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THE INTERMEDIATE POTASSIUM SYSTEM - A
RANKINE-CYCLE TEST FACILITY

M. M. Yarosh
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OAK RIDGE NATIONAL LABORATORY

operated by

UNION CARBIDE CORPORATION

for the

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Reactor Division

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OCTOBER 1968

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FOREWORD AND ACKNOWLEDGMENTS

The work described in this report covered the design, construction, and operation of a liquid-metal test facility, designated the intermediate potassium system, which served the Medium-Power Reactor Experiment program at the Oak Ridge National Laboratory. The majority of the program funds were discontinued in June 1966, but reduced operation of the test facility continued until the middle of September 1966. At that time the facility was placed in a standby condition. No postoperation examination of the system has been conducted following the final operating period.

The contributions of many individuals during the design, construction, and operation of this facility were invaluable. In particular we wish to acknowledge the work of C. J. Claffey and R. D. Stulting for much of the mechanical and electrical design; B. Squires for design and maintenance of the instrumentation; B. Fleischer for metallurgical analysis; and J. W. Reece for operations.

The authors also wish to acknowledge the extensive contribution and responsibility of Jack Zasler in the design and operation of the intermediate potassium system. He participated in all phases of the work on this facility. Only his departure from ORNL precluded his participation as a co-author of this report.

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ABSTRACT

A comprehensive description of the design, operation, and analysis of a single-loop boiling-potassium Rankine-cycle system is presented. The system was designed to be a 36% power mockup of the Medium-Power Reactor Experiment, with electrically heated rods used to simulate the fuel elements of the reactor. A potassium-vapor turbine driving a centrifugal pump provided the feed to the boiler, and a valve was used as an adjustable orifice to simulate the turbine of a turbine-generator. The vapor was condensed in a tapered-tube space-type radiator from which the heat was transmitted by thermal radiation to the inner wall of a double-walled compartment. Air was circulated between the walls.

The stability and control characteristics of the system were investigated over a power range from zero to 335 kw. It was found in startup that nucleation rings on the heater rods helped initiate boiling at temperatures as low as 900°F, which for potassium corresponded to a boiler operating pressure of less than 1 psia. System power was varied from 80 to 335 kw with no external control required to maintain normal system operation. The liquid inventory distribution in the system was controlled through a variation in the degree of cavitation occurring in the centrifugal feed pump. Step changes in power of 25% were imposed, and the system responded with no external control required.

The system was operated under boiling conditions for 4400 hr, of which over 3600 hr were with the turbine-driven feed pump in operation. The boiler outlet temperature ranged from 900°F at the initiation of boiling to over 1500°F at maximum power. Examination of the turbine pump parts after 2577 hr of turbine operation revealed no significant deleterious effects from operation. After the initial shakedown work, much of the operation was endurance and system testing with a two-man crew during the days and no crew during nights and weekends except for two technicians who also attend the numerous other test rigs in the building. Thus the system appears to have promise for long periods of unattended operation. This makes it potentially attractive for space power generation or for other applications where access for maintenance and repair is limited.

1. INTRODUCTION

The anticipated requirements for increasing quantities of electric power for space applications resulted in a study by Oak Ridge National Laboratory staff members of the potential of fission reactors as power sources for space applications.¹ The conclusion of the study was that a boiling-potassium reactor with a Rankine-cycle power-conversion system was the most suitable power plant for immediate development. The Medium-Power Reactor Experiment (MPRE) program was a major step in the development program that stemmed from that conclusion.

The MPRE concept proposed was a 1-Mw(t) single-fluid potassium system in which vapor generated in the reactor core passed directly to a turbine to drive an electric generator and then discharged to a condenser. A portion of the vapor bypassed the condenser to supply a turbine-driven centrifugal feed pump, which returned the liquid from the condenser to the reactor. The liquid-vapor mixture leaving the reactor core passed through a separator, where the entrained liquid was removed and recirculated to the reactor inlet through an auxiliary jet pump. Figure 1.1 presents the basic flow diagram for the system.

The novel design represented by the proposed system required that studies be carried out early in the program to determine the feasibility and to examine its characteristics experimentally. A series of test systems patterned after the proposed concept was planned. Analysis indicated that the physical properties of low-temperature water were sufficiently similar to those of high-temperature potassium to permit early studies to be conducted on systems designed to operate with water. These systems served as useful prototypes for subsequent potassium systems.

Later, a series of electrically heated potassium test facilities was designed and constructed. The test systems included a seven-rod 28-kw boiler system, a 91-rod 364-kw boiler system, and a 1000-kw full-scale electrically heated mockup. The intermediate potassium system (IPS), which was the 91-rod 364-kw system, served as an intermediate step between the small seven-rod potassium system facilities and the final full-scale mockup of the MPRE. The IPS design was based in part on information

Fig. 1.1. Single-Loop MPRE System with Jet Pumps to Recirculate Liquid Through the Boiler and the Direct-Condensing Radiator.

developed from the intermediate water system,² which was identical in electrical capacity to the IPS and was operated with water as the working fluid.

The primary objectives of the IPS were

1. to investigate system stability and control problems of the potassium Rankine cycle,
2. to study the operation of a boiler core of significant size at power densities and temperatures comparable with those of the MPRE,
3. to observe the behavior and performance of individual components of the system, such as the jet pumps, turbine pump, radiator, and liquid feed heater,
4. to investigate effects of extended operation of the feed pump under cavitating conditions,
5. to determine whether turbine-bucket erosion would occur under the prescribed design conditions,
6. to study the extent of mass transfer and corrosion in the system,

7. to investigate the effectiveness of the instrumentation, and
8. to gain operating experience and obtain a substantial amount of operating time on a potassium Rankine-cycle system of the order and magnitude of the MPRE.

A basic flow diagram for the IPS is shown in Fig. 1.2, and isometric drawings showing the physical relationship of the components are presented in Figs. 1.3 and 1.4. A vapor-throttling valve was installed to simulate the turbine-generator assembly, and the system was operated initially without a valve installed upstream of the turbine pump. Basic system operation was similar to that for the MPRE system.

The control concept for the IPS required no automatic control devices on the system for the boiler to operate over a wide range of power levels. The system accommodated the change in power level by a variation in the

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Fig. 1.2. Intermediate Potassium System Simplified Flow Diagram.

Fig. 1.3. Isometric Drawing of Intermediate Potassium System Piping and Components.

Fig. 1.4. Enlarged Section of Fig. 1.3 Showing Boiler and Associated Piping of Intermediate Potassium System.

degree of pump cavitation that occurred as a result of minor shifts in system inventory with a change in power level.

The IPS operated a total of 4400 hr at power levels up to 4 kw per rod and boiler temperatures above 1500°F. Over 3600 hr of this operation was with the turbine-driven pump operating. System power was varied over a wide range with no operator or controller action required when the centrifugal pump was under cavitation conditions. An examination of the pump and turbine impellers after 2577 hr revealed no visible evidence of damage from operation of the pump under cavitation-control conditions.

2. SYSTEM CONTROL CONCEPTS

The concept proposed for control of the MPRE, and incorporated in the IPS, depends on matching the characteristics of the system and the various components in such a way that there is no need to operate any throttle valves over a wide range of system power outputs. Since all the vapor passes through critical pressure-drop orifices before condensation, the boiler exit pressure is almost a linear function of boiler power over the operating power range for the system. To accomplish the control objectives, it is essential that the liquid inventory distribution within the system be regulated in order to insure that adequate liquid is available to the boiler under all power conditions. The liquid inventory distribution in the system is controlled by allowing the degree of cavitation occurring in the turbine-driven feed pump to vary and thus accommodate the mismatch that occurs between feed pump output and the system requirements. Since many of the studies covered in this report were concerned with the control of the liquid inventory distribution in the test system, an understanding of the mechanism by which this control is achieved is fundamental to an understanding of system behavior.

Pump Cavitation Control

The concept of controlling inventory by varying the degree of cavitation in a pump has been employed in conventional power plants. A technique called "suction submergence control" allows the liquid level in the "hot well" of a condenser in a power plant system to be regulated by the degree of cavitation in the condensate pumps.³ The degree of cavitation of the pump, and hence its capacity, is a function of the net positive suction head (NPSH) on the pump. The value of the NPSH is in turn a function of the liquid inventory in the hot well, and thus the pump capacity is made a function of the system liquid inventory distribution.

The turbine-driven pump in the IPS has performance and cavitation characteristics similar to those of the pump curves shown in Figs. 2.1 and 2.2.⁴ These pump characteristics obtained with water are essentially identical to those for the pumps used in the IPS.

Fig. 2.1. Results of ORNL Tests of Aeronutronic Mark I Turbine Pump with Impeller Vane-to-Housing Axial Clearance of 0.057 in. and the Turbine Driven by Steam. Data on pump head versus pump flow at various speeds were obtained from operation with water.

Fig. 2.2. Results of Typical Cavitation Tests of Aeronutronic Mark I Turbine Pump. Data were obtained from operation with water.

The cavitation and performance data from Figs. 2.1 and 2.2 can be represented by the typical pump curves of Fig. 2.3. Also shown on this graph are two typical resistance curves for a hypothetical system. In steady-state system operation, the pump head must match the system resistance. This is represented by operation at point Y on the graph for a system resistance represented by line 1. The pump speed is assumed to be 8000 rpm. With a change in power in the boiler of the assumed system, changes in pump flow and head are required to establish a new equilibrium condition, and the new pump head and flow must match the new pressure drop and flow requirements. The total liquid inventory in the system is assumed to be constant.

In the assumed system, consider a reduction in power such that the pump flow required is reduced from that at point Y to that at point X on Fig. 2.3. For equilibrium conditions, the new pump head must match the system resistance at point X. For this to occur without pump cavitation,

Fig. 2.3. System Control by Pump Cavitation.

the pump speed must be adjusted to give the curve $n-n'$ on the graph or, if the pump speed is held constant, an adjustment must be made in the system resistance. In the latter case, equilibrium is established at constant pump speed with the new system resistance (line 2) at point Z on the graph. Either adjustment, however, requires instrumentation to sense the required change and controls and/or valves to effect the required change.

In the system control concept employing pump cavitation, a power change is controlled in the following manner. Again the system is assumed to be operating at Y under equilibrium conditions with a fixed system resistance. A reduction in boiler power results in less vapor leaving the boiler and thus, initially, since the pump head is momentarily unchanged, there is a tendency for the liquid inventory to shift to the boiler and hence deplete the amount of liquid in the condenser (see Fig. 1.2, the simplified flow diagram for the IPS system). In normal system operation, two liquid-vapor interface points exist; one is in the vapor-separator expansion tank and the other is at the condenser discharge. The system is designed so that minor changes in liquid inventory distribution cause a change in the NPSH on the turbine-driven boiler feed pump. Thus the minor shift in inventory to the boiler during the power change in the example reduces the NPSH on the feed pump and causes it to pass into cavitation. This change is represented by the change in NPSH in moving between points Y and X on Fig. 2.3 (NPSH A and B). As the pump performance moves toward point X along the system resistance line, the liquid inventory continues to transfer toward the boiler until the pump flow returning to the boiler matches the lower boiler vapor flow at X. The initial mismatch results in liquid inventory transfer, which reduces the NPSH on the pump. The reduced NPSH and ensuing pump cavitation degrade the head rise across the pump until, at point X, the pump performance and system requirements are in equilibrium. At this point transfer of the liquid inventory ceases. The transfer of more liquid inventory to the boiler tends to degrade the pump head rise below that required, and the transfer of liquid inventory reverses because the pump head is insufficient to supply the flow necessary to match the net efflux rate from the boiler.

Note that the pump speed is not externally controlled and the line n-n' does not indicate the pump speed, since the pump is no longer operating on the noncavitating pump performance curve. The reduction in boiler power, and hence vapor flow rate, tends to slow the turbine-driven feed pump and bring the liquid flow closer to that required for boiler feed. The relationship in terms of pressure rise versus power is shown on Fig. 2.4. Additional power changes cause further shifts in system inventory and further changes in the degree of turbine-pump cavitation.

In the system described, the minor shift in system inventory changes the NPSH on the pump. This, in turn, affects the degree of pump cavitation until the pump head rise matches the new head required.

Fig. 2.4. Pump Performance Versus System Resistance Curves.

The Role of the Condenser Jet Pump in System Control

The liquid inventory distribution may affect the NPSH in several different ways. In the IPS, the operating liquid inventory may be adjusted so that at full-power conditions an adequate liquid level is maintained in the expansion tank above the boiler when a small mass fraction of vapor is leaving the radiator tubes and entering the condensate lines leading to the condenser scavenging jet pump (see Fig. 1.2). A minor shift in system inventory will change the vapor carryover to these lines, and thus the line pressure drop will change and cause a change in the jet pump sideport suction pressure (suppression head). By introducing the condenser jet pump into the circuit ahead of the turbine pump, the sensitivity of the system to minor shifts in inventory is increased. This can be seen from Fig. 2.5, which presents typical jet pump performance curves.⁵

The pressure ratio N is defined as

$$N = \frac{P_d - P_{sp}}{P_n - P_d},$$

Fig. 2.5. Effect of Cavitation-Suppression Head on Jet Pump Performance.

where

P_d = pressure at jet pump discharge,

P_{sp} = pressure at jet pump sideport,

P_n = pressure at jet pump nozzle.

The flow ratio ϕ is defined by

$$\phi = \frac{\text{Sideport flow}}{\text{Nozzle flow}} .$$

The cavitation-suppression head on the sideport of the jet pump represents the pressure head available to suppress cavitation in the jet pump. This suppression head is influenced by the static elevation existing in the sideport lines to the jet pump, the degree of liquid subcooling, and the pressure drop occurring in these lines. A slight decrease in the sideport cavitation-suppression head may cause a large change in the jet pump performance by forcing the jet pump into cavitation. As a result, the head rise across the pump drops sharply, and this in turn reduces the NPSH available at the boiler feed pump suction. For example, a minor shift in inventory away from the condenser decreases the liquid inventory in the jet pump feed lines. This increases the vapor void fraction and hence the line pressure drop and results in a decrease in cavitation-suppression head on the jet pump and a shift in jet pump operating conditions toward cavitation. It is in this way that the jet pump serves to amplify the effect of minor shifts in system inventory. It is important to recognize that this mechanism for controlling the liquid inventory distribution is sensitive to slight liquid-to-vapor ratio changes because of the very large vapor-to-liquid specific volume ratios for potassium and the strong effect of vapor on the two-phase-flow pressure drop.

Under normal gravity operating conditions, the jet pump scavenges liquid from the radiator along with vapor that is carried into the jet pump feeder lines. Any tendency to shift inventory into the condenser tends to reduce the vapor carryover into the feeder lines, and this acts to bring the jet pump out of cavitation, increase the condenser jet pump discharge pressure, and increase the NPSH on the turbine pump. This, in turn, reduces the degree of cavitation of the turbine pump, increases its

pumping capacity, and acts to reverse the inventory shift. The control of the inventory distribution within the system is thus dependent on the performance characteristics of the jet pump-turbine pump combination and not on the operation of sensing and control devices. The variation in pressure rise available from the jet pump alone is insufficient to provide the system conditions required for the range of power desired. Thus cavitation in the turbine-driven centrifugal pump exercises the major control action.

A more comprehensive discussion of system control characteristics is included in the program literature.⁶

3. SYSTEM DESIGN AND CONSTRUCTION

Design Criteria

The intermediate potassium system was designed to simulate the reactor system in materials of construction and operating parameters. The flow diagrams for the system are presented in Figs. A.1 and A.2 of Appendix A. The turbine-generator to be used in the reactor system was simulated with a throttling valve (HV2). An electromagnetic pump was installed for use during cleaning operations and startup. Provisions to sample the potassium in the system were made, and vents were installed to remove any accumulation of noncondensable gas. Although fixed restrictions would be used in the piping of a flight system, valves were used in a number of places to provide a more flexible arrangement for exploring the control characteristics of the system. Table 3.1 lists the major operating parameters for the MPRE and the IPS and shows that design operating pressures and temperatures were identical.

The system was designed to minimize elevation differences in order to reduce the effects of gravity on system operation. The radiator was designed with tubes nearly horizontal and was elevated only slightly above the turbine pump suction to prevent a high gravity suction head on the pump. Vapor lines were sized so that velocities at full design power would be less than one-fourth sonic velocity. Abrupt changes in velocity were avoided. Vapor volumes and line lengths were kept as small as possible to provide tight coupling of the system and minimize the fluid circuit transit time. The liquid lines were sized to minimize system inventory requirements, to permit effective pressure recovery between the radiator and the pump suction, and to provide balanced flow from the two wings of the radiator. In the MPRE system design for space application, dynamic and capillary forces would control the liquid inventory distribution and permit system pressure recovery in the pump suction line. In the IPS design, gravitational forces were still dominant.

Table 3.1. Comparative Design Parameters for IPS and MPRE

	IPS	MPRE
Core power, kw	364	1000
Core geometry		
Diameter, in.	6.041 ^a	9.5 ^b
Heated length, in.	10.0	11.625
Free flow area, in. ²	14.08	18.7
Heater element outside diameter, in.	0.495	0.5
Center-line spacing of heater elements, in.	0.625	0.5625
Total length of heated elements, in.	27.187	19.875
Total number of heated elements	91	241
Fuel element heated surface area, ft ²	9.83	30.55
Average heat flux, Btu/ft ² ·hr	126,000	111,700
Average core power density (heated section only), w/cm ³	70	79.5
Free flow area (heated section only), %	44.6	28.3
Equivalent diameter, in.	0.347	0.198
Fluid flow data (nominal operating conditions are full power and 20% vapor quality at exit)		
Vapor flow at core outlet, lb/sec	0.414	1.111
Vapor volume flow leaving core, ft ³ /sec	7.13	19.13
Liquid flow entering core, lb/sec	2.070	5.555
Recirculation flow, lb/sec	1.656	4.444
Core exit temperature, °F	1540	1540
Core exit vapor quality, %	20	20
Core exit vapor density, lb/ft ³	0.058	0.058
Core exit superficial vapor velocity, ^b ft/sec	72	152
Core inlet liquid velocity, ^c ft/sec	0.52	1.05
Core exit pressure, psia	29	29
Condensing temperature, °F	1040	1040
Condenser pressure, psia	1.5	1.5

^aIPS dimension across hexagonal flats.

^bMPRE dimension across corners.

^cAt heated section.

Component Design

Boiler Assembly

The boiler assembly, shown in cross section in Fig. 3.1, consists of a core heated with 91 cartridge-type heaters 1/2 in. in diameter and

Fig. 3.1. Cross Section Through Boiler, Vapor-Separator, and Jet-Pump Assembly of Intermediate Potassium System.

21 in. long with a heated length of 10 in. A separator region above the core removes entrained liquid from the vapor and is designed to produce a vapor outlet quality above 99%. The entrained liquid is recirculated to the bottom of the core by four parallel jet pumps equally spaced around the boiler. Boiler feed flow from the condenser is supplied by the turbine pump through the nozzles of the boiler jet pumps. The jet pump discharge flow passes down an outer annulus and enters the core at the bottom of the boiler. The liquid passes up through the core, where boiling occurs, and the liquid-vapor mixture enters the separator region.

Core and Heater Element Design. The MPRE core as proposed from the results of physics and heat transfer calculations has 1/2-in.-diam fuel elements spaced on 9/16-in. centers. The IPS boiler has 1/2-in.-diam heater elements spaced on 5/8-in. centers. This difference was made to assist with early fabrication problems of the IPS core by providing thicker ligaments between holes in the header sheet and thus easing the welding problems. Subsequent development work now permits fabrication of the tighter core. A second boiler design with the 9/16-in. spacing was completed but was not fabricated.

The initial boiler core employed one set of tubular spacers that was located in the flow channel formed by the three proximate heater elements (see Fig. 3.1). Special nucleation rings were also installed on the lower part of the heated length of the heater elements to facilitate nucleation in the core.⁷ The rings were split, placed over the rods, and then welded so that weld shrinkage served to attach the rings to the rod. The crevice remaining between the ring and rod provided numerous active nucleation sites. Five rings were installed on each rod, as shown in Fig. 3.2.

Magnesium oxide insulation was used throughout the heater. The 5/8-in.-diam heater elements were reduced to 3/8 in. in diameter at the bottom end to facilitate assembly into the flange plate and to provide a less restricted flow path to the heated portion of the core. Reduction to the smaller diameter was obtained by welding a smaller diameter tube to the terminal end of the heater. The power and thermocouple leads that extended through the smaller tube were insulated with alumina beads. Attachment of the heater to the support flange was accomplished by welding a larger diameter sleeve to the heater extension and the support flange.

Fig. 3.2. Heater Elements for Cores 1, 2, and 3.

Thermocouples were installed inside the heater to measure the internal temperature of the insulation. These thermocouples served as indicators of high fuel element temperature for heater-element protection. The couples were 0.010-in.-diam Chromel and Alumel wires enclosed in a 0.32-in.-OD stainless steel sheath.

The heating element consisted of four spirally wound Nichrome wire coils electrically in parallel and equally spaced in a 10-in. heated length (see Fig. 3.20). Nickel lead wires were attached to the heated coils. Spacing of the rods at the top was accomplished by brazing a 3/8-in.-OD tube to the heater. Spacing at the bottom was provided by the bottom nucleation ring.

A modification to the heater elements made for the second and subsequent cores is shown in Fig. 3.2. The heater sheath was swaged down to the required smaller diameter. Internal insulation between the heater wire coil and the sheath was changed to boron nitride to provide higher thermal conductivity from the heater coil to the sheath and greater electrical resistance at the operating temperatures. The central core that insulated the heater leads was magnesium oxide. A 0.030-in.-diam Chromel vs Alumel thermocouple bent into a hairpin was swaged in the element. Heater spacers were changed to consist of six pins spaced peripherally around the heater element and located at two elevations. These pins were attached by brazing. The new spacer design provided more positive positioning of the heater rod. Three nucleation rings and a single orifice ring located below the heated section were used in this element.

Figure 3.3 is a photograph of the principal parts of the boiler assembly, including heater elements, jet pumps, separator, and the hexagonal shroud. Figure 3.4 shows the completed assembly.

Critical heat flux considerations for the core elements limited the maximum permissible exit vapor quality for the design heat flux within the boiler. This limit was not well defined for boiling potassium, and therefore a conservative exit quality of 20% at maximum design power was selected. This design exit quality determined the recirculation ratio for the boiler and permitted estimates of the core pressure drop and design of the boiler jet pumps.

Fig. 3.3. Principal Parts of Boiler.

Preliminary calculations for the core pressure drop were based on a homogeneous flow model and a correlation proposed by Owens and modified to yield equations written in terms of friction, acceleration, and elevation losses through the core.^{8,9} These are given as Eqs. (1), (2), and (3) in Appendix B. Figure 3.5 is a graph of the results of the calculations for several values of total power and vapor exit qualities. These calculations were based on a bare heater element and did not include effects of spacers and other attachments to the heater rods. These early estimates of core pressure drop were conservative for the bare rod calculation, but neglected the effects of spacers and nucleation attachments. The calculations, based on a core slip ratio of one, yield pressure drops above those which would be obtained for a higher slip ratio. Later studies made with pressure drop data from water test facilities included a factor for slip ratio to derive correlation equations for computing the effect

Fig. 3.4. Boiler Assembly.

of the higher slip ratios.¹⁰ The pressure drop values predicted for potassium from the revised equations could not be checked in the IPS because accurate measurements of exit vapor quality were not possible and the core pressure drop is strongly dependent on the core vapor quality.

Accurate temperature measurements of the potassium liquid entering and leaving the boiler jet pump could be used to determine the core exit vapor quality if the system were operating at equilibrium conditions and the separator removed all the liquid from the exit vapor stream. The temperature measurements would permit a heat balance to be made around the jet pump from which the expression for quality given below can be derived:

$$X_e = \frac{T_2 - T_3}{T_2 - T_1} ,$$

Fig. 3.5. Calculated Total Pressure Drop Through Tube Matrix of Boiler.

where

- X_e = exit vapor quality,
- T_1 = jet pump nozzle inlet temperature,
- T_2 = jet pump sideport temperature,
- T_3 = jet pump discharge temperature.

Unfortunately, because of difficult geometry restrictions with the IPS boiler, sufficiently accurate temperature measurements could not be obtained to yield meaningful values for the quality.

Early studies of boiling potassium demonstrated a difficulty in initiating and maintaining stable boiling in potassium systems.¹¹ To promote nucleation, a device termed a "hot finger" was installed on the IPS boiler. The hot finger consists of an 8-in.-long 3/4-in.-diam rod welded to the boiler barrel and protruding into the potassium pool below the heated core. Electric clamshell heaters are used to heat the rod external to the boiler. The heat is conducted through the rod into the boiler and generates vapor in a drilled cavity at the end of the rod. The vapor generated serves as sites for additional vapor generation in the core. Three hot fingers are spaced around the boiler periphery, as shown in Fig. 3.1.

Separator and Expansion Tank Assembly. The separator design for the IPS was the product of investigations conducted as part of a separator test program.^{12,13} Tests on an air-water system yielded exit qualities of greater than 99.8% when inlet qualities were varied from approximately 10 to 30% at power levels up to 4 kw per rod. The results of the tests are shown in Fig. 3.6.

As stated earlier a direct measurement of quality was not possible in the IPS. The integral design of the jet pump with the boiler housing prevented sufficiently accurate temperature measurements of the jet pump flow streams to permit calculation of the boiler exit vapor quality. The

Fig. 3.6. Exit Vapor Quality for Various Power Levels with a Constant Separator Inlet Throat Opening Width of 1 5/32 in.

subsequent boiler design (IPS Boiler II) would have provided for more accurate temperature measurements, and the quality calculation could then have been made.

During normal operation, the liquid level in the expansion tank remains below a fixed baffle (see Fig. 3.1). The baffle prevents reentrainment of surface liquid in the high-velocity vapor stream. The vapor stream passes above the baffle and discharges from the boiler assembly. The entrained liquid accumulates in the torroidal trap and passes down a vertical tube to the liquid volume below the baffle. This liquid is returned to the boiler core through the four boiler jet pumps.

Radiator and Heat Sink

Views of the vapor condenser (radiator) are shown in Figs. 1.3 and 3.7. The condenser consists of two banks of seventy-two 7-ft-long tapered tubes. Fins are attached to each of the tubes to increase the heat transfer to the heat sink. The large diameter ends of the tubes are welded to a tapered vapor header and the small ends are connected to a manifold in banks of 12 tubes each. These manifolds discharge vapor and condensed liquid through feeder lines to the sideport of the radiator jet pump.

The condenser is completely enclosed by the double-walled air-cooled heat sink shown in Figs. 1.3 and 3.8. A variable-speed centrifugal blower controls the air flow through the annulus, which in turn controls the temperature of the inner shroud. Regulation of the air flow through a control damper allows system operation with a controlled heat sink temperature at any power level. Pyrex glass windows are provided on each side of the heat sink to allow visual observation and inspection of the condenser. Infrared film photographs used to give an indication of the temperature distribution through the radiator were taken through these windows. These photographs are discussed in Chapters 4 and 5.

Design operating temperature for the radiator is 1040°F at full power (364 kw). The calculated variations in radiator temperature and shroud temperature as a function of power level and air flow are shown in Fig. 3.9.

Under normal conditions the radiator is designed to operate with approximately a 1% vapor quality mixture leaving the small end of the tapered

Fig. 3.7. Intermediate Potassium System Radiator.

Fig. 3.8. Heat Sink. Viewing ports are evident on inclined side.

Fig. 3.9. Variations in Radiator and Shroud Temperatures as a Function of Air Flow.

tube and entering the jet pump feeder lines. Because of the large specific volume ratio (10,000 to 1) of vapor to liquid at radiator conditions, the 1% vapor represents a 99% volume fraction of the mixture leaving the tube. Under normal operating conditions, the vapor is fully condensed in the jet pump feed lines.

Calrod heaters are provided in the heat sink volume for heating the radiator prior to system startup.

Jet Pumps

Jet pumps are used in the system to assist in returning condensed liquid from the radiator to the boiler and to return separated liquid

from the boiler expansion tank to the boiler core. One jet pump is installed in each wing of the radiator and four jet pumps are used for returning liquid from the separator to the boiler. Design parameters of these pumps are given in Table 3.2.

Initial design of the jet pumps was based on the data and recommendations of Cunningham,¹⁴ which covered operation of jet pumps under non-cavitating conditions. It was necessary to perform water tests on models of the proposed IPS jet pumps to determine the pump performance under cavitating conditions.

A comprehensive review of jet pump tests for the MPRE program is given in an earlier ORNL report.⁵ The sensitivity of the pressure rise across the jet pump to the net cavitation-suppression head on the sideport of the jet pump is demonstrated by experimental data in the report.

Boiler Jet Pumps. The design requirements for the boiler jet pump were determined by the flow ratio imposed on the pump by the design exit quality from the core and by the estimates of the core pressure drop at the full-power condition. The sideport suppression head for the jet pumps is limited to approximately 6 in., which represents the maximum static head of liquid in the expansion tank above the center line of the jet

Table 3.2. Design Parameters of IPS Jet Pumps

	Boiler Jet Pump	Radiator Jet Pump
Nozzle diameter, in.	0.098	0.060
Throat diameter, in.	0.300	0.212
Nozzle-to-throat area ratio	0.107	0.080
Throat length-to-diameter ratio	4.01	5.90
Diffuser included angle, deg	8	8
Nozzle-to-throat spacing, in.	0.125	0.125
Design flow ratio	4	1
Design head rise, psia	1	(a)

^aThe IPS radiator jet pump could operate cavitating at full design power and thus operate with a zero or negative head rise.

pump. Since the liquid in the expansion tank is at essentially saturation conditions, there is little additional thermal suppression head provided by subcooling. Earlier work shows that both sideport and nozzle subcooling of the liquid aids in the suppression of pump cavitation.⁵

For low-power operation of the boiler, the pressure drop through the boiler core is small. Since a gravity recirculation head of approximately 1 ft of liquid is available, little or no pressure boost is required from the boiler jet pump under low-power operation.

Radiator Jet Pumps. The radiator jet pump is designed to permit scavenging of the liquid from the radiator tubes and to provide a suction pressure boost over a narrow power range for the turbine-driven centrifugal pump. Characteristics of the condenser jet pump would normally be determined by the suction pressure requirements and cavitation characteristics of the turbine pump. These turbine pump characteristics were not defined at the time the jet pumps were being designed. The jet pump design was therefore established on preliminary data from operation of the intermediate water system¹⁵ and from iterative calculations of the IPS system.

As stated in Chapter 2, the condenser jet pump amplifies the sensitivity of the turbine pump to slight shifts in system inventory. Thus, any variation of liquid level in the radiator manifolds and variation in the pressure drop in the feeder lines as a function of the liquid-to-vapor ratio affects the suppression head on the jet pump. Figure 2.5 indicates the extent to which small changes in the cavitation-suppression head affect jet pump performance. The jet pump performance is then reflected as a change in NPSH on the turbine pump.

The feeder lines leaving the radiator liquid headers were welded to a manifold at the sideport of the jet pump as shown in Fig. 1.3.

Turbine Pump

In order to meet the accelerated schedules established for the IPS facility, rather general design requirements for the turbine pump were established. The long lead time required for design and procurement of a turbine-driven potassium pump did not permit detailed analysis, and the design requirements were set for a broad range of system operating

conditions. The vapor-generation rate for the system far exceeded the anticipated turbine requirements and a high turbine pump efficiency was not a requirement for the unit or the system. Table 3.3 gives the early design criteria for the units.

The turbine pump unit, shown in cross section in Fig. 3.10, has a type 316 stainless steel casing, Mo-5% Ti (TZM) rotor and bearing support structure, and tungsten carbide sleeves brazed to the shaft. These tungsten carbide sleeves operate against a tungsten carbide journal. The turbine wheel is of a single-stage partial-admission impulse type. Vapor enters the turbine casing through five nozzles and impinges on the wheel. The pump impeller is of the open radial-vane type, with four vanes, and is on a common shaft with the turbine wheel. A photograph of the disassembled pump is shown in Fig. 3.11.

A number of design changes were made to the turbine pump after it was received from the vendor, the Aeronutronic Division of Philco-Ford Corporation.^{16,17} In the original design, the bearing lubricant was directed through small holes to the center of each journal bearing. The method of lubrication was modified to provide potassium flow between the

Table 3.3. Specifications for IPS
Turbine Pump

Liquid flow rate through pump, gpm	5-8
Pump head, psia	40-50
Available NPSH, psi	4
Maximum vapor flow available to the turbine, lb/min	26.8
Turbine inlet pressure, psia	17
Turbine outlet pressure, psia	2
Steam unit	
Turbine inlet temperature, °F	220
Pump inlet temperature, °F	126
Potassium units	
Turbine inlet temperature, °F	1450
Pump inlet temperature, °F	1040

Fig. 3.10. Aeronutronics Turbine Pump I-K.

two journal bearings to the thrust bearings to insure a more positive supply of potassium to the thrust faces. The dams were added at the ends of the radial grooves in the thrust bearing (see Fig. 4.4) in order to minimize leakage across the thrust face. The turbine wheel was lightened to shift the center of gravity of the rotating assembly between the bearings. Small radial vanes were machined on the back side of the pump impeller to reduce the hydraulic axial thrust loads. A more comprehensive description of the design and modification of the turbine pump unit is contained in the program literature.⁴

Performance curves for the pump are shown in Figs. 2.1, 2.2, and 4.19. The 10- μ stainless steel filter shown in Fig. 3.12 was installed in the bearing-lubricant supply line to insure particle-free potassium in the bearings.

Fig. 3.11. Turbine Pump Parts.

Fig. 3.12. Filter for Turbine Pump Bearing-Lubrication Line After 2800 hr of Operation.

Feed Heater

A regenerative shell-and-tube vapor-to-liquid liquid-feed preheater, shown in Figs. 3.13 and 3.14, was installed in the feed line leading to the boiler jet pump to avoid excessive thermal stresses in the region of the boiler jet pumps and to demonstrate a technique for improving the cycle thermal efficiency. A side stream of vapor from the boiler enters the shell side of the preheater and is expanded through a series of critical pressure drop orifices.¹⁸ Liquid on the tube side is preheated by the vapor at each of the staged pressure levels on the vapor side. The vapor condensed in each section is expanded through the orificed baffle to the next lower pressure stage and, in the last stage, liquid is discharged to the radiator inlet header.

Sampling System

The performance of the IPS is influenced by the purity of the potassium. The level of oxygen contamination is known to contribute to the degree of system corrosion, and an increased oxygen level in the system may be an indication of a containment leak. It is important that leaks be detected as early as possible so that the corrosion effects to the system are minimized. To establish the initial purity within the system, potassium samples are withdrawn and analyzed, and this process is repeated both during and after the experimental operation.

The potassium is analyzed for oxygen by two techniques: the amalgamation method¹⁹ and the "zirconium-gettering" vacuum-fusion method.²⁰ These methods are capable of determining oxygen levels to less than 50 ppm. Analysis of "as received" potassium by both methods gave agreement to within 10 ppm. Analysis after extended operation showed that the vacuum-fusion method gave results two to three times higher than the amalgamation method.

Taps were installed across the turbine pump to permit onstream sampling and at the bottom of the boiler to determine information on product accumulation. The pressure differential across the turbine pump furnished the driving head to force potassium through the onstream sample system, while samples from the boiler were obtained by gravity flow through the

Fig. 3.13. Regenerative Feed Heater.

Fig. 3.14. Component Parts of Regenerative Feed Heater.

sample container to a temporary waste tank. Fill and drain lines in the sump tank were also used for sampling. Valving was provided for draining and purging the sample system after use.

Figure 3.15 shows the sampling-system facilities schematically. All lines are 3/8-in.-OD 0.065-in.-wall tubing.

Valves

The MPRE concept for system operation proposes that no valve manipulation is required to accommodate power changes. Potassium vapor from the boiler (or reactor) portion of the MPRE must flow through orifices or nozzles sized to operate critically. Because of the experimental nature

Fig. 3.15. Potassium-Sampling Systems.

of the IPS, it was prudent to use valves in place of critical orifices for the studies planned. A series of iterative system calculations determined the valve-port sizes required to maintain a critical pressure ratio across the valve under maximum and minimum power conditions. The valves used in the IPS facility are listed and illustrated in Appendix C.

Electrical Design

Power Supply

An electrical-distribution system furnishes power for the boiler, for preheating the system piping to operating temperatures, for instruments and controls, and for auxiliary services. Figure 3.16 shows a schematic diagram of the boiler power supply system. The secondary windings of a 455-kva transformer bank are connected in a grounded "Y." Individual heaters are connected across the 208-v phase leads of the Y connection. A three-phase circuit breaker and a three-phase disconnect switch are provided on the primary circuit for protection and isolation.

Auxiliary transformers connected to the 440-v building electric supply system provide 200 kw of 115/230-v power to seven separate variable-

Fig. 3.16. Schematic Diagram of Boiler Power Supply.

transformer control cabinets. These variable transformers are used for system preheating. Auxiliary transformers connected to the building system also provide power to an auxiliary power distribution center and provide a supply for instruments and controls.

Variable power for the boiler heaters is provided by silicon-controlled rectifiers (SCR). Six separate groups of heaters are provided with zone controls. These groups or zones may be operated at different power levels, and the six zone controls may be transferred to a common or group control to permit operation of all heaters at the same power level. Figure 3.16 also shows the connections for one of the zone controls.

The SCR system used to control the boiler core power level was a result of a development program conducted by Reactor Division personnel. Details are given in Appendix D. The system was devised to control the number of cycles per second in which the heater was energized. This variation in the energized period controls the power level. Full voltage is applied to the heater in the ON period and all switching from ON to OFF is made at zero voltage.

Power level measurements are made by Leeds and Northrup thermoverter-type measuring devices. Current and voltage transformers are installed to reduce the voltage and currents to values acceptable for operation with the thermoverters. The proportional dc signal from the thermoverters is connected to a Brown-type recorder for display and record purposes. Power measurements were also obtained by counting the percent ON time of the SCR's. By using this percentage of the power calculated from the voltage and current readings, accurate power measurements were obtained.

Instrumentation

Control and monitoring of the IPS were accomplished in a U-shaped control area (see Fig. 3.17) located directly adjacent to the test facility. The control panel is composed of nineteen 2- by 2- by 7-ft individual cabinets joined together to enclose the control area. Control devices and monitoring equipment are mounted on the front of the cabinet, and doors are provided at the back of the cabinets for easy access to instruments and associated wiring and terminal blocks.

Fig. 3.17. Control Panel Layout.

One corner of the U was designed primarily as the boiler power control and system monitoring area. Boiler power control, pertinent system pressures, temperatures, and flows are centered on six adjacent panels forming a right angle of the U. Preheat variac controls and temperature monitors, auxiliary controls, and indicators are located on the remaining 12 cabinets. Two eight-channel Sanborn recording instruments are provided on movable mounts to permit the study of system transient responses. Figure 3.18 is a photograph of a portion of the control panel showing the primary control area. Table E.1 in Appendix E lists the instrumentation associated with the principal system components and the main system parameters. The flowsheets, Figs. A-1 and A-2 in Appendix A, provide a more complete picture of the instrumentation.

High-temperature sensors (Taylor Instrument Company, Catalog No. 740N and Foxboro ECI) are provided for all pressure measurements. These instruments are of the diaphragm type. Accuracies of $\pm 1\%$ can be expected.

Fig. 3.18. Control Panels.

The Taylor instruments are rated for temperatures up to 1500°F. They are installed as closely as practical to the pressure taps. These instruments do not produce a signal for pressures lower than approximately 5 psia and are used only for monitoring higher pressures.

The Foxboro ECI pressure monitors are rated for a maximum of 400°F. The monitor is connected to the pressure tap by 3/8-in.-OD 0.065-in.-wall tubing. The instrument lines are controlled to a 600°F maximum temperature and the monitoring unit is air cooled. Pressure ranges as low as 3 in. H₂O were measured successfully, but the reproducibility of measurements was uncertain.

A difficulty common to many of the pressure-measuring devices is associated with the uncertainties involved in the corrections required for liquid accumulation in the cooled instrument lines. In general, those vapor lines where cooling was required were located so that condensed vapor could drain back to the system without interfering with the pressure

measurement. In cases where a liquid trap was unavoidable an exact height of the liquid trap was required, and special drain provisions were made to remove the liquid potassium during shutdown.

High-temperature pressure instruments operated well, in general, but the accuracy of pressure measurements less than 3 psia was often suspect because of uncertainties of liquid levels in the lines leading to the sensing instruments. Results of instrument calibration checks made before and after high-temperature runs are presented in Appendix E.

The potassium-level-measuring devices for the boiler and miscellaneous tank volumes are an ORNL-developed I-tube adaptation.²¹ Level differences are determined by the measurement of a low-level electric signal, which changes as the potassium level shorts a path between the level probe and the grounded tanks. Figure 3.19 shows the construction of the probe and describes schematically the signal circuit.

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Fig. 3.19. I-Tube Potassium-Level-Measuring Device Used in Intermediate Potassium System.

Small differential-pressure measurements are accomplished by especially designed potassium manometers.^{22,23} These manometers were fabricated by welding an I-tube level probe vertically inside 1 1/4-in.-OD tubing. Pressure taps to the system were located at appropriate points in the piping and connected to the top and bottom of the 1 1/4-in. tubing. The difference in liquid level, as a result of the pressure differential, is reflected in the output signal of the I-tube circuit. Measurements of $\pm 1/4$ -in. potassium are indicated and recorded. These special devices were used to measure boiler and radiator pressure drop. Difficulty was encountered with liquid and vapor transfer through the manometer lines giving erroneous and spurious readings. Control of line temperatures to minimize such fluid transfer was essential to correct readings and was not always possible.

A type 740 Taylor Company differential transmitter was also used for measuring the small pressure drop across the boiler. This unit is a special adaptation of two Taylor pneumatic-transmitting pressure-sensor elements that transmit to a common Foxboro pneumatic recorder. A diaphragm in the modifier unit is positioned by the difference in pressures. An electronic signal proportional to the differential pressure is amplified and indicated on a standard Foxboro ECI recorder.

General Electric Company Catalog No. MA27-G-10 electromagnets were adapted by the ORNL Instrument Division to provide flow indications.²⁴ These units were not precalibrated, and thus flow measurements were only correct within approximately 15%.

Carbon-type microphones are attached to the boiler, turbine pump, and jet pumps on the end of long extension rods. The vibrations sensed by these devices are amplified and connected to a speaker system to permit qualitative determinations on items such as boiling frequency and turbine pump cavitation operation. Tape recordings of the boiler operation during startup with and without hot-finger operation made possible an assessment of the performance of the hot fingers in promoting nucleation during startup.

The turbine pump speed is measured by a magnetic-pulse type of counter circuit. Westinghouse Electric Company Hyperco No. 27 magnet rods are

imbedded in the pump vanes, and an Electro-Products magnetic coil designed for 800°F operation is installed in the pump housing. A cooling air line was installed to keep the coil below the design temperature. As the magnet rods pass the magnetic coil, an electrical impulse is generated and transferred through a Transistor Specialities, Inc., Model 360 amplifier to a digital readout counter.

An attempt was made to provide a signal representative of vapor quality by the use of Taylor electronic equipment. The instrument system worked well but, as previously discussed, close coupling of the boiler and jet pumps made it impossible to obtain the proper temperature measurements, and the vapor-quality-measuring scheme was not successful. A design employing thermocouples in wells or using jet pumps not integral with the boiler would permit accurate measurements to be made.

The internal temperature of the boiler heater elements is measured by Chromel vs Alumel thermocouples installed in the magnesium oxide insulation in the heater. Monitoring of the 91 thermocouple signals is provided by an Advanced Technology Laboratories Model 220 ATL profile monitor, which normally provides a calibrated display in bar-graph form of 48 independent signal voltages. The unit was modified to display 96 signals. Each thermocouple voltage is sampled 20 times per second by an ATL mercury-fed centrifugal Delta switch and is displayed as a verticle pulse (or line) on the display tube.

A signal proportional to the insulation resistance of each heater rod is monitored by a ground-current-measuring circuit and is displayed on an ATL monitor. This protective scheme is discussed later in this section.

Pertinent data on system operation is processed with a Beckman DEXTIR data system. Process signals are connected to analog or digital converters and transmitted via a party line to a central processor where the signals are recorded on magnetic tape. These magnetic tapes are then used in digital computers for the printout of raw data and are used as the input for a computer program for data reduction (see Appendix F). Process information is scanned at a rate of eight analog or eight digital channels per second. Seventy-five selected process signals are transmitted to the DEXTIR processing system.

A Fortran program was written for use with a Calcomp plotter to plot temperature profiles and system parameters over a range of powers (see Figs. 4.40, 5.4, and Fig. F.1 in Appendix F).

Heater Protection

Figure 3.20 is an x-ray photograph of typical heater rods used in the boiler. Ninety-one of these heaters are arranged as shown in Fig. 3.1. Construction of these heaters is discussed earlier in this section.

The heaters are operated at surface temperatures up to 1540°F and internal temperatures to 1700°F. To protect the integrity of the heater sheath in the event of insulation breakdown, a protection scheme consisting of two independent circuits, as shown in Fig. 3.21, is employed. One circuit employs a sensitive current relay in the neutral of the transformer bank to detect any current leakage to the heater sheath. The circuit is completed to the transformer neutral through the ground bus. The relay operates to open the boiler power circuit breaker. In operation it has been extremely difficult to determine the proper setting for this relay, since any transformer load unbalance appears as transformer neutral current and because the sum of small leakage currents cannot be distinguished from one large leakage current. A 1300-ohm resistor is placed in the neutral of the transformer in an effort to limit the fault current. Fuses are inserted in each phase lead to protect against faults between leads. These fuses are rated slightly above the operating current and do not always open quickly enough to serve as protection for the heater sheath against failures.

In an attempt to anticipate possible short circuits, an additional protective system was added. Since the center of the heater is the hottest portion of the insulation and, since the insulation conductance rises sharply with temperature, a low-voltage direct-current detection circuit is installed between the thermocouple and the heater leads. Any decrease in resistance (increase in conductance) in the insulation results in a change of current in the 6-v battery circuit. The voltage across a resistor as a result of the current flow is displayed on a multipoint ATL electronic scanner. Trip settings from predetermined values of heater

PHOTO 74022

Fig. 3.20. X-ray Photograph of Typical Heater Elements.

Fig. 3.21. Heater Protection System.

resistance are used to open the boiler power circuit breaker. This circuit also measures the resistance from the heater sheath to the thermocouple and heater leads and thermocouple and sheath or the combination of these resistances. A current-limiting resistor is installed to control the magnitude of the current flow.

These protective schemes have time lags which, under fast developing fault conditions, may be too slow to prevent heater sheath damage. In general, however, for low-resistance conditions they have offered reasonable protection.

Construction

The fabrication and assembly of components and the assembly of the units required approximately 18 months. Design, component fabrication, and facility construction proceeded concurrently.

Piping and component materials were of type 316 or 347 stainless steel. Material selections were made in accordance with the following:

<u>Item</u>	<u>ASTM Specification</u>
Plate	A 240
Bar	A 240
Forgings	A 182
Pipe	A 312
Tubing	A 269
	A 213
Fittings	A 312

All material required a minimum of 0.04% carbon content and certification of chemical analysis and physical properties. Check chemical analyses were made on each shipment of material. Liquid penetrant and ultrasonic examinations were made on all material in accordance with ORNL specifications. A wet-chemistry check analysis was made on each shipment of material.

The construction period extended over approximately 25 weeks. All welds were made by the Heliarc method. Inspection included dye checking of the root and final weld pass and radiographic examination.* Seven percent of the welds made were rejected for excessive oxidation, excessive or lack of penetration, undercut, excessive reinforcement, pinholes or craters, lack of fusion, general misalignment, cracks, or porosity.

Special cleanliness techniques were invoked for all material installed in the main piping system. Each component, sections of pipe or tubing and fitting, was cleaned in a perchloroethylene vapor bath. This vapor bath was followed by wiping and rinsing with an alcohol- or acetone-soaked cloth. Figure 3.22 is a photograph of the main piping after completion

*ORNL Metallurgy Specification MET-NDT-3.

Fig. 3.22. Main Piping System.

of welding and prior to installation of heaters, thermocouples, and thermal insulation.

Approximately 185 calrod heaters and 300 thermocouples were installed on the pipe and components. Connections from each of the heaters and thermocouples to control and readout devices on the instrument panels were routed through conduit and cable trays.

The piping and components were covered with approximately 2 in. of Johns Manville Kaowool Refractory Insulation. The heat sink was insulated with 2-in.-thick magnesium silicate block insulation rated for 1500°F service.

Safety considerations for the high-temperature liquid metals in the event of a pipe or component failure required that the complete liquid-metal system be enclosed in a protective housing fabricated from 1/16-in. mild steel sheets. Removable panels permitted access to the equipment for maintenance. Two air blowers were installed on the roof of the structure to induce air cooling to limit the temperature inside the housing to 150°F. Exterior operating handles were attached to special extension rods which passed through the housing and were then attached to the normal operating valve handles.

Where equipment interferences required offsets to the handle extensions, McMaster Corporation Universal gear joints were used. Some of the applications required two offsets before the handle extension could be taken through the housing.

Prior to assembly into the system, all pressure-containing components were given a pneumatic pressure check equivalent to the design rating. Variations in the design temperatures for the piping prohibited a pressure check of the completed system at lower temperatures that would yield stresses equivalent to those at design conditions. A helium leak check to a 10^{-10} -cm³/sec criteria at a 15-lb pressure difference at room temperature was given to the completed system. The temperature of the completed system was then increased to near operating conditions, and the system was again given a helium leak check.

4. SYSTEM OPERATION

The test facility logged a total of 3618 turbine pump hours and 4403 boiling hours. During all operations three different turbine pump units and three different boiler cores were used. Table 4.1 lists the turbine pump unit and boiler core used for each operating run.

The principal operating history of the facility is described graphically in Figs. 4.1, 4.2, and 4.3. The operating period extended over 21 months, starting in January 1965, but all significant data were obtained

Fig. 4.1. Intermediate Potassium System Operating History - Run 2, July Through September 1965.

Table 4.1. Turbine Pump Unit and Boiler Core Unit for IPS Operating Runs

Operating Run	Turbine Pump Unit		Boiler Core	
	No.	Period of Operation ^a (hr)	No.	Period of Operation ^b (hr)
1	1	18	1	361
2	2	2570	2	2801
3	2	7	1	128
4	3	1023	3	1113
^a Total turbine pump unit 1 18 hr				
Total turbine pump unit 2 2577				
Total turbine pump unit 3 1023				
Total turbine pump operation 3618 hr				
^b Total boiling in core 1 489 hr				
Total boiling in core 2 2801				
Total boiling in core 3 1113				
Total boiling time 4403 hr

Fig. 4.2. Intermediate Potassium System Operating History — Run 2, October Through December 1965.

Fig. 4.3. Intermediate Potassium System Operating History - Run 4.

during two operating periods covering a total of 8 months. The maximum power level was 335 kw or 4.35 kw per rod, and the peak boiler exit temperature was 1508°F. Maximum vapor flow to the condenser section was approximately 20 lb/min. The second turbine pump unit was started a total of 46 times during its operation before it was removed for examination. An instantaneous power reduction of 52 kw and an instantaneous

power increase of 36 kw were accomplished without control action. Power was varied from 80 to 335 kw with the turbine pump unit in cavitation control throughout the range.

Run 1

Startup

Prior to the first operation of the facility, the system was held under a $1\text{-}\mu$ vacuum and at a system temperature ranging from 500 to 1200°F for approximately one week to permit outgassing. A cleaning charge of potassium was then added, and a reflux cleaning cycle of the piping and components was initiated. Vapor generated in the boiler was condensed in the radiator and returned by an electromagnetic pump to the boiler. Initiation of boiling was accomplished by using the hot fingers located on the boiler. As power was applied to the boiler heater elements, the bubble evolution rate increased, and reasonably smooth boiling was reached at approximately 800°F. Microphones attached to the boiler provided a means of detecting boiling within the core. The reflux cleaning operation was continued for approximately 24 hr under the following conditions.

Power input to boiler core, kw	65
Power per rod, kw	~0.75
Boiler liquid temperature, °F	1180
Condenser temperature, °F	990
Boiler feed flow, gpm	0.6

Analysis of the cleaning charge after the cleaning cycle indicated that the oxygen content had increased from 29 to 50 ppm. The cleaning charge was drained and the system was refilled with clean potassium (~30 ppm O_2). The boiler was then operated with the electromagnetic pump providing boiler feed flow until the turbine was started.

The system was designed to permit throttling of the potassium vapor downstream of the turbine. A bypass valve installed in parallel with the turbine permitted diverting a fraction of the total vapor around the turbine. The positions of these two valves determined the boiler outlet pressure and the turbine speed at a given power level. During the cleaning cycle the turbine downstream valve was closed and all the vapor

generated was bypassed around the turbine. The discharge of the electromagnetic pump was connected to the bottom of the boiler.

The turbine pump was started by increasing boiler power in small steps (~5 kw) until steady boiling was established. The turbine bypass valve was partially closed to raise the boiler pressure. At approximately 65 kw the turbine downstream throttle valve was opened and the turbine started. The following conditions were typical at startup:

Power per heater rod, kw	0.75
Boiler liquid temperature, °F	1180
Condenser temperature, °F	~990
Boiler feed flow, gpm	~0.6

When the turbine pump was rotating, the valve between the electromagnetic pump and the bottom of the boiler was closed, and the turbine pump discharge was diverted through the nozzles of the boiler jet pumps. The speed of the turbine pump was very sensitive to the position of the downstream throttle valve. The pressure in the turbine cavity reflected almost directly the boiler outlet pressure and resulted in a high end load on the turbine thrust bearings. This condition was predicted some months prior to initial startup, but the system had already been welded and leak tested, and a decision was made to operate with the system as built. With the piping arrangement of run 1, the turbine pump bearing flow could not be established until the turbine was running.

On the initial startup a differential pressure across the turbine nozzles of 20 in. H₂O was required to start rotation. However, for each subsequent startup, the value became progressively higher, and it reached nearly 100 in. H₂O on the final start. The reasons for these higher values and the inability to control the speed are discussed in subsequent paragraphs.

Operation

The pump operated at a speed of approximately 11,000 rpm, well above the design value of 8,000 rpm. Attempts to reduce the speed by vapor throttling were unsuccessful, so it was left at the high value. After approximately 18 hr of operation, however, the speed suddenly decreased from 11,000 to 8,000 rpm. An attempt was then made to increase the speed

by closing the turbine bypass valve, but overheating (or shorting) of the boiler heater rods occurred, and the boiler power was automatically disconnected. After the heater rod overtemperature condition had been corrected, the turbine was momentarily restarted, but speed could not be maintained. Several more attempts to start the turbine were unsuccessful. The potassium was then drained into the fill and drain tank, and the residual potassium was vacuum distilled at temperatures ranging between 1100 and 500°F.

Examination of Turbine Pump

The turbine pump was removed from the system for examination. The interior appeared to be clean and the shaft could be rotated; however, the bearings felt rough. The unit was disassembled and it was found that rubbing had occurred around the outer periphery of both rotor and stator surfaces of the labyrinth at the turbine end; apparently both tilting and wobbling of the shaft had occurred during rotation.²⁵ There was evidence that a foreign particle had been lodged in the narrow space between the back of the impeller and the end of the TZM bearing mount. One of four tungsten dams in the turbine end thrust bearing had become dislodged (Fig. 4.4) and was not found. The bearing surfaces at the turbine and impeller ends are shown in Figs. 4.5 and 4.6.

The surface of the impeller-end journal (Fig. 4.7) showed some evidence of metal pickup near the in-board edge, but it had a considerably better appearance than the turbine-end journal. Both showed evidence of end loading at the outboard end. The impeller-end thrust surface showed no evidence of rubbing, which is consistent with a differential pressure gradient forcing the rotary assembly toward the pump end.

Condenser Operation

During the 18 hr of pump operation, infrared photographs were taken of the condenser by using the infrared light spectrum emitted from the radiator. Figures 4.8 and 4.9 were taken looking up from below the radiator. These pictures indicate that uniform scavenging of the radiator was occurring and that the radiator temperature was uniform. An indication

Fig. 4.4. Thrust Surface Showing Radial Slot with Missing Dam from the Turbine End of the Bearing Used in the Turbine Pump for Run 1.

Fig. 4.5. Journal Surface at Turbine End of the Bearing Used in the Turbine Pump During Run 1.

of the sensitivity of the infrared photography technique for temperature measurement is given by the fact that a color gradient on the fins is evident where the temperature differential between the roots and tips of the fins was estimated at only 25°F.

By partially closing a valve in the line between the radiator and the suction end of the turbine pump, liquid was allowed to accumulate in the radiator tubes. Figures 4.10 and 4.11 show the result. A sharp

Fig. 4.6. Journal Surface at the Impeller End of the Bearing Used in the Turbine Pump During Run 1.

Fig. 4.7. Journal Sleeves, Impeller End (Left) and Turbine End (Right), Used in the Turbine Pump During Run 1.

Fig. 4.8. Photograph Made with Polaroid Infrared Film Looking up from Below at the Intermediate Potassium System Radiator Running at About 850°F Under Cavitating Conditions. No liquid accumulation evident.

Fig. 4.9. Infrared Photograph of Radiator Tubes During Cavitation Control.

Fig. 4.10. Infrared Photograph of the Radiator at Essentially the Same Power and Temperature Conditions as for Fig. 4.8; However, the Valve in the Condensate Return Line Was Partially Closed so that the Liquid Inventory Distribution in the System Shifted and the Outer 8 in. of the Radiator Tubes Operated Full of Liquid.

Fig. 4.11. Closeup Infrared Photograph of the Outboard Ends of the Tubes and the Condensate Manifolds Under Operator Control.

difference in temperature was indicated by the thermocouples installed on either side of the light to dark transition zone. Figure 4.11 shows a thermocouple installed at the liquid-vapor interface.

Additional discussion of the infrared photograph technique for temperature measurement is included in Chapter 5 of this report and in the references.²⁵

Heater Rod Examination

At the completion of the run the boiler heater rod bundle was removed and examined. The external surfaces of the rods were extremely clean and showed no signs of damage after approximately 361 hr of boiling. These heater rods had been internally insulated with magnesium oxide and, since MgO has such a low insulating value above 1200°F, many of the rods were inoperative or faulty in the final stages of the run.

Runs 2 and 3

System Modifications

Figure 4.12 shows the modifications made to the system piping between runs 1 and 2. A vapor-throttling valve was installed upstream of the turbine to permit turbine startup without full boiler pressure in the turbine cavity. This reduced the thrust load on the turbine, since the turbine cavity was then subjected only to the low condenser pressure. The addition of this valve also increased system flexibility and permitted better manual control of turbine speed.

The piping around the electromagnetic pump was changed so that the electromagnetic pump could supply turbine pump bearing flow during startup of the turbine pump. With the original arrangement, the bearing flow could not be established until the turbine pump was up to speed. Additional piping changes allowed the electromagnetic pump to supply the boiler feed liquid through the boiler jet pump nozzles or through the bottom of the boiler while the turbine pump operated in its own closed loop.

Fig. 4.12. System Modifications Made Between Runs 1 and 2.

A valve was added to the bearing filter assembly to make it possible to vent any accumulated noncondensable gases. A pressure transmitter to read the absolute turbine cavity pressure was also installed. New lines were added to permit potassium sampling from the turbine pump discharge and the bottom of the boiler during operation.

Turbine Pump Modifications

A number of modifications were made to the turbine pump unit. Radial vanes were added on the back shroud of the pump impeller to reduce the

axial forces.^{4,26} Dams fabricated of TZM alloy were brazed in the radial slots of the thrust bearing, and minor changes were made to the labyrinth and bearing mount. The weight of the turbine wheel was reduced to move the center of gravity of the rotary assembly to between the journal bearings. This latter modification resulted in reduced bearing loads and avoided tilting of the shaft in the bearings, which in turn reduced edge loading.

Heater Rod Redesign

A newly designed boiler heater element was used. This element was fabricated with boron-nitride replacing the MgO insulation between the heater coil and sheath. The magnesium oxide insulation in the center of the heater was not replaced. These rods and a new spacer arrangement are described in Chapter 3, Core and Heater Element Design.

Startup

The outgassing and cleaning procedure used for run 1 was repeated because of the possibility of contamination during the many modifications to the system. Analysis of potassium samples did not, however, indicate an appreciable increase in the oxygen contamination, so the cleaning charge was left in the system as the operating charge.

The method of startup on these runs was considerably different from that used originally. The downstream and upstream turbine valves were both closed until just prior to startup. The downstream valve was closed in an attempt to keep condensate from flowing back and accumulating in the turbine housing. The upstream valve was used as the turbine cutoff valve. Power was increased to approximately 100 kw with the electromagnetic pump furnishing boiler feed flow through the boiler jet pumps. A turbine pump bearing flow of approximately 0.2 gpm was provided by the electromagnetic pump. The turbine downstream valve was opened fully, and then the turbine upstream valve was open partially to start the turbine. The turbine pump operated in a closed loop with the discharge through the radiator jet pumps. The turbine pump unit and the electromagnetic pump had a common suction line. The pressure differential across the turbine at the time of start was approximately 38 in. H₂O.

After the turbine pump had reached a speed of approximately 4500 rpm, the appropriate valves (see Fig. 1.2) were positioned to (1) provide the boiler feed and radiator jet pump flow from the turbine pump discharge and (2) valve the electromagnetic pump out of the main system. The turbine pump was then feeding the boiler and providing its own bearing lubricant. The electromagnetic pump electrical circuit was deenergized.

Operation

Tests were run to determine the operating characteristics of the system in order to obtain data for design of future Rankine-cycle systems. The operating history for run 2 is shown in Figs. 4.1 and 4.2. The boiler was operated up to a maximum power of 260 kw or 3 kw per active rod and to a maximum temperature of 1500°F.

Effect of Heat Sink Temperature. Several tests were made to determine the effect of heat sink temperature on the system temperature profile and operating parameters. Figure 4.13 shows the temperature profile

Fig. 4.13. Temperature Profile for Constant Power and Various Heat Sink Temperatures.

and Fig. 4.14 indicates the effect of heat sink temperature on pump discharge pressure, boiler outlet pressure, turbine pump speed, and cavitation coefficient.

Effect of Boiler Outlet Pressure. The data of Fig. 4.15 were obtained in a series of runs during which the boiler power was varied while no external control was exercised on the system; that is, no valve changes or other adjustments were made. The data are for two different settings of the vapor valves (turbine inlet and turbine bypass). The valve settings established a different boiler outlet pressure for the same turbine speed at a given reference power level. The tests were run to determine the effect of boiler outlet pressure on the control of the system. The cavitation data show that increasing the boiler pressure decreases the span over which the system remains in control. An examination of the curves for boiler outlet and turbine discharge pressures in Fig. 4.15

Fig. 4.14. Comparison of Operating Data at Two Heat Sink Temperatures.

Fig. 4.15. Effect of Boiler Outlet Pressure on System Characteristics.

shows the effect on the position and slope of the cavitation curves with power level.

Effect of Turbine Pump Speed. Figure 4.16 shows the effect of turbine pump speed on the performance of the system. Tests were run by establishing different turbine speeds at 170 kw while holding an identical

Fig. 4.16. Effect of Turbine Pump Speed on Performance of System.

boiler discharge pressure at the 170-kw power level. The adjustments were made by manipulation of the turbine inlet and bypass valves. For each speed setting, power was varied over the range shown, and no other control actions were taken.

Effect of Power on Temperature Profile. Tests were run to determine the temperature profile around the system as a function of power level. Figure 4.17 presents the data. The heat sink temperature was held

Fig. 4.17. Temperature Profile at an Approximate 550°F Heat Sink Temperature and Various Power Levels.

constant by varying the air flow through the heat sink annulus at the different power levels.

Evaluation of Hot Fingers. In order to establish the value of the hot fingers in the initiation of nucleation, startup tests were run with and without them. The hot fingers are heated independently and are raised to temperatures high enough to initiate boiling in the cavity. The vapor bubbles emitted from the cavity help initiate boiling in the core. A sound pickup on the side of a boiler was connected to a tape recorder to provide an audible record of the startup. Figure 4.18 shows the results of this test. The bottom curve in each case indicates the frequency and duration of the boiling bursts prior to initiation to steady boiling. These tests demonstrated that steady boiling is initiated at a lower power and lower pool temperature with the hot fingers operating.

Cavitation Mode of Control. Several tests were run to investigate the cavitation mode of control, and the results are given in Figs. 4.14,

Fig. 4.18. System Startups With and Without Hot Fingers.

4.15, and 4.16. The term "cavitation coefficient" shown on the graphs is defined as

$$C = \frac{\text{Measured pressure rise across the pump}}{\text{Pressure rise at measured speed and flow from pump performance curve (Fig. 4.19)}}$$

Pump performance tests were conducted with water prior to installation of the pump in the system, and the data were converted to performance with

potassium. The predicted pump performance with potassium is shown in Fig. 4.19.

The cavitation coefficient is a measure of head degradation occurring across the pump and thus for a noncavitating pump has a value of one. Figures 4.14, 4.15, and 4.16 show that the shape of the cavitation coefficient curve for a range of power is concave downward. Projections of the points plotted would cross 1.0 at both ends of the curve. When in the cavitation control mode, the system is in equilibrium and the pumped liquid flow to the boiler is equal to that leaving the boiler as vapor. This condition results in a steady inventory distribution within the system.

A test was conducted to show that the turbine was operating under cavitation control. Figure 4.20 shows the results of this test. Changes were made in the flow resistance in both the vapor and boiler feed control valves; the turbine was slowed; and liquid accumulated in the radiator. This resulted in a change in the net positive suction head, and the

Fig. 4.19. Turbine Pump Head Versus Flow for Operation with Potassium at 1040°F. Calculated from water test data given in Ref. 16.

Fig. 4.20. Typical Transition from Cavitation Control.

pump went out of cavitation, as evidenced by agreement between the actual and noncavitating pump head rises. Repositioning of the valves returned the system to the earlier condition.

Figure 4.11 shows the infrared photograph taken of the condenser while the pump was not in a cavitating mode. Figure 4.9 is a photo of the same condenser area while the system was under cavitation control. As discussed previously, the color change in Fig. 4.11 is assumed to be a result of liquid potassium accumulating in the condenser tubes because of the shift in system inventory to the condenser. More complete descriptions of the cavitation mode of control are given in Chapters 2 and 5.

Effect of Inventory on Control. A program to investigate the effect of total inventory on system control was planned and, although some tests were conducted, these were not run to completion. Since power level was so limited with the heater bundle for this run, the minimum inventory at a maximum power could not be established.

Run Termination

After approximately 90 days of operation, the frequency of scrams initiated by the ground-current heater rod protection system increased. After each scram the faulty heater was disconnected and operation again resumed. After a scram on the 135th day of operation, attempts to restart the system were unsuccessful. Power was applied to the boiler core after the faulty heater was disconnected, but the potassium flow rate from the radiator to the boiler through the electromagnetic pump proved insufficient to maintain the liquid level in the boiler expansion tank. The boiler power level required to start the turbine could not be reached before the liquid level in the boiler expansion tank dropped below the minimum safe operating level. Subsequent examination revealed that the electromagnetic pump lugs had developed a short circuit during the thermal cycling of the system.

Since the rate of heater rod failures indicated that the heater element life expectancy may have been reached, a decision was made to terminate the test and remove the heater bundle for examination. The examination of the bundle disclosed that six heaters had failures in the sheath. Subsequent examination revealed that these failures were a result of electrical arcing. Aside from the six holes, the rod bundle was extremely clean and bright. Figures 4.21 and 4.22 show the condition of the rod bundle immediately after removal from the system.

System Examination

Corrosion. Potassium samples indicated a low oxide content (36 to 40 ppm). Trace indications of iron, nickel, and chromium were found in the potassium from the drain tank. The relative amounts of the contaminants indicated that they were probably corrosion products.

PHOTO 72948

Fig. 4.21. Side View of Rod Bundle 2 Shortly After Removal from System Following 2801 hr of Operation.

PHOTO 72949

Fig. 4.22. Top View of Rod Bundle 2 Shortly After Removal from System Following 2801 hr of Operation.

Portions of the line between the electromagnetic pump and the boiler were examined for possible plugs. No evidence of plugging was found in the main feed line to the boiler, but a bypass line in this circuit, which also serves as the drain line for the boiler, contained a mass of potassium, some K_2O , pieces of stainless steel-MgO agglomerate, and a small amount of two types of mass-transfer crystals. The potassium present was believed to be that retained on the other deposits as a result of surface tension effects, and the potassium oxide was believed to have resulted from reaction of the potassium with air when the line was cut to remove it from the system. The stainless steel-MgO agglomerates were presumably the solidified mass generated by the electric arcing mentioned above. A very small amount of one type of mass-transfer product precipitated on the walls as small sandy-like nodules. It is likely that this precipitation resulted from the thermal gradient that existed between the boiler and the drain line during the extended periods when the electromagnetic pump was not in operation. Examination of the second type of mass-transfer crystal showed dendritic formations and structures which suggested that the material had broken off elsewhere, been trapped in the low-velocity region at the bottom of the boiler, and subsequently deposited in the drain leg when the boiler was drained. Light deposits of this type were found on the boiler jet-pump nozzles, and microscopic examination of the nozzles revealed surfaces which suggested that some of the deposits had broken off.

During operation of the system, there were indications that the pressure drop through the nozzles had varied. It is worth noting that this variation in pressure drop was not enough to affect the system operation materially. The deposits on the nozzles were probably due to the presence of oxygen in the system, and the operation of the jet pump under conditions such that a vapor "plume" that existed at the nozzle discharge left the deposition at the nozzle.⁵ In addition, if feed liquid at a temperature well below boiler temperature entered through the jet pumps, the nozzle would tend to act as a "cold finger" in the boiler loop and precipitate out corrosion products on the nozzle tip.²⁷ Figures 4.23 and 4.24 are photos of the boiler jet pump nozzles after operation for 2801 hr. More details of the corrosion-product examination and analysis

Fig. 4.23. Boiler Jet Pump Nozzle After 2801 hr of Operation.

Fig. 4.24. Closeup of Boiler Jet Pump Nozzle After 2801 hr of Operation.

are covered in the Medium-Power Reactor Experiment quarterly progress reports.^{26,28} It is important to note that the potassium analyses indicated that there had not been a serious oxygen-contamination problem.

Heater Rod Failures. Evidence that potassium had leaked from the system through the holes in some of the heaters was given by severe external corrosion that caused deterioration of the weld closures at the bottoms of several heaters and had attacked the outside of the heater header plate (see Fig. 4.25). The inside of the header plate was clean,

Fig. 4.25 Bottom of Header Plate of Rod Bundle 2 Shortly After Removal from System Following 2801 hr of Operation.

and the only evidence of attack was in the sleeve that attached the heater to the plate. It appears that the potassium reaction with air on the outside of the header plate had created a tight seal that limited both the leakage of oxygen into the system and the potassium out of the system.

The failures in the electric heaters were detected by low resistances across the magnesium oxide or boron nitride insulation. During the test, as the resistances reached a value of approximately 1 megohm, the appropriate heaters were disconnected from the power source, often without interrupting the test.

The resistance is a function of temperature and decreases as the rod temperature increases. Approximately 20 heaters were removed from service within a few weeks after system operation began when the resistances dropped to approximately 1 megohm as the temperature was increased. The resistance on the remaining heaters was above the 1-megohm value at the higher temperatures and remained within acceptable limits for several months, until a sharp increase in heater failure rate began. The latter failures occurred as a sudden resistance decrease from normal values to the scram point of the protective equipment. The system was automatically scrammed, and the rod involved was disconnected before power was restored to the boiler. Figure 4.26 shows the failure rate of the heaters. The increase in failures in the latter portion of the test was an indication

Fig. 4.26. Heater Failures in Boiler Bundle 2 During Operation.

that the life expectancy of the heaters had been reached. The failure rate versus operating life for the heater elements matches the classic "bathtub" curve for new equipment operation. At the time it was decided to terminate the test, after 2801 hr of boiling operation, 57 of the 91 heaters had been disconnected.

The postoperation examination of the heaters indicated damage to the heater leads of the type shown in Fig. 4.27, which is an x-ray photograph. No acceptable explanation for the apparent attack on the nickel leads was available, and it was recognized that additional heater element development work was needed. Accordingly metallurgical studies were initiated at ORNL and the problems were discussed with the heater manufacturer.

System Restart

Because of procurement uncertainties for new heaters, the original MgO boiler bundle used in run 1 was reinstalled in the boiler. This core had been removed because of its limited power; however, it was felt that sufficient power could be achieved to permit additional running time on the turbine unit. Subsequent operation of the boiler revealed that the MgO bundle had deteriorated since its removal from the test system, and only 18-kw maximum power was available at safe operating conditions for the heater rods. However, the turbine pump unit was restarted after the system had been degassed and reflux cleaned. By completely closing the turbine bypass valve and completely opening the turbine inlet valve, the pump was run for an additional 6.5 hr at the 18-kw maximum power level. Because of the limited power the decision was made to remove the turbine pump for examination and await delivery of heaters for a new boiler bundle.

Turbine Pump Operation and Examination

The turbine pump had been operated a total of 2577 hr under cavitation control and had been started 46 times. Table G.1 in Appendix G shows the history of starts and Fig. 4.28 gives information on system power, pump speed, and pump cavitation for each day of operation. Examination of the unit disclosed no corrosion or erosion of the turbine buckets or pump impeller (See Figs. 4.29 and 4.30). Visual examination

Fig. 4.27. Typical Lead Damage and Sheath Failure of Heater Rods in Bundle 2.

LEGEND (DEVIATIONS FROM CAVITATING MODE OF OPERATION)

- A SYSTEM STARTUP INADVERTENT NONCAVITATING MODE
 B INADVERTENTLY RAISED POWER ABOVE CONTROL RANGE
 C NOT KNOWN INVENTORY SHIFTED INTERMITTENTLY DURING THIS DAY PROBABLY DUE TO LOW SYSTEM INVENTORY
 D NOT KNOWN OCCURRED ON WEEK END PROBABLY POWER VARIATIONS WITH SYSTEM AT EXTREME END OF CONTROL RANGE

LEGEND (DEVIATIONS FROM CAVITATING MODE OF OPERATION)

- E POWER DELIBERATELY LOWERED BELOW CONTROL RANGE
 F INVENTORY TRANSFERRED TO CONDENSER BY THROTTLING BOILER FEED FLOW TURBINE PUMP SUCTION PRESSURE RAISED BY LIQUID LEVEL IN RADIATOR AND TURBINE PUMP FORCED OUT OF CAVITATION
 G TURBINE PUMP SPEED REDUCED AND ADDITIONAL K INTRODUCED INTO SYSTEM INVENTORY INVENTORY SHIFTED TO CONDENSER BY THROTTLING BOILER FEED FLOW TURBINE PUMP SUCTION PRESSURE RAISED BY LIQUID LEVEL AND TURBINE PUMP FORCED OUT OF CAVITATION

NOTES

- 1 TURBINE PUMP FLOWS BETWEEN 2 AND 4 gpm AT ALL TIMES
 2 ONE TYPICAL SET OF DATA SHOWN FOR EACH DAY OF OPERATION

Fig. 4.28. Selected Parameters Related to Cavitation During Operation of Turbine Pump 2.

Fig. 4.29. Impeller from Turbine Pump Unit 2 After 2577 hr of Operation.

of the bearings revealed very small indications of wear. The turbine-end journal had an approximately 0.2-mil taper, and both bearing diametral clearances had increased by approximately 0.3 mil. Some very light metallic pickup was visible on both journal surfaces. The journal run-outs were of the order of 0.4 mil.

Run 4

Run Objectives and Overall Achievements

A new boiler core bundle was installed and a new turbine pump unit was assembled and installed prior to the final operating run. The assigned objective of this run was to establish that the heater elements

Fig. 4.30. Wheel from Turbine Pump Unit 2 After 2577 hr of Operation.

and boiler assembly could perform at power ratings up to 4.0 kw per rod and to establish that the system could be operated under cavitation control at these conditions.

Prior to the orderly shutdown of the system September 15, 1966, 1113 additional hours of boiling and 1024 additional turbine pump hours were accumulated. The heater elements were operated up to 4.32 kw per element, and a total power level of 335 kw was reached. The system operated smoothly and, within the electrical resistance limitations of the individual heaters, demonstrated that a heat flux of 136,000 Btu/ft²·hr (approximately 20% above the average specified for the MPRE core) was well within the capabilities of the boiler system. Figure 4.3 shows the maximum and minimum power levels and the maximum and minimum boiler outlet temperatures for each day during run 4. Figure 4.31 shows the power per heater for each day of operation, and Fig. 4.32 shows the average cavitation level of the pump and the average pump speed for each day of operation.

Fig. 4.31. Maximum Daily Power per Heater Rod in Run 4.

Fig. 4.32. Selected Parameters Related to Cavitation During Operation of Turbine Pump 3 in Run 4.

Heater Bundle

The new heater bundle was assembled from carefully tested heater rods in which the internal insulation was a combination of MgO and BN. A new swaging technique and additional quality controls were initiated by the heater rod vendor at ORNL's request. Each rod was tested for

internal insulating resistance at 1500°F prior to assembly into the bundle. Values below 1 megohm at temperature were cause for rejection. This screening process proved effective in rejecting most of the poor heaters prior to assembly into the heater bundle. Figure 4.33 shows the number of heaters operating during this run. Comparison of this curve with Fig. 4.26 (failure rate of bundle 2) shows that the number of heaters disconnected in the early stages of operation was much less than for bundle 2. Bundle 3 was not operated as long as bundle 2, and a lifetime limitation was not reached. A steady failure rate of the type experienced with bundle 2 after approximately 2000 hr of operation had not developed at termination of run 4 after some 1100 hr of boiler operation. The increase in the number of operable heaters in Fig. 4.33 represents heaters put back in service after having been removed.

Turbine Pump, Piping, and Instrumentation

Turbine pump 3 was assembled from the original turbine wheel and modified housing of pump 1 and a new rotary assembly, including the shaft, pump impeller, and bearings. The pump was essentially identical to pump 2. The main piping system was identical for operating runs 2, 3, and 4. A new Taylor Company differential-pressure-measuring unit was added in parallel with the original boiler pressure drop manometer in an effort to provide an accurate check on the use of a manometer for pressure drop

Fig. 4.33. Heater Failures in Bundle 3 During Run 4.

measurements. A Foxboro differential-pressure transmitter was added to measure the pressure drop in the line between the radiator jet pump discharge and the turbine pump suction. This latter unit was added in an attempt to determine whether the measured pressure drop could be used to confirm that vapor was being carried through the radiator jet pump to the turbine pump suction.

Startup

The system was outgassed and reflux cleaned in the manner previously described; however, the outgassing operation was shortened from approximately 7 days to 30 hr. The potassium used for operating runs 2 and 3 was used for the cleaning operation. The oxygen content increased from approximately 53 to 135 ppm during the 28-hr cleaning run. This potassium was drained and the system was recharged with clean potassium (<30 ppm O_2).

Startup of the turbine pump was identical to the startup for runs 2 and 3. The electromagnetic pump was used to provide the initial bearing lubrication and boiler feed flow, and the turbine pump was started and allowed to operate in its own closed loop. The differential pressure required to start was 32 in. H_2O . This starting pressure drop was slightly lower ($\sim 17\%$) than that for pump 2.

Shortly after startup a leak across the seat of a valve between the system and the dump tank was discovered, so the boiler power was turned off and the valve seat was thoroughly cleaned by raising and lowering the potassium between the system and the dump tank. Attempts to restart the turbine pump by the previously used method were unsuccessful. The turbine would not start with a pressure drop of over 100 in. H_2O . The turbine downstream valve was therefore closed and the upstream valve was opened. Power was raised to approximately 100 kw and the downstream valve was slowly opened. The turbine then started at a pressure drop of approximately 32 in. H_2O . Closing the downstream valve and opening the upstream valve could have accomplished two things. Condensed vapor could have accumulated in the turbine cavity and dislodged any foreign material from the bearing surfaces or the increase in turbine cavity (discharge) pressure

could have caused the shaft assembly to move away from a thrust surface and thus reduce bearing drag (see Chap. 5). Several subsequent starts were made by the "normal" method. Some of these starts required more than the 32 in. H₂O pressure drop, but all required less than 45 in. H₂O.

Operation

During the first few days of operation, the turbine pump parameters tended to oscillate, but venting the top of the gas trap and the bearing filter reduced the oscillations. This indicated that the system was not completely outgassed and that noncondensable argon cover gas used during system shutdown was probably accumulating at the high spots of the system. It is possible that the longer period of outgassing used in previous runs was necessary to remove all the noncondensable gases. By the end of the first week of operation, the venting was no longer necessary.

Power Level Variations. To accomplish the assigned objectives for the run and at the same time determine the range of power over which the system would remain in cavitation control, a decision was made to set the cavitation coefficient near 1.0 at the maximum power available. The maximum power was to be determined by the maximum safe operating resistance of the largest number of operable rods. Power was gradually increased to a maximum of 319 kw.

At each increment of power, the pump cavitation coefficient was changed to a value near, but still below, 1.0. This was accomplished by opening the turbine inlet valve to increase the pump speed. The change in position of the inlet valve resulted in a departure of the boiler outlet pressure from the design condition, and a change was necessary in the turbine bypass valve to bring the pressure back to the desired value. Several days were required to make the necessary adjustments and arrive at a power of 319 kw. As power was increased, the boiler outlet temperature increased and caused higher internal temperatures in the boiler heaters. A limit of 1-megohm minimum resistance had been established as a safe operating level for the heaters and, as the heaters reached this value, they were disconnected and the power was adjusted upward for the remaining heaters. Eleven rods had been disconnected when a power level of essentially 4 kw per rod, or a total power of 319 kw, was reached.

The pump speed was then adjusted to give a cavitation coefficient of approximately 0.85, since data analysis indicated that a cavitation coefficient of 0.85 at this power level would be sufficient to keep within the cavitation control range at the full design power of 364 kw. The pump speed at this time was between 7800 and 7900 rpm, which indicated that at full design power (364 kw) the pump speed would be less than 10,000 rpm.

The power level was then gradually decreased to approximately 100 kw, at which point the cavitation coefficient was approximately 0.90. Because the boiler discharge pressure was within 1 lb of the pump discharge pressure, it was not felt prudent to proceed to lower power. The power was then gradually increased to approximately 310 kw. This series of tests demonstrated that the power could be varied by a factor of 3.2 without valve control of any kind. The heat sink temperature was held constant at 500°F as the power level was increased until the temperatures of the radiator (condenser) reached the design point. When this condition was reached, the air cooling was increased as power continued up to maintain the radiator temperature at design values within the limits of the air-cooling system. This technique resulted in heat sink temperatures as low as 380°F at the higher powers. At maximum power levels the air-cooling system was unable to maintain radiator temperatures at the design value, and the radiator temperature rose slightly.

A color photograph (Fig. 4.34) was made of one wing of the radiator with the latter operating at 1120°F. This photograph was made in the natural glow of the tubes. The inlet header can be seen in the lower portion. The spider-like object is the radiator jet pump.

Pump Speed Variations. The foregoing operation disclosed an abrupt change in the pump speed curve between 135 to 150 kw. Two more runs were therefore made over the power range at different turbine inlet and turbine bypass valve settings, and in each case the break in the speed curve was encountered. The valves were not operated during the power variations; however, after each power run, the pump speed was raised or lowered to a new value at 150 kw by adjusting the turbine inlet valve. The boiler outlet pressure was then readjusted by positioning the turbine bypass valve to match the previous pressure at that power. Figure 4.35 shows the effect of pump speed on system parameters. Although the

Fig. 4.34. Radiator During Operation at 1120°F.

Blank

* THESE DATA NOT AT SAME HEAT SINK
TEMPERATURE AS OTHER DATA

Fig. 4.35. Effect of Speed on Performance of Turbine Pump 3.

data of September 9, 1966 to September 13, 1966 were taken at a heat sink temperature different from that of the other data shown, the effect of the heat sink change is small.

The change in slope of the pump speed curve between 130 and 150 kw was not readily explainable, particularly since such behavior was not recorded during run 2. During run 2 a change in the pressure drop between the pump discharge and the boiler outlet pressure was experienced but, as described earlier, this change was attributed to partial plugging of the boiler jet pump. The pressure drop condition was experienced intermittently during operation of run 2 but was not encountered during runs 1, 3, or 4. Figure 4.36 shows a comparison of the system parameters for data obtained during run 2 where the liquid line pressure drops at equivalent power levels differed substantially. This difference in pressure drop did not affect the control of the system.

System conditions were then adjusted to match a typical set of conditions in run 2, and the new data were plotted to obtain a comparison of the speed curves of the two pumps. The results are shown on Fig. 4.37. The marked changes in pump speed and cavitation coefficient are discussed in Chapter 5.

Temperature Profile of System. A test was run to obtain a comparison of system temperatures (boiler outlet, turbine inlet, turbine outlet, radiator inlet, radiator outlet) when the radiator outlet temperature was held above values calculated to represent minimum temperatures for normal radiator operation at the corresponding power levels. This test was run over a power range of 80 to 335 kw without valve control action. During this run the radiator outlet temperature was held above the values shown in Fig. 4.38 by adjustment of the cooling air to the heat sink. Temperatures recorded during this test are shown in Fig. 4.39. A complete system temperature profile for four different power levels during this test run is shown in Fig. 4.40. The increase from a maximum power of 319 kw (established earlier) to 335 kw maximum power was obtained by increasing the voltage on the heater rods through changing of the taps on the main transformers. The power reached represented 4.35 kw per rod.

Fig. 4.36. Effect of Change in Pressure Drop in Liquid Line in Run 2.

A test to determine the effect of heat sink temperature on the system temperature profile was run at 150 kw. During this test the heat sink temperature was varied from 502 to 226°F by adjustment of the cooling air flow, and data were taken to provide the temperature profiles shown in Fig. 4.41.

Fig. 4.37. Comparison of System Data from Operation of Pumps 2 and 3.

Fig. 4.38. Minimum Radiator Exit Temperatures During Power Test at 80 to 335 kw.

Fig. 4.39. Pertinent Temperatures Over Power Range 80 to 335 kw.

Fig. 4.40. Temperature Profile at Various Power Levels.

Fig. 4.41. Comparison of Temperature Profiles for Constant Power and Varying Heat Sink Temperatures.

Tests with Instantaneous Changes in Boiler Power. Instantaneous step changes in boiler power were made to simulate a sudden increase or decrease of the connected load to a reactor power system. An instantaneous reduction of power from 199 to 147 kw (52 kw) and an instantaneous increase in power from 150 to 186 kw (36 kw) were accomplished without control action. The automatic data-acquisition system (DEXTIR) was allowed to scan the system parameters continuously during and after the instantaneous change in power. Each parameter was read and recorded once each 9 sec. A graph and discussion of the data appear in Chapter 5 of this report.

Summary. The third turbine pump unit was operated 1024 hr and the boiler core was operated 1113 hr at boiling conditions. Nine starts were made during this period. Table G.2 in Appendix G shows the operating history of the pump. During most of this run the pump was in deep cavitation (less than 0.5 of the noncavitating head). Figure 4.28 shows a daily record of the pump operation. Also shown are the pump speed and power associated with the pump heads. The noncavitating head plotted was obtained from the actual test curve shown in Fig. 4.19.

On September 15, 1966, the boiler power was manually decreased to 0, and the test of the intermediate potassium system was terminated. The potassium was drained into the dump tank and the piping temperatures were reduced to ambient. An overpressure of argon was placed in the dump tank and system, and the system was placed in standby condition.

5. RESULTS AND ANALYSIS

Operation of the intermediate potassium system was carried out for several thousand hours at power levels up to 335 kw. The system operated successfully over a wide power range with no external control required to match the boiler feed pump performance to the system requirements. The significant results from system operation were obtained during the two extended periods of loop operation. The first period commenced in late July 1965 and extended through December 1965, with an interim one-month shutdown period. The second period of operation extended from July to mid-September 1966, at which time the operating program was terminated as a result of cancellation of the MPRE program.

Operation during 1965 was concerned primarily with determining the basic feasibility of a boiling-potassium Rankine-cycle system designed to operate with cavitation control for accommodating changes in system power. Because of the program termination, limited objectives were assigned to the 1966 test operation. These objectives were to demonstrate that the boiler core could be operated at heat flux levels equivalent to 4 kw per rod and to establish that cavitation control was effective over a wide range and up to power levels near the maximum design power. Additional information on the effects of system parameters on system control was also obtained.

System Control

Response to Changes in Power Level

The responses of major parameters of the IPS to changes in power level are shown by the data of Fig. 5.1. The linear relationship of boiler outlet pressure with power on the logarithmic graph is a manifestation of the coupling of the system through orifices (or valves) that operate with a critical vapor velocity at the point of restriction. The vapor flow rate from the boiler is proportional to the boiler power, while the flow rate through flow restrictions is proportional to the pressure upstream of these restrictions.

Fig. 5.1. System Performance Under Cavitation Control.

The effects of power, radiator temperature, pump speed, and inventory distribution on system control were studied. The system response to slow power changes when under cavitation control was recorded on a Sanborn high-speed eight-channel recorder, and a typical trace is shown in Fig. 5.2. The boiler pressure, turbine speed, and boiler feed flow may be seen to follow the power change with a time constant of less than 1 min. Actual system time constants for a similar Rankine-cycle system operating on water were determined to be about 40 sec.⁶

Cavitation Control

The term "cavitation coefficient" shown on the graph and discussed in Chapter 4 was defined as the measured pressure rise across the pump divided by the noncavitating pressure rise at the same flow and speed. The noncavitating characteristics were obtained from water test data.

As explained in Chapter 2, the degree of pump cavitation serves to accommodate the difference between system requirements and pump capabilities. From Fig. 2.4 the shape of the cavitation coefficient to accomplish this can be anticipated. The coefficient will have a value of one at the intersection of the curves (where the pump passes out of cavitation) and will have a minimum value somewhere between these intersection points. The data shown in Fig. 5.1 confirm this. From the explanation of system control given previously, it is apparent that when the cavitation coefficient of the turbine-driven feed pump reaches a value of one and the turbine pump remains out of cavitation, the pump is no longer capable of exercising control on the system in response to changes in power.

The behavior of the system under conditions where the cavitation coefficient reaches a value of one is shown in Figs. 5.3 and 5.4. Figure 5.3 is a graph of the system operating initially under conditions of cavitation control but very close to a cavitation coefficient of one. Increasing the power caused a minor shift of inventory to the radiator, as indicated by the reduced level of the liquid in the separator expansion tank. An abrupt rise in pump suction pressure occurred that forced the turbine pump unit out of cavitation, and the value of the cavitation coefficient reached one. The decrease in pump speed is a result of the

Fig. 5.2. Response to Power Change During Cavitation Control.

Fig. 5.3. Effect of Cavitation Coefficient of Close to One on System Performance.

Fig. 5.4. Transition to Operator Control.

increased hydraulic loading on the pump. At the reduced speed the pumped flow to the boiler no longer matched the boiler requirements. Operation of the system with a noncavitating turbine pump required periodic operator action, termed operator control.

The response of the system under conditions where the turbine pump passes out of cavitation is shown by the high-speed Sanborn chart recorder trace in Fig. 5.4. The minor shift in inventory (see separator level) increased the cavitation suppression head on the condenser jet pump and brought the jet pump out of cavitation. The pressure rise across the jet pump increased abruptly, and the effect on the turbine pump suction pressure is evident from the recorder chart.

Pressure profiles for the system during cavitation control and during noncavitation operation are shown in Figs. 5.5 and 5.6. The data were taken immediately before and immediately after indications that the turbine pump had entered a condition of cavitation operation. From the graph the major effect of being out of cavitation is seen to be a rise in the condenser jet pump discharge pressure, and hence a rise in the turbine pump suction pressure, which is seen as a sharp rise in turbine pump discharge pressure. Note that the pressures through the boiler, turbine, and radiator inlet remain unchanged. Figure 4.20 showed a graph of data on pump speed and suction and discharge pressures during a transition to operator control at approximately constant power. The data in that graph can be seen to reflect the same response as the high-speed recorder charts for the similar transition shown in Fig. 5.4.

Accurate measurements of the low pressures that existed at the condenser discharge proved difficult through much of the test program. Since the performance of the jet pump is sensitive to changes of even a few inches of liquid in the sideport suppression head and because changes of this magnitude could not be detected accurately with the available instrumentation, the data on jet pump performance were erratic.⁵

The line pressure drop and liquid subcooling are both effective in determining the cavitation suppression head on the condenser jet pump.⁶ Since these values are sensitive to line temperatures, the technique of using infrared sensitive film was employed to provide additional information on the temperature distribution in the radiator discharge lines.

Fig. 5.5. Calcomp Graphs of System Pressures During Cavitation Control.

Fig. 5.6. Calcomp Graphs of System Pressures During Noncavitation.

From this information, qualitative estimates of the liquid inventory distribution in the system could be made. The location where vapor condensation was complete was assumed to be the location of the abrupt change in system temperatures where liquid subcooling began and reduced the line temperatures. In Fig. 4.9 the infrared photograph indicates no abrupt color change in the discharge headers of the radiator and suggests that vapor is being carried through the headers into the condenser jet pump feeder lines. The photograph shown in Fig. 4.11 reveals a sharp color gradient and suggests that liquid is backed up into the ends of the radiator tubes. Because of the sensitivity of the condenser jet pump to the cavitation suppression head, the presence of liquid in the radiator tubes forces the jet pump out of cavitation, and the resultant relatively high discharge pressure of the jet pump in turn brings the turbine pump out of cavitation. These conditions are reflected in Figs. 5.3, 5.4, and 5.6 and the photograph in Fig. 4.11. Under all conditions, when the system operated in cavitation control, there was no evidence of liquid accumulation in the radiator tubes; rather, the infrared photographs indicated that vapor was carried into the jet pump feeder lines.

A composite infrared photograph of the condenser discharge manifold and the feeder lines leading to the condenser jet pump, with the system under cavitation control, is shown in Fig. 5.7. Note particularly the color gradient along the feeder line leading from the radiator liquid header to the jet pump suction. Temperatures taken with the thermocouples located along this line are indicated on the figure. The high sensitivity of temperature with slight pressure drop under saturation conditions at these pressures accounts for some of the temperature change. The technique of using infrared photographs for such applications has the potential of permitting a complete temperature mapping of the radiator and feed lines without employing many hundreds of thermocouples. The use of this technique of temperature measurement, however, permitted only a qualitative inference as to the presence of liquid or vapor in the lines. Quantitative measurements of liquid-to-vapor ratios would have required the installation of density sensitive detectors that could scan the process lines of interest.

Fig. 5.7. Composite Infrared Photograph of Radiator Discharge Headers and Jet Pump Feed Lines During Cavitation Control.

Effect of Heat Sink Temperature

The effect of the heat sink temperature was investigated. The test results shown in Fig. 4.14 were obtained at constant power and varying air flow to the heat sink to control the heat sink temperature. The data graphed are for almost identical settings of vapor and liquid restrictions. A slight increase in pump speed and reduction in pump cavitation coefficient occurred with reduced radiator temperature, and this

can be attributed to the slight change in pump suction pressure. Under the conditions of Fig. 4.14, the pump suction pressure was extremely low. Normally, changes in radiator conditions can affect system operation only by affecting the net positive suction head on the turbine pump and, over the power interval for the cases shown, these effects proved to be very small.

Effect of Pump Speed

The effect of pump speed on the cavitation coefficient curve was studied in order to determine its effects on the range over which the turbine pump exercises control action on the system. The pump speed was preset at an initial power level of 150 kw for each of two different pump speeds. Boiler pressure at each initial condition was identical, and in each case power was varied from the initial condition. An increase in pump speed at specified system conditions reduced the cavitation coefficient, and this lower coefficient extended the power interval range for which the cavitation coefficient remained less than one. As may be seen in Figs. 5.8 and 5.9 the actual system data agree with the data obtained from an analog computer study of the control characteristics of a similar single-loop Rankine-cycle system.²⁹

Effect of Boiler Pressure

The effect of a change in boiler pressure at a particular set point was also examined in the analog study of the system. A comparison of the effect on the cavitation coefficient curves for the analog and from the IPS studies can be seen by a comparison of curves A and B of Fig. 5.9 with the data of Fig. 4.15. Note that in both graphs an increase in pump speed tends to shift the left end of the cavitation curve to the left, while a decrease in boiler pressure tends to shift the right end of the curve to the right. The increase in pump speed at the reference power level forces the pump into deeper cavitation, as indicated on the graphs. At higher power levels, pump speed is a weak function of power. Thus the effect of increased pump speed on the cavitation coefficient, which is apparent at low power levels, is much less apparent at high power levels. At reduced power levels the increased pump speed is seen

Fig. 5.8. Effect of Turbine Pump Speed on System Performance.

Fig. 5.9. Results of Analog Computer Study of the Effect of Turbine Speed and Boiler Pressure on Cavitation Coefficient for Single-Loop Rankine-Cycle System.

to be effective (see Figs. 5.8 and 5.9) in keeping the pump in cavitation to lower powers.

The increase of boiler pressure increased the pump discharge pressure (hence, pump pressure drop) and tended to bring the pump out of cavitation. The effect of this on the cavitation coefficient at higher power levels is clearly seen on Figs. 4.15 and 5.9. As power is reduced the absolute boiler pressure and pump discharge pressures are substantially lower, and hence the boiler pressure effect is reduced. The speed, however, at low power levels is a stronger function of power, and its effect becomes dominant in the low-power range. Thus the boiler pressure is less of an influence at low power levels, as indicated by the data shown. These effects demonstrate how the initial conditions of pump speed and boiler pressure at a given power level influence the range over which cavitation control is effective.

Results of Run 2 Operation

In summary, run 2 operation demonstrated the feasibility of controlling a Rankine-cycle system operating with potassium by affecting the degree of cavitation in the boiler feed pumps and the necessity for maintaining the pump in cavitation. Run 2 also demonstrated the effects of heat sink, pump speed, and boiler pressure on system performance.

A major limitation on operation during run 2 was imposed by the boiler heater element capability. In general, maximum total power was limited to just over 200 kw. This prevented satisfactory investigation of system performance at higher power levels and a determination of the control range for the system under various initial conditions. Run 4, performed during the late summer of 1966, utilized a new boiler with improved heater elements which permitted operation to a maximum total power of 335 kw.

Difference in System Response to Power Changes in Run 4

Early in run 4 it was apparent that a difference in system response to power changes was occurring at a power level of about 150 kw. Figure 5.10 shows data from run 4, and a comparison of these data with data of run 2 is given in Fig. 5.11. The primary differences are in the higher

Fig. 5.10. System Characteristics During Typical Period in Run 4.

Fig. 5.11. Typical Data for Tests During Runs 2 and 4.

turbine cavity pressure and the marked change in pump speed over a narrow range of power, as may be seen in the comparison of the speed curves of Fig. 5.11. The pump speed was observed to drop off sharply on power reduction and, similarly, to increase on a power increase. The effect was reproducible over a wide range of system conditions but almost always occurred at a system power range of approximately 130 to 150 kw. The data of Fig. 5.10 are typical of several power test runs made to study the speed change phenomenon. The effect of the "speed break" on the cavitation coefficient is shown as a discontinuous curve because the abrupt change in pump speed with power precluded getting data at interim pump speeds.

Figure 5.12 is a linear graph of all the pertinent system parameters associated with the run shown in Fig. 5.10. No inconsistencies appear in any of the data at the power level where the large speed change occurs.

An examination was carried out for determining the effects of the convergent-divergent turbine nozzles on turbine performance. The turbine nozzle dimensions are given in Fig. 5.13, and the nozzle exits can be seen in Fig. 3.11. Normal system operation occurs with the potassium vapor at sonic velocity in the nozzle throat.

The convergent-divergent nozzle operating at conditions for which it was designed follows a pressure curve, $P_i P_c$, as shown in Fig. 5.14. Flow in the divergent section of the nozzle is supersonic for normal operation. An increase in the turbine back pressure above pressure P_c results in a pressure shock in the diffuser section of the nozzle and subsonic flow downstream of the shock. This is represented by 1, 1', 1'' and 2, 2', 2'' as typical pressure shock curves. The position of the shock front is a function of the cavity pressure and physical properties of the fluid.

A series of calculations were made to determine the required exit-to-throat area ratio leaving the diffuser section as a function of the pressure ratio across the turbine. The calculations were based on the ideal gas equation for full adiabatic expansion from an upstream pressure P_i to the cavity pressure P_c :

Fig. 5.12. System Parameters Versus Power During Tests with Speed Break.

Fig. 5.13. Convergent-Divergent Turbine Nozzle Dimensions.

Fig. 5.14. Effect of Back Pressure Variation on Pressures in Diffuser Section of Convergent-Divergent Nozzle.

$$\frac{A_t}{A_e} = \left(\frac{k+1}{2} \right)^{1/(k-1)} \left(\frac{P_c}{P_i} \right)^{1/k} \left\{ \frac{k+1}{k-1} \left[1 - \left(\frac{P_c}{P_i} \right)^{(k-1)/k} \right] \right\}^{1/2},$$

where

A_t = throat area,

A_e = nozzle exit area,

k = potassium vapor specific heat ratio,

P_c = cavity pressure,

P_i = initial pressure.

The results of the calculations for several values of k are graphed in Fig. 5.15. The graphs show the pressure ratio required at various area

Fig. 5.15. Area Ratio Required for Complete Expansion in Nozzle Versus Overall Pressure Ratio for Turbine Pump.

ratios and k values for complete isentropic expansion in the nozzle. For the area ratio of the present turbine nozzle, the pressure ratio is seen to range from 3.3 to 4.6. Results from turbine operation with potassium vapor indicate the k value to be about 1.4, and thus the present turbine should operate with a pressure ratio of about 4.³⁰ The actual operating pressure ratio for the IPS in almost all cases was well below this value (see Figs. 5.12 and 5.16) and indicated that the cavity pressure was above that required for full expansion. The result was that the nozzle tended to "overexpand" the vapor and a pressure shock occurred downstream of the throat with a pressure rise to the final cavity pressure, as suggested by Fig. 5.14 (see Ref. 31). Finding the exact position of the shock front and determining its effects are complicated by the facts that almost certainly the shock front is not normal but oblique to the nozzle axis and that the ideal gas equations, the expansion exponent, and assumptions of isentropic expansion are all only approximations. The results, however, indicate that a shock does occur in the divergent section of the nozzle. If a pressure shock were to pass across the turbine bucket as the system power was changed, an effect on turbine speed should be evident. This seems unlikely from the values of the pressure ratio data. The nozzle diffuser specified for the subsequent Mark II turbine units has only a 1° divergence.

Fig. 5.16. Turbine Pressure Ratio Versus Speed for Several Tests of Run 4.

An attempt to relate the speed change with the operating turbine pressure ratio was made. Figure 5.16 shows a graph of data points taken for different tests in which the turbine pressure ratio (1) remained constant, (2) increased, and (3) decreased for a speed change in a given direction. The data indicate no clear relationship of pressure ratio to speed change. In those cases where the system remained in cavitation control and the pressure ratio remained critical, the pressure ratio did not reflect an effect during the speed change (see Fig. 5.12 and data of August 9, 1966 on Fig. 5.16).

The lack of evidence of a speed effect due to pressure shocks in the nozzle or due to turbine pressure ratio changes led to further analysis of the data for those tests during which a "speed break" occurred. This analysis revealed that in most cases the turbine speed remained high when the pressure difference between the turbine cavity and pump suction was high and fell off when the pressure difference was low, as shown by the data points in Fig. 5.17. Note the steep speed change over a narrow change of pressure difference. A composite plot of run 4 data on this basis is shown in Fig. 5.18. The general trend of speed with the difference in turbine cavity-to-pump suction pressures is indicated by the

Fig. 5.17. Typical Speed Change as a Function of Pump Suction-to-Turbine Cavity Pressure Difference.

Day	Time	Power
♦	8/9 8 08-20 39	Decreasing
○	8/19 9 49-15 04	Decreasing
□	8/22 8 23-9 31	Increasing
✱	8/23 11 43-13 24	Decreasing
✱	8/23 14 31-17 08	Increasing
■	8/25 12 13-13 20	Decreasing
◆	8/25 13 20-15 56	Constant, rising heat sink temperature*
◇	8/26 10 15-11 23	Constant, throttled HV-2 and opened again
◇	8/26 8 58&9 11	Constant, only air flow varied
●	8/31 8 28&9 30	Raised P_{cav} until P_r went way down
△	8/31 10 03-11 18	Decreasing
▲	8/31 12 15-13 41	Constant, opening HV-2
●	9/13 8 15-15 30	Constant, HV-2 throttled, pump coming out of cavitation, P below 20, high pump discharge

*For clarity, not all points included

Fig. 5.18. Composite Data on Effect of Turbine Cavity Pressure on Speed in Run 4.

dashed curves. The speed change normally occurred at a pressure difference range of 1.1 to 1.4 psi. The effect of an increase in this value is to increase the thrust load on the turbine-end thrust bearing and unload the pump-end thrust bearing (see Fig. 3.10). This suggested that an uneven bearing drag may have occurred on the two thrust bearing faces. In no case did a speed break occur when the pressure differential remained below 1.0 (as in almost all of run 2) or above 1.5.

To determine whether this pressure differential influenced the pump speed, the system was operated at a power level of 138 kw with a pump speed of approximately 4800 rpm. With the power level held constant, the downstream turbine valve (HV2) was partially closed to increase the turbine cavity pressure. No other action was taken, and as seen in Table 5.1 (data of August 26) the turbine speed increased about 27%. Opening the downstream valve then resulted in a reduction in pump speed. These results seemed to confirm that the difference in cavity-to-pump suction pressure influenced the speed. This difference could result from a difference in drag forces between the two bearing thrust faces.

There was, however, evidence from the data that other factors influenced the occurrence of the speed change. Such factors include the pump discharge pressure, the operating pressure ratio, and the system inventory distribution. The data of Table 5.1 and Fig. 5.16 (data of August 26, 10:15-11:23) show the speed increasing with a decreasing turbine pressure ratio, but in this case the pressure ratio remained above the critical value (2.0 for potassium vapor). From the data of August 31 on Fig. 5.16 and in Table 5.1, the pump speed is seen to decrease with throttling of the downstream valve. In this case, however, the turbine nozzle pressure ratio becomes subcritical and throttling of the downstream valve then effectively reduces the vapor flow through the turbine and causes a speed decrease. In addition, for this case, as the speed dropped off with decreasing pressure ratio, the pump was caused to come out of cavitation and the additional hydraulic loading also served to slow the pump. The data of August 9, 1966 (see Figs. 5.10 and 5.16) were obtained by changing only boiler power, and they show an almost constant turbine pressure ratio, which indicates that a change in turbine pressure ratio need not accompany

Table 5.1. Effects of Throttling of Downstream Turbine Valve HV2

	August 26, 1966				August 31, 1966	
Time	10:15	10:23	10:33	10:48	08:28	09:30
Position of downstream valve HV2	Open	Partially closed	Partially closed	Open	Open	Partially closed
Boiler power, kw	138	138	138	137	150	148
Turbine speed, rpm	4811	6123	6247	5029	7072	4537
Boiler outlet pressure, psia	11.89	11.83	11.83	11.79	12.78	12.62
Turbine inlet pressure, psia	3.612	3.622	3.603	3.576	3.943	3.943
Pressure ratio at upstream valve	3.292	3.266	3.283	3.297	3.241	3.201
Turbine cavity pressure, psia	1.439	1.674	1.688	1.420	1.492	2.268
Turbine pump suction pressure, psia	0.224	0.224	0.224	0.225	0.238	0.234
Turbine cavity less pump suction pressure, psi	1.215	1.450	1.464	1.195	1.254	2.034
Turbine pump discharge pressure, psia	14.5	14.3	14.3	14.8	15.8	16.2
Radiator inlet pressure, psia	0.497	0.434	0.459	0.449	0.610	0.588
Turbine pressure drop, psi	2.17	1.95	1.92	2.16	2.45	1.67
Pressure ratio at downstream valve	2.895	3.857	3.678	3.163	2.446	3.857
Bearing flow, gpm	0.38	0.34	0.33	0.38	0.35	0.40
Total turbine pump flow, gpm	2.49	2.48	2.42	2.57	2.60	2.74
Turbine pressure ratio	2.51	2.16	2.13	2.52	2.64	1.74
NPSH	0.32	0.33	0.33	0.32	0.37	0.85

a speed change. In most cases during normal system operation, the pressure ratios across the vapor valves were critical.

The data indicate that when the pump remains in cavitation and the turbine pressure ratio remains above the critical value, a speed change can be effected by a change in the lateral loading on the thrust bearings. This suggests that the bearing drag forces on the pump-end thrust face may be higher than those on the turbine-end thrust face. Hydraulic loading of the pump and/or subcritical turbine nozzle operation tends to mask the previous effect and, indeed, may reverse the speed change.

The presence of the speed change during operation did not prevent operation of the system over a wide range of power without external controls on the system. Figure 5.19 presents data covering operation of the system from 80 to 335 kw without external system control. The data show the characteristic speed change between 130 and 150 kw seen in almost all data of run 4. Because of the abrupt nature of the change, the speed and cavitation coefficient curves are drawn discontinuously at the appropriate power interval. Below a power of approximately 85 kw the speed curve is seen to fall off rapidly with a power decrease. Reference to the cavitation coefficient indicates that the pump is coming out of cavitation. The boiler outlet and pump discharge curves show that the pump discharge pressure is falling below the boiler pressure, and thus the boiler feed flow will not be sustained at the low power levels. Note that at power levels above 150 kw the pump speed is relatively insensitive to power, and a break is shown in the speed curve. Also shown on the graph is a plot of the cavity-to-pump suction pressure differential. At a differential pressure value just above 1.0 the transition of pump speed occurs. Similar differential pressure measurements had been made in a previous test system.²⁶

Because of the program cancellation no postoperative examination of the turbine pump was conducted, so the bearings were not examined. However, if the difficulty was uneven bearing drag forces, it should be easily correctable.

Fig. 5.19. System Parameters During Maximum Cavitation Control Power Change in Run 4.

Response of System to Step Changes

The response of the single-loop Rankine-cycle system to step changes in power was previously examined in a test loop operating with water.⁶ Results of those tests indicated that step changes of 25% in power could be successfully accommodated.

The IPS was accordingly subjected to step changes in power of approximately 25%. Data on system response were recorded by placing the DEXTIR on a continuous mode so that it was possible to obtain a complete sweep of all system data every 9 sec. Figure 5.20 presents a graph of the data indicated for a step power increase, and Fig. 5.21 gives similar

Fig. 5.20. Transient Response of System to Step Increase in Power.

Fig. 5.21. Transient Response of System to Step Decrease in Power.

data for a 25% step power decrease. The data of Fig. 5.20 indicate that the response time for system parameters is generally less than 20 sec. Extremely rapid response is difficult to determine because of time lags in the instrumentation, the 9-sec cycle time for the data-gathering system, and the fact that normal parameter variation may mask the initial change occurring in the parameter. The data of Fig. 5.21 permit an evaluation of the zero position prior to the step change. Small changes in system parameters where absolute values are also small, such as the pump suction and radiator inlet pressures, show no clear initiation nor end of the transient. The degree of fluctuation of a parameter prior to

the step change also influenced the apparent lag time for the response. Thus pump flow and pump speed under equilibrium conditions show a small fluctuation; this fluctuation masks the initial response by one data cycle (9 sec). An example of the effect of the 9-sec data-gathering cycle time can be seen by a comparison of Figs. 5.20 and 5.21, which indicate lag times on the pump speed of 16 and 2 sec, respectively. This difference of 14 sec in lag time can be accounted for in part by speed fluctuations that occur during the data cycle time and by the difference in elapsed time between the power change and the time of the data recording. The data of Fig. 5.21 indicate time constants to be shortest for the pump speed parameter (approximately 10 sec after initiation of the pump change), while pump suction and radiator inlet values have not yet reached equilibrium $2\frac{1}{2}$ min after the perturbation.

It may be concluded that the system demonstrated a high degree of stability in response to the step changes in power.

6. CONCLUSIONS

The successful operation of the intermediate potassium system answered the questions on the feasibility of operation and control of a single-loop Rankine-cycle system using potassium as a working fluid. Information obtained during the extensive operation of the facility will make substantial contributions to the design and operation of still larger facilities associated with the Rankine-cycle proposed for the MPRE. The most important of the findings are summarized below.

1. Control of system inventory distribution by pump cavitation was effective over a wide range of power. No external controls were required under conditions of changing power. The system operated many hundreds of hours completely unattended.

2. The system will remain under cavitation control within a power interval in which the cavitation coefficient is less than one.

3. The power interval for cavitation control can be adjusted by the design values selected for pump speed and upstream turbine pressure. The design of the turbine-driven centrifugal pump might be tailored to produce a cavitation characteristic more suitable to the system requirements.

4. The system responded with good stability to step changes of approximately 25% in power level.

5. The control characteristics observed were consistent with those observed for water systems and confirmed the usefulness of testing prototypes of such systems with water.

6. A boiler core of approximately 35% of the MPRE capacity was operated to power levels per rod that were 20% above the average MPRE design power, and no indication of deficiency of boiler operation was detected.

7. The boiler operated stably, with no indication of cyclic boiling or liquid superheating within the power range studied.

8. The three hot fingers were effective in reducing the temperature at which stable boiling could be achieved during system startup conditions.

9. Meaningful pressure drop correlations could have been obtained with a boiler design that incorporated a means for accurate exit quality determination.

10. The monitoring of ground-current leakage in the heater elements was an effective means for protecting the heaters and the system against electrical failure in the heater rods. Electrical heater rods with high reliability for operation at design temperature and power levels were not available at the time of program termination.

11. The boiler system appeared clean and free of corrosion after several thousand hours of operation. This gave additional support to the hypothesis that boiling systems reduce mass transfer and corrosion. Although a small deposit was noted on the boiler jet pump nozzle, there was no detectable effect on system performance.

12. No indication of corrosion or erosion was observed on the turbine pump unit after 2577 hr of operation under cavitation conditions. There was no indication of turbine bucket erosion.

13. Examination of the potassium-lubricated bearings after run 2 revealed a small amount of wear that indicated a need for further improvements to insure long life. Improvements that had been incorporated in the turbine pump for this run included (a) lightening the turbine wheel to more equally distribute journal bearing loads, (b) reducing the hydraulic load on the thrust bearings, and (c) redirecting the lubricant flow paths.

14. During a 2577-hr run there was no detectable change in either turbine or pump performance. During 4400 hr of boiling operation there was no detectable change in system behavior due to mass transfer or corrosion.

15. The technique of using infrared-sensitive film for a qualitative determination of the radiator temperature was used successfully. It permitted determinations of the liquid distribution in the radiator and feed lines that could be made quickly and were inexpensive and effective. A quantitative measure of liquid-to-vapor ratios in the feeder lines may be accomplished with density-sensitive detectors.

16. The infrared technique indicated no accumulation of liquid in the radiator tubes when the system was operating under cavitation control.

17. The system startup technique that allowed the pump to operate within a closed loop prior to feeding the boiler worked well. Switchover from the closed loop to the boiler feed circuit was accomplished smoothly.

18. The measurement of low absolute pressures and low pressure drop values proved difficult. Careful design considerations for pressure measurements in vapor lines subject to condensation are required to limit excessive recirculation through vapor condensation and thermal circulation.

19. The instruments developed for continuous liquid-level measurement in the boiler expansion tank worked very well.

20. The automatic data-acquisition and data-reduction program employed for the first time on a large liquid metal facility at ORNL worked exceptionally well.

21. Turbine performance and turbine pump efficiency data could not be obtained in detail on the IPS. A facility designed for this purpose would be necessary.

Postoperation examinations of the system piping, the boiler rods, and particularly the system pumps and major components have not been conducted because of termination of the program. Such examination could provide additional information on materials compatibility and component reliability.

A review of the data and operating experience from the IPS indicates that more specialized facilities and tests for certain of the system components are needed to obtain design information for future systems. These facilities should be designed primarily to obtain data on the specific components. Tests designed to carefully control flow and inlet conditions to tapered tubes and to accurately measure pressure drop and vapor qualities would permit a more closely defined radiator design. Similarly, tests designed specifically to obtain more detailed information on the turbine and pump operation in potassium are required.

Early recognition of the necessity for more detailed information on system components resulted in the design and construction of some new facilities and the modification of existing facilities to perform the more specialized tasks. A facility for conducting tests on jet pumps operating in potassium was constructed.²⁶ Similarly, a facility for calibrating potassium electromagnetic flowmeters was built.³² Modifications were made to an existing facility to permit more detailed study of the turbine pump operation, and another facility was modified to permit a more intensive study of boiling and critical heat flux in the core. The need for these and other studies was confirmed by the IPS experience.

The primary objectives initially outlined for the facility were accomplished. Some questions concerning component and system behavior and system control remain to be answered.

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APPENDICES

Blank

Appendix A
SYSTEM FLOW DIAGRAMS

Blank

LEGEND

X = LINE SIZE
Y = SCHEDULE OR TUBING (T)

W	P/sec
P	PSIA
T	°F

FLOW DATA ON BASIS OF 4KW/200

PARTS LIST

PART NO.	DWG. NO.	NO. REQ.	DESCRIPTION (NAME, SIZE, ETC.)	MATERIAL
1	207	1	275	
2	208	2	276	
3	228	3	278	
4	211	4	276	
5	209	5	206	
6	212	6	255	
7	237	7	250	
8	214	8	209	
9	209	9	201	
10	204	10	208	
11	208	11	202	
12	255	12	255	
13	256	13	256	
14	261	14	261	
15	266	15	266	
16	270	16	270	

PT. NO.	TE. NO.
1	205C18
2	203018
3	203A18
4	211
5	209
6	212
7	237
8	214
9	209
10	204
11	208
12	255

PT. NO.	TE. NO.
1	207
2	208
3	228
4	211
5	209
6	212
7	237
8	214
9	209
10	204
11	208
12	255

PT. NO.	TE. NO.
1	275
2	276
3	278
4	276
5	206
6	255
7	250
8	209
9	201
10	208
11	202
12	255
13	256
14	261
15	266
16	270

PT. NO.	TE. NO.	PT. NO.	TE. NO.	PT. NO.	TE. NO.
1	210	15	245	29	259
2	240	16	246	30	260
3	241	17	249	31	261
4	242	18	250	32	262
5	243	19	251	33	263
6	244	20	252	34	264
7	245	21	253	35	265
8	246	22	254	36	266
9	247	23	255	37	267
10	248	24	256	38	268
11	249	25	257	39	269
12	250	26	258	40	270

PT. NO.	TE. NO.
1	225
2	255A
3	238
4	253C
5	254A
6	255B
7	256C
8	257A
9	258A
10	259A
11	260A
12	261A
13	262A
14	263A
15	264A
16	265A

PT. NO.	TE. NO.
1	2323
2	202A
3	202B
4	202C
5	202D
6	202E
7	202F
8	202G
9	202H
10	202I
11	202J
12	202K
13	202L
14	202M
15	202N
16	202O

PT. NO.	TE. NO.
1	204
2	202A
3	202B
4	202C
5	202D
6	202E
7	202F
8	202G
9	202H
10	202I
11	202J
12	202K
13	202L
14	202M
15	202N
16	202O

PT. NO.	TE. NO.
1	204
2	202A
3	202B
4	202C
5	202D
6	202E
7	202F
8	202G
9	202H
10	202I
11	202J
12	202K
13	202L
14	202M
15	202N
16	202O

PT. NO.	TE. NO.
1	204
2	202A
3	202B
4	202C
5	202D
6	202E
7	202F
8	202G
9	202H
10	202I
11	202J
12	202K
13	202L
14	202M
15	202N
16	202O

PT. NO.	TE. NO.
1	215
2	216
3	217
4	218
5	219
6	220
7	221
8	222
9	223
10	224
11	225
12	226
13	227
14	228
15	229
16	230

PT. NO.	TE. NO.
1	225E
2	255A
3	238
4	253C
5	254A
6	255B
7	256C
8	257A
9	258A
10	259A
11	260A
12	261A
13	262A
14	263A
15	264A
16	265A
17	266A
18	267A
19	268A
20	269A

PT. NO.	TE. NO.
1	202C
2	202A
3	202B
4	202C
5	202D
6	202E
7	202F
8	202G
9	202H
10	202I
11	202J
12	202K
13	202L
14	202M
15	202N
16	202O

PT. NO.	TE. NO.
1	2045
2	2046
3	2047
4	2048
5	2049
6	2050
7	2051
8	2052
9	2053
10	2054
11	2055
12	2056
13	2057
14	2058
15	2059
16	2060

PT. NO.	TE. NO.
1	2045
2	2046
3	2047
4	2048
5	2049
6	2050
7	2051
8	2052
9	2053
10	2054
11	2055
12	2056
13	2057
14	2058
15	2059
16	2060

PT. NO.	TE. NO.
1	2045
2	2046
3	2047
4	2048
5	2049
6	2050
7	2051
8	2052
9	2053
10	2054
11	2055
12	2056
13	2057
14	2058
15	2059
16	2060

PT. NO.	TE. NO.
1	2045
2	2046
3	2047
4	2048
5	2049
6	2050
7	2051
8	2052
9	2053
10	2054
11	2055
12	2056
13	2057
14	2058
15	2059
16	2060

PT. NO.	TE. NO.
1	72
2	73
3	74

NOTE
ALL THERMOCOUPLES LOCATED WITHIN RADIATOR AREA AND IDENTIFIED WITHOUT INSTRUMENT NUMBERS ARE INDICATED ON T1-1003 (EXAMPLE - POINT 24) SEE TABLE ABOVE

SEE COMPANION DWG. N2 E-48907

THIS DRAWING SUPERSEDES D-46528

UNITS ON DIMENSIONS UNLESS OTHERWISE SPECIFIED:
FRACTIONS: 1/16, 1/8, 1/4, 3/8, 1/2, 5/8, 3/4, 7/8, 1, 1 1/8, 1 1/4, 1 1/2, 1 3/4, 2, 2 1/8, 2 1/4, 2 1/2, 2 3/4, 3, 3 1/8, 3 1/4, 3 1/2, 3 3/4, 4, 4 1/8, 4 1/4, 4 1/2, 4 3/4, 5, 5 1/8, 5 1/4, 5 1/2, 5 3/4, 6, 6 1/8, 6 1/4, 6 1/2, 6 3/4, 7, 7 1/8, 7 1/4, 7 1/2, 7 3/4, 8, 8 1/8, 8 1/4, 8 1/2, 8 3/4, 9, 9 1/8, 9 1/4, 9 1/2, 9 3/4, 10, 10 1/8, 10 1/4, 10 1/2, 10 3/4, 11, 11 1/8, 11 1/4, 11 1/2, 11 3/4, 12, 12 1/8, 12 1/4, 12 1/2, 12 3/4, 13, 13 1/8, 13 1/4, 13 1/2, 13 3/4, 14, 14 1/8, 14 1/4, 14 1/2, 14 3/4, 15, 15 1/8, 15 1/4, 15 1/2, 15 3/4, 16, 16 1/8, 16 1/4, 16 1/2, 16 3/4, 17, 17 1/8, 17 1/4, 17 1/2, 17 3/4, 18, 18 1/8, 18 1/4, 18 1/2, 18 3/4, 19, 19 1/8, 19 1/4, 19 1/2, 19 3/4, 20, 20 1/8, 20 1/4, 20 1/2, 20 3/4, 21, 21 1/8, 21 1/4, 21 1/2, 21 3/4, 22, 22 1/8, 22 1/4, 22 1/2, 22 3/4, 23, 23 1/8, 23 1/4, 23 1/2, 23 3/4, 24, 24 1/8, 24 1/4, 24 1/2, 24 3/4, 25, 25 1/8, 25 1/4, 25 1/2, 25 3/4, 26, 26 1/8, 26 1/4, 26 1/2, 26 3/4, 27, 27 1/8, 27 1/4, 27 1/2, 27 3/4, 28, 28 1/8, 28 1/4, 28 1/2, 28 3/4, 29, 29 1/8, 29 1/4, 29 1/2, 29 3/4, 30, 30 1/8, 30 1/4, 30 1/2, 30 3/4, 31, 31 1/8, 31 1/4, 31 1/2, 31 3/4, 32, 32 1/8, 32 1/4, 32 1/2, 32 3/4, 33, 33 1/8, 33 1/4, 33 1/2, 33 3/4, 34, 34 1/8, 34 1/4, 34 1/2, 34 3/4, 35, 35 1/8, 35 1/4, 35 1/2, 35 3/4, 36, 36 1/8, 36 1/4, 36 1/2, 36 3/4, 37, 37 1/8, 37 1/4, 37 1/2, 37 3/4, 38, 38 1/8, 38 1/4, 38 1/2, 38 3/4, 39, 39 1/8, 39 1/4, 39 1/2, 39 3/4, 40, 40 1/8, 40 1/4, 40 1/2, 40 3/4, 41, 41 1/8, 41 1/4, 41 1/2, 41 3/4, 42, 42 1/8, 42 1/4, 42 1/2, 42 3/4, 43, 43 1/8, 43 1/4, 43 1/2, 43 3/4, 44, 44 1/8, 44 1/4, 44 1/2, 44 3/4, 45, 45 1/8, 45 1/4, 45 1/2, 45 3/4, 46, 46 1/8, 46 1/4, 46 1/2, 46 3/4, 47, 47 1/8, 47 1/4, 47 1/2, 47 3/4, 48, 48 1/8, 48 1/4, 48 1/2, 48 3/4, 49, 49 1/8, 49 1/4, 49 1/2, 49 3/4, 50, 50 1/8, 50 1/4, 50 1/2, 50 3/4, 51, 51 1/8, 51 1/4, 51 1/2, 51 3/4, 52, 52 1/8, 52 1/4, 52 1/2, 52 3/4, 53, 53 1/8, 53 1/4, 53 1/2, 53 3/4, 54, 54 1/8, 54 1/4, 54 1/2, 54 3/4, 55, 55 1/8, 55 1/4, 55 1/2, 55 3/4, 56, 56 1/8, 56 1/4, 56 1/2, 56 3/4, 57, 57 1/8, 57 1/4, 57 1/2, 57 3/4, 58, 58 1/8, 58 1/4, 58 1/2, 58 3/4, 59, 59 1/8, 59 1/4, 59 1/2, 59 3/4, 60, 60 1/8, 60 1/4, 60 1/2, 60 3/4, 61, 61 1/8, 61 1/4, 61 1/2, 61 3/4, 62, 62 1/8, 62 1/4, 62 1/2, 62 3/4, 63, 63 1/8, 63 1/4, 63 1/2, 63 3/4, 64, 64 1/8, 64 1/4, 64 1/2, 64 3/4, 65, 65 1/8, 65 1/4, 65 1/2, 65 3/4, 66, 66 1/8, 66 1/4, 66 1/2, 66 3/4, 67, 67 1/8, 67 1/4, 67 1/2, 67 3/4, 68, 68 1/8, 68 1/4, 68 1/2, 68 3/4, 69, 69 1/8, 69 1/4, 69 1/2, 69 3/4, 70, 70 1/8, 70 1/4, 70 1/2, 70 3/4, 71, 71 1/8, 71 1/4, 71 1/2, 71 3/4, 72, 72 1/8, 72 1/4, 72 1/2, 72 3/4, 73, 73 1/8, 73 1/4, 73 1/2, 73 3/4, 74, 74 1/8, 74 1/4, 74 1/2, 74 3/4, 75, 75 1/8, 75 1/4, 75 1/2, 75 3/4, 76, 76 1/8, 76 1/4, 76 1/2, 76 3/4, 77, 77 1/8, 77 1/4, 77 1/2, 77 3/4, 78, 78 1/8, 78 1/4, 78 1/2, 78 3/4, 79, 79 1/8, 79 1/4, 79 1/2, 79 3/4, 80, 80 1/8, 80 1/4, 80 1/2, 80 3/4, 81, 81 1/8, 81 1/4, 81 1/2, 81 3/4, 82, 82 1/8, 82 1/4, 82 1/2, 82 3/4, 83, 83 1/8, 83 1/4, 83 1/2, 83 3/4, 84, 84 1/8, 84 1/4, 84 1/2, 84 3/4, 85, 85 1/8, 85 1/4, 85 1/2, 85 3/4, 86, 86 1/8, 86 1/4, 86 1/2, 86 3/4, 87, 87 1/8, 87 1/4, 87 1/2, 87 3/4, 88, 88 1/8, 88 1/4, 88 1/2, 88 3/4, 89, 89 1/8, 89 1/4, 89 1/2, 89 3/4, 90, 90 1/8, 90 1/4, 90 1/2, 90 3/4, 91, 91 1/8, 91 1/4, 91 1/2, 91 3/4, 92, 92 1/8, 92 1/4, 92 1/2, 92 3/4, 93, 93 1/8, 93 1/4, 93 1/2, 93 3/4, 94, 94 1/8, 94 1/4, 94 1/2, 94 3/4, 95, 95 1/8, 95 1/4, 95 1/2, 95 3/4, 96, 96 1/8, 96 1/4, 96 1/2, 96 3/4, 97, 97 1/8, 97 1/4, 97 1/2, 97 3/4, 98, 98 1/8, 98 1/4, 98 1/2, 98 3/4, 99, 99 1/8, 99 1/4, 99 1/2, 99 3/4, 100, 100 1/8, 100 1/4, 100 1/2, 100 3/4, 101, 101 1/8, 101 1/4, 101 1/2, 101 3/4, 102, 102 1/8, 102 1/4, 102 1/2, 102 3/4, 103, 103 1/8, 103 1/4, 103 1/2, 103 3/4, 104, 104 1/8, 104 1/4, 104 1/2, 104 3/4, 105, 105 1/8, 105 1/4, 105 1/2, 105 3/4, 106, 106 1/8, 106 1/4, 106 1/2, 106 3/4, 107, 107 1/8, 107 1/4, 107 1/2, 107 3/4, 108, 108 1/8, 108 1/4, 108 1/2, 108 3/4, 109, 109 1/8, 109 1/4, 109 1/2, 109 3/4, 110, 110 1/8, 110 1/4, 110 1/2, 110 3/4, 111, 111 1/8, 111 1/4, 111 1/2, 111 3/4, 112, 112 1/8, 112 1/4, 112 1/2, 112 3/4, 113, 113 1/8, 113 1/4, 113 1/2, 113 3/4, 114, 114 1/8, 114 1/4, 114 1/2, 114 3/

Appendix B

CORE PRESSURE DROP CALCULATIONS

Pressure drop equations for homogeneous flow through the boiler core were based on a modified version of a correlation proposed by Owens.⁸ The calculations computed the pressure loss separately for the frictional component, acceleration losses, and elevation changes, as indicated by the equations below.

For friction loss,

$$\Delta P_f = -F_L \frac{L}{Bx_e} \left[\left(1 - \frac{A}{B}\right) \ln (1 + Bx_e) + Ax_e \right] ; \quad (1)$$

for acceleration pressure drop,

$$\Delta P_a = \frac{AC}{B} \ln (1 + Bx_e) ; \quad (2)$$

and for elevation drop,

$$\Delta P_e = \frac{\sin \theta}{v_L} \frac{L}{(A - B)x_e} \ln \frac{1 + Ax_e}{1 + Bx_e} . \quad (3)$$

In these equations,

$$-F_L = \frac{FG^2 v_L}{2g} ,$$

$$A = \frac{v_G}{v_L} - 1 ,$$

$$B = \frac{dv_G}{dP} \left(\frac{G^2}{g} \right) ,$$

$$C = \frac{G^2}{g} v_L ,$$

and

f = Blasius friction factor, dimensionless,
 g = gravitational acceleration, $4.17 \times 10^8 \text{ ft/hr}^2$,
 G = mass velocity, $\text{ft/hr} \cdot \text{ft}^2$,
 L = overall length, ft,
 P = pressure, lb/ft^2 ,
 v = specific volume, ft^3/lb ,
 x = quality, dimensionless,
 θ = angle of inclination of flow passage, deg.

The subscripts are

e = exit,
 G = gas,
 L = liquid.

The equations were initially applied to correlate data taken on the intermediate water system.² This work indicated that the calculated estimates of core pressure drop were above the values being measured. Because the initial estimates were based on homogeneous flow through the core (a slip ratio of one), expressions were derived that included an effect due to slip.¹⁰ The data from the intermediate water system were then used in conjunction with the correlating equations to determine the slip ratio required to match the measured pressure drop. A relationship was found between the slip ratio determined and the density ratio in the core. With quality measurement from the intermediate potassium system, this relationship could then be used to compute slip and pressure drop and to determine whether the measured pressure drop agreed with the calculations. The method is covered more fully in the referenced reports.^{2,10}

Appendix C

SYSTEM VALVES

The valves used in the intermediate potassium system are described below and Fig. C.1 presents photographs of them.

Figure C.1(a) shows a 2 1/2-in. bellows-sealed valve manufactured by the Mason-Neilan Company of Norwood, Massachusetts. The valve is capable of operation at 1540°F and 25 psig. Two of these are used in the IPS; one controls bypass vapor flow around the turbine, and the other permits vapor throttling downstream of the turbine.

Figure C.1(b) shows a 1/2-in. bellows-sealed valve, Model HY-473, Mod. 57, manufactured by Hoke, Inc. of Cresskill, New Jersey. The valve is designed for a 50-psi pressure at 1600°F. These valves are used as liquid throttle and shutoff valves.

Figure C.1(c) shows a 1/2-in. bellows-sealed valve, Model HY-474, Mod. 13, manufactured by Hoke, Inc., for 50-psi 1600°F operation. These valves are used as liquid throttle valves in locations where a low pressure drop across the valve is desired.

Figure C.1(d) shows a 3/8-in. bellows-sealed valve, Model TY-442, manufactured by Hoke, Inc., for 1200°F operation. These valves are used in the sampling system and as shutoff valves for instrument lines.

Figure C.1(e) shows a 3/8-in. bellows-sealed valve, Model TY-445, made by Hoke, Inc., for 600°F service. The valve is used primarily for sample container service.

Figure C.1(f) shows a 1 1/2-in. brass, bellows-sealed, teflon-seated valve, Model R-150S, manufactured by VEECO. It is used in the vacuum system for low-temperature service only.

(a)

(b)

(c)

(d)

(e)

(f)

Fig. C.1. Valves Used in Intermediate Potassium System.

Appendix D

TRANSIENT-FREE POWER CONTROL WITH SILICON-CONTROLLED
RECTIFIERS (SCR) AS USED IN THE INTERMEDIATE
POTASSIUM SYSTEM*

Electrical transients have long plagued the electrical industry, and the periodic transients associated with power control devices have been the worst offenders. By definition, periodic transients are those frequencies other than 60 cycles that occur as a result of the operation of the power control device. Frequencies as high as 40 megacycles have been detected. In general, power control devices can be classified as either digital (ON-OFF) or analog (variable), and either type can cause severe electrical transients.

With digital control contactors, relays or circuit breakers are normally used. In addition, thyratrons, ignitrons, and silicon-controlled rectifiers (SCR's) can be used. In each case, when the device is actuated to energize the circuit, there is no recognition of zero voltage, and transient currents of large values can flow due to application of stepped voltage. These transients cause interference with other equipment unless expensive, and sometimes cumbersome, filter equipment is used. While thyratrons, ignitrons, and SCR's normally open the circuit only at current zero, this is not true of other devices used where large transients can be generated on circuit interruption, especially on inductive or capacitive circuits.

With analog control, variable transformers or saturable reactors are normally used. Variable transformers are not harmonic or transient generators and consequently are acceptable. However, saturable reactors, with their associated nonsinusoidal wave shapes, are harmonic generators of the first order. In some cases where the loads controlled by the saturable reactors are a significant part of the total load of the plant, power companies have threatened to drop the load or to charge higher rates because of the load-current wave shape. Other disadvantages of this type of control device exist but are not of concern here.

*This material was prepared by D. L. Clark of the Reactor Division of the Oak Ridge National Laboratory.

With zero-voltage switching of SCR's, the power circuit is energized only at that point on the cycle where the voltage wave is going through zero. Since a stepped function of voltage is not applied, there can be no transient current on resistive or inductive circuits. Even on highly capacitive circuits, the inrush current is of small consequence. Further, since the SCR units inherently cut off at current zero, there is no transient associated with opening the circuit. Power control is obtained by a series of ON and OFF voltage conditions with the ratios and duration required by the particular process. Control devices to provide the ON and OFF periods have been developed in conjunction with the actual zero-voltage switching control.

Zero-Voltage Switching

The requirements for a zero-voltage switching control (ZVS control) came about as a result of the frustration from months of chasing transients on the 450-kw silicon-controlled rectifier (SCR) installation for the intermediate water system. Transients associated with this installation ranged from 250 kc to 40 mc in frequency and up to 200 v in amplitude (230-v system). These transients affected a considerable amount of instrumentation both on the particular system involved and on other equipment in the area. Considerable time and money were expended in efforts to reduce these transients and their effects without much success. Considerable effort had also been expended in reducing transients associated with contactors, thyratrons, ignitrons, saturable reactors, and other power control devices.

The idea of eliminating the cause of the transients, rather than reducing the transients and their effects, led to the concept of an ac circuit that would (1) become energized as the voltage wave increased from the zero voltage point and (2) become deenergized as the current wave approached zero. In this way transient currents would be effectively eliminated and initial currents would be of the same magnitude as the steady-state currents. The SCR was an obvious choice with which to work because of its fast turn-on time, low losses (consequently low voltage drop), and its ability to satisfy the second requirement above.

With this control unit the method of power control used a series of ON and OFF voltage conditions of ratios and duration required by the process. In conjunction with the ZVS control development, a manual control device was developed (by others) to provide the ON and OFF periods in accordance with settings on two control potentiometers.

Early in the design stages on this type of control, the following criteria were established:

1. The control should function regardless of the phase angle of the load, including variable phase-angle loads.
2. The control should be capable of turning ON the voltage for as little as one-half cycle but should be capable of full-cycle control (multiples of full cycles) that would be required by inductive loads such as transformers under certain circumstances.
3. The control should consist of solid-state devices.
4. The control should offer full range of control from zero output to full ON.
5. The control should be capable of controlling multiple series and parallel arrangements of SCR's without requiring complex circuitry.
6. The control should be capable of being grouped with other similar controls for operation from a single master controller. Typical requirements would be three-phase circuits with one or more ZVS controls per phase.
7. The control should be capable of controlling any SCR on the market.
8. The control should be competitive with other types of gate drives for SCR's.

The circuit shown in Fig. D.1 meets all the criteria and requirements and is the circuit that was utilized. The wave shapes and phase relationships of the currents and voltages of interest are shown in Fig. D.2. The combination of resistor R1 and capacitor C1 causes the voltage across points E and F to lead the line voltage by the desired electrical degrees (44 on the present control). When the control SCR (SCR-C) is ON, the voltage across points E and F is limited to the forward voltage drop of the bridge rectifier and SCR-C combination, which is an approximately

ORNL-DWG 67-9295

Fig. D.1. Schematic Diagram of Zero-Voltage Switching Control.

ORNL-DWG 67-9296

Fig. D.2. Voltage Waves Produced by Zero-Voltage Switching Control.

3-v peak. This voltage is less than the excitation voltage required by transformer T2 and, consequently, there is no secondary voltage, and the load SCR units are not turned ON. (Gating source for SCR-C should be a 3.0-v minimum to terminals C- and C+.) When the gating source for SCR-C is removed, SCR-C will conduct for the balance of that half cycle (thus providing synchronization with the line) before going into a blocking state (see point B in Fig. D.2). With SCR-C in a blocking state, design voltage (22 v) will be impressed across the primary of transformer T2. As the primary voltage builds up from zero (point B in Fig. D.2), sufficient secondary voltage for gating SCR-1 (diode D6 will block signals of this polarity from the gate of SCR-2) will be produced in advance of the point (point C on Fig. D.2) where line voltage passes through zero. With the gate of SCR-1 thus energized, load current will flow as the cathode becomes slightly negative (less than 1.0 v by test with C-35E and C-50E SCR units).

This satisfies requirement 1 of the original concept, and requirement 2 is satisfied automatically by the utilization of SCR units for the control. On the opposite half of the cycle, operation of SCR-2 will be similar, with diode D5 blocking signals of that polarity from the gate of SCR-1. The utilization of the ac cycle for transformer T2 (rather than rectified ac), plus the subsequent polarization accomplished by diodes D5 and D6, eliminates the need for precise synchronization of gate signals with the line voltage.

Power Control

Power control is obtained by the proper series of ON and OFF power periods. For certain types of ON-OFF (digital type) controllers, this series is a function of the instrument and cannot be adapted to suit the process. For other types of ON-OFF controllers, such as the L&N Series 60, type D.A.T., the time between the start of ON periods (hereinafter called the "impulse period") can be adjusted to suit the process requirements. This impulse period is normally set at the maximum time period that does not cause detectable process changes. For one process system this may be six cycles (1/10 sec), whereas for another process system,

it may be 5 min or more. Within this impulse period the "percent ON time" is varied as required to maintain process conditions. The manual control (which is basically an unsymmetrical multivibrator) has been developed (by others) to provide variable impulse periods and variable percent ON time as determined by the settings of two potentiometers. The impulse period is adjustable from six cycles to several minutes (or longer) by adjustment of one potentiometer, and this impulse period is independent of the setting of the other potentiometer. The percent ON time is adjustable from 0 to 100% by adjustment of the second potentiometer, and its setting is determined by process requirements.

If the automatic control device to be used is an analog type (variable current), the signal from it can be used to duplicate the operation of the percent-ON-time potentiometer. The circuit can be arranged for any analog controller. As previously stated, within reason, any desired number of ZVS controls can be paralleled from any given controller.

Appendix E

INSTRUMENT TABULATION AND CALIBRATION

A tabulation of the important system instrumentation is given in Table E.1. Figure E.1a, b, and c shows typical calibration curves obtained with the system under gas pressure at elevated temperature.

Table E.1. Partial List of Intermediate Potassium System Instruments

Application	Sensor				Display		Remarks
	Manufacturer	Type	Temperature Tolerance (°F)	Range	Manufacturer	Type	
Boiler							
Outlet pressure	Taylor	NaK filled	0-1500	5-50 psia	Foxboro	ECI	
Differential pressure	ORNL	K manometer	0-1540	0-48 in. K	Foxboro	ECI	See Ref. 24
Differential pressure	Taylor	NaK filled	0-1500	5-50 in. H ₂ O	Foxboro	Pneumatic	
Level	ORNL	K manometer	0-1540	0-6 in. K	Brown	Ser 15	See Ref. 24
Boiling noise	ORNL	Carbon microphone					
Flow	ORNL	EM flowmeter	0-1500	0-3.56 gpm	Foxboro	ECI	See Ref. 24
Temperature (preheat)		Thermocouple	0-2000	0-2000°F	Brown	Ser 15	
Temperature (boiler heater)		Thermocouple	0-2000	0-2000°F	ATL	220 profile monitor	See ORNL drawing Q-2122-190
Boiler heater protection	ORNL	Insulation leakage			ATL	220 profile monitor	
Power	L&N	Thermoverter		0-455 kw	Brown	Ser 15	
Turbine pump							
Turbine discharge pressure	Foxboro	ECI	0-400	0-10 psia	Foxboro	ECI	
Turbine pressure drop	Foxboro	ECI	0-400	0-200 in. H ₂ O	Foxboro	ECI	
Pump suction pressure	Foxboro	ECI	0-400	0-10 psia	Foxboro	ECI	
Pump discharge pressure	Taylor	NaK filled	0-1500	5-100 psia	Foxboro	ECI	
Shaft speed			800	0-10,000 rpm	TSI	Counter	
					Foxboro	ECI	Electro-Products pickup
Turbine noise	ORNL	Carbon microphone					
Pump noise	ORNL	Carbon microphone					
Vibration	CEC	Y-121		45-1500 cps	CEC		
Flow to jet pumps	ORNL	EM flow	0-1500	0-4.26 gpm	Foxboro	ECI	See Ref. 23
Flow to pump bearings	ORNL	EM flow	0-1500	0-0.76 gpm	Foxboro	ECI	See Ref. 23
Temperatures		Thermocouple	0-2000	0-2000°F	Brown	Ser 15	
Radiator							
Inlet pressure	Foxboro	ECI	0-400	0-3 psia	Foxboro	ECI	
Differential pressure	ORNL	K manometer		0-48 in. K	Foxboro	ECI	See Ref. 24
Airflow	Taylor	Pitot venturi	0-1540	0-5 in. H ₂ O	Foxboro	Pneumatic	
Temperatures		Thermocouple	0-2000		Brown	Ser 15	
Fill and drain tank							
Pressure	Taylor	NaK filled	0-1500	5-50 psia	Brown	Ser 15	
Pressure	Fulton Sylphon		0-200	30 in. Hg, 0-30 psi	Local		
Level	ORNL	I tube	0-1500	0-21 in.	Brown	Ser 15	See Ref. 24
Degasser level	ORNL	I tube	0-1500	0-8.3 in.	Brown	Ser 15	See Ref. 24
Expansion tank level	ORNL	I tube	0-1500	0-13.5 in.	Brown		See Ref. 24
Catch tank level	ORNL	I tube	0-1500	0-11.5 in.	Brown		See Ref. 24
EM pump flow	ORNL	Magnet	0-1500	6.1 gpm	Foxboro	ECI	See Ref. 23
Pressure ratio, pump discharge to boiler outlet	Taylor	740	0-1500	1-3	Foxboro	ECI	Special design, see ORNL drawing Q-2122-160
Vapor quality		Thermocouple	0-2000		Taylor	Electronic recorder	Special design, see ORNL drawing Q-2122-179

Fig. E.1. Instrument Calibration Curves.

Appendix F

DATA REDUCTION

A data-reduction program was prepared to process information received from the intermediate potassium system.* Three data-gathering DEXTIR boxes, each with a capacity for 25 analog and 25 digital inputs, were installed. Table F.1 lists the data inputs to each of the boxes. Data received at the input boxes were transferred to a central data processor and stored on magnetic tape. Minor corrections, such as elevation corrections on pressure measurements, density corrections on the vapor and liquid, flow corrections, and instrument zero corrections, were incorporated on the input tape at the central processor. This information then served as the permanent log of the raw data taken on the system, and a printout of this information could be obtained at any time. This data also served as the input for the data-reduction program.

The data-reduction program computed the performance of the various system components and tabulated the output information as shown in Tables F.2 and F.3. Table F.2 is a typical printout of the raw data output; each point of each data box is printed and the time the data is taken is listed. The data printed under each box corresponds to the listing in Table F.1. The reduced data for the corresponding time (hour 11:06 for the case shown) is tabulated on a separate sheet, as shown in Table F.3.

In addition, graphs such as those in Figs. 4.41, 5.5, and F.1 were originally machine plotted from the raw data.

*This program was initially prepared by B. J. Walker and later modified by the authors.

Table F.1. IPS DEXTIR Data-Acquisition Sheet Showing Analog and Digital Box and Channel Number

Analog Point	Digital Point	Transmitter	Parameter Measured
DEXTIR Box No. 1			
1		PT-50	Boiler outlet pressure
2		PdT-60	Boiler pressure drop
3		PdT-61	Turbine pressure drop
4		PT-52	Turbine pump suction pressure
5		PT-53	Turbine pump discharge pressure
6		PT-51	Radiator inlet pressure
7		PdT-62	Radiator pressure drop
8		PdT-63	Jet pump pressure drop
9		FE-81	Boiler feed flow
10		FE-85	Bearing flow
11		FE-82	Radiator jet pump flow
12		LT-70	Separator level
13		SR-80	Turbine speed
14		XwR-1007	Total power
15		TE-211	Base of boiler temperature
16		TE-200	Boiler outlet temperature
17		TE-212S	Turbine inlet temperature
18		TE-237S	Turbine exhaust temperature
19		TE-214S	Downstream turbine exhaust valve temperature
20		TE-203S	Radiator inlet (south) temperature
21		TE-302	Radiator top manifold temperature
22		TE-299	Radiator top manifold temperature
23		TE-273	Radiator tube temperature
24		TE-274	Radiator tube temperature
25		531A	Pressure transmitter 51 riser
	1	Dial 1	Number of rods
	2	Dial 2	Cooling air inlet
DEXTIR Box No. 2			
1		TE-275	Radiator tube temperature
2		TE-276	Radiator tube temperature
3		TE-261	Radiator outlet manifold temperature
4		TE-262	Radiator outlet manifold temperature
5		TE-263	Radiator outlet manifold temperature
6		TE-264	Radiator outlet manifold temperature
7		TE-265	Radiator outlet manifold temperature
8		TE-266	Radiator outlet manifold temperature
9		TE-267	Radiator outlet manifold temperature
10		TE-268	Radiator outlet manifold temperature
11		TE-297	Radiator outlet manifold temperature
12		TE-270	Radiator outlet manifold temperature
13		TE-271	Radiator outlet manifold temperature

Table F.1 (continued)

Analog Point	Digital Point	Transmitter	Parameter Measured
14		TE-272	Radiator outlet manifold temperature
15		TE-320	South condensate feed line temperature
16		TE-321	South condensate feed line temperature
17		TE-322	South condensate feed line temperature
18		TE-323	South condensate feed line temperature
19		TE-324	South condensate feed line temperature
20		TE-204	Radiator outlet (north) temperature
21		TE-205	Radiator outlet (south) temperature
22		TE-207S	Jet pump (south) temperature
23		TE-208S	Jet pump (north) temperature
24		TE-228S	Jet pump north discharge temperature
25		564B	Pressure transmitter 610 riser
DEXTIR Box No. 3			
1		TE-229S	Jet pump south discharge temperature
2		TE-325	Jet pump discharge south temperature
3		TE-326	Jet pump discharge south temperature
4		TE-327	Jet pump discharge south temperature
5		TE-328	Jet pump discharge south temperature
6		TE-206S	Turbine pump suction temperature
7		TE-235S	Turbine pump discharge temperature
8		TE-519CS	EM pump suction temperature
9		TE-525B	EM pump discharge temperature
10		TE-224S	Liquid preheater inlet temperature
11		TE-209S	Liquid preheater outlet temperature
12		TE-288S	Vapor preheater inlet temperature
13		TE-290	Preheater vapor temperature
14		TE-291	Preheater vapor temperature
15		TE-292	Preheater vapor temperature
16		TE-293	Preheater vapor temperature
17		TE-202S	Preheater vapor outlet temperature
18		TE-303D2S	Jet pump 4 sideport temperature
19		TE-201S	Boiler jet pump inlet temperature
20		TE-303A2	Jet pump 4 discharge temperature
21		TE-316	Air leaving radiator temperature
22		PT-610	Turbine cavity pressure temperature
23		TE-213	PdT 60 temperature
24		TE-549A	Line to bottom of PdT-60 temperature
25		536A	Pressure transmitter 52 riser

Table F.2. Typical Output of Data Boxes for Intermediate Potassium System

	1134,003	1146,685	1148,378	884,209	1117,966	1101,942	1104,050	1079,177	1042,925	1032,386
	1067,373	,259	986,821	1042,503	627,062					
BOX 3										
	1031,542	1027,326	1036,180	1028,591	1028,591	1021,001	1040,817	630,943	517,010	1034,915
	1485,750	1498,987	-688,000	1392,921	1360,573	1307,261	1318,421	1509,113	1485,350	1552,780
	226,517	4,247	1459,455	1012,987	395,144					
BOX 1, DAY 256 = 9/13/66,										
10 43	25,129	,173	4,158	1,652	44,708	2,684	*3,841	,278	3,382	,671
	1,256	3,544	7243,478	326,080	1509,113	1501,628	1276,855	1169,589	1158,547	1163,640
	1163,640	1160,244	1166,188	1164,489	582,469	79,000	85,000	6063,900	6243,000	0
BOX 2										
	1162,751	1153,883	1144,146	1142,032	1144,146	320,106	1119,653	1142,878	1147,954	1141,609
	1134,425	1147,108	1148,801	884,631	1117,966	1103,629	1105,315	1080,863	1043,346	1031,964
	1066,530	,188	986,821	1042,925	627,925					
BOX 3										
	1031,542	1028,169	1036,180	1028,591	1029,434	1021,844	1041,239	630,512	514,818	1035,758
	1487,548	1498,987	-688,000	1395,514	1360,573	1307,690	1317,562	1510,434	1485,350	1552,780
	226,952	4,262	1455,953	1012,565	395,590					
BOX 1, DAY 256 = 9/13/66,										
10 48	25,274	,174	4,190	1,692	45,543	2,869	*4,236	,277	3,418	,663
	1,258	3,546	7235,444	327,360	1509,993	1501,628	1277,711	1171,290	1160,668	1165,339
	1164,914	1162,366	1167,888	1165,764	582,469	79,000	85,000	6063,900	6243,000	0
BOX 2										
	1164,914	1158,123	1146,685	1143,723	1145,838	316,951	1121,341	1144,569	1150,071	1142,032
	1134,848	1148,378	1150,071	888,001	1119,653	1104,472	1105,737	1080,441	1044,190	1033,650
	1069,481	,180	991,043	1044,190	627,925					
BOX 3										
	1033,650	1030,699	1038,288	1030,278	1031,121	1023,953	1042,925	633,529	515,695	1037,866
	1486,229	1499,427	-688,000	1395,082	1362,727	1310,693	1321,427	1510,434	1486,229	1553,665
	227,388	4,371	1459,893	1022,688	395,590					
BOX 1, DAY 256 = 9/13/66,										
11 6	25,755	,185	4,261	1,878	46,710	3,144	*4,570	,277	3,468	,635
	1,317	3,596	7211,342	335,200	1514,839	1507,792	1282,416	1177,676	1169,589	1174,695
	1173,844	1171,290	1177,250	1175,547	582,903	77,000	85,000	6063,900	6243,000	0
BOX 2										
	1173,844	1167,038	1156,002	1152,612	1155,155	319,204	1129,359	1154,307	1159,395	1151,765
	1139,455	1156,426	1159,395	890,950	1128,936	1112,905	1115,435	1089,716	1053,463	1038,288
	1079,177	,191	996,953	1051,777	628,356					
BOX 3										
	1042,503	1039,131	1047,140	1037,866	1038,288	1031,542	1050,091	639,559	522,707	1045,033
	1491,946	1505,150	-688,000	1399,406	1366,604	1314,986	1326,154	1515,720	1491,507	1558,976
	228,258	4,595	1465,591	1109,110	396,484					
BOX 1, DAY 256 = 9/13/66,										
11 15	25,878	,181	4,247	2,192	46,641	3,197	*5,431	,077	3,459	,602
	1,310	3,459	6865,880	334,400	1515,279	1507,792	1282,843	1178,102	1172,141	1176,824
	1176,358	1172,992	1179,381	1177,250	582,469	77,000	85,000	6063,900	6243,000	0
BOX 2										
	1175,972	1168,739	1157,699	1154,731	1156,850	319,204	1130,625	1156,002	1161,093	1154,307
	1129,781	1159,819	1161,517	896,427	1131,469	1115,014	1117,966	1093,089	1057,257	1040,396
	1081,285	,202	1000,330	1055,571	627,925					
BOX 3										
	1045,876	1042,082	1049,669	1040,396	1041,660	1034,494	1053,042	643,003	524,896	1048,405
	1491,067	1504,709	-688,000	1401,136	1369,621	1318,850	1329,592	1516,601	1491,507	1558,976
	229,130	4,446	1465,591	1127,670	396,484					
BOX 1, DAY 256 = 9/13/66,										

Table F.3. Typical Reduced Data Output Sheet from Intermediate Potassium System Computer Program

IPS SYSTEM, DAY 256 # 5/13/66, 11 HR, 6 MIN.			PAGE NO. 8
1. BOILER POWER, KW.	335.	9. COMPLETATIONS.	
2. PREHEATER POWER, KW.	22.	A. POWER/ROD, KW/RCC	4.35
3. TURBINE SPEED, RPM.	7211.	B. BOILER SUBCOOLING, F.	-51.2
4. SEPARATOR LEVEL, INCHES	3.4	C. QUALITY	-0.863
5. NUMBER OF RODS	77.	D. TURBINE PUMP PERFORMANCE.	
6. PRESSURES, PSIA.		1. MEASURED PSI	0.631
A. BOILER OUTLET	25.80	2. MEASURED PHI	0.637
B. TURBINE INLET	8.858	3. CALCULATED PSI (NON-CAVITATING)	0.652
C. TURBINE CAVITY	4.597	4. HEAD COEFFICIENT RATIO	0.97
D. TURBINE PUMP SUCTION	1.878	5. PUMP WORK, FT-LBS/HR.	282146.
E. TURBINE PUMP DISCHARGE	46.7	6. TURBINE PRESSURE RATIO	1.93
F. RADIATOR INLET	3.142	7. TURBINE VAPOR FLOW, LBS/SEC.	0.041
G. TURBINE DELTA P.	4.26	7B. TURBINE VAPOR FLOW (B), LBS/SEC.	0.066
H. JET PUMP DELTA P.	-0.277	8. NET POSITIVE SUCTION HEAD, FT. OF K.	1.59
I. BOILER DELTA P.	NOCALC	9. SUCTION SPECIFIC SPEED, RPM.	11885.
J. RADIATOR DELTA P.	-0.729	E. CONDENSER JET PUMP PERFORMANCE.	
K. BOILER JET PUMP SIDEPORT	25.954	1. HEAD RATIO, N	-0.006
L. RADIATOR JET PUMP SIDEPORT	2.25	2. FLOW RATIO, PHI	3.138
M1. PUMP SUCTION LINE FRICTION LOSS	0.184	F. NOZZLE SUBCOOLING, DEG. F.	
M2. PUMP SUCTION LINE PRESS. CHANGE	0.058	1. BOILER JET PUMP	192.
N. BOILER DELTA P.	-0.582	2. CONDENSER JET PUMP	660.
7. FLOWS.		G. JP SP CAV. SUPPRESSION HEAD, IN. OF K.	
A. BOILER FEED FLOW, GPM.	3.49	1. BOILER JET PUMP	8.56
B. CONDENSER JET PUMP DISCHARGE NORTH, GPM.	2.725	2. CONDENSER JET PUMP	30.80
C. RADIATOR JET PUMP FLOW, GPM.	1.32	H. MASS VEL-BOIL HEATED SECT., LB/SEC-SQ FT.	3.229
D. BEARING FLOW, GPM.	0.64	I. VAPOR VEL. AT SEPARATOR THROAT, FT/SEC.	52.63
E. SUM OF TURBINE PUMP FLOWS, GPM.	5.45	J. VAPOR VEL. ENT. RADIATOR, FT/SEC.	94.05
AA. BOILER FEED FLOW, LB/SEC.	0.317	K. HEAT FLUX, BTU/SQ FT-HR.	137580.
BB. CONDENSER J P DISCHARGE NORTH, LB/SEC.	0.255	L. TURBINE NOZZLE EXIT VAPOR VEL., FT/SEC.	1657.81
CC. RADIATOR JET PUMP FLOW, LB/SEC.	0.130		
DD. BEARING FLOW, LB/SEC.	0.063		
8. TEMPERATURES, DEG. F.		(TEMPERATURES, CCNT.)	
A. BOILER OUTLET	1508.	L. COOLING AIR CUT	228.
B. BOILER INLET	1559.	M. BOILER JET PUMP NO. 4 SIDEPORT	1516.
C. TURBINE INLET	1282.	N. TURBINE PUMP IN ET	1032.
D. TURBINE DISCHARGE	1178.	O. TURBINE PUMP DISCHARGE	1050.
E. RADIATOR OUTLET	1038.	P. PT-610 LINE	628.
F. RADIATOR INLET	1175.	Q. PT-52 LINE	396.
G. RADIATOR MANIFOLD AVG.	1059.	R. PT-51 LINE	583.
H. CONDENSATE	1038.	S. PDT-60 BODY	1466.
I. PREHEATER INLET	1045.	T. PDT-60 LINE	1109.
J. PREHEATER OUTLET	1492.	U. PDT-62 BODY	1139.
K. COOLING AIR IN	85.

Fig. F.1. Typical Calcomp Graph from Intermediate Potassium System Computer Program.

Appendix G

HISTORY OF RUNS

Table G.1. Runs 2 and 3 with Turbine Pump 2

Date	Time		Starts	Turbine Operating Time (hr)
7/27/65	09:07	Started turbine	1	
	09:10	Shutdown - high ground current		
	09:32	Started turbine	1	
7/28/65	09:05	Shutdown - high ground current		23 1/2
	09:10	Started turbine	1	
7/30/65	15:35	Shutdown system		6 1/2
	11:21	Started turbine	1	
8/2/65	10:00	Shutdown - high ground current		70 1/2
	10:22	Started turbine	2	
8/4/65	03:47	Shutdown - high ground current		41 1/2
	04:47	Started turbine	1	
8/5/65	13:00	Shutdown		32
	13:10	Started turbine	1	
8/20/65	11:04	Shutdown - spurious		358
	11:16	Started turbine	1	
	12:40	Shutdown - power outage		145 1/2
	12:55	Started turbine	2	
9/24/65	22:00	Shutdown - SCR failure		706
10/20/65	10:50	Started turbine	1	
	14:30	Shutdown - spurious		3 1/2
	14:55	Started turbine	1	
	16:36	Shutdown - spurious		1 1/2
10/21/65	13:20	Started turbine	1	
10/24/65	10:00	Shutdown - high ground current		69
	12:47	Started turbine	1	
11/12/65	06:30	Shutdown - high ground current		450
	18:30	Started turbine	3	
11/15/65	21:30	Shutdown - high ground current		3
	09:12	Started turbine	2	

Table G.1 (continued)

Date	Time		Starts	Turbine Operating Time (hr)
11/19/65	17:45	Shutdown - high ground current		104 1/2
	19:24	Started turbine	2	
11/20/65	16:30	Shutdown - high ground current		21
11/23/65	09:06	Started turbine	1	
11/24/65	22:15	Shutdown - high ground current		37
	23:35	Started turbine	3	
11/25/65	00:20	Shutdown - high ground current		1
	00:40	Started turbine	1	
11/26/65	16:35	Shutdown - high ground current		40
11/27/65	10:20	Started turbine	1	
11/29/65	10:30	Shutdown - high ground current		48
		Started turbine	1	
11/29/65	22:00	Shutdown - high ground current		11 1/2
11/30/65	09:10	Started turbine	1	
12/1/65	21:50	Shutdown - high ground current		37 1/2
12/2/65	00:00	Started turbine	1	
12/2/65	22:10	Shutdown - high ground current		22 1/2
12/3/65	08:40	Started turbine	1	
12/4/65	16:40	Shutdown - high ground current		32
12/6/65	08:26	Started turbine	1	
12/8/65	01:10	Shutdown - high ground current		41
	08:30	Started turbine	1	
12/10/65	04:37	Shutdown - high ground current		44
	08:33	Started turbine	1	
12/11/65	07:05	Shutdown - high ground current		22 1/2
12/13/65	08:24	Started turbine	1	
12/18/65	08:55	Shutdown - high ground current		120 1/2
12/20/65	08:37	Started turbine	1	
12/21/65	02:20	Shutdown - high ground current		18 1/2
	09:20	Started turbine	1	
12/21/65	20:15	Shutdown - high ground current		11
12/22/65	08:53	Started turbine	1	
12/24/65	01:57	Shutdown - high ground current		41
12/27/65	08:26	Started turbine	1	

Table G.1 (continued)

Date	Time		Starts	Turbine Operating Time (hr)
12/27/65	09:32	Shutdown - high ground current		1/2
	09:35	Started turbine	1	
12/27/65	09:39	Shutdown - high ground current		
	09:41	Started turbine	1	
	09:45	Shutdown - high ground current		
	10:00	Started turbine	1	
12/27/65	15:30	Shutdown - low separator level		5 1/2
	15:38	Started turbine	1	
	15:42	Shutdown - low separator level		
3/28/66	15:00	Started turbine	1	
	15:00	Manual shutdown - no bearing lubrication		3 sec
4/1/66	08:35	Started turbine	1	
	15:00	Manual shutdown - 18-kw maximum power available		6 1/2
		Total	46	2577
Total boiling - 2929 hr				

Table G.2. Run 4 with Turbine Pump 3

Date	Time		Starts	Turbine Operating Time (hr)
7/26/66	10:09	Started turbine	1	
	23:00	Shutdown - HV-6 leaking		13.0
7/28/66	09:57	Started turbine	1	
	13:55	Shutdown - reducing power		4.0
7/29/66	08:40	Started turbine	1	
8/3/66	11:45	Shutdown - loose wire and connections shorted		123.0
8/4/66	13:12	Started turbine	1	
8/11/66	18:45	Shutdown - power outage		173.5
8/15/66	09:50	Started turbine	1	
8/16/66	08:12	Shutdown - instrument maintenance		22.5
8/17/66	12:55	Started turbine	1	
8/24/66	01:55	Shutdown - high ground current		157.0
	09:50	Started turbine	1	
	09:56	Shutdown - valve open		
	10:05	Started turbine	1	
9/12/66	13:40	Shutdown - high ground current		459.5
	15:20	Started turbine	1	
9/15/66	14:14	Shutdown - manual		71.0
Total			9	1023.5
Total boiling - 1113 hr				

Blank

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