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630A

MARITIME NUCLEAR STEAM GENERATOR
PROGRESS REPORT NO.7

March 16, 1964

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United States Atomic Energy Commission

Contract No. AT(40-1)-2847

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1. INTRODUCTION and SUMMARY

Introduction

This is progress report No. 7 on the 630A nuclear steam generator program covering the period from November 15, 1963 to February 15, 1964.

Summary

The current status of the 630A nuclear steam generator program and highlights during this period are presented below.

1. A study indicated that the most desirable type of blower drive turbine is one using main turbine throttle steam conditions and exhausting to the main turbine cross-over line.
2. Preliminary planning for the initiation of a dynamic structural analyses of the over-all steam generator was completed.
3. External pressure loading and thermal stress calculations show that the calandria has a suitable design margin.
4. A revised fuel latch operable from the rear face of the core was designed.
5. A study was initiated to determine the feasibility of substituting Zircaloy for the stainless steel tubing within the active core.
6. Preliminary sizing of control rod extensions and gang plates was completed.
7. Initial loading of the second configuration of the 630A critical experiment reactor was completed.
8. Detailed power distributions were measured in the 11 typical positions.
9. Subcritical and critical rod worth curves were obtained in the critical experiment with up to 132 shim rods in the core.
10. Moderator temperature coefficient measurements were made and agreed well with analytical data.
11. Critical experiment correlation of fine radial power calculations in the revised mockup showed good agreement.
12. Performance specifications were prepared for a 1-mw_e power plant.
13. Parametric thermal analyses, based on various tube sheet thicknesses and heating rates, were completed for the top and bottom tube sheets.
14. The shield plug was redesigned to accommodate additional shim rods and to facilitate fabrication and moderator flow adjustments.
15. Calculations of the operating and shutdown dose at the nuclear sensor location was started.
16. Studies were performed to determine the size of the port openings needed to prevent buckling in the containment vessel in the event of ship sinking.
17. A blower shaft static seal was built and tested with satisfactory results.

18. Manufacture and procurement of all parts for two developmental dynamic rod actuators were completed.
19. The two-tube boiler was modified and operated satisfactorily on automatic control down to 20 percent power.
20. A reactor control system that meets response and accuracy requirements was simulated on the analog computer.
21. Investigation of a possible test site for power testing the prototype reactor was continued with cost estimates and a facility layout drawing being made for the IET facility at ITS.
22. Specimen ORF-4 was removed from test after 4800 hours of actual operation at 1300°F, equivalent to approximately 15,000 hours of full-power 630A operation. An estimated 40 percent burnup of the U^{235} atoms occurred.
23. Creep tests on warm-finished fuel sheet have accumulated more than 12,000 hours at 630A stress and temperature conditions with no greater than 0.75 percent plastic creep. Cladding-stock creep tests have surpassed 15,000 hours at various combinations of stress and temperature with satisfactory results.

2. PROPULSION SYSTEM INVESTIGATION

A study indicated that the most desirable type of blower drive turbine is one using main turbine throttle steam conditions and exhausting to the main turbine cross-over line.

A 630A duty cycle was assumed to establish a logical number of temperature cycles to be expected during the core life.

Performance specifications were prepared for a 1-mw_e power plant.

Preliminary planning of a dynamic structural analysis of the 630A was completed.

THERMODYNAMIC CYCLE

The heat balance diagram of a 27,300 SHP marine power plant with a maximum power rating of 30,000 SHP (Figure 2.1) has appeared in previous progress reports and is repeated here for reference purposes. For 27,300 SHP output, a reactor total power of 67.4 mw_t is required. The overall thermal efficiency is 30.2 percent.

The propulsion system consists of a conventional cross-compound steam turbine and associated auxiliaries including low-pressure steam generator, evaporator, auxiliary turbine generator, and desuperheaters. The feedwater heating cycle consists of demineralizers, a moderator heat exchanger, boiler feed pump, and three regenerative feedwater heaters.

The thermodynamic cycle is a nonreheat, regenerative cycle utilizing modern steam equipment conditions of 850 psig and 950°F temperature. The nuclear steam generator is, however, wholly suitable for extant plants designed for reduced steam conditions.

630A ASSEMBLY

The present concept (Figures 2.2 and 2.3) of the 630A is presented in detail in the fifth progress report.* A list of significant design point operating conditions is presented in Table 2.1.

TABLE 2.1

DESIGN POINT OPERATING CONDITIONS ^a	
Reactor power, mw	67.4
Air flow rate, lb/sec	352
Air pressure at blower exit, psia	400
Air temperature at reactor exit, °F	1200
Air temperature at boiler exit, °F	550
Blower power, BHP	2600
Fuel element maximum average temperature, °F	1335

^aThese data are consistent with the heat balance for 27,300 SHP shown in Figure 2.1.

*"630A Maritime Nuclear Steam Generator Progress Report No. 5," CE-NMPO, CEMP-215, June 28, 1963.

Fig. 2.1—The 27,300-SHP marine power plant heat balance at normal sea conditions

Fig. 2.2—Artist's concept of the 630A Nuclear Steam Generator (AS-201)

Fig. 2.3 - 630A Nuclear Steam Generator (Dwg. 219R996)

Fig. 2.3 - (Continued)

SECTION D-D

SECTION E-E

SECTION F-F

GE-NMPO

Fig. 2.3 - (Continued)

Optimization Study of Steam Turbine Auxiliary Drives

A study of the cost and performance characteristics of several possible steam turbine auxiliary drives for the secondary loop was essentially completed. The estimated costs and efficiencies for the blower, auxiliary generator, and boiler feed pump turbine drives for various throttle and exhaust steam conditions were obtained from the Small Steam Turbine Department of the General Electric Company.

Pending the results of overall power plant heat balance calculations, the optimum blower drive turbine appears to be one that operates at the same throttle steam conditions as those for the propulsion turbine: 850 psig and 950° F. Steam would exhaust from this unit to the cross-over pipe of the propulsion unit to take advantage of the high efficiency of the propulsion turbine. The auxiliary generator drive turbine will also utilize throttle steam; being a condensing unit, however, it will operate independently of the propulsion turbine. Using throttle steam rather than extracted or desuperheated steam for these units should not increase the turbine costs. There is a higher cost, however, in using throttle steam rather than desuperheated steam in the boiler feed pump turbine drive; this added cost, therefore, leads to the selection of a turbine using reduced steam temperature. The exhaust steam from this unit would also be returned to the propulsion unit cross-over pipes.

Methods of improving performance at the normal load point of 27,300 SHP with a minimum sacrifice of efficiency at full load are being investigated. The heat balances will be computed over the load range upon completion of these investigations.

DUTY CYCLE

A duty cycle for the 630A was established, assuming a 30-knot cargo vessel that makes thirteen 14,000-mile round trips. About 70 percent of each 28-day round trip is spent at sea. The duty cycle is shown schematically in Figure 2.4 and in tabular form in Table 2.2. This cycle results in the equivalent of 5720 full-power hours per year. Based on a core life of 15,000 hours at full power, refueling would occur about every 2.6 years, depending upon the frequency of shutdown for boiler cleaning, other maintenance, and power plant inspection.

In addition to the assumed power fluctuations, allowances for accidental scrams will be included in the planned fatigue analyses of components.

DYNAMIC ANALYSIS

A study was completed of the methods available to perform a dynamic analysis of the 630A. The purpose of the analysis would be to evaluate the structural design under the loadings imposed by the motion of the ship. The analysis involves establishing mass - spring systems that represent the actual components of the power plant. A preliminary mass - spring model of the 630A was prepared and is shown in Figure 2.5. The individual masses and springs of the model are given in Tables 2.3 and 2.4, respectively. After the model is completed, the stiffness and mass matrices are calculated. The response of each component to the imposed load is then determined from the mocked-up mass - spring system. A computer code is available at the Advanced Technology Laboratories of the General Electric Company to complete this portion of the analysis. This code was developed for the Navy's dynamic shock analysis but can be modified for the 630A by substituting the ship's motion for the Navy shock spectrum. The analysis is completed by using the accelerations calculated by the dynamic analysis computer program to evaluate the stresses in each component.

Fig. 2.4 - 630A duty cycle

TABLE 2.2
 POSTULATED 630A POWER LEVELS FOR ONE ROUND TRIP

Operating Mode	Days	Fractional Power	Full Power Days
Shutdown	2.5	0	0
Loading	3.0	0.10	0.30
Maneuvering	1.4	0.91	1.27
Reduced speed	1.5	0.70	1.05
Normal speed	13.6	0.91	12.38
Full power	3.0	1.0	3.00
Unloading	3.0	0.10	0.30
	28.0		18.30

Note: See Tables 1 and 2 for identities of the individual masses and springs.

Fig. 2.5 – Proposed dynamic analysis model

TABLE 2.3
SUMMARY OF MASSES FOR THE 630A DYNAMIC MODEL^a

Mass No.	Components	Approximate Weight (W), thousands of lb	Approximate Mass (m),
			$\frac{\text{lbf-sec}^2}{\text{in.}} \left(m = \frac{W}{386} \right)$
1	1/2 Foundation + 1/2 external side shield + support ring	190	492
2	Containment lower lead + 1/8 containment shell	20	52
3	1/2 Pressure vessel + 1/2 blower shields + 1/2 internal side shield	137	355
4	Forward (blower + 1/2 blower shield + 1/2 blower nozzle)	11	28
5	Forward (1/2 blower containment + auxi- liary blower containment)	11	28
6	Aft (blower + motor + 1/2 blower shield + 1/2 blower nozzle)	13	34
7	Aft (1/2 blower containment + auxiliary blower containment)	13	34
8	1/2 External side shield	85	220
9	1/8 Pressure vessel + lower pressure vessel head	16	41
10	Steam generator + diffuser	60	155
11	1/8 Pressure vessel	8	21
12	1/2 Containment shell + 1/2 blower containment nozzles	45	117
13	1/4 Pressure vessel + 1/2 shield plug	43	111
14	1/4 Shield plug	13	34
15	Actuators	2.5	6
16	1/4 Shield plug	14	36
17	Calandria	24	62
18	Expansion tank	3	8
19	3/8 Containment shell + 1/2 upper containment shell	66	171
20	1/2 Internal side shield	104	269
21	1/2 Upper containment shell + upper head	50	130
22	Heat exchanger	13	34
23	Pump + pipe	1.5	4
24	Pump + pipe	1.5	4
Totals		945	2446

^aAll masses have four degrees of freedom except No. 18, which has six.

TABLE 2.4
SUMMARY OF SPRINGS IN 630A DYNAMIC MODEL

-
1. Foundation
 2. Containment shell below the foundation
 3. Pressure vessel support ring
 4. Blower nozzle - forward
 5. Blower (containment) nozzle - forward
 6. Blower nozzle - aft
 7. Blower (containment) nozzle - aft
 8. External side shield tank
 9. Pressure vessel - flange to lower head
 10. Bottom head and steam generator supports
 11. Pressure vessel - flange to top of cone
 12. Pump support - forward
 13. Pressure vessel - top of cone to top of vessel
 14. Shield plug - upper cylinder
 15. Actuators
 16. Shield plug - lower shell
 17. Calandria support (tubes)
 18. Expansion tank - aft support
 19. Expansion tank - forward support
 20. Internal side shield tank
 21. Containment shell - foundation to top flange
 22. Upper containment shell
 23. Heat exchanger structure
 24. Pump support - aft
-

In addition to running the computer program, personnel from the Advanced Technology Laboratories would provide consultation on establishing the mass - spring systems and interpreting the results from the dynamic analysis computer program.

WORK PLANNED FOR NEXT QUARTER

The secondary loop component operating conditions will be optimized and the results integrated into the propulsion and feedwater heating system.

Investigation of the various structural and pressure levels in the reactor, pressure vessel, and containment vessel will be continued, with the objective of determining deflections and stress levels.

3. REACTOR

The stress analysis of the reactor calandria vessel was continued to determine the optimum thicknesses. Calculations showed that the combined thermal and mechanical stresses due to external pressure loading in the bottom tube sheet fall within allowable ASME Boiler and Pressure Vessel Code limits.

A revised fuel cartridge design, incorporating the latching mechanism at the bottom end of the calandria to facilitate the remote handling of the cartridge, was studied. Other improvements in the design of the fuel cartridge were also proposed.

A study was initiated to determine the feasibility of substituting Zircaloy for some of the stainless steel components in the active core region.

Preliminary sizing of control rod extensions and gang plates was completed. Control rod buckling loads, gang plate stresses, and internal pressure in the shim rods due to helium release from the B_4C poison material were calculated. The mechanical connection between the shim and dynamic rod extensions and the gang plates was designed.

Correlation of five radial power calculations showed good agreement with the critical experiment. Moderator temperature coefficient measurements were made and agreed well with calculated data. Studies of nuclear heating and radiation damage of core components were continued.

Parametric thermal analyses were completed for various thicknesses and heating rates for the top and bottom tube sheets.

A thermal analysis of the dummy tube sheets was completed.

MECHANICAL DESIGN

Calandria Stress Analysis

Preliminary stress analysis of the calandria vessel continued. These efforts were primarily concentrated on studies of stresses in the bottom tube sheet (Figure 3.1) because these stresses are generally higher than those in the top tube sheet. Primary membrane and bending stresses in the bottom tube sheet were determined for an external pressure loading of 430 psig, the relief valve setting for the primary-air loop. A summary of these stresses is given in Table 3.1.

Bottom tube sheet thermal stresses, both membrane and bending, resulting from internal heat generation were also determined and are summarized in Table 3.2. For this analysis, a heating rate of 0.0025 watt/g-mw at the center of the inside surface of the bottom tube sheet was used. The heating rate was assumed to be distributed in a pattern which decreases exponentially through the thickness and parabolically across the radius. The solid portion of the tube sheet was assumed to be a fixed boundary and thermal stresses were computed in the perforated region only.

TABLE 3.1
BOTTOM TUBE SHEET MECHANICAL STRESS SUMMARY FOR AN EXTERNAL PRESSURE LOADING OF 430 PSI

Radial Distance From Centerline, inches	Top Surface ^a						Bottom Surface ^a					
	Radial Bending Stress	Radial Membrane Stress	Radial Stress Total	Tangential Bending Stress	Tangential Membrane Stress	Tangential Stress Total	Radial Bending Stress	Radial Membrane Stress	Radial Stress Total	Tangential Bending Stress	Tangential Membrane Stress	Tangential Stress Total
2.17	-5,615	-4,500	-10,115	-955	-4,500	-5,455	+5,615	-4,500	+1,115	+955	-4,500	-3,545
4.9	-9,150	-4,500	-13,650	-365	-4,500	-4,865	+9,150	-4,500	+4,650	+365	-4,500	-4,135
9.2	-12,715	-4,500	-17,215	-1,940	-4,500	-6,440	+12,715	-4,500	+8,215	+1,940	-4,500	-2,560
13.7	-14,030	-4,500	-18,530	-2,245	-4,500	-6,745	+14,030	-4,500	+9,530	+2,245	-4,500	-2,255
18.2	-16,165	-4,500	-20,665	-2,300	-4,500	-6,800	+16,165	-4,500	+11,665	+2,300	-4,500	-2,200
22.0	+5,265	-2,290	+2,975	+945	-4,000	-3,055	-5,265	-2,290	-7,555	-945	-4,000	-4,945
24.0	+14,610	-1,925	+12,685	+1,665	-4,000	-2,335	-14,610	-1,925	-16,535	-1,665	-4,000	-5,665
24.0 ^b	+1,345	-180	+1,165	-360	-2,755	-3,115	-1,345	-180	-1,525	+360	-2,755	-2,395
33.025	-7,435	-785	-8,220	-2,385	-2,145	-4,530	+7,435	-785	+6,650	+2,385	-2,145	+240

^a+indicates tension and - indicates compression.

^bA stress discontinuity occurs at this radius as tube sheet transitions from the perforated region to the solid region.

TABLE 3.2
BOTTOM TUBE SHEET THERMAL STRESS SUMMARY FOR A HEATING RATE OF 2.5 WATTS/G-MW
AT THE CENTER OF THE INSIDE SURFACE WITH AN EXPONENTIAL DISTRIBUTION
THROUGH THE THICKNESS AND A PARABOLIC DISTRIBUTION RADially

Radial Distance From Centerline, inches	Top Surface ^a						Bottom Surface ^a					
	Radial Bending Stress	Radial Membrane Stress	Radial Stress Total	Tangential Bending Stress	Tangential Membrane Stress	Tangential Stress Total	Radial Bending Stress	Radial Membrane Stress	Radial Stress Total	Tangential Bending Stress	Tangential Membrane Stress	Tangential Stress Total
2.17	+3,665	-2,485	+1,180	+3,660	-2,475	+1,185	-5,635	-2,485	-8,120	-5,640	-2,475	-8,115
4.9	+3,725	-2,455	+1,370	+3,685	-2,385	+1,300	-5,375	-2,455	-7,830	-5,415	-2,385	-7,800
9.2	+3,915	-2,370	+1,545	+3,755	-2,125	+1,650	-4,565	-2,370	-6,935	-4,705	-2,125	-6,830
13.7	+4,240	-2,225	+2,015	+3,920	-1,680	+2,250	-3,185	-2,225	-5,410	-3,495	-1,680	-5,175
18.2	+4,685	-2,015	+2,670	+4,140	-1,060	+3,080	-1,260	-2,015	-3,275	-1,805	-1,060	-2,865
22.0	+5,165	-1,795	+3,370	+4,370	-400	+3,970	+190	-1,795	-1,605	-605	-400	-1,005
24.0	+5,450	-1,660	+3,790	+4,505	0	+4,505	+2,020	-1,660	+360	+1,075	0	+1,075

^a + indicates tension and - indicates compression.

Fig. 3.1 - 630A calandria bottom tube sheet configuration

Similarly, thermal stresses were also computed for a 1.375-inch-thick perforated central region of the bottom tube sheet. The solid portion of the tube sheet was again assumed to be a fixed boundary, and thermal stresses were computed in the perforated region only. Two cases were evaluated under these conditions; in the first case, a heating rate of 0.004 watt/g-mw was assumed at the center of the inside surface of the bottom tube sheet. The heating rate was assumed to be distributed in a pattern which decreases exponentially through the thickness and parabolically across the radius. These stresses are summarized in Table 3.3.

In the second case, the assumptions were identical except that the heating rate was assumed to remain constant across the radius. The stress for the top surface is +19,150 lb/in² (tension) and for the bottom surface is -8,430 lb/in² (compression). In this case, the

TABLE 3.3
BOTTOM TUBE SHEET (1.375-INCH-THICK) THERMAL STRESSES
FOR A 0.004 WATT/G-MW^a HEATING RATE AT THE CENTER OF THE INSIDE SURFACE

Radial Distance From Centerline, inch	Top Surface ^b		Bottom Surface ^b	
	Total Radial Stress, lb/in. ²	Total Tangential Stress, lb/in. ²	Total Radial Stress, lb/in. ²	Total Tangential Stress, lb/in. ²
2.17	+ 5,605	+ 5,630	-21,835	-21,805
4.9	+ 5,945	+ 6,085	-20,915	-20,780
9.2	+ 7,025	+ 7,485	-17,990	-17,530
13.7	+ 8,860	+ 9,885	-13,040	-12,015
18.2	+11,405	+13,220	- 6,130	- 4,320
22.0	+14,120	+16,775	+ 1,215	+ 3,865
24.0	+15,760	+18,915	+ 5,640	+ 8,795

^aHeating rate assumed to be distributed in a pattern which decreases exponentially through the thickness and parabolically across the radius.

^bPlus indicates tension and minus indicates compression.

radial stress equals the tangential stress at both the top and bottom surfaces. These thermal stress levels appear to be within acceptable limits.

Combining the stress values shown in Tables 3.1, 3.2, and 3.3 provides the maximum combined stress for comparison with the maximum allowable stresses as specified by the ASME Boiler and Pressure Vessel Code. This comparison will be made when stresses resulting from all loading conditions, mechanical and thermal, have been determined. Analysis of other loading conditions, such as the pressure drop across the calandria, has been initiated.

Combined axial and bending stresses for the moderator tubes and control rod guide tubes were determined for the same primary-air pressure loading of 430 psi and are shown in Tables 3.4 and 3.5. These stresses do not include the effects of gross thermal deformations nor thermal stresses in the tubes themselves. However, these effects are not expected to significantly affect the total stresses in the tubes.

TABLE 3.4
MODERATOR TUBE STRESSES

Radial Distance Of Tubes From Centerline, inches	Combined Axial And Bending Stress, lb/in. ²
Center tube	2,570
4.9	4,650
9.2	3,695
13.7	3,760
18.2	5,860
22.0	6,780

TABLE 3.5
CONTROL ROD GUIDE TUBE STRESSES

Radial Distance Of Tubes From Centerline, inches	Combined Axial And Bending Stress, lb/in. ²
2.17	765
6.7	1,010
11.3	1,290
15.9	2,050
20.0	2,960

The 2.75-inch-thick calandria shell established for the present configuration satisfies the required thickness of a shell under external pressure, as specified in Section VIII of the ASME Boiler and Pressure Vessel Code. The criteria of proposed Section III of the Code is to be used for the design; under Section III, the required thickness can be reduced from 2.75 to 2.0 inches. To evaluate the merits of this reduction, the resulting changes in geometry were incorporated into the computer program. The program computations were completed, but the results have not yet been evaluated.

Fuel Cartridge Design

A revised fuel cartridge design, shown in Figure 3.2, was studied during this report period. The improvements under consideration are (1) a new latching mechanism, (2) one-half-length dead stage at the bottom of the active stages, (3) insulation on the tail piece,

Fig. 3.2—Typical fuel cartridge arrangement (Dwg. 221R155)

and (4) a redesigned nose piece that provides a dead air gap between the top tube sheet and the nose piece.

The latching mechanism is located at the bottom of the fuel cartridge. The spring and the latch fingers are part of the tail piece of the fuel cartridge assembly. Thus, a new latching mechanism is provided each time a fuel cartridge is replaced. The spring and the latch fingers can also be replaced individually. Positive locking is achieved by means of a spring-loaded sleeve.

Thermal insulation, 0.200 inch thick, was added to the tail piece of the fuel cartridge to limit the heat input to the bottom tube sheet. Similarly, a 0.10-inch dead air gap was provided between the nose piece and the tube sheet to restrict the heat input to the top tube sheet.

A half-length, 80Ni - 20Cr unfueled stage was added behind the ninth-stage fuel element. The purpose of this stage is to reduce the fast neutron dose rate to the boiler.

Zircaloy Investigation

A literature search was conducted to determine the available technology on Zircaloy, particularly under environmental conditions similar to those in the 630A core. Data were compiled on the mechanical and physical properties, radiation effects, corrosion rates, and manufacturing processes.

Zircaloy is being considered as a replacement for the stainless steel in the moderator support tubes and the center moderator water return tubes. Initial data indicate that Zircaloy is compatible for this application and has the required strength and structural integrity, and that substantial savings could be attained in initial fuel loading if the support tubes were made of Zircaloy. A comparison between the allowable stresses of Zircaloy and Type 304 stainless steel is shown in Figure 3.3.

Fig. 3.3—Allowable stress intensity versus temperature of Zircaloy and Type 304 stainless steel

Galvanic Corrosion

Information was requested from Argonne National Laboratory and Phillips Petroleum Company at the NRTS on their experience with the effects of aluminum - stainless steel galvanic corrosion. Argonne reported that they have observed no corrosion problems between aluminum specimens in contact with stainless steel for as long as 5 years. During this period the pH of the demineralized water varied from 8 to 4.5 and the temperatures went as high as 485°F. Phillips reported that there have been no corrosion problems between aluminum and stainless steel in approximately 12 years of operation at the MTR. The pH at the MTR is normally controlled at about 5.

Control Rods

To size the control rod extensions and ganging plates, a control rod analysis was initiated. The first step in sizing the control rod extensions was to calculate critical buckling loads for the control rod tip and extension which were assumed to be a continuous tube with a constant outside diameter. The diameters and cladding thicknesses of four control rod tips are shown in Figure 3.4. The wall thickness of the safety rod extensions was assumed to be 0.060 inch, and of the shim and dynamic rod extensions, 0.065 inch. All these rods were assumed to be fixed at the gang plate and free at the opposite end. The total length of the shim and safety rods is 81 inches; the length of the dynamic rod is 73.5 inches. The effective length of the shim and dynamic rods was assumed to be 49 inches due to the restraint imposed by the guide tubes.

The buckling loads compared to the maximum force that can be exerted by the actuators are summarized in Table 3.6.

TABLE 3.6
CONTROL ROD LOADING ANALYSIS

	Buckling Load, lb	Maximum Actuator Force, lb
Safety rod		
Type 8	1700	275
Type 0	1800	275
Type 10	975	275
Type 11	710	275
Shim rod	265	52
Dynamic rod	315	175

Although this analysis shows these tubes to be satisfactory, it is desirable to design and use a single extension tube on all four sizes of safety rods. With a wall thickness of 0.060 inch, a 1.2-inch-OD extension rod would be satisfactory from a buckling standpoint. Further analysis will be done before the size is established.

The maximum pressure that could result from the helium generated in the shim rod is estimated to be 1150 psi. This is based on the assumption that all the helium generated is released from the B₄C poison into the extension tube; the void volume in the poison tip is neglected. Since the maximum allowable pressure in the shim rod is 1560 psi, using ASME Boiler and Pressure Vessel Code criteria, this design appears to be conservative.

Gang Plates

The results of a preliminary stress analysis of gang plates (Figure 3.5) indicate that a thickness of 1 inch with localized stiffening webs should be satisfactory.

Fig. 3.4 - Fifth iteration fuel element and control rod dimensions (Dwg. 848D853)

Fig. 3.5 – Control rod pattern and gang plate configuration (Dwg. 221R151)

Each actuator drive rod is located at the center of gravity of its gang plate. Five different safety-rod gang-plate shapes are required, whereas only one gang-plate configuration is necessary for the shim and dynamic rods. The method of fastening the shim and dynamic rods to the gang plates is shown in Figure 3.6.

The effects of vibration and ship's motion on the control rods are being evaluated. A model (18 mass - 36 degrees of freedom) of the shim-rod system was established to determine the natural frequency using a digital computer program. Although initial results have been received, they have not yet been evaluated.

NUCLEAR DESIGN

Critical Experiment Correlation

Fine Radial Power Distribution - The technique used for analytically determining the fine radial power distribution was modified to improve the correlation between the predicted and measured values, using the same computer codes. For computing system reactivity, all the material in the fueled region, including the combs and the unfueled leading and trailing edges of the clad fuel, formerly was homogenized and distributed over the active core length. This homogeneous mixture was then used in the fine radial analysis when predicting power distribution. In the modified technique, the fueled region consists of only the fuel and cladding on one stage, homogenized over the fueled length of the

Fig. 3.6 - Control rod gang plate-extension rod assembly study (Dwg. 221R152)

stage. This is not the same as the fueled region used for reactivity correlation; both the fuel density and the fuel-to-cladding ratio are higher. Using this modified technique, the predicted power distribution is considerably closer to the measured value than it previously was.

The correlation between predicted and measured power distribution for the central cartridge in the first critical experiment configuration using the old analytical technique is shown in Figure 3.7.* In Figures 3.8 through 3.12 are shown the correlation in the present critical experiment with the modified analytical technique. Four layers of borated foil are in place, and the measurements are taken at stage 7, below the region affected by the shim rods. The best 11-ring correlation is for the center cell in the reactor (position 00), shown in Figure 3.8. The poorest 11-ring cartridge is in position 421 (Figure 3.9). The 10-, 9-, and 8-ring cartridges are closer to the side reflector; the correlation appears to get worse at the positions closer to the reflector.

Interstitial Control Rod Worth - The critical experiment measurements to date have indicated that the 132 interstitial rods are worth about 12.5 percent $\Delta k/k$, which is considerably greater than had been predicted. The predictions did not take into account the substitution of aluminum for the steel guide tubes, which partially shielded the rods in the original configuration. It now appears that 96 interstitial rods would be adequate for both the shim and dynamic requirements. Accordingly, the number of rods in the critical experiment was reduced.

Moderator Temperature Reactivity Coefficient - The change in reactivity due to the change in moderator temperature from 68°F to 200°F was computed for the fifth iteration design at the start and at the end of life. At the start of life, this temperature change produces an increase in reactivity of 1.0 percent $\Delta k/k$. At the end of life, the increase is 0.65 percent $\Delta k/k$. The smaller effect at the end of life is a result of burnup of the burnable poison during the operating lifetime.

The measured critical experiment data indicate that the change in moderator temperature increases reactivity by from 0.95 to 1.03 percent $\Delta k/k$ at the start of life.

Prototype and Other Design Studies

Isotopic History Trace Program No. 431 - To calculate end of life reactivity, cross sections were defined for the ten most significant, slowly saturating fission products, and for the four most significant, rapidly saturating fission products. The reactivity worth of the undefined "average" remaining fission product is less than 30 percent of all the fission products. In addition, cross sections were added so that the plutonium chains can be followed. Some difficulty was encountered in the Pm-Sm chain which will be cleared up in the next reporting period.

Beryllium Heating and Fast Neutron Damage - Some analytical work was done on the beryllium side reflector. The average fast neutron dose in the side reflector is about 0.8 to 1.0×10^{21} nvt ($E_n \geq 1$ Mev) per fuel cycle. The peak value, on the inner surface at the core midplane, is about three times this average value. The average gas formation (He^4 and H^3) is about 3×10^{19} atoms per cm^3 of beryllium per fuel cycle, or $1.1 \text{ cm}^3/\text{cm}^3$. Again, the peak value is about three times this average.

*Other correlations based on the old technique are contained in:

"630A Maritime Nuclear Steam Generator Progress Report No. 4," GE-NMPO, GEMP-175, January 31, 1963, pp. 100-102; and "630A Maritime Nuclear Steam Generator Progress Report No. 5," GE-NMPO, GEMP-215, June 28, 1963, pp. 100-102.

Fig. 3.7—Radial power distribution, stage 5, fuel cartridge 00.
Original configuration using old analysis technique.

Fig. 3.8—Radial power distribution, stage 7, fuel cartridge 00, second
loading, run No. 1517

Fig. 3.9—Radial power distribution, stage 7, fuel cartridge 421, second loading, run No. 1520 (11 ring cartridge)

Fig. 3.10—Radial power distribution, stage 7, fuel cartridge 441, second loading, run No. 1518 (10 ring cartridge)

Fig. 3.11 – Radial power distribution, stage 7, fuel cartridge 452, second loading, run No. 1519 (9 ring cartridge)

Fig. 3.12 – Radial power distribution, stage 7, fuel cartridge 451, second loading, run No. 1517 (8 ring cartridge)

It was reported elsewhere* that, with a fast neutron dose of 10^{22} nvt and a gas formation of about $23 \text{ cm}^3/\text{cm}^3$, the beryllium did not swell upon annealing at 440°C , although swelling did occur upon annealing at higher temperatures. Thus, swelling in the beryllium due to gas formation should not be a problem in the 630A.

The average nuclear heating rate in the beryllium is 0.01295 watt/g-mw. This is 0.619 percent of the total heat generated in the 630A. One-third of it is from neutron energy loss and two-thirds from gamma heating. At the midplane of the core, the average rate is about 0.02 watt/g-mw. The peak value (on the inner surface) is 0.03; the minimum (on the outer surface) is 0.0116.

Using the average heating rate at the core midplane, and assuming the heat to be generated uniformly throughout the thickness, the maximum temperature at the center of the beryllium slab was calculated to be about 310°F ; at the surface, it was calculated to be about 285°F .

Using these same assumptions and including the effect of the parabolic temperature distribution along the longitudinal axis of the slab, the maximum thermal stresses in the beryllium were calculated to be about 5500 psi. This stress is tension on the surface and compression in the center. With the present support method, the mechanical stresses are insignificant.

THERMAL DESIGN

Bottom Tube Sheet

The bottom tube sheet temperatures were computed for various heating rates and tube sheet thicknesses. This study was carried out by using the Transient Heat Transfer (THT) computer program to solve a 351-node problem. The node numbering system for the geometry selected is shown in Figure 3.13. The tube sheet segment that was analyzed consists of one-half the triflute-shaped segment located between three adjacent fuel cells. Also included in the problem were the 80Ni - 20Cr support tube, insulation, and moderator tube segments in the vicinity of the tube sheet. Insulation was also assumed over the lower face of the tube sheet adjacent to the air plenum at the reactor exit. The primary method of heat removal was by forced convection to moderator water flowing radially over the tube sheet.

Computations were made for internal heat generation rates of 0.002, 0.005, and 0.010 watt/g-mw for assumed tube sheet thicknesses of 1.0, 1.5, and 2.0 inches. Temperature plots were prepared for moderator tube axial temperature variation, tube sheet top surface ligament temperatures, and tube sheet temperature variation from the bottom surface to the top surface. Figures 3.14, 3.15, and 3.16 show these temperature maps for the 1.5-inch-thick tube sheet. These results are now being analyzed.

Top Tube Sheet

A similar study is being made for the top tube sheet. The THT program simulation of the top tube sheet and surrounding components has 168 nodes arranged as shown in Figure 3.17. The segment of the tube sheet analyzed consists of one-sixth of the triflute section between adjacent fuel cells. Heat removal to cool the tube sheet is accomplished by heat transfer across the tube sheet to the moderator flowing through the shim rod guide tube and also by convection to the moderator at the lower surface of the tube sheet. Heat transfer from the primary coolant and bellmouth at the fuel cell entrance is included.

*C. E. Ellis, "The Swelling of Beryllium from Neutron Induced Gases," Chalk River, Ontario, CRMET-809, November 1958.

Fig. 3.13 - Node numbering system for bottom tube sheet (THTC-2A analysis)

Fig. 3.14 – Moderator tube temperatures near the bottom tube sheet for a tube sheet thickness of 1.5 inch

Fig. 3.15 – Bottom tube sheet ligament temperatures on the top surface for a tube sheet thickness of 1.5 inch

Fig. 3.16—Bottom tube sheet temperatures for a tube sheet thickness of 1.5 inch

Figures 3.18 through 3.22 present the results for the 1.0-inch-thick tube sheet at each of three assumed heating rates. The profile of these temperature plots is the same as that of the bottom tube sheet. The studies performed to date show that, for heating rates below about 0.008 watt/g-mw, nucleate boiling should not occur at the tube sheet surfaces.

Dummy Tube Sheets

A THT analysis was run on a 5-inch-thick top dummy tube sheet with a heating rate that decays from 0.015 to 0.003 watts/g-mw. All four surfaces of the dummy tube sheet were cooled with 215°F water, the top and bottom by free convection, and the moderator and guide tube annuli surfaces by forced convection. The results showed a maximum temperature of 254°F, with a maximum surface temperature of about 235°F, well below the level that would produce surface boiling.

A calculation for the bottom dummy tube sheet is not believed to be necessary since the largest anticipated change in the top dummy tube sheet is a temperature increase of about 30°F due to a 30°F temperature increase in the ambient cooling water. However, a full 30°F increase is not anticipated since all the heat transfer coefficients are larger for the bottom dummy tube sheet.

GE-NMPO

Fig. 3.17—Node numbering system for top tube sheet (THTC-2A analysis)

Fig. 3.18 - Moderator tube temperatures near the top tube sheet for a tube sheet thickness of 1 inch

Fig. 3.19—Moderator tube temperatures near the top tube sheet for a tube sheet thickness of 1 inch

Fig. 3.20 - Moderator tube temperatures near the top tube sheet for a tube sheet thickness of 1 inch

Fig. 3.21 - Top tube sheet ligament temperatures on bottom surface for a top tube sheet thickness of 1 inch

Fig. 3.22 – Top tube sheet temperatures for a tube sheet thickness of 1 inch

WORK PLANNED FOR NEXT QUARTER

The stress analysis of the calandria will be continued to evaluate the merits of the reduction in thickness of the calandria shell. Other loading conditions will be reviewed and incorporated into the analysis.

A more detailed study will be made on the substitution of Zircaloy for stainless steel in the 630A reactor core. This evaluation will be based on parameters such as mechanical design, nuclear analysis, economics, and compatibility with the system design.

The analysis of the effects of vibration and the ship's motion on the control rods will continue.

Work will be completed on Isotopic History Trace Program No. 431.

Work will continue on critical experiment correlation including:

1. Fine radial power distribution for remaining typical positions
2. Fine radial power distribution for cartridges with a fifth layer of borated foil
3. Longitudinal effect of control rods
4. Gross radial power distribution.

4. SHIELD PLUG

The configuration of the shield plug was revised to incorporate additional shim rods and to reduce the shield plug diameter. Other changes in the configuration to facilitate manufacture and assembly are being evaluated.

MECHANICAL DESIGN

The configuration of the shield plug was revised to incorporate the additional 59 shim rods and to reduce the shield plug diameter as discussed in the previous report.* The revised configuration is shown in Figure 4. 1 and reflects the deletion of 36 shim rods as discussed in section 3. This configuration also includes changes being considered to facilitate manufacture and assembly. As delineated in this figure, the central portion of the flange plate is a separate and removable piece. This permits the lower head and skirt, the flange ring, and the upper cylinder to be fabricated and inspected as a unit. Assembly of the shield plug consists of welding the control rod guide tubes and the water inlet and exit tubes extending between the can and plug to the lower dished head. The shim rod guide tube extensions are then threaded onto the guide tubes. The lead shield slab is then bolted in place. The central portion of the flange plate is then installed. First, the four sections which are keyed to the flange ring and, when installed, form a complete ring are installed. Seal welds between these sections prevent leakage. The center flange plate is then bolted to the ring formed by the four sections. The control rod guide tubes are fastened and sealed to the flange plate with mechanical fasteners.

THERMAL ANALYSIS

Thermal analysis by means of the Transient Heat Transfer Computer Program has been initiated for the lower dished head portion of the shield plug.

WORK PLANNED FOR NEXT QUARTER

Additional changes in the configuration to facilitate manufacture and assembly will be developed and evaluated. The thermal stress analysis of the lower dished head will be started.

Thermal analysis of the lower dished head will be continued, and temperature maps will be drawn. Analyses of the flange region of the shield plug and the pressure vessel will be started.

*"630A Maritime Nuclear Steam Generator Progress Report No. 6," GE-NMPO, GFMP-257, December 31, 1963, p. 45.

Fig. 4.1 – Revised shield plug assembly (Dwg. 221R144)

5. SHIELDING

Evaluation of the operating and shutdown dose to establish sensor design and location was started.

RADIATION LEVELS AT NUCLEAR SENSORS

It is planned to locate the nuclear sensors in the inner side shield at a radius of 50 inches from the core centerline. Analysis of both the operating and shutdown radiation levels at this location, which are required for sensor design work, was started.

The operating fast neutron dose and primary gamma dose above the core midplane at a radius of 50 inches are shown in Figure 5. 1. The thermal flux at a radius of 50 inches on the core midplane is 9×10^9 nv.

The photoneutron production after shutdown will be a function of the gamma dose rate in the beryllium reflector. The radial and longitudinal distribution of the operating primary gamma dose is shown in Figures 5. 2 and 5. 3. This data will be used in calculating the shutdown levels.

WORK PLANNED FOR NEXT QUARTER

The nuclear heating rate distribution in the core calandria and pressure vessel will be determined. Shutdown gamma doses in the beryllium reflector and the resulting photoneutron sources will be computed. The magnitude of the gamma source in the moderator water using the shorter delay times in the revised moderator water circuit will be investigated. Use of depleted uranium as a shielding material will be evaluated.

Fig. 5.1 – Longitudinal distribution of calculated primary dose rates at location of nuclear sensors

Fig. 5.2 – Radial distribution of calculated gamma dose in beryllium reflector at core midplane

Fig. 5.3 – Longitudinal distribution of calculated gamma dose in beryllium reflector

6. CONTAINMENT VESSEL

Studies were performed to determine the port opening required to prevent containment vessel buckling in the event of ship sinking.

If a ship sinks, it is necessary to maintain containment vessel integrity in spite of external water pressure buildup. This can be accomplished by admitting sea water from the reactor compartment to the containment vessel through properly sized ports in the vessel wall. The ports would close automatically when the pressure differential was reduced by pressurization of the trapped containment air, thereby maintaining containment integrity. Several calculations were made to determine the port opening area required to admit sea water rapidly enough to prevent buckling of the containment vessel. The maximum code-allowable external pressure differential is 150 psi.

A C-4 cargo vessel of 21,000 load displacement tonnage was assumed as a basis for calculations. Load density was assumed to be 500 lb/ft³. A drag coefficient during sinking of 2.0 on the hull only was considered. To provide minimum buoyancy and maximum rate of sinking, the ship was assumed to be wholly flooded before sinking starts. Two modes of ship sinking were considered: (1) bow first with 88 ft/sec terminal velocity and (2) keel first or settling in the normal ship attitude with 26 ft/sec terminal velocity.

For the first case, a containment port opening of 5 ft² was investigated and found to be more than adequate to limit the pressure differential to 150 psi. As shown in Figure 6. 1, the maximum calculated pressure differential is 96 psi for a valve opening at a 200-foot depth and 70 psi for a valve opening at an 80-foot depth. These results are conservative not only because of the assumptions stated above, but it would be nearly impossible for the ship to sink bow first during its initial descent.

For the second case, a port opening of 1 ft² was found to be adequate to limit the pressure differential to 149 psi if the port opens at the 200-foot depth or 117 psi if it opens at the 70-foot depth. Pressure differentials are shown in Figure 6. 2 as a function of time during sinking.

It is concluded that about 1 ft² of port opening is adequate to prevent containment vessel buckling, considering more realistic sinking rates of 26 ft/sec or lower.

WORK PLANNED FOR NEXT QUARTER

Details of typical installations of piping and instrumentation through the upper containment head will be investigated.

The interface between the containment vessel, side shield, and the pressure vessel will be defined in regard to the bearing loads and radial support structure.

Fig. 6.1—Ship sinking incident with 5-square-foot containment port opening

Fig. 6.2—Ship sinking incident with 1-square-foot containment port opening

7. CONTROLS

REACTOR POWER RANGE CONTROLS

The reactor power range control system was simulated on an analog computer, optimized to give the best response to changes in load demand, and was investigated for response to many different transient conditions. A component hardware block diagram, based on the computer simulation, was made. The resulting control system was then subjected to a Bode analysis which indicated that the control was stable over the anticipated operating range of 10 to 100 percent.

The reactor power range control system is shown in Figure 7. 1. The control is essentially a flux control with a temperature trim. In operation, an airflow signal either from the throttle or from a tachometer on the air blower will be fed into the flux error amplifier as a demand signal. If the resulting change in flux does not establish the controlled air temperature, then the temperature trim loop will force a slight readjustment in flux level to re-establish the correct temperature. This control was simulated on an analog computer. The computer simulation included the reactor kinetics, the reactor heat transfer characteristics, the control rod position loop, the flux control loop, and the temperature control loop. The shim rods were also included in the simulation. The shim rods were simulated to only move when the dynamic rods were displaced ± 5 inches from their mid-position.

The original control did not include an airflow signal as a demand into the flux loop, and it proved to be very sluggish in its response. Analog recordings of its response were shown in the previous quarterly report* along with a comparison recording of the control with the airflow signal incorporated. It was shown that using airflow as a flux demand speeded up the response and minimized the transient temperature error. Further evaluation of the control without an airflow signal established that it was unstable at low-power levels. Stability was established by changing the temperature loop from an integral control to a proportional control. The disadvantage of this system was that any error between the flux demand and the flux feedback had to be offset by a temperature error. The temperature control was then changed to a proportional plus integral and found to be stable and accurate at all power levels from 10 to 100 percent. The proportional portion of the loop assured stability and gave good transient response while the integral portion assured that there would not be a temperature error. This modified control was subjected to numerous transient operating conditions.

The response of the control for a 10 percent step decrease in power from 100-percent power is shown in Figure 7. 2. Figure 7. 3 shows the response for a 10 percent step increase from 10-percent power. Figures 7. 4 and 7. 5 are analogous cases using one instead of two dynamic actuators. As can be seen, response is good except that there is a tendency to oscillate in the 10-percent power case. This is due primarily to the thermocouple time

*"630A Maritime Nuclear Steam Generator Progress Report No. 6," GE-NMPO, GEMP-257, December 31, 1963, Figures 7.2 and 7.3, pp. 55-56.

Fig. 7.1 – Reactor temperature control system

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Fig. 7.2—Transient response for 10 percent step decrease from 100 percent power

GE-NMPO

Fig. 7.3—Transient response for 10 percent step increase from 10 percent power

GE-NMPO

Fig. 7.4 – Transient response for 10 percent step decrease from 100 percent power using one dynamic actuator

GE-NMPO

Fig. 7.5 – Transient response for 10 percent step increase from 10 percent power using one dynamic actuator

lag and it is felt that the 5-second time constant assumed is pessimistic. Response with one dynamic actuator appears good for transient steps of 20 percent or less. Transient operation without the use of shims was also tried and found to give good results on transient steps of less than about 30 percent.

Figure 7. 6 shows the response to a ramp increase in power from 10 to 100 percent in about 60 seconds. Figure 7. 7 is the same transient operation but without shims. It can be seen on either figure that the maximum transient temperature error is 25°F.

A Bode analysis of the reactor power range was performed at various power levels between 10- and 100-percent power. The analysis indicated that the control was stable even though the reactor gain is a function of the power level and the heat-transfer time constant is a function of airflow. These changes tend to be somewhat compensating and result in a stable system. There is less phase margin (22 degrees) at 10-percent power than is considered desirable; the desired phase margin is 40 degrees. The most significant improvement in the phase margin was obtained by using a reactor outlet air thermocouple faster than the 5-second response which was assumed in this analysis.

TWO-TUBE BOILER TEST

The single-flute inner valve was received and incorporated into the feedwater control valve. This inner valve will allow testing down to one-half the flow possible with the two-flute inner valve. A new feedwater flowmeter permitting readings down to less than 10 percent of rated flow was also installed. The boiler, with these modifications, was operated on manual control from 16 to 35 percent flow. Operation on either automatic or manual control was found to be stable and the operating point was held in spite of the flow oscillations in the individual tubes. On automatic control, a step change was made from the 35-percent operating point to a 20-percent operating point without problems. The boiler required about 4 minutes to reach steady state; however, the temperature remained within 50°F of the set point, and the pressure remained within 50 psi during this time. Transient operation at lower flow levels is now being conducted.

The design details and operating results of the two-tube boiler were discussed in detail with representatives of the Babcock & Wilcox Company. Their recommendations for additional testing to increase the available information about nuclear plant once-through boiler operation and control have been received and were evaluated. The present program plan does not provide for procurement of this additional operating data.

CONTROL ROD ACTUATORS

Safety Rod Actuator

The revised safety actuator drawings were checked. During their checking, redesign of several parts was made in order to improve the design or lower the manufacturing costs. All of the drawings have been revised.

A series of tests on the miniature latch and scram valves proposed for use in the safety actuator was completed. These tests were run at pressures from 0 to 200 psi, temperatures from 68° to 275°F, and voltages up to 30 vdc to check the effects of the various parameters on valve operation. The tests showed that these valves will be satisfactory if the solenoids are of sufficient power to operate the valves under all expected conditions of temperature, pressure, and voltage. The solenoids will be adequate if they are wound with heavier wire and Class H insulation.

A scram latch-release piston made of Type 440C stainless steel hardened to Rc 55 to 60 was tested for 3400 scram cycles with no sign of wear. Therefore, this piston will replace

Fig. 7.6—Ramp increase in power using shims

Fig. 7.7—Ramp increase in power without shims

the more expensive Carboloy grade 608 (R_C 68), which is presently used in the safety actuator.

Actuator Test Rig

Drawings for the actuator test rig were released and work was started on its fabrication. Dynamic Rod Actuator

Fabrication of the parts for two development model dynamic actuators was completed and the actuators are now being assembled. Figure 7.8 shows all of the parts laid out prior to assembly.

Fabrication of a control box to drive the dynamic actuator during bench testing was started. This control box will be used to drive the actuator until a power supply and amplifier is purchased.

An alternate seal design for the dynamic actuator was completed and three test specimens were fabricated. The seal design is shown in Figure 7.9. This seal consists of four chevron-type Teflon seals with a male and female adapter and preloaded with 45 pounds of spring force. The tapered sides of the chevron seals provide for take-up of wear. This type of seal has been very successful in other applications and appears to be well adapted to this application. One of the test specimens was sent to Idaho for testing in their test rig. The other two test specimens will be used in the development model dynamic actuators now being assembled.

ACTUATOR SEAL TESTS

The control rod drive actuator seal tests continued throughout this report period. The tests were conducted utilizing modified HTRE-3-type shim and safety rod actuators. Test conditions simulated those expected to occur during actual operation.

Test results to date indicate that the seal proposed for use with the safety rods is very adequate. Six safety rod seal tests were completed utilizing a predicted duty cycle of one scram every 10 hours for the total 15,000 hour life. All six of these tests exceeded seal life requirements without visible signs of leakage. The tests were conducted while maintaining test water conditions at 240°F and 65 psig at all times. The rods were cycled only during normal working hours and were held stationary alternately in the withdrawn and inserted positions during off-duty periods. The only parameter of actual operating conditions not being duplicated in this series of tests is the aging of the O-ring and Kapseal combinations.

The shim rod actuator seal tests were also continued throughout this report period. Two tests (Nos. 7 and 8) were started at the beginning of this report period and have not failed to date. Test No. 7 has now completed 17,987 cycles in approximately 944 hours of actuator operation. Test No. 8 has completed 15,013 cycles in 798 hours of actuator operation. These tests are cycled only during normal working hours, but temperature and pressure conditions are maintained 24 hours a day, 7 days a week (except during required maintenance shutdowns). More tests will be performed to determine whether or not these two tests are representative.

The carbon seal tests were discontinued on completion of the 6,338 cycles representing a 15,000 hour life. A leakage rate of approximately 8 cm³/cycle was the maximum leak obtained. This leakage occurred even while the rod was stationary and therefore was approximately 160 cm³/hour. No further tests are planned on the carbon seals.

A chevron-type seal was manufactured in Evendale and delivered to Idaho for tests. This seal was placed on a shim rod actuator and will be put on test February 21, 1964.

Neg. P64-2-25a

Fig. 7.8-630A development model dynamic actuator parts

Fig. 7.8—(Continued)

Neg. P64-2-25b

Fig. 7.9—Proposed seal and bearing modification for dynamic actuator

SHIM ROD ACTUATOR CONTROL CIRCUIT

A preliminary schematic of a manual shim rod actuator control, Figure 7.10, was prepared. The following provisions were incorporated:

1. Actuators can be operated as a group.
2. Actuators can be operated individually for alignment or test purposes.
3. Defective actuators can be locked or held in an "off" position.
4. Actuators other than those in the off position or already fully inserted will automatically be fully inserted on scram follow-up, for scram conditions other than loss of either 115-volt, 60-cycle drive power or 28-volt, d-c dynamic braking and holding power.
5. Actuators are locked in place on loss of either 115-volt, 60-cycle drive power or 28-volt, d-c dynamic braking and holding power.

Normal Operation

The mode of operation of each shim rod actuator during normal operation is determined by the position of the individual selector switches. The following is a description of this operation.

Group Operation - Actuators with selector switches in the group position can be operated as a group by depression of the group insert or group withdraw pushbuttons. The pushbuttons are electrically interlocked and, if possible, are also mechanically interlocked so that only an insert or withdraw operation can be performed at one time. After partial insertion or withdrawal (except when limits of travel are reached) release of pushbuttons will present coasting by applying 28-volt, d-c dynamic braking power to the insert winding of the shim rod actuator motor. When limits of travel are reached, microswitches are

Fig. 7.10—Wiring diagram of manually operated shim rod actuator control (Dwg. 848D867)

transferred, which automatically cut off a-c driving power and supply 28-volt, d-c dynamic braking and holding power to stop actuator travel and hold the actuators in position.

Individual Operation - Actuators may be operated on an individual basis to bring them into alignment with the group or other desired position by switching the individual selector switches to the individual position and depressing the master individual insert or master individual withdraw pushbuttons. The interlocking of pushbuttons, braking, holding, and microswitch operations are the same as for group operation. The position of the individual actuators can be determined by a single actuator position indicator and selector switch. Full insert lights located on the main console indicate when the individual shim rod actuators are fully inserted.

Off Operation - The off position of the individual shim rod actuator selector switches will normally be used for locking defective shim rod actuators in position. In the off position, the d-c brake on the shim rod actuator motor is de-energized, locking the actuator in position.

Under scram conditions, other than loss of either 115-volt, 60-cycle drive power or 28-volt, d-c dynamic braking and holding power, all of the shim rod actuators not already fully inserted or with individual actuator selector switches in the off position will automatically be fully inserted. Loss of either 115-volt, 60-cycle drive power or 28-volt, d-c dynamic braking and holding power will cause the shim rod actuators to be held locked in position.

Emergency Operation

The following is a description of circuit operation under scram conditions with drive power and 28-volt, d-c dynamic braking and holding power and with the loss of either.

Scram Operation - Under scram conditions, the scram relays are de-energized and their contacts are as indicated in Figure 7. 10. On scram, contacts SCR 1, 2, and 3 open to re-release the applied 28-volt, d-c braking and holding power to all shim rod actuators other than those actuators already in the fully inserted position or with individual selector switches in the off position. Contacts SCR 4 and 5 open to prevent applying withdraw power during scram. Contacts SCR 6 and 7 close to automatically apply insert power (shim rod follow-up) to all shim rod actuators with individual selector switches in the group and individual positions. Contacts SCR 8 and 9 close to insure a 28-volt, d-c dynamic braking and holding circuit for the shim rod actuators when fully inserted.

Scram Operation (Power Failure) - The shim rod drive power relay will become de-energized on loss of 115-volt, 60-cycle shim rod drive power and through its normally-open contact will de-energize the d-c brakes on the shim rod drive actuator motors, locking the shim rod drive actuators in position. Loss of the 28-volt, d-c dynamic braking and holding power will also de-energize the d-c brake and lock the actuators. The d-c power relay through its normally-open contact prevents application of drive power on loss of 28-volt, d-c dynamic braking and holding power so the locked actuator motors are not damaged by overheating.

WORK PLANNED FOR NEXT QUARTER

Operation of the two-tube boiler test rig to investigate control methods and parallel flow path stability will be continued. Assembly of the dynamic rod actuators will be completed and testing will be started. Fabrication of actuator test rig will be continued. Actuator shaft seal testing will be continued.

8. PRIMARY COOLANT SYSTEM

BLOWER SHAFT SEALS

Continuation of further development work on the blower shaft seal at the Stein Seal Company is awaiting approval of the AEC. A supplemental contract has been drawn up and approved by GE-NMPO and Stein Seal Company.

Chicago Rawhide Company has proposed an alternate design for the blower shaft seal utilizing a double carbon ring with buffer pressure between them. A carbon ring with a 400-psi differential is floated on a cushion of air while a carbon ring with a 3-psi differential is a direct rubbing seal. The vendor will supply a cost estimate on this seal, and the proposal will be evaluated when the cost estimate is received.

Four manufacturers of high-pressure air compressors were contacted concerning an air compressor to supply 45 scfm air at 500 psi for the blower shaft seals. Both oil-lubricated and oil-less models were considered since the presence of oil in the air is undesirable. The oil-lubricated type is recommended by the vendors for marine applications because of the internal rusting problems associated with the oil-less type. Oil consumption of a unit of this type is approximately 3 drops per minute, with the aftercoolers removing about 90 percent of the oil from the air. Filters are available which will remove virtually all of the remaining oil.

An investigation was also conducted on the availability of a pressure regulator to control the air to the blower shaft seal. The Fisher Governor Company makes a valve which meets these requirements.

BLOWER SHAFT STATIC SEAL TESTS

The blower shaft static seal provides positive isolation of the air in the containment vessel in the event a leak develops in the primary loop pressure vessel. This seal is used in conjunction with a dynamic seal. The dynamic seal provides the seal while the blower shaft is turning, and the static seal is activated only after the blower shaft has stopped. The combination static and dynamic seal is shown in Figure 8.1.

A test seal was fabricated to prove the feasibility of the proposed design to operate without binding and to evaluate its sealing capability. A drawing of the test fixture and the seal is shown in Figure 8.2. The static seal is operated by applying air pressure to the back side of the piston so that the piston moves forward to contact the inner face of the buffer ring which simulates the shaft collar adjacent to the static seal.

A seal is then effected by compressing the seal ring between the collar and the piston. The chamber which contains the springs simulates the containment vessel. Once the seal is actuated, it can be held in an actuated position either by external pressure or by containment vessel pressure through check valves, as can be seen on the test schematic in Figure 8.2. Therefore, once the seal is energized, it is no longer dependent on an external pressure source.

Fig. 8.1—Combination dynamic and static blower shaft seal (Dwg. 848D839)

The test fixture was assembled using Sohimil 3278* grease on all of the O-rings and the piston. The number of springs for retracting the piston was increased to 12 (70 pounds) when it was found that six springs (33.6 pounds) would not cause the piston to operate properly. Satisfactory operation of the piston was obtained. At this level of spring load, a pressure 9.1 psi above the containment vessel pressure was required to actuate or extend the seal piston. The piston retracted when the pressure difference fell below 3.45 psi. This holding pressure normally would be supplied from an external source, but since this pressure is trapped by check valves, it will maintain itself automatically.

The assembled test fixture was energized with 250 psig air and leak tested by submerging it in water. The test assembly was leak tested for 18 hours and found to be bubble tight.

The test fixture was stored in a de-energized condition and will be operated periodically to evaluate the effects of long periods of inactivity.

Figure 8.3 shows the test assembly in various stages of disassembly.

The following conclusions and recommendations are made as the result of these tests:

1. The tests were successful and indicated that a design of this type will provide adequate seal of the blower shaft at the containment vessel penetration.

*Trademark, Standard Oil Company.

Fig. 8.2—Seal test fixture assembly (Dwg. 692E641)

GE-NMPO

Neg. P64-1-15E

GE-NMPO

Neg. P64-1-15C

Fig. 8.3—Face seal and return spring details

2. The seal should be actuated periodically, such as each time the blower is stopped, to assure proper operation.
3. The system which controls this seal must be interlocked to prevent actuation of the seal while the blower shaft is turning.

MAIN BLOWER DRIVE

A study was made to determine the type of main blower drive to be used. The results are discussed in section 2, "Propulsion System Investigation."

WORK PLANNED FOR NEXT QUARTER

Blower shaft seal development will be continued. The blower lube system configurations will be investigated. Specifications for blower development will be prepared.

9. CRITICAL EXPERIMENT TEST OPERATIONS

Initial loading of the second configuration of the 630A critical experiment reactor was completed during this report period. Detailed power distribution measurements were obtained on the 11 typical positions. The results showed an 18 percent spread in power from the center to the outside of the reactor. Further adjustments will be made to reduce this spread and get the peak down to about 1.07 times the average value. The moderator temperature coefficient measurement was completed and fine radial power distribution changes were measured in three representative positions at an elevated temperature of about 150°F.

INITIAL LOADING

Initial loading of the second configuration of the 630A critical experiment reactor continued until the full complement of 85 fuel cartridges was installed. The number of control rods in the reactor was increased twice before the initial loading was completed. This change was necessary to assure the required shutdown of two dollars with the moderator tank full and the control rods scrambled. With 77 fuel cartridges in the reactor the number of control rods was increased from the original 57 to 100. After the 80th fuel cartridge was placed in the reactor, the total number of control rods was increased to the maximum of 132. Of the 132 rods, 70 were thin-wall rods clad with 0.025-inch-thick stainless steel rather than the 0.049-inch-thick stainless steel cladding used on the regular rods. The outside diameter of all control rods is 0.5 inch. The location of the 132 control rods and the actuator arrangement is shown in Figure 9.1. Excess reactivity was measured as each fuel cartridge was added to the reactor. A summary of these data is shown in Table 9.1.

CONTROL ROD CALIBRATIONS

Early in the experiment, it was seen that the subcritical control rod worth curves were yielding much higher rod worths than expected. It was also found that excess reactivity values were not consistent with those obtained from the critical rod worth curve when using only a few rods in the reactor. Further study showed that by using the critical curve for several combinations of control rods the excess reactivity values were in good agreement. Therefore, the subcritical rod worth curves were considered to be invalid and only the critical curve was used to evaluate control rod worths and excess reactivity.

After the reactor was fully loaded and 132 control rods were on the control system, another rod worth curve was measured using actuator 431. This actuator contained six thin-wall rods. The curve was the same as the one presented in the previous progress report for actuator 631.*

The worth of the original 57 control rods was 6.20 percent $\Delta k/k$, the 100 rods were worth 9.46 percent $\Delta k/k$, and the 132 rods were worth 12.50 percent $\Delta k/k$. These rod worths as well as the previously quoted excess reactivity values had a standard deviation of about 5 percent.

*"630A Maritime Nuclear Steam Generator Progress Report No. 6," GE-NMPO, CEMP-257, December 31, 1963, p.94.

OPERATIONAL SUMMARY

A summary of the critical experiment operation for this report period is presented below.

1. Fifty-five reactivity measurements were made and 20 foil exposures were completed representing 3324 power map data points in the 630A critical experiment.
2. The total reactor operating time was approximately 57 hours.
3. The control system wiring was modified to prevent more than two actuators from being withdrawn at the same time when the frame selector switch is in the manual position. This modification was required to assure that the 0.025 percent $\Delta k/k$ per

Fig. 9.1 – Modified actuator layout

TABLE 9. 1
EXCESS REACTIVITY MEASUREMENTS OF
CRITICAL EXPERIMENT REACTOR DURING
INITIAL LOADING OF SECOND CONFIGURATION

Fuel Cartridge Added	Total In Reactor	Excess Reactivity, ^a % $\Delta k/k$	Fuel Cartridge Worth, % $\Delta k/k$
252	68	1.587	0.235
552	69	1.859	0.272
652	70	2.156	0.297
352	71	2.442	0.286
452	72	2.694	0.252
152	73	2.836	0.142 ^b
251	74	3.111	0.275
551	75	3.461	0.350
351	76	3.919	0.458
651	77	4.251	0.332
151	78	4.605	0.354
451	79	4.718	0.113 ^c
154	80	5.055	0.337
454	81	5.291	0.236
354	82	5.734	0.443
654	83	6.001	0.267
554	84	6.356	0.355
254	85	6.726	0.370

^aMost of the excess reactivity values were obtained with a few actuators equally inserted in the critical position and the remaining control rods withdrawn. The partially inserted rods were bumped, their worth extrapolated from the resulting period, and excess reactivity determined from the worth of the rods and their critical position. The critical rod worth curve was used in all cases to extrapolate total rod worth and to calculate excess reactivity.

^bThe moderator temperature was 10°F less on this measurement than with 72 fuel cartridges in the reactor.

^cThe moderator temperature was 15°F less on this measurement than with 78 fuel cartridges in the reactor.

second reactivity insertion rate was not exceeded when the gang frame and additional poison tips were added to the reactor. The frame selector switch was deactivated so that actuators were operable only in the manual mode. The system was further modified to permit operation of an additional actuator.

4. The rod drop time was remeasured with the added weight of the gang frames and additional poison tips.

POWER DISTRIBUTION MEASUREMENTS

The reactor was critical with all 132 control rods equally inserted near the top of stage 5. Fine radial power distribution measurements were obtained from the center of stages 3 and 7 on the 11 typical fuel cartridge positions. The relative circumferential power distribution on the outer fuel layer for stages 3 and 7 on each of the typical positions is presented in Figures 9.2 through 9.12. These data indicate that there is considerable power fluctuation around the fuel cartridges near the outside of the reactor and particularly on stage 3 where there are fewer than six control rods around the fuel cartridges. It should be pointed out, however, that after integrating the longitudinal and fine radial power over the peak areas, the magnitude of the peaks at the periphery of the reactor will be no higher than at the center of the reactor if the radial fuel loading for each fuel cartridge is optimum. (The power distribution for stage 7 is typical for all axial locations below the rod tip, and that of stage 3 is typical for all axial locations that are "shadowed" by the rods.)

Fig. 9.2 - Circumferential power distribution, second loading, fuel cartridge 00, outer layer of fuel

Fig. 9.3 - Circumferential power distribution, second loading, fuel cartridge 410, outer layer of fuel

Fig. 9.4 - Circumferential power distribution, second loading, fuel cartridge 420, outer layer of fuel

Fig. 9.5 - Circumferential power distribution, second loading, fuel cartridge 421, outer layer of fuel

Fig. 9.6—Circumferential power distribution, second loading, fuel cartridge 430, outer layer of fuel

Fig. 9.7—Circumferential power distribution, second loading, fuel cartridge 431, outer layer of fuel

Fig. 9.8—Circumferential power distribution, second loading, fuel cartridge 440, outer layer of fuel

Fig. 9.9—Circumferential power distribution, second loading, fuel cartridge 441, outer layer of fuel

Fig. 9.10 - Circumferential power distribution, second loading, fuel cartridge 442, outer layer of fuel

Fig. 9.11 - Circumferential power distribution, second loading, fuel cartridge 451, outer layer of fuel

Fig. 9.12 – Circumferential power distribution, second loading, fuel cartridge 452, outer layer of fuel

The fine radial power distributions for the 11 typical fuel cartridges are given in Figures 9.13 through 9.23. The analytical fine radial power distribution was available only for position 00 and is presented in Figure 9.13. The analytical curve assumed no rods inserted and should, therefore, be comparable to stage 7 results.

The longitudinal power distribution was measured on positions 00, 421, 440, 441, 451, and 452; these data are presented in Figures 9.24 through 9.29. These curves are based on the average circumferential counts from the outer fuel layer at the positions or stages indicated on the curves. The effect of the rods is clearly shown by each of the longitudinal power distribution plots (the control rods were inserted to the midposition of the active core).

To determine the gross radial power distribution, it was assumed that the fine radial power distribution on stage 3 was valid from stages 1 through 5 and that stages 6 through 9 were the same as stage 7. It was also assumed that the difference in fuel loading between the various fuel cartridges (8, 9, 10, and 11 ring) was directly proportional to the change in heat transfer area. The resulting gross radial power distribution, based on these assumptions, is presented in Figure 9.30. It can be seen that there is a spread of about 18 percent in power per unit heat transfer area from the center of the reactor to position 452. This is a significant improvement over the original spread of about 33 percent with the initial loading of the 630A reactor. Steps are presently being taken to improve the gross radial power distribution by removing the 36 rods which are between the outer two rings of fuel cartridges. In addition, the upper 5.5 inches of boron steel will be removed and placed over the remaining 22 inches of the active core to provide a total of 5 instead of 4 layers over the area containing boron steel. Following these changes, another complete power map will be obtained.

MODERATOR TEMPERATURE COEFFICIENT

The moderator temperature coefficient of reactivity was measured from an average temperature of 64° to 147°F. The results are presented in Figure 9.31. The overall average for the measured temperature range was 0.008 percent $\Delta k/k$ per °F increase in

Fig. 9.13—Radial power distribution, second loading, fuel cartridge 00

Fig. 9.14—Radial power distribution, second loading, fuel cartridge 410

Fig. 9.15 - Radial power distribution, second loading, fuel cartridge 420

Fig. 9.16 - Radial power distribution, second loading, fuel cartridge 421

Fig. 9.17—Radial power distribution, second loading, fuel cartridge 430

Fig. 9.18—Radial power distribution, second loading, fuel cartridge 431

Fig. 9.19—Radial power distribution, second loading, fuel cartridge 440

Fig. 9.20—Radial power distribution, second loading, fuel cartridge 441

Fig. 9.21 – Radial power distribution, second loading, fuel cartridge 442

Fig. 9.22 – Radial power distribution, second loading, fuel cartridge 451

Fig. 9.23—Radial power distribution, second loading, fuel cartridge 452

Fig. 9.24—Longitudinal power distribution, fuel cartridge 00, run No. 1527

Fig. 9.25—Longitudinal power distribution, fuel cartridge 421, run No. 1529

Fig. 9.26—Longitudinal power distribution, fuel cartridge 440, run No. 1530

Fig. 9.27—Longitudinal power distribution, fuel cartridge 441, run No. 1529

Fig. 9.28—Longitudinal power distribution, fuel cartridge 431, run No. 1527

Fig. 9.29 – Longitudinal power distribution, fuel cartridge 452, run No. 1528

Fig. 9.30 – Gross loading power distribution, second loading

Fig. 9.31 – Moderator temperature coefficient of reactivity, second loading

moderator temperature. At the lower temperature levels, around 60°F, the temperature coefficient was about 0.012 percent $\Delta k/k$ and at the upper levels, around 150°F, it was approximately 0.007 percent $\Delta k/k$ per °F increase in water temperature. The overall average is essentially the same as measured for the original 630A critical experiment assembly whose average was 0.007 percent $\Delta k/k$ per °F increase in moderator temperature.

A fine radial power distribution measurement was obtained on stage 7 of fuel cartridges 00, 430, and 451. Measurements were taken at both upper and lower temperature levels. The results are presented in Table 9.2. Each set of data shows a decrease at the center and outside with respect to the average at the higher temperature. This result was expected because of the neutron spectrum shift with temperature.

TABLE 9.2
RADIAL POWER DISTRIBUTION
STAGE SEVEN

Fuel Cartridge Position	Weighted Average ^a	
	56°F	147°F
00	0.709	0.719
430	0.633	0.668
451	0.652	0.675

^aAverage to outer bend.

WORK PLANNED FOR NEXT QUARTER

Power distribution measurements will be made with the borated foil removed from the top 20 percent of the core and the outer 36 shim rods removed.

Other measurements (e.g., scram rod effectiveness, heating rates, dose ratio) will be made as indicated in the test program.

10. HANDLING EQUIPMENT

Preliminary layout drawings were completed on the cask, doors, and door-closing components of the core transfer cask assembly. The required cooling system and lifting assembly are being designed.

Preliminary sketches of the control rod removal cask and proposed pickup devices are shown in Figure 10.1. No major problems are anticipated with this part of the power plant disassembly operation. The final design of this equipment will be completed upon completion of the design of the control rod gang plates.

WORK PLANNED FOR NEXT QUARTER

The design of the cooling system and lifting arrangement will be continued.

Fig. 10.1 - Control rod removal cask and assembly

11. FACILITIES AND EQUIPMENT

Current facility requirements for testing the 630A prototype were reviewed. The existing Initial Engineering Test Facility (IET) at the NRTS was examined for possible use in connection with 630A testing. Cost estimates and a preliminary facility layout drawing (Figure 11.1) were prepared in connection with this study. The required facilities were estimated to cost about \$810,000 if existing IET facilities are utilized, as compared to about \$2,051,000 if completely new facilities are constructed at an undeveloped site.

This recent re-evaluation of the cost of modifying the IET for 630A testing was conducted on the basis of a minimum acceptable facility. For example, a substantial cost saving resulted from the substitution of a bridge crane in place of the gantry crane estimated in a previous study.

A similar study is now being performed in connection with the possible utilization of the remaining SL-1 facilities. Preliminary results of this study indicate that similar or slightly greater cost savings would result if this facility were used instead of the IET. The SL-1 site is also more suitable for expansion and has an existing area suitable for crew training purposes.

Fig. 11.1 - Proposed layout of 630A test facilities at IET

12. FUEL MATERIALS

Radiochemical analysis of 80Ni - 20Cr fuel sheet specimen EB-8 from in-pile test ORF-3 indicated a burnup of 22.6 percent of the U^{235} atoms occurred during 2758 hours at 1450°F.

Test ORF-4 was terminated after 4800 hours at an indicated maximum specimen temperature of 1300°F. It is estimated that 40 percent burnup of the U^{235} atoms occurred. This is a revised estimate since the burnup rate decreases with fuel depletion.

The temperature and burnup conditions experienced by ORF-3 and ORF-4 are equivalent to the most severe conditions anticipated for 630A fuel elements approaching the end of 15,000 hours operation at full power.

Assembly of the next scheduled test capsule, ORF-5, is complete although test plans are indefinite.

Creep test times in excess of 12,000 hours have been accumulated on 80Ni - 20Cr fuel sheet with no greater than 0.75 percent elongation measured. Oxidation penetration in unstressed 80Ni - 20Cr alloy after 10,000 hours at 1600°F was less than 0.002 inch.

ORR TESTS

In this series of in-reactor tests, flat plate specimens of 80Ni - 20Cr fuel sheet are being evaluated for resistance to high burnup of the U^{235} atoms in the ORR F-2 facility. Test ORF-4 was terminated after 488 hours. Test capsule ORF-5 has been assembled, but the test plans on this capsule are indefinite.

ORF-3

Chemical analysis for U^{235} burnup in specimen EB-8 was performed. Based on gamma counting of Cs^{137} , the total burnup was 22.6 percent of the U^{235} atoms. The estimated burnup in this specimen had been 30 percent. Test ORF-3 was operated 2758 hours at 1450°F indicated temperature. This is equivalent to more than the 15,000 hour design life of the 630A fuel. It is estimated that approximately 2 percent of the fuel surface will experience burnup and temperature conditions approaching the maximum perturbations.

ORF-4

Test ORF-4 was terminated after 4800 hours at an indicated temperature of 1300°F. This is equivalent to approximately 15,000 hours of full power 630A operation. Burnup of the U^{235} atoms in this test is estimated to be about 40 percent. A relatively small percentage of the fuel is expected to be exposed to a combination of high burnup and high temperature. The average burnup throughout all the fuel is estimated to be 23 percent. The fuel sheet specimen, EB-18, was removed from the capsule at GE-NMPO for post-irradiation examination.

During operation, a temperature spread developed in data from the five thermocouples attached directly to the specimen backup plate. This trend can be seen in Figure 12.1, a plot of principal thermocouple data throughout the test. The sequence of fission product trapping is also shown in Figure 12.1. Table 12.1 presents the results of trap analyses.

TABLE 12.1
RESULTS OF ORF-4 TRAP ANALYSES

Trap No.	End Of Trapping Period ^a		Detected Percent Of Fission Products Generated				
	Test Time, hr	Trap Time, hr	I ¹³¹	I ¹³³	Xe ¹³³	Xe ¹³⁵	Ba-La ¹⁴⁰
12 ^b	2061	17.2	0.012	ND ^c	0.0799	0.1	0.01
13	2082	19.3	0.000963	ND	0.0339	ND	0.00892
14	2216	5.5	0.0157	ND	0.0903	0.00663	ND
15	Continuous warm trap - gamma scanned only						
16	2383	5.4	0.0187	0.0834	0.0659	0.0605	ND
17	2562	6.3	0.0110	ND	0.0881	ND	ND
18	2707	6.6	0.00023	ND	0.00370	ND	ND
19	2881	6.9	0.00014	ND	0.0034	ND	ND
20	3048	4.5	0.00024	ND	0.578	ND	ND
21	3186	7.2	0.00042	ND	ND	ND	ND
22	3220	17.0	0.00070	ND	0.0011	ND	ND
23	3401	5.5	0.00098	ND	0.0831	ND	ND
24	3707	5.6	0.0002	ND	0.059	ND	ND
25	4013	22	0.0002	ND	0.0002	ND	ND
26	4191	19.7	0.0002	ND	1.08	ND	ND
27	4340	6.1	0.0001	ND	0.014	ND	0.0002
28	4643	2.5	0.0008	ND	0.009	ND	ND
29	4691	8.0	0.0003	ND	0.002	ND	ND
30	4755	6.5	0.0003	ND	0.0014	ND	ND
31	4774	1.6	0.0017	ND	6.83	ND	ND
32 ^d	Cool-down	21	0.0001	ND	0.98	ND	ND

^aSee Figure 12.1 for trapping sequence and operating temperature.

^bNo fission products detected in traps 1 through 11.

^cNot detected.

^dOperated during post-test cool-down of capsule.

Post-test visual examination revealed specimen EB-18 had developed a definite swelling, particularly near its midpoint. The cladding, however, is apparently intact and contains no definite cracks, pinholes, or other visual defects. Figure 12.2 illustrates the condition of the specimen after irradiation. Evaluation will continue and will include metallography and burnup analyses.

PROPERTY STUDIES

The laboratory test program for the evaluation of certain properties of 80Ni - 20Cr alloy has a twofold purpose: (1) to initially assess the suitability of 80Ni - 20Cr as a fuel sheet material for 630A application and (2) to provide base line data for judging the degree of improvement attained by advancing the technology of 80Ni - 20Cr fuel sheet preparations. A number of long-term tests is in progress to determine creep properties of 80Ni - 20Cr fuel sheet and cladding stock and the resistance of this alloy to oxidation in air and superheated steam.

Creep Tests of Warm-Finished Standard Fuel Sheet

Test times in excess of 12,000 hours have been accumulated in the 630A operational stress-level creep tests. Plastic creep has not exceeded 0.75 percent, in 2-inch gage length, in any specimen. Table 12.2 presents the results obtained to date.

Principle thermocouple locations
on specimen EB-18 and capsule

All fission product traps operated iced to -40°F except
long time traps numbered 2 and 15 which were ambient
temperature. See Table 12.1 of this report for analysis.

GE-NMPO

Fig. 12.1 - Plot of principle thermocouple readings, reactor operations and
fission product trap data for in-pile test ORF-4

Fig. 12.2—Oblique view of air-cooled cladding surface of 80Ni-20Cr fuel sheet specimen EB-18 after 4800 hours at 1300°F indicated temperature during in-pile test ORF-4. Specimen shows some swelling. Total burnup is estimated to be 40 percent of the U^{235} atoms. (RML-2203L)

TABLE 12.2

LOW-STRESS CREEP TEST DATA ON WARM-FINISHED STANDARD 80Ni - 20Cr FUEL SHEET

Specimen	Temperature, °F	Stress, psi	Total Time, hr	Cycles	Elongation, %		Elapsed Time At Last Measurement, hr
					In 2 in.	In 6 in.	
<u>Isothermal Tests</u>							
C-1	1600	300	12,364	-	0.60	0.51	11,936
C-2	1450	300	12,482	-	0.30	0.22	12,054
C-4	1300	300	12,022	-	0.20	nil	11,594
B-2	1300	300	11,017	-	-	nil	10,589
C-8	1450	600	11,989	-	0.25	0.23	11,561
W-2	1450	600	9,907	-	-	0.12	9,479
<u>Thermal Cycle Tests</u>							
C-3	1600	300	11,984	209	0.75	0.66	11,556
C-5	1300	300	12,030	210	nil	nil	11,602
B-1	1300	300	11,305	199	-	nil	10,877
C-6	1450	600	11,419	202	0.35	0.17	10,991
C-7	1450	300	12,014	209	0.40	nil	11,586

Creep Tests of Experimental Fuel Sheet

Creep tests are continuing on recently produced warm- and cold-finished fuel sheet with stresses an order of magnitude greater than anticipated for 630A operation. Initial tests at various combinations of stress and temperature showed warm-finished sheet generally superior to cold-finished sheet in 5 percent creep, 1000-hour parameter tests.* Tests on several specimens have been continued beyond 1000 hours; the results are shown in Figure 12.3. The data indicate the ability of this fuel sheet to withstand temperature and stress excursions for several thousand hours with less than 5 percent creep.

Cladding Stock Creep Tests

Three specimens of 80Ni - 20Cr cladding stock surpassed the 15,000 hour milestone and were removed from test. Figure 12.4 presents the status of the six original specimens. Three new specimens have been added to duplicate those removed, but data are not yet available. The new specimens will be tested until second-stage creep rate is well established.

The metallographic and microscopic evaluation of the specimens removed from test is reported in the following section.

Oxidation Studies of 80Ni - 20Cr Alloy

No change was observed in the integrity of the fuel sheet cladding during the long-term creep tests (see Table 12.2). No blistering nor increase in alpha count was detected.

Disc specimens of 80Ni - 20Cr from a series included at representative temperatures with the long-term fuel sheet creep specimens were removed from test for oxidation evaluation after 10,000 hours. Metallographic cross sections of 0.016-inch-thick discs showing typical oxidation effects are shown in Figure 12.5. Penetration has been less than 0.002 inch at 1600°F.

Oxidation penetration measurements were obtained from creep specimen RD-3 tested more than 15,000 hours at 1750°F, in excess of 630A operational temperature,† stressed to 300 psi. Figure 12.6 shows photomicrographs of the specimen taken at the midpoint of the gage sections. Metal loss was 0.002 inch from the original 0.030 inch thickness. Average oxide stringer penetration was 0.004 inch with maximum penetration to 0.008 inch. Internal oxidation was not identified, nor has positive identification been made of a second phase faintly visible throughout the central area of the strip. A similar microstructure in the back-plate of specimen EB-6‡ after 7055 hours at 1600°F was determined to be a Cr-rich phase by electron microprobe analysis.

Bend tests were performed on transverse and longitudinal segments of specimen RD-3 adjacent to the metallographic sections in the gage length. Both segments were fully ductile through a complete U-bend over a 1/16-inch radius, equivalent to approximately 50 percent elongation of the outer fibers.

Figure 12.7 shows both surfaces at the 2-inch gage length of cladding stock specimens RD-1 and RD-2 which were recently removed from test (see Figure 12.4). Both specimens were tested at temperatures above the 630A maximum operational temperature. The surface condition of RD-1 is a typical gray-green high-temperature oxide. Specimen RD-2, however, shows the occurrence of spalling, a characteristic not generally observed. Metallographic evaluation will be performed.

*"630A Maritime Nuclear Steam Generator Progress Report No. 5," GE-NMPO, GEMP-215, June 28, 1963, pp. 139-141.

†The maximum anticipated 630A fuel element operating temperature was exceeded by 300°F, since the 1300° to 1450°F range had not been defined at the time these tests originated.

‡"630A Maritime Nuclear Steam Generator Progress Report No. 6," GE-NMPO, GEMP-257, Figure 15.15, p. 122.

Fig. 12.3—High-stress creep-rupture test results on experimental warm- and cold-finished 80Ni–20Cr fuel sheet

Fig. 12.4 - Low-stress creep data on 0.030-inch-thick 80Ni - 20Cr cladding stock

Neg. 4016-A, 100 X, UNETCHED
1300°F

Neg. 4016-B, 100 X, UNETCHED
1450°F

Neg. 4018, 100 X, UNETCHED
1600°F
ISOTHERMAL

Neg. 4017, 100 X, UNETCHED
1600°F
THERMAL CYCLE

Fig. 12.5—Photomicrographs showing cross sections of 0.016-inch-thick 80Ni-20Cr discs exposed for 10,000 hours to still air at the indicated temperatures

Neg. 3890, 100 X, ETCHED

MODIFIED MARBLES ETCH

Section at mid-point of gage length. Metal loss was 0.001 inch from each surface with 0.004 inch additional intergranular oxide penetration. Grain structure did not change significantly from original cold-worked condition.

Neg. 3891, 100 X, ETCHED

MODIFIED MARBLES ETCH

Section from cold end of specimen showing original microstructure.

Fig. 12.6—Photomicrograph of cladding stock creep test specimen RD-3 after 15,240 hours at 1750°F and 300 psi stress. Total plastic creep was 2.3 percent in 2 inches.

~1.5 X

RD-1 CREEP TESTED AT 1750°F AND 300 PSI
FOR 15,730 HOURS

~1.5 X

RD-2 CREEP TESTED AT 1600°F AND 600 PSI
FOR 15,500 HOURS

Fig. 12.7—Macro photographs showing surfaces of gage lengths of 80Ni–20Cr cladding stock specimens RD-1 and RD-2 (Neg. P64-2-16D)

Steam Corrosion Tests of 80Ni - 20Cr

A summary of the weight changes and surface metal loss measurements from specimens exposed for 2400 hours to superheated steam at 1020°F and 3000 psig and 5000 psig pressures are presented in Table 12.3. This completes this test program at GE-HAPO.

Results of tests conducted at GE-NMPO in 1350°F superheated steam are presented in Table 12.4. The corrosion effect on 80Ni - 20Cr alloy appears minimal on an overall weight change basis. Measurements of uniform metal loss indicate less than 0.001 inch with less than 0.0005 inch intergranular penetration.

WORK PLANNED FOR NEXT QUARTER

Post-irradiation analysis will be conducted on test capsule ORF-4. Creep, oxidation, steam corrosion, and water corrosion studies will be continued.

TABLE 12.3
WEIGHT CHANGE AND SURFACE METAL LOSS DATA^a FOR
80Ni - 20Cr AND OTHER ALLOYS TESTED FOR 2400 HOURS
IN DEOXYGENATED STEAM SUPERHEATED TO 1020°F

Alloy	Preparation	Weight Change, mg/dm ²		Metal Loss, mils/side	
		3000 psig	5000 psig	3000 psig	5000 psig
80Ni - 20Cr (Nb)	Pre-oxidized	-	5 ^b	-	-
Lot X-1	Bright annealed	44	59	0.04	0.06
80Ni - 20Cr (Nb)	Pre-oxidized	-	16 ^b	-	-
Lot F-317	Bright annealed	-	74	-	0.08
80Ni - 20Cr	Pre-oxidized	-	2 ^b	-	-
Unmodified	Bright annealed	-	61	-	0.05
Hastelloy X	Annealed - pickled	9	9	0.03	0.01
446 SS	Annealed - pickled	9	42	0.02	0.07
Incoloy 800	Annealed - pickled	10	11	0.01	0.01
Inconel X	Annealed - pickled	12	14	0.02	0.02
PDRL-102	Annealed - pickled	38	13	0.06	0.03
17-4-PH	Annealed - pickled	56	49	0.10	0.13
Hastelloy N	Annealed - pickled	59	35	-	0.04
316L SS	Annealed - pickled	231	195	-	0.20
304L SS	Annealed - pickled	277	184	0.34	0.14
430 SS	Annealed - pickled	280	277	0.31	0.33
406 SS	Annealed - pickled	335	185	0.33	-

^aTests conducted at GE-HAPO.

^bShowed negative weight change initially (-8, -5, and -12, respectively).

TABLE 12.4
GENERAL CORROSION DATA^a FOR ALLOYS
EXPOSED TO 730°C SUPERHEATED STEAM AT 3 ATMOSPHERES PRESSURE

Alloy	Preparation	Total Test	Total Weight	Weight Change Rate,
		Time, hr	Change, mg/dm ²	mg/dm ² -100 hr
80Ni - 20Cr	A-P ^b	2536	45	1.8
(Nb modified)	A-VB ^c	2108	27	1.3
Hastelloy X	A-P	1257	51	4.0
	A-VB	1257	37	2.9
Inconel 600	A-P	1257	269	21.3
	A-VB	1257	71	5.6
Inconel 702	A-P	1257	266	21.1
	A-VB	1257	135	10.7
Incoloy 800	A-P	1257	139	11.0
	A-VB	1257	33	2.6
NMO-2541	A-P	2536	34	1.3
	A-VB	2108	29	1.4
NMO-2541 (tubing)	Brite Ann'L	1257	5	0.4
NMO-1541	A-VB	1257	49	3.9
304 SS	A-P	2536	6 ^d	.d
	A-VB	2108	65	3.1
	A-P	1257	149	11.2
	A-VB	1257	40	3.2
446 SS	A-VB	500	28	5.5

^aTests conducted at GE-NMPO.

^bAnnealed - pickled.

^cAnnealed - vapor-blasted.

^dAverage of two specimens having individually +358 and -346 mg/dm² total weight change.

13. SAFETY ANALYSIS

A computational logic diagram was prepared for the digital computer startup excursion analysis.

An investigation was started to determine if the analog computer equipment capabilities at GE-NMPO are adequate to perform certain specified safety analyses.

TRANSIENT ANALYSIS OF THE STARTUP EXCURSION

A computational logic diagram, similar to the wiring diagram for an analog computer, was prepared for the boiler, reactor, blower, and ducting analytical expressions which will be used in the cold startup excursion analysis. Preparations are underway to run the reactor temperature and pressure drop equations alone on the 7090 computer to simulate a study performed during the preliminary hazards analysis with the core thermal transient program. This will serve to verify that program input data have been prepared properly.

Illustrations of the methods used for reactor temperature equations are shown in Figure 13.1. Identification of the boxes in Figure 13.1 is given in Table 13.1; nomenclature and equations relating to the digital computer simulation are given in Table 13.2.

TABLE 13.1
630A CLOSED LOOP - REACTOR BOX IDENTIFICATION

(Z's are system variables used as box inputs)

Function
Number

- 1 constant input
 - 5 time-dependent input
 - 8 non-linear integrator
 - 9 summer
 - 11 multiplier
 - 13 divider
 - 14 tabular variable dependent on system variable
 - 19 variable to a constant power
-

Fig. 13.1 - Reactor simulation for digital computer safety analyses

TABLE 13.2

IDENTIFICATION OF SYSTEM PARAMETERS FOR REACTOR DIGITAL ANALYSIS

Nomenclature

- A Heat transfer area, ft²
 ΔX Distance from node centroid to surface, ft
 k Thermal conductivity, Btu/sec-ft-°R
 h_{cv} Convective heat transfer coefficient, Btu/sec-ft²-°R
 ρ Density, lbm/ft³
 C_p Specific heat at constant pressure, Btu/lbm-°R
 V Nodal volume, ft³
 W Fluid weight flow, lbm/sec
 D_h Coolant passage hydraulic diameter, ft
 q''' Volumetric heating rate, Btu/sec-ft³
 T Temperature, °R
 z Factor relating fluid node centroid and discharge temperature,
 T_{discharge} = z T_{centroid}
 A_{ff} Cross section flow area for coolant, ft²
 τ Time, sec

Subscripts

- f Reactor fuel
 c Reactor coolant
 n Reactor stage index
 in Inlet conditions
 r Reactor in overall sense

Identities

$$Y \equiv \frac{A}{\frac{\Delta X}{k} + \frac{1}{h_{cv}}}$$

$$B \equiv \rho C_p V$$

$$M \equiv W C_p$$

$$Q''' \equiv q''' V$$

Miscellaneous equations

h_{cv} for air

$$h_{cv} = \frac{1.396 \times 10^{-4}}{D_h^{0.2}} \left(\frac{W}{A_{ff}} \right)^{0.8} \left(\frac{T_{air}^{0.8}}{\left(\frac{T_{air} + T_{wall}}{2} \right)^{0.56}} \right)^{0.56}$$

$$z_{c,n} = 2 - \frac{z_{c,n-1} T_{c,n-1}}{T_{c,n}}$$

$$\bar{z}_{c,o} T_{c,o} \equiv T_{c,in}$$

$$q''' = q'''(\tau)$$

Lumped parameter heat balances (based on centroid temperatures)Fuel

$$Y_{r,n} (T_{c,n} - T_{f,n}) + Q'''_{f,n} - B_{f,n} \frac{dT_{f,n}}{d\tau} = 0$$

Coolant fluid

$$Y_{r,n} (T_{f,n} - T_{c,n}) + \frac{M_{c,n}}{2} (z_{c,n-1} T_{c,n-1} - z_{c,n} T_{c,n}) = 0$$

where it has been assumed that $B_{c,n} \ll B_{f,n}$

Computational forms of heat balance equationsFuel

$$B_{f,n} \frac{dT_{f,n}}{d\tau} + Y_{r,n} T_{f,n} - Y_{r,n} T_{c,n} - Q'''_{f,n} = 0$$

Coolant fluid

$$T_{c,n} = \frac{Y_{r,n} T_{f,n} + \frac{M_{c,n}}{2} z_{c,n-1} T_{c,n-1}}{Y_{r,n} + z_{c,n} \frac{M_{c,n}}{2}}$$

ANALOG COMPUTER SAFETY SIMULATION

To investigate the dynamic response of the 630A power plant during postulated accidents resulting from equipment failure, it is necessary to define all the interacting power, flow, pressure, and temperature processes with appropriate equations. The major classification of these equations and their principal interconnections are shown in Figure 13.2. Control system equations include both normal controllers used during operation and the safety system which initiates a scram. Reactor power, the coolant pressure, and flow determine the thermodynamic state of the coolant and the temperature of the fuel element. Temperature of the coolant and fuel channel then, in turn, determine the pressure drop flow and reactivity changes.

Fig. 13.2 – Classification of analytical expressions for analog simulation of the 630A.

The relationships shown in Figure 13.2 apply to accidents in which the reactor coolant (compressed air) and the steam or water from the moderator or boiler do not mix in the primary loop. Mixing of the fluids will result if a moderator structure failure or boiler breach occurs. When these latter accidents occur, there are obviously more equations and interconnections between the primary loop and the boiler or moderator loops. These interconnections will result in more complicated diagrams for these failures.

Many of the postulated accidents can be investigated by an analog simulation represented by Figure 13.2, or parts thereof.

With limited analog computer equipment, these accidents will be investigated first, and, then later, the more difficult boiler breach problem will be considered.

Startup accidents can be simulated with the least investment in analog computer equipment. Direct breaches to the containment vessel require an intermediate number of equations, maps, and empirical relations and hence, equipment. Finally, the boiler breach to the primary loop requires rather elaborate thermodynamic tables, graphs, and complex mixed flow relationships. With certain assumptions, it may also be possible to represent these accidents with the present analog computer facility.

Work is continuing on the analysis of several postulated accidents to determine whether the associated processes can be adequately simulated on analog computers and, if so, to establish analog computer requirements. Reactor kinetics were simulated for the startup excursion.

WORK PLANNED FOR NEXT QUARTER

Digital and analog simulation of the 630A and analyses of various transient conditions will be continued.

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