liner. Figure 5 illustrates these effects for the three liner thicknesses with 0.5 inch inside diameter tubes. These results indicate that the liner, although it acts to effectively suppress armor spall, can, in some cases, spall itself under the correct conditions of impact and thickness.

In order to investigate the possible existence of an optimum value of liner thickness to prevent perforation damage without spall, a series of tests was performed on tubes with 0.5 inch inside diameters and liners with thicknesses of 0 (no liner), 0.015, 0.028, and 0.065 inch. As before, the armor thickness was varied to define the required total thickness (armor and liner) to prevent perforation. From these results it was possible to deduce an incipient perforation line as a function of liner thickness by plotting the total effective weight of armor and liner (per inch of length) for the particular observed damage modes against effective liner thickness.

The effective thicknesses (armor and liner) from which the effective weights were determined, represent values normalized to standard impact conditions at 25,000 feet per second. These effective thicknesses were determined from the relation

$$t_{\text{eff}} = t_a \left(\frac{P_{\infty \text{ eff}}}{P_{\infty a}} \right) \quad (2)$$

where the subscripts eff and a refer to effective and actual, respectively. By using equation (1) for P_{∞} ,

$$t_{\text{eff}} = t_a \left(\frac{V_{\text{eff}}}{V_a} \right)^{2/3} \quad (3)$$

where V_{eff} is set equal to 25,000 feet per second.

Figure 6 shows the results of this series of tests in the form of a plot of effective tube weight against effective liner thickness for the conditions of dimple, spall, and perforation. The minimum value of tube specific weight, as obtained from the faired curve, indicates the existence of an optimum liner thickness between 0.025 and 0.050 inch for incipient perforation of the liner without spall for these particular conditions if impact. Previous tests had shown that at failure the tube

would close completely without rupturing the inner liner. This immediately raised the possibility of allowing a wide "degree" of dimple or closure damage not possible in unlined tubes. Thus, if a partial or nearly full closure of the fluid carrying passage without rupturing of the inner liner can be accepted in the design, a sizable weight saving may be possible.

A series of tests was run in which tubes with optimum liners (0.028-in. thick) were impacted with varying thicknesses of armor. The armor thickness was varied so that the complete range of internal damage was observed (i.e., complete closure to no dimple). The results of the tests are shown graphically in figure 7, where the damage thickness factor for the aluminum armor t^*/P_{∞} is plotted against the ratio of dimple height to tube inside diameter. The damage factors are seen to vary from 0.9 for complete closure to 2.5 for no dimple. It appears that for these lined tubes a substantial armor weight saving is possible if a degree of damage is allowed beyond no dimple. For example, an allowable dimple height of only 25 percent of the tube inside diameter results in reducing the thickness of armor required to suppress dimple damage by 50 percent.

It should be remembered however, that these factors and discussions refer only to the particular liner-armor combinations tested and the particular energy level of impacts employed in these tests. The nature of the liner-armor bond may have an important effect in the type of inner-surface damage observed. As yet, the relative importance of a physical interface as compared with an integral bond between the liner and armor (impedance mismatch) is unexplored under the conditions of hypervelocity impact. The cast aluminum over steel liner targets used in these tests had physical interfaces rather than integral bonds. Caution must therefore be used when these results are extended to other liner-armor combinations.

BERYLLIUM IMPACTS

The targets tested in this series of experiments consisted of tubular beryllium armor 1.25-inch outside diameter and a 0.375-inch wall thickness surrounding a 0.50-inch outside diameter liner tube of AISI-316 stainless steel or columbium - 1-percent zirconium. The liner thicknesses were 0.028 and 0.050 inch for the stainless steel and columbium alloy, respectively. The 0.375-inch beryllium armor thickness was selected to prevent perforation of the liner tube but to permit internal dimpling.

The tubular beryllium armor was bonded to the liner tube by three different methods: (1) by placing solid extruded beryllium tubing around the liner tube, (2) by cold-pressing beryllium powder around the liner tube and sintering by the simultaneous application of heat and pressure, or (3) by casting beryllium directly around the liner tube. Specific details of these fabrication processes together with the chemical composition of the beryllium armor are found in references 13 and 14.

Figure 8 illustrates the external impact damage on targets fabricated by the different processing methods. All the targets shown on the figure were impacted at a nominal temperature of 1300° F. The hemispherical crater surrounded by a spalled region is characteristic of impact damage encountered in beryllium targets (ref. 12). As shown on the figure, the beryllium armor remained intact around the liner tube even though extensive cracking damage occurred.

The damage to the inner surface of the liner tubes for all the targets shown on figure 8 was confined to a dimple inside the tube bore. Little difference in the pattern of observed damage shown on figure 8 resulted when beryllium targets were tested at room temperature (ref. 12). In reference 12, one beryllium tube target was also impacted by two equal-energy

impacts in the same plane approximately 120° apart. No large armor fragments were dislodged by the impacts, and only a connecting fracture was observed between the two impact points. The average materials coefficient obtained for the beryllium impact tests in reference 12 was 2.28.

The cracking damage encountered on the beryllium targets prompted the fabrication of tubular beryllium targets reinforced with small randomly oriented stainless steel fibers or with two concentric cylinders of wire mesh. The details pertaining to the fabrication of the reinforced targets are presented in reference 15. Impact tests conducted on the reinforced targets did not reveal any major tendencies toward reduction of the external damage as shown in figure 9. The figure shows that the general features and severity of the external damage were similar to those encountered on targets without reinforcements.

In order to gather more complete information on the effect of the internal reinforcements, the steel-mesh and steel-fiber reinforced beryllium targets and the impact target armored only with sintered powder were cut and polished for examination. These polished cross sections are shown in figure 10. The internal-damage features are approximately the same for all targets shown. The reinforced configurations, however, had less cracking damage adjacent to the liner tube under the impact point. None of the cross sections had any observable bond remaining between the armor and liner after impact, and several localized bond failures are evident in several beryllium steel-reinforcement interfaces. Radial cracks are also visible diametrically opposite the impact point in the rear portions of the sintered powder armor and the fiber-reinforced armor.

These limited observations indicate that some potential benefits can be gained by reinforcing beryllium armor; however, sounder, more integral

bonds are required. Also, the extensive, internal-cracking damage in the beryllium armor can severely reduce the structural capability of the armor, depending on the particular design application.

RADIATOR WEIGHT COMPARISONS

The experimentally obtained values of damage factors for dimple, spall, and perforation for the specific materials investigated can now permit a more definite comparison of the specific weights of radiators with these materials as meteoroid armor. Computations similar to those in reference 4 were made for a flat-plate, direct-condensing radiator with a central fin-tube cross section by using the procedures and computer program described in reference 9. The particular cycle conditions chosen for the comparison were 300-kW-output Rankine cycle system, turbine inlet temperature of 1850° F, potassium working fluid, mission time of 500 days, probability of no meteoroid critical damage equal to 0.95, and a surface emittance of 0.90. The materials included in the comparison were columbium - 1-percent zirconium, Inconel 718, L-605, 316 stainless steel, TZM molybdenum, vanadium, ATJ graphite, and beryllium. The graphite- and beryllium-armored radiators included an inner liner tube of columbium - 1-percent zirconium. The balance of the materials were used without internal liners.

A comparison of the calculated radiator weights for the design condition of no allowable spall, because of potential damage to rotating components, is shown in figure 11. The damage thickness factors used for each material to prevent inner-surface spall were taken as the experimentally determined factors at incipient dimpling. For the lined tubes, the damage factor used corresponded to an allowable liner dimple height of around 25 to 30 percent of the diameter. The results of the calculations shown on figure 11 indicate that beryllium and graphite continue to maintain their potential low-specific-

weight advantage for armor. A graphite-armored radiator is potentially now only 20 percent heavier than a beryllium armored radiator (because of the low value of γ for graphite). These results, however, are based on equal values of liner-damage factors, which have not yet been verified. Also, the radiator-vehicle support structure is not included. The poor showing of the columbium - 1-percent-zirconium alloy, results from its large value of spall damage factor.

A considerable reduction in radiator weight can be obtained with monometallic radiator tubes if the prevention of perforation rather than the prevention of spall can be adopted as the critical design criterion. Such a situation is possible with an all-liquid coolant radiator in conjunction with a heat exchanger vapor condenser and a nonrotating (e.g., electromagnetic) circulating pump. An indication of the calculated weight saving when designing to prevent perforation with stainless steel and columbium - 1-percent zirconium is shown in figure 12.

The results of figures 11 and 12 vividly indicate the individual behavior of each armor material and accent the need for further detailed impact-test data for specific radiator materials, constructions, and applications.

CONCLUSIONS

The following are the major results of the hypervelocity impact tests conducted on space radiator materials:

- (1) Each of the materials tested had a unique cratering behavior as evidenced by the wide range of values obtained for the materials coefficient γ used in the correlating expression for crater depth. The required armor thickness defining the onset of dimple, spall, or perforation damage was also found to vary with the target material for flat-plate targets tested

at room temperature. Tubular targets of 316 stainless steel and 2024-T6 aluminum each displayed a slight tendency toward reduced dimple and spall damage thickness factors as tube inside diameter decreased.

(2) For tubes with thin-walled liners, only dimple and perforation remained as active damage modes. For cast 356-T51 aluminum armor over 316 stainless steel liners, the liner wall thickness for minimum total weight to prevent liner perforation ranged from 0.025 to 0.050 inch. Sizable reductions in armor weight are possible if increased dimple heights are allowed in the internal tube liner. For beryllium tubes, the use of steel fibers or mesh reinforcements imbedded in the beryllium armor tend to reduce cracking damage only slightly.

(3) Calculated radiator specific weights with the experimental-materials constants and damage-thickness factors resulted in wide variations in radiator weight depending on the damage mode and the armor material used. There is a need for further impact data on specific-armor materials and radiator constructions as applied to specific power systems.

REFERENCES

1. Loeffler, I. J., Lieblein, S., and Clough, N.: Meteoroid Protection for Space Radiators. Progress in Astronautics and Aeronautics, Vol. 11 Power Systems for Space Flight, Academic Press, New York, N. Y., p. 551-579, 1963.
2. Loeffler, I. J., Clough, N., and Lieblein, S.: Recent Developments in Space Power Systems Meteoroid Protection. Paper presented at Aerospace Power Systems Conf., AIAA, SAE, ASME, IAPE, Electron Div. Group IEEE, Phila. (Penn)., Sept. 1, 1964.
3. Clough, N., and Lieblein, S.: Significance of Photographic Meteor Data in the Design of Meteoroid Protection for Large Space Vehicles. NASA TN D-2958, 1965.
4. Diedrich, J. H., and Lieblein, S.: Materials Problems Associated with the Design of Radiators for Space Powerplants. Progress in Astronautics and Aeronautics, Vol. 11 - Power Systems for Space Flight, Academic Press, New York, N. Y., p. 627-653, 1963.
5. Bjork, R. L.: Meteoroids vs. Space Vehicles. ARS Journal, Vol. 31, No. 6, June 1961, p. 803-807.
6. Eichelberger, R. J., and Gehring, J. W.: Effects of Meteoroid Impacts in Space Vehicles. ARS Journal, Vol. 32, No. 10, Oct. 1962, p. 1583-1591.
7. Summers, J. L., and Charters, A. C.: High-Speed Impact of Metal Projectiles in Targets of Various Materials, Vol. I of Proc. of the Third Hypervelocity Impact Symposium, Armour Res. Foundation, Feb. 1959, p. 101-113.
8. Summers, J. L. Investigation of High-Speed Impact: Regions of Impact and Impact at Oblique Angles. NASA TN D-94, 1959.

9. Krebs, R. P., Haller, H. C., and Auer, B. M.: Analysis and Design Procedures for a Flat, Direct-Condensing, Central Finned-Tube Radiator. NASA TN D-2474, September, 1964.
10. Saule, A. V., Krebs, R. P., and Auer, B. M.: Design Analysis and General Characteristics of Flat-Plate Central-Fin-Tube Sensible-Heat Space Radiators. NASA TN D-2839, June 1965.
11. Lieblein, S., Clough, N., and McMillan, A. R.: Hypervelocity Impact Damage Characteristics in Armored Space Radiator Tubes. NASA TN D-2472, 1964.
12. Diedrich, J. H., Loeffler, I. J., and McMillan, A. R.: Hypervelocity Impact Damage Characteristics in Beryllium and Graphite Plates and Tubes. NASA TN D-3018, 1965.
13. Diersing, R. J., Hanes, H. D., and Hodge, E. S.: Fabrication of Beryllium-Clad Tubular Hypervelocity Impact Targets by Gas Pressure Bonding. NASA CR-54058, 1963.
14. Patenaude, C. J., and Santschi, W. H.: Casting of Beryllium - Stainless Steel and Beryllium - Columbium Impact Target Composites. NASA CR-54144, 1964.
15. Diersing, R. J., Carmichael, D. C., Hanes, H. D., and Hodge, E. S.: Gas-Pressure Bonding of Stainless Steel-Reinforced Beryllium Hypervelocity Impact Targets. NASA CR-54222, 1964.

TABLE I. - DESCRIPTION OF TARGETS TESTED

Target material	Configuration			Test temperature, °F				
	Tube		Flat plate	Room	400°	700°	1300°	2000°
	Lined	Unlined						
Aluminum (7075-T6)			x	x		x		
(356-T51)	x		x	x		x		
(2024-T6)		x	x	x	x	x		
Cobalt (L-605)			x	x				
Columbium (Cb-1Zr)			x	x		x	x	x
Molybdenum (TZM)			x	x				x
Nickel (Inconel 718)			x	x				
Stainless steel (316)		x	x	x		x	x	
Steel (A286)			x	x				
Tantalum			x	x			x	x
Vanadium			x	x			x	x

TABLE II. - THICK PLATE TEST RESULTS

Target material	Specific gravity	Impact, round number	Projectile ^a				Target			Calculated Materials coefficient r
			Diameter, d, in.	Material	Mass, g	Impact velocity V, ft/sec	Temperature, °F	Crater depth P _{co} , in.	Modulus of elasticity E _t , lb/ft ²	
Aluminum (7075-T6)	2.80	221	1/8	Pyrex	0.0409	25,900	Room	0.311	15.00×10 ⁸	1.93
		223	1/8	Pyrex	.0411	25,700	700	.362	5.76	1.68
		239	1/8	Aluminum	.0414	26,900	Room	.316	15.00	1.92
Aluminum (356-T51)	2.68	355	3/32	Pyrex	0.0178	24,600	700	0.302	9.08×10 ⁸	2.21
		99	1/8	Pyrex	.0418	20,300	Room	.305	15.00	2.26
		127	1/8	Pyrex	.0413	25,800	Room	.466	15.00	2.92
		128	1/8	Pyrex	.0413	24,700	Room	.396	15.00	2.57
		164	1/8	Pyrex	.0413	22,100	700	.403	9.08	2.38
		176	1/8	Pyrex	.0416	24,200	700	.405	9.08	2.33
Aluminum (2024-T6)	2.77	446	1/8	Pyrex	0.0412	24,800	Room	0.321	15.00×10 ⁸	1.97
		449	1/8	Pyrex	.0412	25,000	700	.375	9.08	2.06
		454	1/8	Pyrex	.0414	25,000	400	.366	12.95	2.25
Columbium-1 percent zirconium	8.66	94	1/8	Pyrex	0.0416	23,700	Room	0.156	23.10×10 ⁸	1.46
		95	1/8	Pyrex	.0431	20,400	Room	.136	23.10	1.41
		97	1/8	Pyrex	.0416	24,000	Room	.158	23.10	1.46
		101	1/8	Pyrex	.0422	16,000	Room	.097	23.10	1.18
		102	1/8	Pyrex	.0419	20,200	Room	.131	23.10	1.37
		103	1/8	Pyrex	.0417	24,200	Room	.157	23.10	1.45
		320	3/32	Pyrex	.0176	24,800	700	.140	22.30	1.67
		353	1/8	Pyrex	.0416	25,000	1300	.207	20.90	1.83
		544	1/8	Pyrex	.0409	25,300	2000	.220	15.85	1.76
Stainless steel (316)	8.01	337	1/8	Pyrex	0.0429	24,200	Room	0.152	40.60×10 ⁸	1.68
		403	1/8	Pyrex	.0422	25,300	Room	.160	40.60	1.71
		338	1/8	Pyrex	.0412	24,700	700	.177	37.40	1.87
		339	1/8	Pyrex	.0412	24,700	1300	.199	30.30	1.95
		350	1/8	Aluminum	.0468	25,300	Room	.156	40.60	1.62
Molybdenum (TZM)	10.2	340	1/8	Pyrex	0.0412	25,400	Room	0.120	66.30×10 ⁸	1.57
		547	1/8	Pyrex	.0421	25,000	2000	.175	49.70	2.09
Tantalum	16.6	384	1/8	Pyrex	0.0410	25,700	Room	0.121	39.90×10 ⁸	1.42
		385	1/8	Pyrex	.0411	25,200	1300	.142	35.30	1.65
		543	1/8	Pyrex	.0411	25,000	2000	.152	32.40	1.71
Vanadium	6.11	386	1/8	Pyrex	0.0414	25,200	Room	0.154	27.40×10 ⁸	1.38
		389	1/8	Pyrex	.0411	25,600	1300	.194	25.40	1.68
		548	1/8	Pyrex	.0422	24,350	2000	.231	18.70	1.80
Steel (A-286)	7.92	1257	3/32	Pyrex	0.0183	25,000	Room	0.129	41.60×10 ⁸	1.99
Cobalt (L-605)	9.13	1253	3/32	Pyrex	0.0178	23,500	Room	0.109	46.80×10 ⁸	1.77
Nickel (Inconel 718)	8.21	1256	3/32	Pyrex	0.0178	24,900	Room	0.103	41.70×10 ⁸	1.55

^a Average calculated specific gravity; 2.50 for 3/32-inch-diameter Pyrex sphere, 2.48 for 1/8-inch-diameter Pyrex sphere, and 2.63 for 1/8-inch-diameter aluminum sphere.

TABLE III. - SUMMARY OF AVERAGE MATERIALS COEFFICIENTS γ

Target material (alloy)	Target temperature, °F				
	Room	400°	700°	1300°	2000°
Aluminum (7075-T6)	1.93		1.68		
(356-T51)	2.58		2.31		
(2024-T6)	1.97	2.25	2.06		
Columbium (1 percent Zr)	1.39		1.67	1.83	1.76
Steel (316)	1.67		1.87	1.95	
(A286)	1.99				
Molybdenum (TZM)	1.57				2.09
Tantalum	1.42			1.65	1.71
Vanadium	1.38			1.68	1.80
Beryllium ^a	2.05		2.38	2.30	
Graphite ^a (ATJ)	1.18			1.13	
Nickel (Inconel 718)	1.55				
Cobalt (L605)	1.77				

^aData from reference 10.

TABLE IV. - THIN PLATE DATA

Round	Target material	Target thickness, t, in.	Projectile mass, g	Projectile velocity V, ft/sec	Penetration depth, in.	Semi-infinite penetration depth, P_{∞} , in.	Rear surface damage ^a	Ratio t/P_{∞}
519	Stainless steel (316)	0.179	0.0163	24,900		0.138	P	1.38
486		.200	.0156	25,200	0.144	.129	S	1.55
480		.224	.0159	24,900	.128	.129	S	1.74
483		.240	.0163	25,300	.130	.130	S	1.85
487		.300	.0156	25,200	.128	.129	D	2.32
488	Aluminum (2024-T6)	0.325	0.0160	24,800		0.203	P	1.65
489		.330	.0157	23,900		.197	P	1.68
514		.375	.0163	24,400	0.219	.201	S	1.87
482		.450	.0157	25,100	.206	.203	S	2.22
481		.470	.0156	24,800	.203	.200	D	2.35
963		.475	.0178	24,600	.190	.205	D	2.32
478		.483	.0157	24,000	.196	.197	D	2.45
1129	Columbium-1 percent zirconium	0.276	0.0421	23,500		0.163	P	1.69
1077		.300	.0415	25,000	0.228	.170	S	1.77
1076		.350	.0415	24,800	.167	.169	S	2.07
1130		.400	.0420	22,800	.160	.160	S	2.50
1131		.454	.0418	23,300	.171	.163	S	2.78
1194		.550	.0418	23,400	.164	.163	S	3.35
1195		.650	.0405	24,000	.159	.164	S	3.96
1196		.750	.0412	24,600	.150	.168	D	4.47
1292	Steel (A-286)	0.186	0.0172	25,000		0.126	P	1.48
1291		.203	.0173	24,400	0.129	.124	S	1.64
1261		.218	0.181	24,700	.128	.128	S	1.70
1257		.230	.0183	25,000	.129	.129	S	1.79
1266		.292	.0174	24,500	.115	.125	D	2.33
1251	Cobalt (L605)	0.182	0.0180	24,100		0.112	P	1.62
1250		.204	.0181	24,200	0.117	.112	S	1.82
1252		.232	.0176	24,300	.104	.112	S	2.07
1253		.254	.0178	23,500	.092	.109	D	2.33
1293		.273	.0177	24,900	.114	.114	D	2.39
1295	Nickel (Inconel 718)	0.193	0.0176	24,700	0.134	0.107	S	1.80
1294		.215	.0177	24,800	.110	.110	S	1.96
1254		.230	.0179	24,000	.109	.107	S	2.15
1255		.263	.0183	24,700	.109	.110	S	2.40
1256		.292	.0180	24,900	.103	.103	D	2.84
1296		.315	.0177	24,800	.107	.107	D	2.94

^a Perforation, spall, and dimple are designated by P, S, and D.

TABLE V. - DAMAGE-THICKNESS FACTORS t^*/P_{∞} FOR FLAT PLATES

Target material	Damage mode		
	Dimple	Spall	Perforation
Aluminum (2024-T6)	2.5	2.3	1.7
Steel (316)	2.4	1.9	1.4
(A286)	~2.4	1.9	1.4
Columbium (1 percent zirconium)	4.5	4.0	1.7
Nickel (Inconel 718)	~3.0	2.5	
Cobalt (L605)	~2.5	2.1	1.7

TABLE VI. - UNLINED TUBE TEST RESULTS

Round	Target material	Tube inside diameter, in.	Target thickness, t, in.	Projectile mass, g	Projectile velocity, V, ft/sec	Impact angle, λ , deg	Calculated semi-infinite penetration depth P_{∞} , in.	Ratio t/P_{∞}	Inner surface damage ^a
599	316-S.S.	0.288	0.151	0.0174	24,400	20	0.125	1.21	P
598			.181	.0173	24,100	30	.117	1.55	S
580			.205	.0175	23,700	32	.115	1.78	D
552	316-S.S.	0.500	0.175	0.0173	24,800	27	0.122	1.44	S
531			.208	.0160	25,100	16	.126	1.65	S
530			.216	.0160	24,800	27	.119	1.82	S
479			.223	.0157	24,200	0	.123	1.78	S
517			.231	.0162	24,700	2	.127	1.82	S
532			.240	.0159	24,800	20	.123	1.95	B
491			.240	.0157	25,600	32	.111	2.16	D
553	316-S.S.	1.00	0.175	0.0172	24,550	14	0.128	1.37	P
560			.190	.0170	25,800	0	.134	1.42	S
525			.224	.0159	24,600	7	.127	1.76	S
559			.230	.0170	25,800	10	.133	1.73	S
551			.239	.0175	24,500	8	.129	1.85	B
550			.239	.0182	24,400	32	.119	2.01	D
558			.245	.0170	26,000	12	.133	1.84	S
614	2024-T6Al	0.250	0.300	0.0175	24,100	28	0.190	1.57	S
612			.300	.0175	23,400	20	.195	1.54	S
583			.325	.0175	24,000	30	.186	1.73	S
573			.361	.0172	25,200	9	.209	1.73	S
602			.375	.0170	23,700	42	.165	2.28	D
568			.482	.0172	24,600	27	.192	2.50	D
579	2024-T6Al	0.500	0.325	0.0174	24,400	5	0.205	1.59	P
572			.361	.0172	25,000	33	.186	1.94	S
601			.379	.0170	24,500	17	.199	1.90	B
595			.422	.0173	24,100	28	.192	2.19	D
567			.482	.0172	25,200	8	.209	2.30	D
575	2024-T6Al	1.00	0.325	0.0172	25,100	9	0.209	1.56	P
571			.361	.0172	25,600	20	.204	1.72	S
570			.361	.0173	24,800	32	.172	2.10	S
600			.397	.0173	24,800	42	.187	2.12	D
594			.422	.0171	25,000	20	.200	2.11	D
560			.482	.0178	25,000	0	.209	2.30	D

^aPerforation, spall, dimple, and broken bubble are designated by P, S, D, and B.

TABLE VII. - LINED TUBE TEST RESULTS FOR 356-T51 CAST ALUMINUM OVER 316 STAINLESS STEEL LINERS

Round	Liner inside diameter, in.	Liner thickness, t_L , in.	Armor thickness, t_a , in.	Projectile velocity, V , ft/sec	Impact angle, λ , deg	Effective liner thickness, t_L^* , in.	Effective armor thickness, t_a^* , in.	Effective target weight, lb/in.	Liner damage descrip- tion ^a
720 805	0.320 .320	0.028 .028	0.188 .162	24,000 25,200	15 20	0.0294 .0290	0.198 .168	0.0359 .0405	D (Severe) C
677 800 717 670 668 669 695 802	0.500 ↓ ↓ ↓ ↓ ↓ ↓ ↓	0.015 .028 .028 .028 .065 ↓ ↓ ↓	0.185 .122 .142 .172 .095 .095 .125 .135	23,500 25,100 23,500 23,100 23,500 23,500 24,000 24,900	30 30 38 23 20 18 28 10	0.0172 .0302 .0342 .0312 .0705 .0702 .0723 .0660	0.212 .134 .174 .192 .103 .103 .139 .137	0.0605 .0462 .0606 .0634 .0627 .0623 .0747 .0700	P C D D P (Severe) P (Severe) S P
809 718 807 799 796	1.195 ↓ ↓ ↓ ↓	0.028 ↓ ↓ ↓ ↓	.174 .192 .224 .250 .274	24,500 24,000 23,500 24,900 25,200	15 14 15 7 0	0.0290 .0294 .0298 .0282 .0279	0.180 .202 .239 .251 .273	0.1196 .1321 .1552 .1594 .1719	P (Severe) D D D D
954 969 967 966 970 971	0.500 ↓ ↓ ↓ ↓ ↓	0.015 .015 .028 .028 .065 .065	0.485 .360 .137 .397 .215 .290	24,800 24,000 24,000 24,500 24,000 24,700	17.5 10 40 15 40 35	0.0156 .0156 .0344 .0290 .0796 .0751	0.503 .374 .168 .411 .264 .335	0.1847 .1237 .0599 .1418 .1332 .1374	D D D (Severe) D ↓
964 968	0.320 .320	0.028 ↓	0.237 .312	24,500 24,300	27.5 0	0.0307 .0285	0.260 .317	0.0683 .0856	↓
960 959	1.195 1.195	↓ ↓	0.379 .500	24,500 24,800	7.5 30	0.0286 .0309	0.388 .553	0.2540 .3820	↓

^aDimple closure, perfection, and spall are designated by D, C, P, and S.

Figure 1. - Definition of damage modes.

C-65-1923

Figure 2. - Cratering, dimple, spall and perforation in columbium - 1-percent zirconium plates after impact of 1/8-inch pyrex projectile at 25,000 feet per second.

Figure 3. - Dimple, spall, and perforation in 1/2-inch-inside-diameter 2024-T6 aluminum tubes after impact of 3/32-inch pyrex projectiles at 25,000 feet per second.

Figure 4. - Threshold dimple, spall, and perforation in unlined tubes of 316 stainless steel and 2024-T6 aluminum impacted by 3/32-inch pyrex projectile at 25,000 feet per second.

Figure 5. - Limiting damage modes in lined 316 stainless steel tubes armored with cast aluminum after impact at room temperature by 3/32-inch pyrex projectiles at 25,000 feet per second.

Figure 6. - Total effective tube weight per linear inch against effective liner thickness for 1/2-inch inside diameter tubes. Cast aluminum armor over 316 stainless steel liners. 3/32-inch pyrex projectile, 25,000 feet per second, room temperature.

Figure 7. - Variation of damage thickness factor with dimple height, cast aluminum armor over 0.028-inch stainless steel liners. 3/32-inch pyrex projectiles, 25,000 feet per second, room temperature.

Figure 8. - Results of impact into beryllium - armored tubes fabricated by different processing methods. Impact velocity, 25,000 feet per second (nominal) with 3/32-inch-diameter pyrex sphere; target temperature, 1300° F.

Figure 9. - Results of impact into AISI 316 stainless steel reinforced beryllium armored tubes. Nominal impact velocity, 24,000 feet per second; 3/32-inch-diameter glass sphere; target temperature, 1300° F.

Figure 10. - Polished cross sections of internally reinforced and plain sintered powder beryllium armored AISI 316 stainless steel tubes after impact. Nominal impact velocity, 24,000 feet per second; 3/32-inch-diameter glass sphere target; temperature, 1300° F.

Figure 11. - Radiator specific weights for various armor materials using experimental materials coefficients γ and damage thickness factors a . No spall allowed.

Figure 12. - Effect of designing for prevention of perforation or spall on radiator specific weights for columbium alloy and 316 stainless steel armor.