ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

DESIGN, ANALYSIS, AND CONSTRUCTION
OF EBR-II PRIMARY TANK
AND ASSOCIATED STRUCTURES

by

R. W. Seidensticker

Quality Assurance Management*

LEGAL NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

September 1970

*Work performed while a member of the Reactor Engineering Division.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT	9
I. INTRODUCTION	9
II. DESIGN	14
A. Primary Tank	17
1. Material Specifications	17
2. Cover	20
3. Walls	28
4. Bottoms	29
5. Insulation	31
B. Support Structure	32
1. Load Analysis	33
2. Materials	36
3. Top Structure	37
4. Columns	38
5. Bottom Beams	38
C. Air-baffle Tank, Blast Shield, and Biological Shield	39
1. Air-baffle Tank	39
2. Blast Shield	40
3. Radial Biological Shield	41
III. FABRICATION	42
A. Primary Tank	42
1. Cover	42
2. Walls and Bottoms	47
B. Support Structure	47
1. Top Structure	47
2. Columns and Bottom Beams	48
IV. ERECTION	49
APPENDIXES	
A. Structural Analysis of Primary Tank	58
1. Cover Structure	58
2. Walls and Bottoms	63

TABLE OF CONTENTS

	<u>Page</u>
B. Structural Analysis of Primary-tank Support Structure . . .	64
1. General Discussion	64
2. Central-ring Analysis	67
3. Stresses in a Circular Ring with Six Equally Spaced Radial Forces	69
C. Thermal Design of Primary Tank.	73
1. Nomenclature for Appendix C	73
2. Introduction	73
3. Cover Thermal Features	74
4. Details of Cover Calculations	77
5. Other Primary-tank Thermal-design Considerations. . .	84
D. Analysis of Thermal Deformations of Primary Tank	87
1. Basic Thermal-deflection Equations.	87
2. Effects of 12-in.-deep Isothermal Beam Stiffeners	89
E. Elevated-temperature Tests	94
1. Dry-heat Test	94
2. The Second Test.	96
ACKNOWLEDGMENTS	98
REFERENCES	99

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1.	EBR-II Plant Arrangement.	10
2.	Reactor Plant	11
3.	Primary System.	12
4.	Primary Tank and Associated Structure.	16
5.	Primary-tank Cover	21
6.	Primary-tank Assembly	23
7.	Primary-tank Hanger Assembly	24
8.	Carbon-steel Plates in Cover Half of Primary Tank.	27
9.	Bottoms of Primary-tank Vessel.	30
10.	Primary-tank Support Structure	33
11.	Pressure-vessel Concept.	34
12.	Blast Shield and Typical Column Detail	40
13.	Model of Primary-tank Cover without Nozzles.	43
14.	Model of Primary-tank Cover with Nozzles.	43
15.	Cover Halves during Fabrication.	44
16.	Machining One Cover Half at Part Line	44
17.	View of Completed Cover Half	45
18.	Closeup View of Completed Cover Half.	46
19.	View of Cover Half Loaded on Rail Car	46
20.	View of T-1 Columns in Place	50
21.	Construction of Biological Shield.	50
22.	Blast-shield Cavity.	51
23.	Inner-vessel Bottom under Construction.	51
24.	Moving Cover Half into Reactor Building	51
25.	Overall View of Erection Work in Reactor Building	52
26.	Lowering Inner Vessel into Outer Vessel	53
27.	Raising of Tank Cover.	53
28.	View of Fully Welded Primary Tank	55
29.	Completing Outer-vessel Insulation.	55

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
30.	Tank Suspended from Temporary Supports.	56
31.	Moving T-1 Central Ring into Reactor Building.	56
32.	Temporary Assembly of T-1 Top Structure	56
33.	Completed T-1 Top Structure, Ready for Lowering	57
34.	Transfer of Primary Tank to T-1 Top Structure	57
A.1.	Normal Loading of Radial Beams of Primary-tank Cover . . .	59
A.2.	Moment Loading of Central Ring of Primary-tank Cover . . .	61
B.1.	Loading and Bending Moment for Typical T-1 Frame for 75-psig Internal Pressure.	64
B.2.	Holddown Arrangement for Rotating Plugs	65
B.3.	Typical Details of T-1 Beam and Column.	66
B.4.	Detail of T-1 Beam-to-Ring Connection.	68
B.5.	Tensile Radial Loads on T-1 Ring Transmitted from Radial Beam	69
B.6.	Circular Ring under Action of Two Equal and Opposite Radial Loads	70
B.7.	Circular Ring under Action of Six Equally Spaced Radial Loads	71
C.1.	Temperatures in Unmodified Primary-tank Cover.	77
C.2.	Additional Temperatures in Unmodified Primary-tank Cover	77
C.3.	Two-node Heat-transfer System.	78
C.4.	Example of Application of Two-node System.	79
C.5.	Temperature Difference across Various Structural Members vs Thickness of Carbon Steel Installed against Each Side of Member	80
C.6.	Temperatures in Typical Nozzle	81
C.7.	Temperatures in Auxiliary Pump-bus Nozzle	81
C.8.	Temperatures in Safety-rod Nozzle	82
C.9.	Temperatures in Shutdown-cooler Nozzle	82
C.10.	Model for Including Radiation across a Gap	83

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
C.11.	Model for Calculating Baffle Spacing Requirement.	84
C.12.	Temperature Change of Flowing Sodium, Δt_c , vs Gap Width, a.	85
D.1.	Thermal Stresses in Radial Beams of Primary-tank Cover. .	89

LIST OF TABLES

<u>No.</u>	<u>Title</u>	<u>Page</u>
A.1.	Load Analysis for Primary Tank	58
A.2.	Section Moduli for Cover Radial Beams	60
C.1.	Comparison of Results for 49-node and 2-node Systems.	79
C.2.	Airflow Requirements for Cooled Nozzles	82

DESIGN, ANALYSIS, AND CONSTRUCTION OF EBR-II PRIMARY TANK AND ASSOCIATED STRUCTURES

by

R. W. Seidensticker

ABSTRACT

The entire EBR-II primary coolant system and fuel-handling system are submerged in about 80,000 gal of sodium at 700°F, all contained in a large double-walled vessel (primary tank). This report describes in detail the design, fabrication, and erection of the primary tank and its support structure. Special features of the design peculiar to this system are emphasized, particularly the fabrication and erection of the several large machined weldments employed in the system. Design philosophy for the structural system is developed, and typical design calculations presented. The thermal deformations of the primary tank at elevated temperatures are treated theoretically. Elevated temperature tests performed both before and after filling the tank with sodium are described, and test results summarized. The thermal design of the primary tank is discussed.

I. INTRODUCTION

The Experimental Breeder Reactor II (EBR-II) is an unmoderated, heterogeneous, sodium-cooled reactor and power plant with a power output of 62.5 MWt. The energy produced in the reactor is converted to 20 MWe through a conventional steam cycle. The reactor is fueled with ^{235}U or plutonium, and the plant includes an integral fuel-processing facility where the irradiated fuel is processed, fabricated, and assembled for return to the reactor.¹ The EBR-II facility (shown in Fig. 1) is located at the National Reactor Testing Station in Idaho.

The reactor, primary coolant system, and associated equipment are contained in the reactor containment building, a gas-tight cylindrical steel shell (as shown in Fig. 2). A unique feature of this plant is that the reactor and the entire primary coolant system, including pumps, heat exchangers, piping, etc., as well as the fuel-handling system components, are contained in a large double-walled stainless steel vessel--the primary tank--and operate completely submerged in sodium (as shown in Fig. 3). Since all

Fig. 1. EBR-II Plant Arrangement. ANL Neg. No. 111-7032.

Fig. 2. Reactor Plant. ANL Neg. No. 111-6388.

Fig. 3. Primary System. ANL Neg. No. 111-6453.

penetrations into the primary tank are located in the top cover of the primary tank (well above the level of the bulk sodium), this "submerged concept" leads to a system that exhibits a high degree of reliability of coolant containment.

The primary tank, its support structure, blast shield, biological shield, and shield cooling system, as shown in Fig. 3, comprise a complex, integrated structural system. The design of many components in this system was strongly influenced by requirements of fabrication and erection, particularly with respect to the actual sequence of field erection. This report describes the design and construction (and subsequent testing) of the primary tank and support structure, which work was carried out from 1957 to 1961.

Successful operation of this structural system was verified prior to wet criticality by two elevated temperature tests. The first test was performed before the tank was filled with sodium and was carried out at about 350°F. Subsequent tests, which occurred after the tank was filled with sodium, were made with the sodium at 650°F. (Normal operating temperature of bulk sodium is 700°F.) In both tests, the thermal expansions of the tank (radially outward) occurred in the predicted uniform fashion, thus assuring concentricity of the primary tank and its support structures. A complete discussion of the conduct of these tests and results are included in Appendix E of this report. Subsequent power operation of the EBR-II reactor system has verified the reliability of this structural system design.

II. DESIGN

The primary tank, support structure, blast shield, biological shield, and the shield-cooling system comprise an integrated structural system. Figure 3 indicates that a full understanding of this complex system requires a rather complete description of each portion of the system and a discussion of the interrelationship of each part to the other. The main purpose of this report is to discuss the primary tank and its support structure. The design of certain other items, however, such as the blast shield and biological shield are closely related to the tank and structure design, and these items are described in subsequent sections of this report.

The basic design objectives of the primary tank and support structure are to: (1) meet static-load requirements; (2) maintain accuracy of alignment of the equipment located in the primary system at various elevated temperature levels; (3) contain the effect of an internal energy release due to a nuclear accident; and (4) provide high reliability of containment of the sodium coolant.

The temperature of the 80,000 gal of bulk sodium in the primary tank is normally 700°F, although it will occasionally vary from 250 to 750°F. Submerged in the bulk sodium is the entire primary cooling system, the fuel-handling system, and various other pieces of equipment and instrumentation.

Operating personnel must have access to the top of the primary-tank support structure (operating-floor level), thus requiring near room temperatures at the top of the nozzles in the primary-tank cover. In addition, all equipment penetrations into the primary tank are located in the tank cover to provide maximum reliability of the bulk-sodium coolant containment. This then results in a difficult design problem associated with the large temperature difference across the top structure (about 600°F), the requirement of accurate positioning of equipment, and a rather limited amount of space for structure available (due to the exceptionally high number of penetrations).

The structural system employed (shown in Fig. 3) resolves this problem by separating the upper structure into two regions. The lower section of the structure--the primary-tank cover--is maintained at the higher temperature and provides support and alignment for equipment that must communicate internally with the primary tank, such as pumps, heat exchanger, and instrument thimbles. The upper section--the top support structure--is maintained at the lower temperature and provides the support for the primary tank and the upper biological shield and, from a central ring in the top support structure, provides support of the rotating plugs. The rotating plugs, in turn, provide support for critical mechanisms such as the control-rod drives and the gripper mechanisms used to insert and remove fuel subassemblies from the reactor.

The primary tank and its support structure are completely independent of each other at all sides except the top. The primary tank and its contents, and those components supported by the tank cover, are supported by hangers (welded to radial beams in the tank cover) which bear on high-strength steel rollers supported by the top support structure. As shown in Fig. 3, the main structure of the top support structure consists of six radial beams, spaced equally at 60° intervals, and a large central ring. The main structure of the primary-tank cover is essentially the same, and in plan view the two "spiderlike" structures are coincident.

The rollers are provided with close-fitting flanges to avoid lateral movement. In addition, lateral movement at the hangers is restricted by thrust-bearing plates mounted on the sides of the top structure beams. Thus, during an increase in temperature of the primary tank, the six rollers move radially outward to accommodate thermal expansion. Since the tank is suspended, vertical thermal expansion is accommodated by providing sufficient clearance below the bottom of the primary tank.

Figure 4 shows the spiderlike structure of the top support structure (the same type of structure in the tank cover lying directly below it), and also shows clearly the interrelationship of the many structural components in the system--the hanger, the separation of the tank and its support structure, and the blast and biological shields. This figure also shows the relationship of the rotating plugs. The rotating plugs are supported by the central ring of the support structure, while the primary tank gas space (above the sodium) is sealed at the rotating plugs by a low-melting-point alloy "freeze seal," which is located on the top of the Z-ring that forms an integral part of the primary-tank cover. (There are two rotating plugs--a large one and a small one; the large plug rotates on ball bearings supported by the top structure, while the small plug--eccentric to the large plug--rotates on ball bearings supported by the large plug.)

The overall design philosophy of the entire primary structure system may be summarized as follows: (1) The primary tank design must accommodate very high static loads at a relatively high temperature; (2) the dead-load deflections and nonuniform thermal deformations of the various structures of the tank (cover and bottoms) must be quite low to minimize equipment alignment problems; (3) the primary tank, support structure, and all other associated structure must be designed with due consideration given to the very difficult field erection problems associated with this structural complex; (4) the design must provide for uniform radial thermal expansion of the primary tank about the vertical centerline of the system; and (5) although the structures involved are massive, the final erection-tolerance requirements must be stringent, and measures must be taken at each phase of fabrication and erection to attain the desired structural system.

Fig. 4. Primary Tank and Ass

sociated Structure

2

The remainder of this section is devoted to detailed discussion of the design of each main component of the system. Section III will then discuss fabrication of these parts, and finally Section IV will describe in detail the stepwise erection of this system.

A. Primary Tank

The primary tank (see Figs. 3 and 4) is of double-wall construction (a vessel within a vessel) to provide maximum reliability of sodium containment. The tank is constructed entirely of Type 304 stainless steel and is of all-welded construction. The tank may be considered as consisting of two vessels, completely independent of each other, suspended from a common head (primary tank cover). The inner vessel has a 26-ft ID and a 1/2-in. sidewall thickness, and the outer vessel has a 26-ft 11-in. ID and a 1/4-in. sidewall thickness. The bottoms of each vessel consist of heavier-gauge plates, reinforced with built-up beams to carry the heavy loads imposed on them. The primary-tank cover is common to both the inner and outer vessels and is the only physical connection between the two tanks. The cover consists of a heavy spiderlike structure of six radial beams and a central ring, numerous stiffener webs, and a heavy bottom plate to which all the nozzles (used to support various items of equipment) are welded. The cover is suspended from the top support structure by a system of six hangers and rollers. This suspension and "floating" support eliminates, both radially and longitudinally, expansion stresses due to the differences in the temperatures of the cover and top support structures. For shielding purposes, the cover is filled to a depth of about 26 in. with carbon steel balls (3/8 in. in diameter). The overall height of the primary tank (top of cover to bottom of outer vessel) is approximately 30 ft 4 in.

The entire primary tank is insulated to minimize heat losses from the 700°F bulk sodium. Numerous thermocouples are located throughout the tank to monitor temperatures of the sodium, the inert-gas blanket above the sodium, the shells of the inner and outer vessels, and the cover.

1. Material Specifications

Because of the large amount of welding, hot-forming, and subsequent stress-relieving of the primary-tank cover, low-carbon stainless steel, Type 304L was specified for this structure. Although this material was not required for the remainder of the primary tank, it was, however, considered desirable to obtain a low carbon content and to control the ratio of chromium content to nickel content. (This latter requirement was intended to enhance weldability and to obtain good mechanical properties.) To attain these objectives, we used the following specifications for all Type 304 stainless steel, except the cover:

"All plate, strip and sheet steel material used in the fabrication of the inner and outer vessels of the primary tank shall be of

Type 304 stainless steel and shall conform to the requirements of ASTM Specification A240-57T. The ratio of chromium content to nickel content for all Type 304 stainless steel shall be a minimum of 1.90 and a maximum of 2.20. The maximum carbon content shall be 0.08 percent with 0.04 to 0.06 percent preferred.*

"In addition to the A240 Specification, the following statement shall be attached to all Type 304 SST Material Specifications:

"The maximum carbon content shall be 0.08 percent, with 0.04 to 0.06 percent preferred, as determined by both a ladle and check analysis, and no process shall be used on the metal which will raise the carbon analysis, particularly in the surface of the metal. All heating and annealing shall be done in an atmosphere which cannot introduce additional carbon in the metal. Cobalt and tantalum are undesirable constituents and shall be minimized.' "

Working in close cooperation with the primary-tank fabricator enabled all steel used to meet the desired requirements.

The importance of the primary tank prompted extensive measures designed to guard against the possibility of poor weld quality. The following weld specifications were imposed on the fabricator and were adhered to:

"1. Electrodes for Manual Welding

"All weld rod material used in the fabrication of the inner and outer vessels of the Primary Tank, including the field welded joints between the two vessels and the Primary Tank cover, shall meet the requirements of Table Q-11.3 weld metal No. A-7 of Section IX, 1956 Edition of the ASME "Welding Qualifications Code," and shall be in accordance with ASTM Specifications A298-55T, Type 308 series. The weld rod metal used shall be in accordance with ASME Filler Metal Specification, Type 308-15 (with lime coating). The weld rod metal used in the field assembly of the Primary Tank cover shall also meet the above requirements. All weld rod metal used in all the above operations shall have a free ferrite content in the range of 4% minimum to 10% maximum.

"For each lot or separate heat of welding electrodes shipped, the Primary Tank Subcontractor shall obtain from the electrode manufacturer certified test reports for chemical analysis and physical properties. These tests shall conform with the requirements of ASTM Specification A298-55T,

*Although the 0.04-0.06 carbon limit was not a firm requirement, all plate material furnished was in this desired range.

Type 308 series. Copies of these test reports shall be submitted to the Laboratory for approval prior to the use of any of the electrodes in a given lot or heat.

"In addition to the above tests, the electrode manufacturer shall be required to prepare for each lot and heat of electrodes, a composite 12 in. x 12 in. x 1 in. test plate (prepared by joining two 12 in. x 6 in. x 1 in. test plates of the same composition as the material specified for the Primary Tank) containing a single vee butt weld made with electrodes from that lot or separate heat, for the purpose of determining the chemical composition and mechanical properties of deposited weld metal. Standard 0.505 in. dia. x 2 in. gage length tensile specimens consisting entirely of deposited weld metal from this test plate shall develop the minimum tensile strength of the base plate at atmospheric temperature (75,000 psi min.) and at 700 F (55,000 psi min.). Failure of these all-welded metal tensile specimens to develop minimum ambient and elevated temperature tensile properties shall disqualify the lot or heat of electrodes. Records of all sources of manual electrodes, lot number, chemical analyses, and results from test plates shall be transmitted to the Laboratory for approval before incorporation of electrodes into weldments.

"Upon receipt of each lot of electrodes by the Primary Tank Subcontractor at his shop, or in the field, three or more weld rods shall be picked at random from each weld rod container, or package, of approximately 50 lbs. Chemical welding test specimens shall be produced and analyzed from the selected samples in accordance with ASTM Specification A298-55T, Type 308 series. Reports of these weld rod analyses shall be submitted for approval to the Laboratory prior to the use of the weld rod material from each container, or package.*

"2. Electrodes for Machine Welding: Semi-automatic, Automatic

"All weld wire used for machine welding shall have a free ferrite content in the range of 4% minimum to 10% maximum, and shall produce deposited weld metal whose composition is approximately the same as that specified above for plate material. Upon receipt of each lot of welding wire by the Primary Tank Subcontractor at his shop, or in the field, a sample shall be cut from one end of each continuous roll for chemical analysis.

*The manufacture of the welding electrodes was witnessed by a Laboratory representative and containers packed with these electrodes were marked by this representative at the time the packages were sealed. No packages of electrodes were used that did not bear this distinguishing mark.

Reports of these analyses shall be submitted for approval to the Laboratory prior to the use of the roll of wire for welding in the Primary Tank.

"After approval by the Laboratory of the proposed process and procedure for machine welding is granted, the Primary Tank Subcontractor shall prepare composite test plates at least 12 in. long x 12 in. wide x 1 in. thick (prepared by joining at least 12 in. long x 6 in. wide x 1 in. thick test plates of the same composition as the material specified for the Primary Tank) containing a single vee butt weld, on each machine for each lot of separate heat of wire procured for use by a given machine, for the purpose of determining the chemical composition and mechanical properties of deposited weld metal. Standard 0.505 in. x 2 in. gage tensile specimens consisting entirely of deposited weld metal from these test plates shall develop the minimum tensile strength of the base plate at atmospheric temperature (75,000 psi min.) and at 700 F (55,000 psi min.). Failure of these all-weld metal tensile specimens to develop the minimum ambient and elevated temperature properties shall disqualify the lot or heat of wire. Records of all sources of wire, lot numbers, chemical analyses, and results from test plates shall be transmitted to the Laboratory for approval before incorporation of the wire into weldments.

"Welding wire which has been disqualified may be re-qualified after corrective measures in welding technique are instituted. Any such measures which involve changes in process or procedure are subject to re-approval by the Laboratory.

"3. Deposited Weld Metal

"Weld metal as deposited during actual construction from manual, or machine, electrodes should preferably contain approximately 5% of free ferrite as calculated from the chemical composition of the deposited weld metal in accordance with the Schaeffler Constitution Diagram for Stainless Steel Weld Metal. The Primary Tank Subcontractor shall use all reasonable means to produce welds which approach this composition as nearly as possible."

2. Cover

The primary-tank cover is an accurately machined structural weldment, which provides support for the various pieces of equipment that enter the primary tank and for the inner and outer vessels. Virtually the entire volume of the cover is filled with 3/8-in.-dia carbon steel balls, which serve as radiation shielding.

The basic structure of the cover consists of a 2-in.-thick bottom plate, numerous web-plate stiffeners, a central circular ring (2 in. thick and 9 ft in diameter), and six radial beams, which frame into the central ring and from which the entire primary tank and its contents are supported. Numerous stepped nozzles of various sizes welded to the cover are used to support the equipment that enters the primary tank. Figure 5 is a photograph of the cover taken inside the reactor building during the primary-tank erection contract. The overall size of the cover (excluding the nozzles and hangers) is 2 ft 8 $\frac{3}{4}$ in. in height and 27 ft in diameter. The assembled cover, without any equipment and shielding, weighs approximately 90 tons.

Fig. 5. Primary-tank Cover

Because of its size, the cover had to be fabricated and shipped in two halves. The parting joint was located along the center of one line of radial beams. These diametrically opposed beams were fabricated as channels, oriented back-to-back, and later assembled by bolting and welding in the field to form a built-up I-beam section. This interesting fabrication and erection job is described in Sections III and IV of this report.

All equipment penetrating the primary-tank cover (except for those items located in the rotating plugs) is supported from the cover and

extended down into the primary tank. Each piece of equipment is secured to a stainless steel nozzle (about 7 ft long), which is welded at its bottom to the 2-in. bottom plate of the cover and extends up through, but is not fastened to, the top support structure (see Fig. 4). Carbon steel sleeves provide sufficient clearance in the top support structure around each nozzle to allow for the radial thermal expansion of the primary-tank cover. Each penetration includes a shield plug, and the entire unit fits into its respective stepped nozzle and is gasketed and bolted to the top flange of the nozzle.

To provide sufficient strength and stiffness in the cover to support equipment and the shielding balls, the spaces in the cover between radial beams is literally laced with vertical web-plate stiffeners (see Fig. 6). These stiffeners carry superimposed loads either directly to the radial beams or to the inner- and outer-ring members of the cover. (To reduce heat losses from the cover, these stiffeners were not run to the top of the cover, but terminated about 4 in. below that point.) As shown in Fig. 6, the outer-ring "girder" is, in reality, a vertical extension of the walls of the inner and outer vessels. A continuous steel plate is placed across the top of the cover. This plate supports the insulation on the cover and, in addition, provides some stiffness.

The central ring assembly consists of two parts. The first is the basic ring, a "boxlike" structure, consisting of a 2-in.-thick, 9-ft-dia circular ring, a series of six vertical web plates arranged in a hexagonal array (see plan view in Fig. 6), and portions of the 2-in. bottom plate and the top cover plate. These pieces, when assembled, form a continuous, closed box-ring girder, which possesses very high torsional resistance and stiffness, and high bending strength and stiffness. The main function of this portion of the central ring, then, is to provide the necessary continuity between opposing radial beams, acting very much as the hub of a spoked wheel when loaded normal to its plane. The second part of the central ring is a Z-shaped, 10-ft-dia ring, approximately 21 in. high. The Z-ring supports the liquid-metal seal trough used to effect a gastight seal between the large rotating plug and the primary tank (see Fig. 4). Although not of a closed-form cross section, the Z-ring adds significantly to the stiffness of the cover. (Since the Z-ring cannot be installed on the cover until the tank is suspended from the top support structure, it does not appear in Fig. 5.)

The entire primary tank is supported by transmitting the entire load of about 900 tons from the six radial beams in the tank cover to the six radial beams in the top support structure (directly above) through six stainless steel hangers (see Figs. 3 and 4). These hangers are welded to both the bottom 2-in. plate and the radial beams in the cover and extend from the tank cover up to the top flange of the radial beams in the top support structure. The hangers are 12 ft from the vertical centerline of the tank. One-sixth of the total load, or about 150 tons, is transmitted from

Fig. 6. Primary-tank

k Assembly

each hanger to each support structure radial beam by a yoke arrangement (Fig. 7), which bears on a high-strength alloy steel roller, which in turn bears on the beam web in the top support structure. This roller support system eliminates high stresses which would occur because of the large temperature difference between the cover and the top support structure (about 600°F). Reasonably equal distribution of the total load was achieved by accurately machined shim plates at each hanger assembly.

Fig. 7. Primary-tank Hanger Assembly

Although the roller supports are carefully machined and installed, the possibility exists for some horizontal loadings on the cover, e.g., seismic effects and reactions from piping entering the primary tank.

To prevent undesirable tangential movements of the cover on the rollers, a pair of aluminum-bronze guide-bearing blocks are located on the web of the support structure beam and the hanger straps (see Fig. 7). Any spreading tendency of the hanger straps is prevented by a tie bolt (located near the top of the hanger) which passes through a clearance hole in the support beam web.

The roller and the lower-support roller-bearing block are made of "Thermold J," a special high-strength steel-alloy material made by the Universal-Cyclops Steel Corporation. The upper-support roller-bearing block is of "T-1" constructional alloy steel and has a dovetailed insert of Thermold J which bears on the roller. Thermold J was selected for its high strength (285,000 psi ultimate tensile strength) and hardness (Rockwell C-56). T-1 was selected for the upper block for its high allowable stress, combined with good ductibility. (T-1 constructional alloy was used for the entire primary-tank support structure; its physical and mechanical properties are given in Section II.B.1.)

Structurally, two diametrically opposed radial beams, their respective hangers, and the continuity afforded by the central ring in the cover comprise, in effect, a simply supported structural beam with overhangs at each end. The exact location of the hanger--12 ft from the vertical centerline of the tank--was selected to minimize the resulting bending moments in the beam and the vertical deflection of the cover at the central ring. The heavy loads transmitted to the cover by the inner and outer vessels are applied "outboard" of the hangers; the majority of the loads carried by the cover (shielding and equipment) are located within the "hanger circle."

The reactor design group desired the maximum dead-load deflection of the cover (at the central ring) to be as small as possible. Since the loads applied at the overhanging ends of the radial beams were high, it was possible to set a design value of 1/16-in. maximum deflection of the radial beam and the central ring. The maximum allowable value of bending stress plus thermal stress was 14,700 psi (Section VIII of the ASME Boiler Code at that time). Because of the very strict requirement on deflection (deflection-to-span ratio of about 1 to 4600), the calculated dead-load bending stresses in the radial beams are quite low. The maximum bending moment due to the mechanical loads occurs at the hangers in the radial beams and results in bending stresses of 4830-psi tension (in top flange) and 3300-psi compression (bottom flange).

Meeting the requirement of very low deflections due to mechanical loads is especially important because of thermal deformations in the cover which are generated by an unavoidable temperature drop across the cover structure. The mechanical deflections and thermal "bowing" deflections are additive within the hanger circle, hence the desire to minimize

mechanical deformation.* (A more complete analysis of the thermal deformations--both theoretical and experimental--is presented in the appendixes of this report.)

The normal operating temperature of the primary-tank bulk sodium is 700°F. Under this condition, a significant heat loss occurs through the tank cover and its associated nozzles to the cooling-system air for the biological shield. This heat loss is accompanied by production of a temperature differential across the tank-cover structure of such sign as to produce downward bowing of the cover. Such bowing, in turn, tilts the cover nozzles so that the lower ends of mechanisms extending through them into the primary tank move slightly radially outward. For many nozzles, such movement is clearly of no consequence. However, if the movement were sufficiently large, certain components suspended from nozzles (or from the bottom plate of the cover), might be undesirably affected from the standpoint of alignment or of increase in stress level.

Several features are incorporated into the cover design to lessen both the temperature drop through the cover and the undesirable deformation effects of this drop:

a. To minimize the temperature drop across the cover, the following design features are used:

1) The thickness of the thermal insulation on top of the cover was increased from 6 in. of Foamglas to 8 in. of Foamglas.

2) Carbon steel grit, similar to that used in "metal-blast" finishing processes was added to the 3/8-in.-dia carbon steel balls. This feature is intended to increase the thermal conductivity of the steel ball matrix; it also leads to a higher specific weight of the shielding mixture, which is in itself desirable.

3) "Loose," 1-in.-thick carbon-steel plates were installed--on edge--in the tank cover around the main structural members and some of the larger nozzles. These serve to increase thermal conductivity locally at those points where the structural behavior of the cover is controlled. Figure 8 shows the placement of these plates in one of the primary-tank cover halves.

b. To lessen the undesirable effects of the resulting temperature drop across the cover, 12-in.-deep stiffening beams are welded on the underside of the 2-in. bottom plate of the cover, directly below each of the six radial beams. Since these stiffeners are completely immersed in the argon-gas blanket above the bulk sodium, they will be essentially isothermal.

*This, in general, need not be the case for alternate designs; adjusting hanger locations would make it possible to have thermal deformations partly offset the mechanical deflections.

Fig. 8. Carbon-steel Plates in Cover Half of Primary Ta

mary Tank (other half similar)

2

They act, therefore, as "strong-back" stiffeners on the cover, resisting the downward bowing. The appendixes of this report include an evaluation of the effectiveness of these stiffeners and of the possible thermal stresses induced in the radial beams (due to the restriction on bowing).

c. In addition to the measures employed above, electrical strip heaters are placed on the top flanges of the radial beams and near the top of the six "hex-ribs" in the central ring assembly. It is not intended that these heaters ever be used. However, if the actual deflections (or rotations of nozzles) in the cover were to exceed desirable limits, the heaters would be used to increase the temperature of the top of the beams, thereby reducing the effective ΔT across the cover structure. To guard against heater failures, an excess of heaters were installed. In addition the heaters, designed for use with a 220-V power source, are connected across a 110-V line, are able to be placed in several combinations of series and parallel arrangements, and will operate at very low watt densities. (See Appendix C for a more complete discussion of thermal design.)

3. Walls

The shells, or walls, of the inner and outer vessels of the primary tank act as tension members transmitting loads carried by the vessel bottoms up to the tank cover. Since the bulk sodium coolant is not pressurized, the walls need only be designed for a hydrostatic loading plus the imposed dead loads.*

The inner-vessel wall has a 26-ft ID and a 1/2-in. thickness, and the outer-vessel wall has a 26-ft 11-in. ID and a 1/4 in. thickness. The outer-vessel wall is stiffened with three circumferential rings to prevent excessive "ovaling" of this shell, particularly during erection. These rings also serve to support the vessel-wall insulation. All joints in the walls are full-penetration double-butt welds.

At the joint to the cover and to the bottom structures, the wall thicknesses are gradually increased to meet the heavier-gauge plates on the bottom and to provide increased bending resistance at the shell to cover joints.

The inner-vessel wall is designed to carry the entire sodium load, reactor and neutron shielding, and the inner-vessel bottom structure. The uniform dead-load tensile stress under these loads is only about 2100 psi. At the joint between the inner-vessel shell and the cover, there exist local bending stresses due, for the most part, to rotation of the cover structure caused by the temperature drop across the cover. The maximum calculated value for these bending stresses in the inner vessel is about

*Although the blanket-gas pressure is normally slightly negative, the vessel can accommodate a few psig increase in this blanket-gas pressure, if this increase should ever be required.

8200 psi for a ΔT across the cover of 175°F. Thus, the maximum combined stress (direct and bending) in the inner vessel wall is approximately 10,300 psi, which is still well below the design value for membrane stress alone at 750°F permitted by the ASME Code (14,700 psi).

The outer-vessel wall is designed to carry the weight of its bottom structure and the entire sodium load in the unlikely event of a leak developing in the inner vessel. Under the full sodium load, the uniform direct tensile stress in the 1/4-in. wall is about 2800 psi. As for the inner-vessel wall, the cover rotations induce local bending stresses in the outer-vessel wall at the cover joint. These are calculated to be about 5700 psi for a cover ΔT of 175°F. The maximum combined stress in the outer vessel is then 8500 psi, again well below the allowable value for membrane stress alone.*

To preserve a reasonably constant thickness of the annuli between the inner- and outer-vessel walls and between the primary-tank wall insulation and the shield-cooling air-baffle tank (see Fig. 4), the specified tolerances for erection on both vessels was $\pm 1/4$ in. on the radii. In addition, the tolerances on the total peripheral, or circumferential, lengths of the shell walls at the juncture with the cover were specified at $\pm 1/4$ in. This latter requirement was necessitated by the fact that the short skirt pieces, forming the transitions from the cover to the 1/4- and 1/2-in. shell walls were installed by the cover fabricator, and the shell walls were fabricated and erected by another contractor.

The calculated allowable external pressure at 700°F on the inner- and outer-vessel shells, are 1.8 and 0.25 psi, respectively. Since a sodium head exists on the inner-vessel shell, the actual safe external pressure for the inner vessel is somewhat greater than 1.8 psi.

4. Bottoms

The bottoms of both the inner and outer vessels consist of heavy-gauge steel plates, reinforced with welded built-up beams to carry the heavy loads imposed on them (Fig. 9).

The inner-vessel bottom consists of a $1\frac{1}{2}$ -in. plate and knuckle piece reinforced with eight parallel beams, about 21 in. deep. Across the tops of these eight beams and forming a common top flange for them is a 3/4-in. plate about $12\frac{1}{2}$ ft wide. Four-in.-dia holes are randomly located in this plate for natural convection of the sodium. Plug-welded to the top of the 3/4-in. plate is a circular leveling plate, $1\frac{1}{2}$ in. thick and 11 ft 10 in. in diameter, forming the support base for the reactor vessel. Shallower

*During the safety review, a hypothetical accident was assumed consisting of complete failure of the inner vessel. The outer vessel was shown to be safe even though the loads were dropped some 5 in. onto it.

Fig. 9. Bottoms of Pri

beams and ribs perpendicular to the main set of eight beams serve mainly to stiffen the $1\frac{1}{2}$ -in. bottom plate to carry the sodium load. Drain-port holes at the bottom of the web stiffeners permit complete pumping out of the sodium, if this should ever be necessary.

Although not a part of the primary tank, the safety-rod support beam and borated-steel plates (neutron shielding below the reactor) are installed during the erection of the primary tank, since the space into which they fit is inaccessible after completion of the tank. These items are indicated in Fig. 9.

The inner-vessel bottom structure supports the full load imposed upon it (approximately 500 tons) with a maximum deflection of $1/4$ in. at a point along the main stiffening beams 3 ft 6 in. from the centerline of the primary tank at a metal temperature of 750°F. This distance is approximately equal to the radius of the reactor vessel, which in turn supports the fuel and blanket subassemblies. Since support of the reactor vessel is along its bottom flange, at a radius of 3 ft 6 in., the amount by which the reactor vessel deflects downward is that deflection in the main eight beams at the same radial location. This deflection was established to minimize misalignment between the reactor and the primary-tank support structure. The maximum allowable combined direct and bending stress was chosen as 14,700 psi. The calculated stresses in the inner-vessel bottom structure are below this value at all locations.

The outer-vessel bottom is of sandwich-plate construction. It consists of two $3/4$ -in. flat plates separated by a series of parallel rib beam stiffeners. The structure is designed to make the top $3/4$ -in. plate and outer vessel wall leaktight. This then minimizes the drop in liquid level if the sodium should ever leak from the inner vessel. This leaktight seal is made with a continuous angle welded to the plate and wall. To prevent pressure buildup, the space between the two plates is vented by a long vertical tube into the annulus between the two vessels.

The outer-vessel bottom structure is designed to support, in addition to its own weight, the uniformly distributed sodium load (320 tons) with an allowable stress in the plates and beams of 14,700 psi. The calculated central (maximum) deflection is $1\frac{1}{4}$ in. This maintains an adequate space between the bottom of the primary tank and the bottom of the air-baffle tank (Fig. 4).

5. Insulation

The entire primary tank is well insulated to minimize heat loss from the system and to minimize effects of temperature drops through the structures. The insulation on the bottom of the outer vessel consists of two 3-in. layers of 18 x 24-in. Foamglas blocks. The first layer of blocks is secured to the outer-vessel bottom plate with stainless steel finger straps

which are spot-welded to the tank. The second layer is fastened to the first layer with stainless steel staples. Joints are staggered both within and between layers.

The two layers of insulation are further supported by a stainless steel pan (1/8 in. thick) stiffened with 16 radial angles. The pan is suspended from the primary tank with 48 studs welded to the bottom of the outer vessel. Each stud fits through a slotted hole in the pan and angle to permit thermal expansion.

The insulation on the sidewall of the tank consists of two 3-in. layers of 12 x 18-in. Foamglas blocks, preformed to fit the curved shell plates. Each layer of blocks is secured with spring-type straps spaced vertically at about 9 in. Flat crown staples are used to prevent the straps from creeping up or down. To reduce attrition between the two layers (as the tank expands and contracts), the first layer of blocks is wrapped with a glass cloth, fastened to the blocks with staples. To contain any "dusting" of the blocks, the second (outer) layer is wrapped with sheets of aluminum foil 0.006 in. thick, over which is wrapped a jacket of 22-gauge Type 304 stainless steel. The joints in the stainless steel jacket are overlapped and interlocked to provide reasonably tight joints. The steel sheets are secured by spring-type straps.

The insulation of the top of the primary tank consists of two 4-in. layers of Foamglas blocks (18 x 24 in.). Considerable cutting and fitting of the blocks around the various nozzles and hangers were required. Glass cloth was placed between layers, for the same reason given for the sidewall insulation. Aluminum foil and a 22-gauge stainless sheet were placed on top of the insulation. A 1-in. radial gap around each nozzle was provided in the insulation and sheets. These gaps were calked with stainless steel wool (commercial fine grade). An expansion joint at the junction of the wall insulation and the top insulation was made of this same stainless steel wool. Split stainless steel collars were placed over the annular gaps around each nozzle and welded to the nozzles.

B. Support Structure

The EBR-II primary-tank support structure consists of a system of columns and beams which transmit the loads incurred on the structure to the main internal building foundation (as shown in Fig. 10). The support structure, in combination with the concrete biological shield, forms a "pressure vessel" surrounding the primary tank, which contains the reactor and all the primary system components. The wall of the "pressure vessel" is formed by the reinforced-concrete radial biological shield lined on its inner surface with a "blast shield." The top closure is formed by the top structure of the primary-tank support structure and the top biological shield. The bottom "closure" of the "pressure vessel" is the internal

building foundation, which is a large mass of concrete, reinforced with steel bars to act as a pressure-vessel head. The top and bottom closures are tied together by six peripherally positioned columns. Figure 11 illustrates the pressure-vessel concept.

Fig. 10. Primary-tank Support Structure. ANL Neg. No. 111-6377.

Since the primary tank contains approximately 640,000 lb of sodium at 700°F, it is imperative that the primary tank support structure contain this sodium in the event of a postulated explosive-type nuclear incident in order to avert the secondary, and undoubtedly more severe, effect of the sodium-oxygen reaction in the Reactor Building. As a consequence of this requirement, the structure is designed not in the conventional manner of proportioning members to resist static gravity loads, but to provide sufficient strength to safely resist high pressure of short duration and to provide a blast-resistant structure in order to preserve the integrity of the "pressure-vessel" enclosure. The combination of close fabrication and erection tolerances and high structural performance under severe impulsive loadings sets this structure apart from the conventional built-up welded structural components normally used in buildings and bridges.

1. Load Analysis

The primary-tank support structure, as shown in Fig. 11, in conjunction with the biological shield and the bottom concrete closure, is designed as a primary container to withstand the energy released in a nuclear accident. It is this function of the primary-tank structure that leads to the "pressure-vessel" concept for the analysis and design of the primary-tank structure.

concept

2

The design of the support structure was based on a nuclear-energy release within the reactor core equivalent to the detonation of 300 lb of TNT. The effects of the 300-lb TNT explosion were analyzed by the Armour Research Foundation under contract with the Laboratory.

The results of this study show that the peak pressure that would be exerted on the top of the support structure by the sodium spray is approximately 25 psig and would exist for approximately 2 msec.

An analysis of the effects of this impulse on the structure would undoubtedly be quite difficult, and many approximations were necessary to arrive at the 25 psig pressure. Consequently, it was decided to choose a value of a steady-state pressure such that no concern be shown to the danger of failure of the primary-tank support structure under the assumed accident.

Consideration of all the uncertainty factors led to a choice of a 75-psig pressure exerted over the entire surface of the top structure. Although this design load appears to lead to an extremely heavy structure, the great depth ($6\frac{1}{2}$ ft) of the structure is dictated by the structural stiffness and biological-shield depth requirements imposed on the structure, aside from internal-pressure considerations. The actual design strength of the top "closure" is much greater than that which would be required for an internal pressure of 75 psig. The stresses associated with normal operations are quite low in the T-1 structure.

The analysis of the 300-lb TNT detonation shows that the peak pressures on the radial biological shield would be much less than the peak pressure on the top structure. This is attributed to the 2-ft-thick radial blast shield, consisting of laminations of mild steel plates, lightweight and aerated concrete, and "Celotex" (see Fig. 4). If high pressures were allowed to occur laterally on the T-1 columns, it would be extremely difficult to provide sufficient structural strength in these columns. Therefore, the radial biological-shield concrete alone is designed to withstand a 75-psig pressure.

No blast shield is used at either the top or the bottom of the system. The argon-gas space directly above the sodium coolant level in the primary tank attenuates the high pressures, while no concern need be shown to blast damage to the bottom "closure," which in reality is a large mass of concrete in contact with the rock foundation. The bottom structure is designed for a 75-psig internal pressure.

In addition to the assumed steady-state pressure of 75 psig, consideration was given to the possibility of high pressures being transferred from the bulk sodium to the top structure through the various shafts and pipes penetrating through the top structure down into the primary tank and reactor vessel, where very high peak pressures of short duration would be present. The structure was first designed to resist the 75 psig allowing

margin for the other loads, and then the effects of the impulsive type loadings--resulting from the transmission of high pressure forces through the shafts to the top structure--were analyzed. Extremely conservative assumptions were made concerning the latter effects, and it was found that the structure is quite adequate for the assumed energy release, and is probably sufficient for an energy release several times that which was used in the analysis.

2. Materials

a. Plate Material. The plates used for fabrication of the primary-tank support structure are of T-1 constructional alloy steel having the following chemical composition:

<u>Element</u>	<u>Composition, wt %</u>
Carbon	0.10-0.20
Manganese	0.60-1.00
Phosphorous	0.040 (max)
Sulfur	0.050 (max)
Silicon	0.15-0.35
Nickel	0.70-1.00
Chromium	0.40-0.80
Molybdenum	0.40-0.60
Vanadium	0.03-0.10
Copper	0.15-0.50
Boron	0.002-0.006

The T-1 plates have the following mechanical properties for all gauges used in the structure:

Yield strength	90,000 psi, minimum
Tensile strength	105,000 psi, minimum
Elongation in 2 in.	17%, minimum
Reduction of area	50%, minimum

The T-1 plates were furnished in the following qualities:

- (1) Regular quality for all plates except those items listed in (2) below.
- (2) Firebox quality for all sections requiring cold bending and forming, such as, but not limited to:
 - (a) Upper (curved) built-up channels;
 - (b) Central ring components

b. High-tensile-strength Bolts. High-tensile-strength bolts were specified to conform to the following:

Bolt, nut, and washer material shall conform to ASTM Specification A-325, latest revision.

Bolt and nut dimensions: ASA Designation B18.2.

Washer dimensions: ASA Designation B27.2.

3. Top Structure

The main structure of the top closure consists of six radial beams framing into a 12-ft-dia central ring (which provides support for the large, or outer, rotating plug) and a $27\frac{1}{2}$ -ft-dia channel-ring beam. A 1-in.-thick continuous steel plate is incorporated across the bottom of the entire top structure. This structure is crisscrossed with numerous vertical plate webs which stiffen the 1-in. bottom plate in carrying the imposed loads. A sectionalized, removable steel floor plate, resting on top of the radial beams, provides for personnel access around the top of the primary system.

The lower 3 ft of the top closure is filled with heavy concrete (about 280-300 lb/cu ft) to form part of the top biological shield. Carbon-steel sleeves are placed around each nozzle and are located slightly eccentric (at the time of installation) to allow for thermal expansion of the tank cover. The tank-cover nozzles terminate about 2 in. above the top of the carbon-steel sleeves.

Each of the six main radial beams is $6\frac{1}{2}$ ft deep and has $1\frac{1}{2}$ -in.-thick webs and 3 by 14-in. flanges. The central ring consists of a 3-in.-thick cylinder with several circular stiffeners and a 5-in.-thick insert cylinder, which has a machined step used to support the large rotating-plug ball-bearing race. The webs and bottom flanges of the radial beams terminate at the inner face of the T-1 columns; the top extends clear across the depth of the column sections. Cutouts and support pads are provided at a radius of 12 ft in the radial beams to receive the primary tank hangers. As indicated in Fig. 4, a splice joint about 2 ft from the central ring is made in each radial beam; these are for shipping purposes and are fully welded during erection of this structure. The entire structure is of all-welded construction, full-strength electrodes (E11016) being used for all splice joints, and E7015 electrodes being used for flange-to-web welds in the beams. The joints between radial beams and columns are full-penetration double-fillet reinforced welds and are intended to provide full joint "fixity."

The machined step in the central ring assembly was limited to the following tolerances:

- a. The diameter of the vertical wall of the step to be a minimum of 138.000 in. and a maximum of 138.030 in.

- b. The maximum ovality of the vertical wall of the step to be 0.005 in.
- c. The horizontal surface of the step not to deviate from a true plane by more than 0.005 in.

An allowable stress of 45,000 psi (combined stress, including bending) was employed for both the top structure and all other T-1 members of the primary-tank support structure.

4. Columns

The six T-1 columns, spaced equally at 60°, transfer the end reactions of the six radial beams under static loads down to the concrete foundation just below the primary tank (see Fig. 4). The columns act as tension members, however, in resisting effects of a nuclear accident, tying the top and bottom "pressure-vessel" closures together.

The overall length of each column is 41 ft and consists of a $\frac{1}{2}$ -in.-thick web and 4-by 14-in. flange plates. The flange-to-web welds are similar to those used in the radial beams.

To prevent significant lateral pressure loading (during a nuclear accident) on the columns, each column is recessed 2 in. from the front of the biological shield, as shown in Fig. 1. Also, to minimize the effects of any movement of the biological shield reflecting in lateral loads on the columns, void spaces in the concrete are provided on the inboard sides of the column flanges. In addition, low bond is ensured by enamel painting of the steel columns before concrete placement.

The columns are fabricated with an integral base plate, $2\frac{1}{2}$ in. thick, 2 ft 2 in. x 4 ft in plan size. The columns are then erected as complete units on a 2-in. bed of nonshrink grout. Each column is secured to the foundation with four $\frac{1}{2}$ -in. anchor bolts. The tolerance on the overall length of each column is $\pm 1/8$ in.

Since the top-structure "spider" radial beams must fit in between the hanger straps on the primary-tank cover, the erection tolerances on the columns are critical. Specified tolerances on the plan-view location of the tops of the columns were a maximum deviation from theoretical position of a $1/2$ in. This then limits the tolerance on the radial beams at the hangers to about $1/4$ in. In addition, the top of each column was specified to be at its given elevation, plus 0, minus $1/2$ in. to permit accurate leveling of the top structure.

5. Bottom Beams

The lower parts of the six vertical T-1 columns are connected together with six horizontal channel beams, welded to the outer flanges of

the columns, and arranged in a hexagonal pattern (as shown in Fig. 10). These six beams transfer the load imposed on the enclosed reinforced concrete hexagonal slab below the primary tank (see Fig. 11) to the columns.

As for the top "closure," the bottom "closure" (the hexagonal concrete slab) is designed for 75 psig. Two rows of bars at right angles to each other, conforming to the shape of the bottom head of the steel containment vessel, are used to reinforce the hexagonal slab. Vertical bar hangers hooked over the horizontal channels extend to the bottom of the slab, where they are bent inward. The entire system, including the six beams, the hanger bars, and the two rows of reinforcement, forms a convenient structural configuration similar to that of an ordinary basket.

The steel channels have an overall length of 19 ft and a depth of $2\frac{1}{2}$ ft. Each channel consists of a 2-in. webplate and 2- x 7-in. flanges welded together to form a continuous structural member.

An 8-in. layer of refractory concrete is cast on top of the regular concrete slab below the primary tank. This is then covered with a $1\frac{1}{2}$ -in. carbon-steel plate (welded to structural tees embedded in the concrete). The refractory concrete is used to insulate the heavy mass of concrete below the primary tank and to absorb a large fraction of the temperature drop in this slab. The steel plate--in conjunction with the innermost steel shell of the blast shield--serves as another barrier for sodium in the very unlikely event that both primary-tank vessels should leak.

C. Air-baffle Tank, Blast Shield, and Biological Shield

The remaining parts of the primary system structure--the air-baffle tank, the blast shield, and the biological shield--form a series of nested concentric cylinders surrounding the primary tank. Although these items do not serve as structural support for the primary tank, each has a definite purpose related to the tank and its support structure and, taken together, forms an integral part of the erection of the primary system structure. For these reasons, each of these component parts is described briefly in this report.

1. Air-baffle Tank

The sole function of the air-baffle tank is to divide the flow of the shield cooling air. As shown in Fig. 4, this open-top tank is suspended from the top of the blast shield and consists simply of a $1\frac{1}{4}$ -in. shell and a flat bottom reinforced with 6-in.-deep beams. The entire tank is fabricated of Type A-7 steel and is of all-welded construction. The bottom of the tank has a 5-ft-dia opening at its center, covered with an orifice plate drilled with several holes.

There are 36 shield-cooling radial ducts located around the periphery of the biological shield, offset to prevent radiation streaming. Alternate ducts terminate at the inner face of the blast shield, and the other ducts extend through the 3-in. gap between the blast shield and the air-baffle tank and are welded to this tank. Air entering from the supply ducts flows downward along the blast shield, down under the bottom of the air-baffle tank, and toward the center of the tank. The flow then is up through the 5-ft orifice plate, radially outward across the bottom of the primary tank, upward along the sides of the primary tank, and finally is exhausted out of the radial exhaust ducts welded to the air-baffle tank. The top of the primary tank is cooled with room air, drawn down through the several annular gaps in the top support structure at equipment nozzles. The air then flows outward across the top of the primary tank and mixes with the air flowing upward along the tank walls.

2. Blast Shield

The inner surface of the radial biological shield is lined with a $26\frac{3}{4}$ -in.-thick annular blast shield. This shield protects the radial biological shield from impulsive loadings generated from a nuclear accident and enables absorption of an appreciably large fraction of the total nuclear-energy release. The design of the shield is apparent from Fig. 12. It is a laminated structure employing three layers of absorbing material, viz.,

Fig. 12
Blast Shield and Typical Column
Detail. ANL Neg. No. 111-6366.

an $8\frac{3}{4}$ -in. thickness of 22-lb/ft³ vermiculite concrete, an $8\frac{3}{4}$ -in. thickness of 16.5-lb/ft³ aerated concrete, and an 8-in. thickness of standard-weight Celotex. The innermost surface of each layer is lined with a continuous Type A-7 steel cylinder of either 1/2- or 3/8-in. thickness as shown.

Ideally, each lamination of the blast shield attenuates the incident energy (and pressure) to a level for which the next lamination will exhibit maximum efficiency in further attenuating the energy and pressure. The efficiency of any one lamination depends principally on its thickness, density, dynamic crushing strength, and ability to absorb large amounts of energy (through crushing deformation). The material, thickness, and relative position of each lamination within the shield was selected to reduce the effects of the maximum credible accident on the radial biological shield in the most efficient and practicable manner for the design chosen.

3. Radial Biological Shield

The radial biological shield is constructed of 6-ft-thick ordinary concrete. A typical horizontal section through the shield is shown in Fig. 12. The concrete has an ultimate compressive strength of 3000 psi at 28 days.

The concrete shield is reinforced to act, if necessary, similar to a pressure-vessel shell. It is reinforced in such a manner as to allow the shield to resist safely an internal pressure of 75 psig. The estimated resultant pressure beyond the blast shield is only a few psig; however, both the top closure and bottom closure possess inherently greater strength than that required to resist the 75-psig pressure, and it was felt desirable to "build in" reserve strength in the radial shield justified on the basis that it did not cause an appreciable increase in total cost of containment.

The reinforcing bars forming the equivalent "pressure vessel" are spaced as three concentric hoops with a mean radius of 21 ft 9 in. Reinforcement for the vertical direction is approximately one-half in cross-sectional area, of that employed for hoop steel. All bars are of intermediate-grade billet steel.

III. FABRICATION

Although many of the items comprising the primary-system structure are of conventional fabrication, certain components, notably the primary-tank cover and the top-support-structure spider, involved unique and difficult fabrication problems. The salient features of the more interesting aspects of fabrication are described briefly here with a view toward a more complete understanding of the overall complexity of this type of structural system chosen for the submerged-primary-system concept.

A. Primary Tank

Because of the size and complexity of the primary-tank cover, the Laboratory had to have the cover fabricated separately from the remainder of the tank. The Rock Island Arsenal in Illinois and, later, the Watertown Arsenal in Massachusetts fabricated the two cover halves complete and shipped them to the site in Idaho. The remaining portions of the primary tank were fabricated by the Chicago Bridge and Iron Company in their Chicago shops. Field erection of the primary tank was by the Chicago Bridge and Iron Company.

1. Cover

Location and alignment are critical for several of the machined nozzles in the primary-tank cover and required accurate positioning and machining in the shop fabrication. In addition, the numerous reinforcing ribs and nozzle guide plates complicated the actual sequence of fabrication. To assist in the planning of the fabrication of the cover, the Central Shops Division of the Laboratory constructed an accurate quarter-size model of the entire cover made of aluminum and wood. Figure 13 is a photograph of the model without any nozzles present, and Fig. 14 shows the completed model. This model proved to be very useful, especially in uncovering difficult and sometimes virtually impossible areas for proper assembly of the many individual pieces. The yoke and pin assemblies at the tops of the hangers are models of the ones used during transfer of the primary tank load from temporary support to the T-1 spider. A full-size wood model of the actual hanger and roller assembly was also constructed to study design refinements and fabrication and erection problems.

The basic structures for each cover half was fabricated at the Rock Island (Illinois) Arsenal and are shown in Fig. 15. The numerous "Swiss-cheese" plates shown were used for the purpose of gross positioning of the nozzles during assembly. The huge 2-in.-thick bottom plates were purchased with the nozzle holes rough cut. A precision theodolite was employed during all phases of assembly to keep close control on any warpage developing during the welding operations.

Fig. 13. Model of Primary-tank Cover without Nozzles. ANL Neg. No. 111-7320

Fig. 14. Model of Primary-tank Cover with Nozzles. ANL Neg. No. 111-7321.

Fig. 15. Cover Halves during Fabrication.
ANL Neg. No. 201-3339.

Upon completion of the basic cover halves, each half was placed in a furnace and a partial stress relief performed at 850°F.* Because of the short time available for fabrication, one half was sent to the Watertown Arsenal (in Massachusetts) for installation of the nozzles and final machining, and the other completed at Rock Island.

Figure 16 is a photograph taken during the facing operation on the mating surface of the split beams. To facilitate field assembly and

Fig. 16. Machining One Cover Half at Part Line

*This temperature was chosen as a value greater than the structure would later experience, but low enough to avoid sensitization of the stainless steel.

ensure a tight fit, the mating surface was prepared to give metal-to-metal contact for a depth of about 6 in. from the top and 6 in. from the bottom of the mating beams.

Although the same template was used by both arsenals in drilling the bolt holes in the mating surfaces at the part line, the risk was too great to completely rely on accurate machining for this crucial field fitup. To solve this problem, the matching bolt holes were drilled about 1/2 in. over-size and stainless steel bushings with eccentric holes drilled through were provided for the fitted bolts. Thus, inserting one bushing in each half of the mating surfaces made it possible to line up the bolt holes in the bushings by merely rotating the bushings within the oversized hole. This feature proved an asset later in the field assembly, which was accomplished quite easily. Dowel-pin holes and scribe marks were used to accurately align the two halves in the field.

The larger nozzles and those not of circular cross section were fabricated of rolled plates. All the remaining nozzles were made by centrifugal casting. The flange of each was machined prior to insertion into the cover assembly. Each nozzle was secured to the cover at the bottom 2-in. plate by extending the nozzle through the plate and welding with a partial penetration groove weld with a fillet reinforcement. Each weld was individually helium leak tested. Figure 17 is a photograph of a completed cover half. The workmen in the right of the picture are completing the helium leak testing of the nozzles. The large circular nozzle at the right is for the primary heat exchanger; the large rectangular nozzle to the left is for one of the two primary pumps. Figure 18 is a closeup view of this cover half with the pump nozzle at the extreme left of the photograph.

Fig. 17. View of Completed Cover Half. ANL Neg. No. 201-3760.

Fig. 30
Tank Suspended from Temporary Supports. ANL Neg. No. ID-103-5773.

Fig. 31
Moving T-1 Central Ring into Reactor Building. ANL Neg. No. ID-103-5777.

Fig. 32. Temporary Assembly of T-1 Top Structure. ANL Neg. No. ID-103-5797.

Fig. 33
Completed T-1 Top Structure,
Ready for Lowering. ANL Neg.
No. ID-103-5803.

Fig. 34. Transfer of Primary Tank to T-1 Top Structure.
ANL Neg. No. ID-103-5809 and -5887.

passing through a threaded hole in the beam top flange. A Watts water-level micrometer was used to check the levelness of the horizontal portion of the machined step in the central ring assembly. Several points along this surface were checked, and the maximum deviation from a perfectly level plane was found to be 0.006 in. (The limits of accuracy on the water micrometer are ± 0.001 in.) The tolerances were checked continuously while the beams were being welded to the columns.

APPENDIX A

Structural Analysis of Primary Tank1. Cover Structure

Two diametrically opposed radial beams, their respective hanger supports, and the continuity afforded by the central ring in the cover comprise, in effect, a simply supported structural beam with overhangs at each end. Table A.1 summarizes the weights comprising the loads on the primary tank. The uniform load for the primary-tank cover was computed on the basis of the total weight of the cover plus all equipment. Admittedly, this is not strictly true, but the complexity of the cover and the assumptions made to analyze the cover are felt to more than justify this approach. Sufficient flexibility of the cover ensures that each hanger receives approximately one-sixth of total load.

TABLE A.1. Load Analysis for Primary Tank

Item	Weight, lb
Cover, including shielding, insulation, and all equipment	924,000
Inner vessel, including sodium, neutron shield, reactor vessel, and subassemblies	1,020,000
Outer vessel, including insulation ^a	72,000
Total load on hangers	2,016,000
Load per hanger	336,000

^aTo account for the unlikely event of sodium leaking from the inner vessel, this load would increase from 70,000 to 710,000 lb, and the inner-vessel load would decrease a like amount.

Figure A.1 shows the loading pattern for the normal case of the sodium weight carried by the inner vessel. All loads are indicated in kip (units of 1000 lb). Because of the circular shape of the cover, the assumed uniform load of the cover appears as a triangular loading. Figure A.1 shows the resulting shear and bending-moment diagrams. The maximum positive moment (compression on top flange) occurs for this loading and is computed as follows by taking moments about the joint between the radial beam and the central ring:

$$\begin{aligned}\Sigma M_0 &= 336 \times 7.5 - 12 \times 9 - 170 \times 8.5 - 0.5 \times 17.2 \times 9 \times 6 \\ &\quad - 8.6 \times 9 \times 4.5 = 161 \text{ kip-ft.}\end{aligned}$$

Fig. A.1. Normal Loading of Radial Beams of Primary-tank Cover

The maximum moment in the beam, which is negative, occurs when the sodium load is carried by the outer vessel (inner vessel leaking) and is located at the center of the hangers. Its magnitude is calculated as follows by taking moments of all loads to the left about the left hanger (total sodium load is 640 kip):

$$\Sigma M_0 = -(12 + 107) \times 1.5 - (170 - 107) \times 1 - 2 \times 1 - 34 \times 0.75 = -270 \text{ kip-ft.}$$

The 12-in.-deep beam stiffeners below the cover start just inside the inner vessel and terminate just before the central ring. Therefore, in calculating the resulting bending stresses at the hanger, we used the section modulus of the stiffened beam, while at the ring we used the section modulus of the unstiffened beam.

Table A.2 summarizes the calculated section moduli for the two locations. The moment of inertia of the unstiffened beam is 10,680 in.⁴, and for the stiffened member is 17,760 in.⁴.

TABLE A.2. Section Moduli for Cover Radial Beams^a

Location	Section Modulus, in. ³
Unstiffened beam, top flange	492
Unstiffened beam, bottom flange	983
Stiffened beam, top flange	670
Stiffened beam, bottom flange	986

^aAssuming that only 14 in. width of the bottom 2-in. plate is effective in the beam section.

The maximum bending stresses, which occur at the hanger with sodium in the outer vessel, are

$$\sigma_{\text{top}} = \frac{270 \text{ kip-ft} \times 12 \text{ in./ft} \times 1000 \text{ lb/kip}}{670 \text{ in.}^3} = 4830 \text{ psi (tension),}$$

and

$$\sigma_{\text{bot}} = \frac{270 \times 12 \times 1000}{986} = 3280 \text{ psi (compression).}$$

The bending stresses in the radial beams at the joint to the ring are

$$\sigma_{\text{top}} = \frac{161 \text{ kip-ft} \times 12 \text{ in./ft} \times 1000 \text{ lb/kip}}{492 \text{ in.}^3} = 3930 \text{ psi (compression),}$$

and

$$\sigma_{\text{bot}} = \frac{161 \times 12 \times 1000}{983} = 1970 \text{ psi (tension).}$$

Although these stresses are low, it is recalled that stiffness of the cover was of prime importance. Furthermore, the presence of the 12-in.-deep "isothermal" stiffeners cause thermal stresses to occur in the radial beams because of their restraining action of cover rotations under a temperature drop across the cover. The value of the thermal stresses are presented in Appendix D, where it is shown that the combined mechanical and thermal stresses are still below the allowable stresses chosen for this structure.

To make the assumption that the central ring does, in fact, afford the necessary continuity between opposing pairs of beams, the ring must be capable of resisting the moments imparted to it at each of the six radial beams. In the analysis of the central ring, it is assumed that no shear forces are transmitted from the beams to the ring. This is tantamount to

assuming that the ring does not carry any of the shielding or equipment load to the cover. Of course, this is not strictly correct, but as pointed out previously, the complicated action of load transfer throughout the cover does not warrant a completely refined analysis. Furthermore, the effects of the six equally spaced moments of 161 kip-ft applied to the ring are predominant in its structural behavior.

The final central ring assembly, consisting of the circular ring, hexagonal ribs, top and bottom plates, and the Z-ring, possesses very large bending and torsional resistance. However, just after the cover is filled with the shielding balls during erection of the primary tank, moments are induced in the radial beams at the central ring of a magnitude almost as large as the final values with all equipment. Also, at the time the shield balls are placed in the cover, the Z-ring is not yet installed. Further, the top closure plates which connect the tops of the hexagonal ribs and circular ring together are also not installed. This latter item renders the hexagonal ribs almost valueless in resisting torsion and the omission of the Z-ring greatly reduces the bending resistance of the ring section. Therefore the most critical loading of the central ring occurs at this time. The bending moment under these loads is calculated as 145 kip-ft and produces compression in the top flange of the beams.

To determine the maximum values of bending moment and torsional moment in the ring requires two free bodies. Since no gravity loads are assumed present on the ring, only the six moments imparted by the radial beams are considered. Figure A.2 shows the moment loading on the ring and the two free bodies required for analyses. All moments are shown as vectors

Fig. A.2. Moment Loading of Central Ring of Primary-tank Cover

directed according to the right-hand rule. Also shown is a cross section of the ring during this loading period. Free body No. 1 is a cut taken along the diameter lying midway between adjacent pairs of the radial beams. Almost by inspection--taking advantage of symmetry--the bending moments at the points in the ring midway between adjacent radial beams are seen to be equal to the value of the applied moment from one radial beam, or 145 kip-ft. It can easily be shown that this is the maximum bending moment (which can also be seen intuitively, since this free body splits the cover in such a fashion as to require the maximum bending resistance from the ring). This section is seen to be free of torsional moment.

Free body No. 2 is a cut taken along the diameter connecting opposing radial beams. Because of symmetry, the applied moment at the cut line is exactly one-half for each half of the ring cut this way. Again--by inspection--the resulting torsional moment is one-half of the applied beam moment, or 73 kip-ft. The bending moment is seen to be equal to 0.866 times the applied beam moment. As the periphery of the ring is traversed, the torsional moment varies from a maximum of 73 kip-ft at the radial beams to zero midway between beams, while the bending moment is seen to vary but little, ranging from 126 kip-ft at the beams to 145 kip-ft midway between beams.

The section modulus of the cross section shown in Fig. A.2 is 658 in.³ for the top flange and 850 in.³ for the bottom flange. (These do not differ much from the values given for the stiffened radial beams.) The maximum bending stresses then are

$$\sigma_{\text{top}} = \frac{145 \text{ kip-ft} \times 12 \text{ in./ft} \times 1000 \text{ lb/kip}}{658} = 2640 \text{ psi (compression)}$$

and

$$\sigma_{\text{bot}} = \frac{145 \times 12 \times 1000}{850} = 2050 \text{ psi (tension)}.$$

The maximum torsional stresses are calculated from³

$$\tau_{\text{max}} = \frac{Tc}{\frac{1}{3} \sum c^3 d} = \frac{Tc}{J_e},$$

where

τ = shear stress, psi,

T = torsional moment, in.-lb,

c = thickness of plate or plates, in.,

d = depth (or width) of plate, in.,

and

J_e = effective polar moment of inertia, in.⁴.

Then

$$J_e = \frac{1}{3} [(1/8)(12) + (8)(31.5) + (8)(12)] = 116 \text{ in.}^4.$$

Maximum shearing stress at center of beam is then

$$\tau_{\max} = \frac{73 \text{ kip-ft} \times \text{in.}/\text{ft} \times 1000 \text{ lb/kip}}{116} = 15,100 \text{ psi.}$$

However, the flanges of the ring are, in fact, infinitely wide at the radial beams, and the maximum shearing stress occurs at the edge of the 14-in.-wide radial-beam flange. The corrected value of maximum shear stress is obtained by reducing the torsional moment by the ratio of one-half the flange width (7 in.) to one-twelfth the perimeter of the 9-ft circular ring (28.2 in.). The corrected value of the maximum shear stress is then

$$t_{\max} = 15,100 - \frac{7}{28.2} (15,100) = 11,300 \text{ psi.}$$

Under this critical loading, the stresses in the ring are seen to be acceptable. The presence of the hexagonal ribs and plates and the Z-ring reduces the stresses given above to much lower values even with the addition of the sodium and all equipment loads. The deflection at the central ring caused by the shielding balls was about 1/16 in., very close to the value calculated using the simply supported beam model.

2. Walls and Bottoms

The stresses in the vessel wall and bottom structure are calculated in a very straightforward manner and are not presented in detail.

Briefly, the calculations for the bottom of the inner vessel were performed by setting the deflections of mutually perpendicular stiffeners equal at the point of intersection. The problem here is similar to the analysis of a grillage of beams and was, in fact, carried out in much the same way. The outer-vessel structure is somewhat simpler, since there are no interacting stiffeners. The beams in this case were analyzed as simply supported beams carrying a uniformly distributed dead load (bottom structure plus sodium load).

Interaction stresses at the junction of each shell to the underside of the cover were found to be quite low. A system such as this size represents, requires setting of shell thicknesses that exhibit reasonable stability during fabrication and erection, rather than stress considerations.

APPENDIX B

Structural Analysis of Primary-tank Support Structure1. General Discussion

Casual inspection of the "pressure vessel" formed by the T-1 support structure and the concrete shields (shown in Fig. 11) makes readily apparent that the bottom closure is the least vulnerable portion of the vessel. In addition to a large quantity of concrete between the bottom of the primary tank cavity and the steel containment vessel directly underneath, the vessel is set on several feet of concrete mass embedded in a lava-rock formation. The bottom closure can be safely considered invulnerable for practically any type of serious structural failure. However, excessive deformation of the material during a nuclear accident may impair the integrity of the steel vessel, which in turn could result in undesirable leaks. To compensate for this possibility, the bottom closure slab is reinforced to carry the 75-psig design pressure and the six T-1 channels designed as simple beams to transmit these loads to the six T-1 columns.

The remainder of the support structure consists basically of the six columns, six radial beams, and the large circular ring assembly to which the radial beams are welded. As for the primary-tank cover, the ring is assumed to provide continuity in the top structure and, therefore, it is possible to analyze the structure as consisting of three interconnected rigid frames. Each frame consists of two columns and radial beams, diametrically opposed, and the continuity afforded by the ring.

Figure B.1 shows the typical frame and the loads imposed upon it by a 75-psig steady-state pressure assumed to exist over the entire bottom surface of the top structure during a nuclear accident. As discussed in Section II.B.3, precautions have been taken to avoid any lateral pressure

Fig. B.1. Loading and Bending Moment for Typical T-1 Frame for 75-psig Internal Pressure

loading on the T-1 columns. The loading on the radial beams consists of the triangular load caused by a uniform load on the sectors including these beams and a pair of concentrated loads of 200 kip each. The concentrated loads are the reactions from the uniform pressure over the bottom surface of the rotating plugs. Figure B.2 shows the holddowns on the plugs that transmit this uniform pressure to the six radial beams. The holddown on the inner plug restrains that plug within the outer plug, and the six holddown keys in the central ring restrain the entire plug assembly against vertical upward loads.

Fig. B.2. Holddown Arrangement for Rotating Plugs.
ANL Neg. No. 112-438 Rev. 1.

Assuming that the circular central ring affords the necessary continuity for the frame, and that its apparent stiffness does not differ appreciably from the stiffness of an equivalent length of a straight beam section, an application of moment distribution methods leads to the bending-moment diagram for the frame, also shown in Fig. B.1.

In the application of the moment-distribution method, the relative stiffnesses of the beams and columns are chosen before proceeding with the indeterminate analysis. The depth of $6\frac{1}{2}$ ft for the top structure was predetermined from consideration of biological shielding and pipe and equipment space between the operating floor and top of the biological shield. Several trial frames were analyzed, the rigid frame shown in Fig. B.1 being selected as the final arrangement.

The frame was analyzed considering both fixed-column bases and hinged-column bases. Since the critical point of the structure appears to be the welded connection of the radial beams to the central ring, the hinged-column analysis was chosen as this leads to the higher bending moment at this connection, its value being approximately 10,700 kip-ft.

Figure B.3 shows the detailed cross section of the typical column and radial beam, and the values of their respective moments of inertia. If the value of 10,700 kip-ft is used as the maximum bending moment in the radial beams, the corresponding maximum bending stresses in the beams are

$$(\sigma_b)_{\max} = \pm \frac{Mc}{I_b} = \frac{(10,700)(12)(39)}{164,700} = \pm 30.4 \text{ kip/sq in.}$$

Fig. B.3

Typical Details of T-1 Beam and Column

The maximum bending moment in the column occurs at the joint between the columns and the radial beams, its value being 2240 kip-ft. The corresponding maximum bending stress is

$$(\sigma_c)_{\max} = \pm \frac{Mc}{I_c} = \pm \frac{(2240)(12)(18)}{31,350} = \pm 15.5 \text{ kip/sq in.}$$

The direct tensile load in each of the columns is 1650 kip. The corresponding direct tensile stress is

$$\sigma_t = \frac{P}{A_c} = \frac{1650}{154} = 10.7 \text{ kip/sq in.}$$

Therefore, the maximum combined tensile stress in the columns at the joint between the columns and the radial beam is

$$\begin{aligned} \sigma_{\max} &= (\sigma_c)_{\max} + \sigma_t \\ &= 15,500 + 10,700 = 26,200 \text{ psi.} \end{aligned}$$

Since the minimum yield stress for the T-1 steel is 90,000 psi at room temperature, it appears that the frame is quite adequate for safely resisting a steady-state pressure of 75 psig.

Neglecting the effects of the additional loads transmitted to the frame by the various shafts penetrating through both the top structure and the primary tank cover, it would seem that a reduction of structural strength would be feasible. However, in addition to being able to safely resist the high internal pressure in the primary system, the support structure must have a low value of central deflection under the normal gravity loadings incurred on the structure. This requirement is necessary so that misalignment of control rods and other associated equipment will be minimized. Using the previously selected sections for the beams and columns, the calculated central deflection of the top structure under normal gravity loads is approximately 0.1 in., a value acceptable to the mechanisms design group.

2. Central-ring Analysis

The main component of the top structure is the large flanged circular ring, $6\frac{1}{2}$ ft deep and approximately 12 ft in diameter, to which the six main radial beams are welded. To be able to make the assumption that the ring affords the necessary continuity for each frame, the bending moments transferred to the ring at each of the six radial beams must be resisted by the ring, much in the same way as for the primary-tank cover. As for the cover-ring analysis it is assumed that no shear forces are transmitted from the beams to the ring. (The six shear keys holding the rotating plugs are located at each radial beam.)

If the cross sections of the radial beams and the ring were rectangular, or shallower, it would be acceptable to consider the central ring as a circular ring subjected to six equally spaced twisting moments, as was assumed for the cover structure. However, because of the deep I-beam cross sections involved, the forces transmitted to the ring from the beams will be heavily concentrated at the top and bottom flanges, and will be much smaller in the webs of the radial beams. To account for the differences in load transfer, the deep ring is divided into several shallower rings, and the effects of the forces transmitted to these rings by the tensile and compressive stress components in bending from the radial beams are analyzed separately for each ring. This approach is conservative inasmuch as the rings do not in fact act independently, but assist each other in resisting the forces imposed by the presence of shearing stresses between two adjacent rings.

The central circular ring has a depth of $6\frac{1}{2}$ ft, of which the lower 3 ft is in direct contact with the heavy concrete biological shielding. Since the lower portion of the ring is essentially under radial compressive forces during the 75-psig internal-pressure loading, the tight-fitting 5-in.-thick ring insert affords more than adequate section to safely resist these compressive forces.

However, the upper portion of the central ring is essentially subjected to radially directed tensile loads under the 75-psig loading, and to resist these forces, several flanges of varying depth, spaced vertically along the ring, were added to the 3-in. ring. Horizontal ribs, spaced to coincide with these ring flanges, are welded to the radial-beam webs. These ribs serve to reduce the maximum stress in the radial beams at the ring joint by increasing the section modulus, and also tend to smooth out the distribution of forces transmitted to the ring. The ribs are wedge-shaped to avoid rapid change in cross-section of the radial beams.

Figure B.4 shows the detail of the radial-beam connection to the central ring. The spacing of the additional flanges on the ring is arbitrarily chosen with consideration made for access in welding operations. Figure B.5 shows the stress distribution in the radial beam at the ring joint. (This is different than for the beams without the "fins" on the web.) Assuming that the tensile forces transmitted to the ring act at the centroids of each corresponding finned section in the beam, the force distribution can be obtained from the areas and the bending stress distribution. The resulting forces on the various flanges of the ring are shown in Fig. B.5.

Considering the six radial tensile forces acting on each flange of the ring independently, it is then only necessary to apply established analytical procedures to determine the maximum combined bending and direct stress in each section. By selecting a suitable allowable stress in the T-1 material, we can establish the necessary depths of the flanges at each prescribed section. These flanges, in fact, act as beam webs for either angle sections or tee sections in the structural action anticipated. A brief derivation of the formulas used to calculate stresses in the rings under the six radially disposed tensile loads is presented next.

Fig. B.4. Detail of T-1 Beam-to-Ring Connection

Fig. B.5. Tensile Radial Loads on T-1 Ring Transmitted from Radial Beam

3. Stresses in a Circular Ring with Six Equally Spaced Radial Forces

Although these stresses can be obtained directly from various handbooks, the structural behavior of the ring is better understood by proceeding through the derivation of the formulas. The scheme used is to analyze each ring under the action of two equal and opposite forces P acting along a common diameter. To obtain the bending moments and forces in the ring under the action of the six radial loads, it is only necessary to superpose the results of the two force analysis twice.

Figure B.6 shows the loading with two equal and opposite loads and the resulting bending-moment diagram. Timoshenko⁴ derives the following expression for the bending moment at any point on the ring:

$$M_{\theta} = \frac{PR}{2} \left(\cos \theta - \frac{2}{\pi} \right), \quad (1)$$

where

P = radial load,

R = radius of ring,

and

θ = angle measured as shown in Fig. B.6.

To obtain the expression for the bending moment in the ring acted on by six equal radial loads spaced at 60° intervals, it is only necessary to write Eq. 1 for each of the three pairs of loads. Using subscripts 1, 2, and 3 for the three sets of loads, we can write the following equations:

$$M_1 = K \left(\cos \theta - \frac{2}{\pi} \right), \quad (2)$$

$$M_2 = K \left[\cos (\theta - 60) - \frac{2}{\pi} \right], \quad (3)$$

and

$$M_3 = K \left[\cos (\theta - 120) - \frac{2}{\pi} \right], \quad (4)$$

where

$$K = \frac{PR}{2}.$$

Fig. B.6

Circular Ring under Action of Two
Equal and Opposite Radial Loads

Adding Eqs. 2, 3, and 4 and combining give the resulting bending-moment expression

$$M = K \left(\cos \theta + 1.732 \sin \theta - \frac{6}{\pi} \right). \quad (5)$$

The loading pattern and bending moment are shown in Fig. B.7. The maximum bending moments occur at the loads, and are reduced in the design work because of the presence of 14-in.-wide stiffeners at the loads.

Fig. B.7. Circular Ring under Action of Six Equally Spaced Radial Loads

The maximum bending moment, which occurs at each load, is obtained by setting θ equal to 90° in Eq. 5 to give

$$M_{\max} = K \left(\cos 90^\circ + 1.732 \sin 90^\circ - \frac{6}{\pi} \right) = -0.176K. \quad (6)$$

The negative sign indicates that the bending moments at the points of load application tend to increase the curvature of the ring while the positive moment between loads tends to decrease curvature.

The direct forces in the ring are obtained by constructing free bodies of the ring and summing up forces in one direction. The maximum direct force obtained is equal to the value of one applied load, P , and occurs midway in the ring between adjacent applied loads. The value of the direct force in the ring at the loads is equal to $0.866P$.

Thus, the maximum stress in the ring is obtained from one of two formulas, namely,

$$\sigma_{\max} = \pm \frac{0.176K}{Z_r} + \frac{0.866P}{A_r} \quad (7)$$

or

$$\sigma_{\max} = \pm \frac{0.092K}{Z_r} + \frac{P}{A_r}, \quad (8)$$

where

Z_r = section modulus of cross section of ring under investigation

and

A_r = cross-sectional area of ring.

All stresses are well below the chosen allowable values of 45,000 psi.

APPENDIX C

Thermal Design of Primary Tank*1. Nomenclature for Appendix C

a	Gap width	t	Temperature
c_p	Specific heat	V	Velocity
C	Heat capacity	x	Distance
D	Diameter	γ	$-d\rho/dt$
f	Friction factor	θ	Temperature difference
g_c	Conversion factor	ρ	Density
h	Height	τ	Time
K	Thermal conductance	<u>Subscripts and Superscripts</u>	
$\dot{m}$	Mass flow rate	e	Equivalent
P	Perimeter	e	Entrained
q	Heat rate	0	Origin
Q	Volumetric flow rate	'	Per unit length
r	Radius	"	Per unit area
R	Thermal resistance		

2. Introduction

At one time, the primary-tank cover was assumed to be substantially isothermal at the temperature of the bulk sodium in the tank (700°F during reactor operation). Reactor-building air was to be drawn downward through the annuli between the cover nozzles and hangers and the biological shield sleeves to cool the nozzles, hangers, and shield. From there, the air, at about 100°F, was to pass through the horizontal gap between the cover insulation and the shield, and enter the shield cooling-system ducts at the periphery of the cover.

*The material in this appendix was the subject matter of a paper presented at the Nov 1966 ASME Annual Meeting in New York by T. R. Bump and H. O. Monson, entitled "Thermal Design of EBR-II Primary Tank Components." It is included here for completeness of this design report.

However, concurrent thermal-design work, on the rotating plugs which fit in the 9-ft-dia central opening in the cover, showed that even limited areas of steel that penetrates insulation can carry enough heat to seriously reduce the effectiveness of the insulation. Therefore, several two-dimensional analyses of the then current cover design were made. This work indicated that the effective temperature difference (Δt) across the cover could be as high as 300°F, since the cover nozzles and hangers act like fins, carrying substantial amounts of heat through the cover insulation. Because an effective Δt across the cover of 100°F would deform the cover enough to produce equipment misalignments of as much as 1/2 in. at the critical positions, and because of thermal-stress considerations, an intensive effort was made to develop design features that would reduce the cover Δt .

This appendix discusses the features incorporated in the primary-tank cover design for thermal reasons, and describes the methods used in the thermal analysis of the cover. In addition, the design of baffles to protect the vertical wall of the primary tank is mentioned, as well as possible devices to prevent an undesirable vertical Δt in the bulk sodium.

3. Cover Thermal Features

Among the first modifications to be made was the reduction of the height of the vertical web stiffeners for the primary-tank cover from 30 to 21 in., to decrease the Δt across them. Also, I-beam stiffeners, which would be essentially isothermal, were welded to the underside of the cover bottom plate directly below each of the six main radial beams in the cover. Preliminary calculations, which assumed that the nozzles were insulated from the cover, air gap, and biological shield, indicated that coolant airflow through the nozzle annuli could be blocked off in most cases without causing the temperatures of the nozzles and their plugs, where they emerge from the biological shield, to be excessive (above 130°F). Accordingly, the annuli airflow was blocked off, wherever possible. Also, since at elevated temperatures concrete loses water and, thus, shielding effectiveness, the biological-shield concrete was cooled by maintaining 200°F air* in the horizontal gap between the shield and the cover insulation. In addition, the hangers were to be insulated from the biological shield, to reduce heat losses from the hangers, and, thus, from the cover.

After the modifications just described were taken into account, coarse two- and three-dimensional calculations were made to determine the effects of adding (1) vertical, mild-steel plates within the cover, adjacent to the hangers, beams, stiffeners, and nozzles; (2) more insulation above the cover; (3) insulation in the cover between the structure and the nozzles; and (4) insulation between the biological shield and the nozzles. On the basis of this

*The source of this air is the cooling system of the rotating plugs and their freeze seals.

work, it was decided to (1) add about 2 in. in thickness of mild-steel plates along each side of critical-cover structural members and (2) increase the thickness of the insulation on top of the cover from 6 to 8 in. The estimated expense of insulating nozzles from cover structure and from biological shield was economically prohibitive.

The 3/8-in.-dia steel balls within the cover had the relatively low thermal conductivity associated with packed beds and thus tended to increase the cover Δt . Accordingly, the possibility of adding smaller particles to the balls, to fill in the voids and thus produce a bed with increased thermal conductivity, was of considerable interest. Because of this, an apparatus was constructed for measuring the conductivities of such mixtures. Since time was short, and since the conductivity of the balls alone was available in the literature, the first measurements were made using mixtures of balls and steel grit. These measurements gave conductivities that appeared to be at least 60% greater than for balls alone, so it was decided to add 30 wt % of 1/16-in. steel grit to the balls, to decrease the void fraction of the bed from about 0.40 to about 0.28. This decision actually resulted in lower cost for this gamma shielding since the grit was cheaper than balls. Because the grit addition increased the density of the shielding, the height of the shielding was decreased from 30 to 26 in., thus maintaining about the same total weight.

Later measurements using balls alone gave conductivities, at the bottom of the bed, insignificantly different from those from the measurements on mixtures; that is, the conductivities given in the literature for balls alone apparently were low. One possible reason for this is that the increase of contact-point loads with depth in the bed has been overlooked.⁶

In any event, it is not clear that addition of the grit to the balls actually decreased the cover Δt significantly. With regard to economics, the lower cost of the grit was partially offset by the increased expense of mixing the grit with the balls during installation, although this was accomplished with minimum difficulty by the contractor.

The results of the calculations mentioned above indicated that the corrective measures already described still might not be adequate to prevent the cover Δt from being so high that unacceptable deformations would occur. Therefore, heaters were installed on the top flanges of the cover beams and along the tops of the hexagonally arrayed stiffeners at the center ring. Coarse three-dimensional calculations showed that 1 kW on top of a beam, with no carbon steel, and surrounded by nozzles 8 in. away on each side, would be sufficient to prevent the average Δt across the beam from exceeding 50°F. The allowable watt densities for heaters at the temperatures involved (700°F) are quite low, so it was found advisable to install as many heaters as there was room for. Doing this made it possible to supply 1 kW per beam under quite conservative watt-density conditions.

The heater installation provided for all the heaters at one location to be connected in series, and one conductor was then connected between every two heaters and brought through the biological shield. This procedure permitted considerable flexibility in connecting the heaters in series and/or parallel combinations, while at the same time decreasing the number of conductors that had to be brought through the shield, and beyond, by almost a factor of two. However, by the time the primary tank and its structure were completed and tests were run on the heaters, a number of shorts and grounds had developed that were more difficult to accommodate than if the heaters had been connected independently. In other inaccessible installations of this type, it is probably better to provide the two conductors for each heater, in the interests of increasing flexibility.

More detailed one- and two-dimensional analyses were made of all nozzles and their plugs, to determine which ones require coolant airflow. The criteria for flow were that (1) the biological shield concrete adjacent to the nozzles must not overheat (exceed 212°F), and (2) the nozzles and their plugs, where they protrude from the biological shield, must not be too hot for personnel to touch (exceed 130°F). Those nozzles found to require cooling had either (1) a large amount of steel per unit length, or (2) internal heat sources. Nozzles in the former category include the sodium throttle valves and the safety-rod drives. Nozzles in the latter category include the auxiliary pump busses, the shutdown coolers, the heat exchanger, the sodium-purification system, and the inert-gas system. Airflow requirements were established which provide adequate nozzle cooling and at the same time allow the air to be heated to the required 200°F before it enters the horizontal gap between the biological shield and the cover insulation.

These airflow requirements were met by installing neoprene boots between each nozzle flange and its mating biological shield sleeve. At those nozzles that require coolant air, fittings capable of holding orifice plates are installed in the neoprene boots. The number and sizes of the orifices used are of course determined by the flow-rate requirements of each nozzle.

Because of the changes discussed above, only a small fraction of the coolant air required for the biological-shield cooling system can be supplied through the passages originally available for that purpose. Accordingly, new openings were provided in the biological shield to supply the shield cooling air that could not be carried in the horizontal gap between the biological shield and the cover insulation. Those openings are located at the outer periphery of the shield in a manner to produce uniformly distributed airflow between the shield and the floor plates above the shield.

The calculations indicated that without the cover heaters in operation the cover Δt could be as high as 190°F after the above design features had been incorporated. (See Fig. C.5, curve No. 1, at thickness of carbon steel equal to 2 in.) Therefore, the fuel-handling equipment was aligned for a 190°F cover Δt . When the tank was actually brought up to temperature, the

most representative value for the cover Δt was 180°F . Because of this perhaps coincidental agreement, no further alignment of the fuel-handling equipment was necessary in this regard. Other tests showed that with low watt densities the cover heaters could, if necessary, easily reduce the cover Δt to 50°F (see Appendix E).

Fig. C.1. Temperatures in Unmodified Primary-tank Cover. ANL Neg. No. 112-6258.

4. Details of Cover Calculations

Data from the early two-dimensional analyses which indicated potential difficulty with the cover are plotted in Figs. C.1 and C.2. The data were generated with an unpublished two-dimensional program written by T. R. Bump in 1958 for the IBM 650 computer, using the Bell Interpretive Routine. Although used in this case for steady-state problems, the pro-

gram is actually based on Dusenberre's finite-difference procedure for solving transient problems.⁷ However, his method is modified so that as soon as the temperature t' at the end of a time interval has been calculated for a node, that temperature is combined with the node temperature t at the beginning of the time interval, to obtain the average temperature $(t + t')/2$ of the node during the time interval. As these average temperatures become available, they are used instead as initial node temperatures, in the calculations for succeeding nodes during the same time interval. This modification is useful in that it changes the stability criterion from $(RC/\Delta\tau) > 4$ (from Ref. 7) to $(RC/\Delta\tau) > 2$. In practice, values of $RC/\Delta\tau$ of unity have been used and no instability has been noticed. This relaxation of the stability criterion to permit use of larger $\Delta\tau$'s, which reduces computation time, is of course not completely advantageous, since the calculation of conditions during each time interval takes longer.

Fig. C.2. Additional Temperatures in Unmodified Primary-tank Cover. ANL Neg. No. 112-6267 Rev. 1.

For the steady-state calculations, an arbitrary ΔT of 1 hr was chosen, and a dummy capacitance, which met the resulting stability criterion $C = 1/R_{\min}$ was developed for each node. An initial estimated temperature was then assigned to each node (a 7 x 7 lattice was used), and the calculations were carried out on a transient basis until the node temperatures no longer changed "appreciably." The asymptotic node temperatures were then taken to be desired steady-state temperatures. Results were compared with one-dimensional calculations to ensure that the two-dimensional results were reasonable.

Fig. C.3. Two-node Heat-transfer System

This program has been successfully transposed for three-dimensional problems, using FORTRAN language.⁸ For steady-state problems, the program automatically calculates the required dummy capacitances. A subroutine exit is available for altering the data as a problem progresses, to get special effects such as those mentioned later in this appendix.

In order to investigate expeditiously the effects on the cover Δt of a number of parameters, a coarse solution, which could easily be handled with a slide rule, was used. This solution is based on the two-node system illustrated in Fig. C.3. For that system, the following equations can be written:

$$[(t_1 - t_A)/R_W] + q_A = [(t_A - t_0)/R_X] + \left\{ \left[\frac{(t_A + t_1)}{2} \right] - \left[\frac{(t_B + t_1)}{2} \right] \right\} / R_{WY} + \left\{ \left[\frac{(t_A + t_0)}{2} \right] - \left[\frac{(t_B + t_0)}{2} \right] \right\} / R_{XZ}; \quad (9)$$

$$(t_1 - t_B)/R_Y = [(t_B - t_0)/R_Z] - \left\{ \left[\frac{(t_A + t_1)}{2} \right] - \left[\frac{(t_B - t_1)}{2} \right] \right\} / R_{WY} - \left\{ \left[\frac{(t_A + t_0)}{2} \right] - \left[\frac{(t_B + t_0)}{2} \right] \right\} / R_{XZ}. \quad (10)$$

Equation 9 states that the heat entering the lower edge of the left two spaces of Fig. C.3, plus the heat entering those two spaces by internal generation or by heat transfer from the third dimension, is equal to the heat leaving the upper edge of those two spaces, plus the heat leaving the right edges of those two spaces. As is evident, linear vertical temperature distributions are assumed. Equation 10 amounts to a similar heat balance on the right two spaces of Fig. C.3.

If temperature differences are defined as

$$\theta = t - t_0,$$

then Eqs. 9 and 10 can be solved to give

$$\theta_A = \left(\frac{[K_W(K_Y + K_Z + K_{AB}) + K_{AB}K_Y] \theta_1 + [(K_Y + K_Z + K_{AB}) q_A]}{[(K_X + K_W)(K_Y + K_Z + K_{AB})] + [K_{AB}(K_Y + K_Z)]} \right) \tag{11}$$

and

$$\theta_B/\theta_1 = \{[K_{AB}(\theta_A/\theta_1)] + K_Y\}/(K_Y + K_Z + K_{AB}), \tag{12}$$

where

$$K_{AB} = [(1/R_{WY}) + (1/R_{XZ})]/2.$$

To test the accuracy of these equations, the problems illustrated in Figs. C.1 and C.2, and another similar problem, were re-solved using the two-node system. Figure C.4 illustrates how a typical physical situation was arranged to fit into the system. In that arrangement, R_Z between t_B and t_0 was calculated to include the fin-type resistance of the hanger where it protruded from the system. Table C.1 compares the results of

Fig. C.4
Example of Application of
Two-node System

TABLE C.1. Comparison of Results for 49-node and Two-node Systems

Physical Situation	Temperatures, °F			
	Plate or Rib Top (Avg)		Fit at Same Elevation	
	49-node	2-node	49-node	2-node
Rotating plug	390	379	320	329
Beam into hanger	470	472	340	346
Left case	320	311	280	298
Right case	360	347	310	320

the 7 x 7 lattice (49-node) computer calculations with those of the corresponding two-node hand calculations. Since the results do not differ by more than 6%, it was concluded that the coarse solution was acceptable, and from

then on all problems of this type were solved using the two-node approach.

Results of the calculations mentioned earlier in this section of this appendix are shown in Fig. C.5. Information found which is difficult to present graphically is summarized here:

1. BEAM: LOSSES INTO HANGER AND INTO NOZZLES 8 in. FROM EACH SIDE
2. BEAM: LOSSES INTO NOZZLES 8 in. FROM EACH SIDE
3. BEAM: LOSSES INTO HANGER AND INTO 40 in. OF STEEL BALLS FROM EACH SIDE
4. BEAM: LOSSES INTO HANGER
5. BEAM: LOSSES INTO RING
6. SHORT STIFFENER: LOSSES INTO NOZZLES 8 in. FROM EACH SIDE
7. SHORT STIFFENER: LOSSES INTO NOZZLES 8 in. FROM ONE SIDE

Fig. C.5. Temperature Difference across Various Structural Members vs Thickness of Carbon Steel Installed against Each Side of Member. ANL Neg. No. 112-6254 Rev. 1.

1. In three beam cases, the thickness of the insulation on top of the steel balls was decreased from 8 to 6 in. In these cases, this decrease resulted in increases in the temperature differences across the beam of 10-17°F (7.2-11.9%).

2. In three beam cases, 0.5 in. of insulation [$k = 0.004 \text{ B}/(\text{hr})(\text{ft})(^\circ\text{F})$] was placed between the beam and adjoining nozzles. In these cases, the addition of this insulation resulted in decreases in the temperature differences across the beam of (optimistically) 21-24°F (13.6-23.9%).

3. In one beam case, the insulation around the hanger was doubled in thickness from 0.25 to 0.50 in. In addition, the conductivity of the insulation was reduced from 0.1 to 0.04 B/(hr)(ft)(°F). Both of these changes resulted in a 14°F (9.2%) decrease in the temperature difference across the beam.

4. In one short-stiffener case, with nozzles 8 in. from one side only of the stiffener, 4 in. of carbon steel on the side of the stiffener towards the nozzles produced about the same temperature difference across the stiffener as 2 in. of carbon steel on each side of the stiffener.

5. In one short-stiffener case, with nozzles 8 in. from one side only of the stiffener, increasing the surface resistance of the fin regions of the nozzles by a factor of 1.84 decreased the temperature difference across the stiffener by 8°F (11.3%).

6. When carbon steel is placed between beams and nozzles, it does not make much difference whether the steel is placed next to the nozzle or next to the beam.

The three-dimensional problems included in this discussion were also solved by Eqs. 11 and 12, using an iterative procedure. The heat transferred to the third dimension was estimated and used as q_A , and after the corresponding temperatures had been calculated, the accuracy of the estimated q_A was checked. If necessary, a new q_A was estimated and the process repeated.

The beam-heater sizing calculations mentioned in Section 3 of this appendix were also made using Eqs. 11 and 12, except that q_A then became merely the heat input of the electrical heaters.

The first nozzle-plug analyses were made using an early version of the "fins-in-series" approach.⁹ Those types of nozzles found to possibly require cooling were reanalyzed using the two-dimensional computer program, and typical results are shown in Figs. C.6-C.9. Airflow requirements for the nozzles requiring cooling are given in Table C.2.

Fig. C.6. Temperatures in Typical Nozzle.
ANL Neg. No. 112-6580.

Fig. C.7. Temperatures in Auxiliary Pump-bus
Nozzle. ANL Neg. No. 112-6579.

Fig. C.8. Temperatures in Safety-rod Nozzle. ANL Neg. No. 112-6581.

Fig. C.9. Temperatures in Shutdown-cooler Nozzle. ANL Neg. No. 112-6264 Rev. 1.

TABLE C.2. Airflow Requirements for Cooled Nozzles

Nozzle	Designation	Airflow Rate, cfm/nozzle
Auxiliary pump bus	I1, I2	100 ^a
Shutdown cooler	L1, L2	36
Inert gas	N1, N2	6
Sodium throttle valve	P1, P2	6
Safety-rod drive	Q1, Q2	6
Heat exchanger	U	64
Sodium purification	V, E-3	16

^aThis air passes through an insulated duct, shrouding the bus between the nozzle and the rectifier, before entering the nozzle annulus.

Fig. C.10

Model for Including Radiation across a Gap. ANL Neg. No. 112-6253.

A subprogram was sometimes added to the two-dimensional program, to make it possible to include the heat transfer by radiation across a gap in which coolant air was flowing. This provision was necessary because, even though the air was kept cooler than the maximum permissible temperature of the biological-shield concrete, radiation across the gap, from a nozzle of interest, could possibly raise the concrete temperature above the air temperature. The subprogram considered the configuration shown in Fig. C.10.

Temperatures of nodes 1, 2, and 3 are calculated by the main program; the air temperature t_2 is either held constant at a periodically rechecked appropriate value, by assigning an infinite capacitance to node 2, or is continuously recalculated

by a separate subprogram outlined later in this section of this appendix. To obtain the gap surface temperatures, which are needed to calculate the radiation heat transfer, the following simultaneous equations were used:

$$K_{1,a}(t_1 - t_a) = K_{a,2}(t_a - t_2) + K_{a,b}(t_a - t_b) \quad (13)$$

and

$$K_{2,b}(t_2 - t_b) + K_{a,b}(t_a - t_b) = K_{b,3}(t_b - t_3), \quad (14)$$

which can be solved to show that

$$t_a = \{ (K_{2,b} + K_{a,b} + K_{b,3}) K_{1,a} t_1 + [(K_{2,b} + K_{a,b} + K_{b,3}) K_{a,2} + K_{a,b} K_{2,b}] t_2 + K_{a,b} K_{b,3} t_3 \} / [(K_{1,a} + K_{a,2} + K_{a,b})(K_{2,b} + K_{a,b} + K_{b,3}) - K_{a,b}^2] \quad (15)$$

$$t_b = \{ K_{1,a} K_{a,b} t_1 + [(K_{1,a} + K_{a,2} + K_{a,b}) K_{2,b} + K_{a,2} K_{a,b}] t_2 + (K_{1,a} + K_{a,2} + K_{a,b}) K_{b,3} t_3 \} / [(K_{1,a} + K_{a,2} + K_{a,b})(K_{2,b} + K_{a,b} + K_{b,3}) - K_{a,b}^2]. \quad (16)$$

The main program requires input of the effective thermal resistance $R_{1,2}$ and $R_{2,3}$ between nodes 1 and 2 and between nodes 2 and 3, respectively. Since these resistances depend on the unknown surface temperature t_a and t_b , it is necessary to continuously recalculate $R_{1,2}$ and $R_{2,3}$ by subprogram, as the problem solution progresses. Those recalculations take the forms

$$R_{1,2} = (t_1 - t_2) / q_{1,a} \quad (17)$$

and

$$R_{2,3} = (t_2 - t_3) / q_{b,3}, \quad (18)$$

where

$$q_{1,a} = (t_1 - t_a)/R_{1,a}$$

and

$$q_{b,3} = (t_b - t_3)/R_{b,3}.$$

The foregoing treatment considers only radiation between opposing nodes, a simplification that is permissible if the gap between nodes is relatively narrow.

Another subprogram was sometimes added to the main program, to include the effects of the coolant air flowing through the nodes. The following relationships were used (see Fig. C.10):

$$t_2 = t_{in} + [K_{a,2}(t_a - t_2) - K_{2,b}(t_2 - t_b)]/2\dot{m}c_p \quad (19)$$

and

$$t_{out} = 2(t_2 - t_{in}) + t_{in} = 2t_2 - t_{in}. \quad (20)$$

Equation 19 can be rewritten as

$$t_2 = [t_{in} + (K_{a,2}t_a + K_{2,b}t_b)/2\dot{m}c_p]/[1 + (K_{a,2} + K_{2,b})/2\dot{m}c_p], \quad (21)$$

and Eq. 20 gives t_{in} for the next node through which the air flows.

5. Other Primary-tank Thermal-design Considerations

The sodium leaving the heat exchanger is often at a different temperature than the bulk sodium in the primary tank, and the difference can become substantial under certain conditions, such as loss of electrical power to the plant. The heat exchanger is located adjacent to the primary tank wall, and the sodium leaving the exchanger tends to impinge directly on the wall; thus sizable temperature differences could be introduced in the wall, causing excessive thermal stresses or distortions of the primary tank.

Fig. C.11
Model for Calculating
Baffle Spacing Require-
ment. ANL Neg.
No. 112-6582.

Because of this, a baffle was installed between the heat exchanger and the tank wall. An adequate gap had to be provided between the baffle and the wall to permit enough bulk sodium to circulate between them to keep the wall at nearly the same temperature as the bulk sodium. To determine the minimum acceptable gap, the model shown in Fig. C.11 was used. The head loss incurred by the flowing sodium was equated with the thermal head to obtain the following relationship:

$$\rho[(fh/D_e) + 1.5](\dot{m}'/\rho a)^2/2g_c = \gamma h(q''h/2\dot{m}'c_p), \quad (22)$$

Fig. C.12. Temperature Change of Flowing Sodium, ΔT_c , vs Gap Width, a . ANL Neg. No. 112-6583.

where the constant 1.5 accounts for entrance and exit losses. After Eq. 22 was solved for $\dot{m}'$, the temperature change of the flowing sodium was found to vary with the gap width a , as shown in Fig. C.12. Since a maximum temperature difference of 50°F was desired in the primary-tank wall, the baffle was placed 3 in. away from the wall. The assumption of uniform heat flux used in the model is valid only for realistic temperature changes in the flowing sodium. The results from this model, which makes use of the mixed-mean coolant temperature at any elevation in the gap, are conservative since the temperature at the insulated wall is always lower than the mixed-mean temperature. In the literature, there are a

number of more elegant solutions for natural-convection situations similar to this one, but none is directly applicable.

To determine how far around the tank wall the baffle should extend, use was made of Rushton's mixing equation¹⁰

$$Q_e = [0.23(x/D_0) - 1] Q_0, \quad (23)$$

which is applicable to a submerged stream issuing from a round opening of diameter D_0 . Since the heat-exchanger opening is circumferential, an attempt was made to restate this equation in more general terms. If conservation of momentum is assumed, Eq. 23 indicates that the effective diameter of the jet is only a function of the distance from the opening.¹¹ Such a condition appears unrealistic; accordingly, the equation was fit by an exponential function

$$Q_e = \{\exp[0.083(x/D_0)] - 1\} Q_0, \quad (24)$$

which is not only valid at small values of x/D_0 but also retains agreement with the original equation to x/D_0 as large as 15. With this function, assuming conservation of momentum indicates that $D/D_0 = \exp[0.083(x/D_0)]$, so that the entrainment in the jet can be expressed in terms of the effective perimeter and initial velocity of the jet as follows:

$$dQ_e/dx = PV_0(0.083/4). \quad (25)$$

Integration of Eq. 25 for the circumferential opening gives

$$Q_e = 0.083r_0[(r/r_0)^2 - 1] Q_0/4h. \quad (26)$$

Since the justification for Eq. 26 is not very substantial, it is reassuring that it gives about the same results as the original equation with D_0 taken equal to the height h of the circumferential opening. The analysis indicated that 10 ft away from the heat exchanger the temperature difference between the sodium leaving the exchanger and that already in the tank would be reduced to about one-fourth of its initial value. This amount of attenuation was considered acceptable for the tank wall, so the baffle to protect the wall was sized accordingly. A second baffle was mounted in front of the main baffle to protect the latter from thermal shocks during transients.

Another consideration in the design of the primary tank was related to the large amount of heat dumped into the bulk sodium near its surface. The sources of this heat are (1) leakage of hot sodium, both through clearance in the reactor vessel cover and through a vent hole at the heat exchanger inlet, and (2) heat losses, both from the pipe between reactor vessel and heat exchanger, and from the heat exchanger. This heat amounts to about 2 MWt during full-power operation, and is potentially adequate to heat the surface of the sodium to 860°F, if thermal conduction to 700°F sodium at the elevation of the primary pump inlets were the only mechanism of heat removal. Since the tank cover was designed to be exposed only to 700°F sodium, other means of removing the heat than by conduction were sought. One possible solution was to insert one or more pumps in spare nozzles, to pump cold sodium from the bottom of the tank to the top. An alternate scheme was to use hollow-valve stems in the throttle valves used for regulating flow to the low-pressure inlet plenum in the reactor vessel. Cold sodium would then be pumped, by the main primary pumps, up the valve stems and be discharged near the sodium surface.

However, these ideas were held in abeyance pending actual operation, since some mixing was bound to occur in the primary tank due to circulation of primary coolant, and that mixing might be adequate to control the vertical Δt in the sodium. To date, at power levels up to three-fourths of plant full power (and 100% of first-core full power), no unreasonable vertical Δt 's have been measured, so no augmentation of mixing has been necessary. One possible reason why the existing mixing appears satisfactory is related to the difference in distance from the heat exchanger to the two primary-pump inlets. Since it was desired to have both pumps always deliver sodium to the reactor vessel at identical temperatures, a vertical baffle was installed between the heat exchanger and the closest pump inlet, to make the distance traveled by the sodium from the exchanger to each inlet the same. The vertical baffle forces some of the circulating primary coolant to the free surface to help keep the surface cool. Also, some of the sodium leaving the heat exchanger is forced to the surface by impingement against the wall baffle.

In future installations of this type, more attention should probably be placed on the installation of baffles, to ensure complete mixing of the primary coolant while passing through the bulk sodium. In this way, the heat capacity of all the bulk sodium is used to stabilize the reactor inlet temperature.

APPENDIX D

Analysis of Thermal Deformations of Primary Tank1. Basic Thermal-deflection Equations

As stated in Section II.A.2 of this report, a temperature drop occurs across the primary-tank cover of such sign and magnitude as to cause a downward bowing of the cover structure at elevated temperatures. Thermal analysis indicates that this ΔT would probably be about 180°F. Although many precautions were taken to lessen the effects of the ΔT (see Appendix C), the cover will undergo bowing deformation. To make intelligent allowances for these movements in adjusting mechanisms and setting equipment, it was highly desirable to know, as close as possible, the magnitude of such movements. A detailed analytical study was made to attempt to define the deformations of the cover. Later, after most of the reactor equipment was in place, two elevated temperature tests were made, one test before sodium filling and one test after the tank was completely filled with sodium. These tests and their results are discussed in Appendix E. This appendix will discuss the analytical work performed to determine deformations that were measured later.

As for the structural analysis of the cover in calculating resistance to the static loads, the behavior of the cover is assumed to be dominated by the behavior of the basic structure--the six hangers, six radial beams, and the central ring. In the analysis immediately following the 12-in.-deep stiffeners below, the radial beams are assumed to be omitted. Their presence will be accounted for later in this appendix.

Consider the familiar structural beam members--two hangers, two radial beams, and the continuity provided by the central ring. Assuming no restraint by the rollers or hangers, a temperature gradient, ΔT , across the cover structure will cause the beam to assume a circular arc of radius R , where R is defined by

$$R = \frac{d}{\alpha \Delta T}, \quad (27)$$

where

d = depth of the radial beam, in.,

α = coefficient of linear thermal expansion, in./in.-°F,

and

Δ = temperature gradient across radial beam, °F.

The resulting deflection curve is identical to one produced by applying end couples, M_0 , at the hangers, on the radial beams, where

$$M_0 = \frac{EI\alpha\Delta T}{d}, \quad (28)$$

where

E = modulus of elasticity of the cover material at temperature

and

I = moment of inertia of the radial beam for the axis about which bending takes place.

Choose a Cartesian coordinate system with its origin at the intersection of the center of the hanger at the left end of the beam with the neutral axis of the beam; x is measured to the right as positive, and y is measured as positive downward. If L is taken as the distance between opposing hangers (24 ft), the deflection curve for the beam will be given by

$$y = \frac{-M_0}{2EI} (x^2 - Lx). \quad (29)$$

Substituting for M_0 from Eq. 28 results in

$$y = \frac{-\alpha\Delta T}{2d} (x^2 - Lx), \quad (30)$$

which defines the shape of the neutral axis of the beam with a temperature gradient ΔT across the depth of the beam.

Differentiating Eq. 30 with respect to x results in a general expression for the slope of any point along the beam; thus,

$$y' = \frac{-\alpha\Delta T}{d} \left(x - \frac{L}{2} \right). \quad (31)$$

To give an idea of the magnitude of these deformations, the deflection of the cover beams at the central ring is calculated for a $\Delta T = 100^\circ\text{F}$ ($x = 7.5$ ft):

$$y = \frac{9.5 \times 10^{-6} \times 100}{2 \times 33} [(7.5 \times 12)^2 - (24)(7.5)(144)] = +0.256 \text{ in. (downward).}$$

(This is 1/4 in. relative to the bottom of the hangers; there is, of course, an additional downward growth of the cover and tank as the average temperature of the hangers is increased.)

The angle of rotation of the cover beams at the hangers for a ΔT of 100°F is

$$\theta = y' = \frac{-9.5 \times 10^{-6} \times 100}{33} \left(0 - \frac{24 \times 12}{2} \right) = +4.14 \times 10^{-3} \text{ radians.}$$

2. Effects of 12-in.-deep Isothermal Beam Stiffeners

As stated in Section II.A.2.a of this report, the 12-in. beam stiffeners were added to partially restrain the bowing of the radial beams due to a ΔT across the cover. In the analysis to follow, these stiffeners are assumed to be continuous across the 9-ft-dia central opening. This assumption is justified on the basis that the central ring probably lends a reasonably equivalent stiffening effect on the cover bowing, particularly with the substantial effects of the Z-ring.

Consider the cross section of a radial beam, a portion of the 2-in. bottom plate, and the 12-in. stiffener subjected to a temperature distribution as shown in Fig. D.1. The 12-in. stiffener is assumed to be at the same temperature as the argon gas blanket above the sodium (700°F); the temperature of the radial beam is assumed to vary linearly from 700°F at the bottom of the 2-in. plate to a lower temperature at the top of the radial beam. Since the temperature distribution is not uniform, thermal stresses will result. The thermal stresses in the cross section for this temperature distribution is shown on Fig. D.1, assuming the ends of the beams to be unrestrained.

Fig. D.1. Thermal Stresses in Radial Beams of Primary-tank Cover

The thermal-stress diagrams in Fig. D.1 are constructed by superposing the stress diagrams for three loading patterns (see Ref. 4). The first stress diagram consists of compressive stresses f_c , which are merely the values of the stresses of any point on the cross section if the beams were completely restrained. These compressive stresses are given by

$$f_c = -\alpha ET(y), \quad (32)$$

where y is measured as positive down from the neutral axis of the cross section, and $t(y)$ is the temperature at any point on the cross section at y .

Secondly, since the beam is free to rotate, these compressive stresses must be counteracted by applying a fictitious tensile force (F_t) at the end of the beam of a magnitude equal to the sum of the compressive stresses acting over the entire area of the cross section. This fictitious tensile force leads to a uniform tensile stress, f_t , over the entire cross section. The value of this tensile stress is given as

$$f_t = \frac{F_t}{A_b} = \frac{\sum (f_{c,i} A_i)}{A_b}, \quad (33)$$

where

$f_{c,i}$ = value of compressive stress as given in Eq. 32 at the i th section,

A_i = area of the i th section,

and

A_b = area of the entire cross section.

Thirdly, the eccentricity of F_t will result in a bending moment in the beam, M_b , which generates bending stresses, f_b , given by

$$f_b = \frac{M_b y}{I_b}, \quad (34)$$

where

I_b = moment of inertia of cross section about neutral axis.

The value of the bending moment, M_b , is given by

$$M_b = F_t \bar{y},$$

where

$\bar{y}$ = distance from the force F_t to the centroid of the cross sections.

Thus, the resulting thermal stresses in the cross section are obtained by adding, algebraically, Eqs. 32, 33, and 34. This will be done for the temperature distribution shown in Fig. D.1.

For convenience, the cross section of the beam is divided into five components: (1) the top flange of the radial beam; (2) the radial-beam web; (3) the bottom flange of the beam and a portion of the 2-in. bottom plate; (4) the web of the stiffener; and (5) the bottom flange of the stiffener. In the calculations that follow, the subscripts 1 through 5 refer to these areas, respectively. The temperature gradient across the radial beam is given as T_0 .

First, the analytical expression for the temperature, $T(y)$, is given for $-26.5 \leq y \leq 6$:

$$T(y) = \frac{T_0}{32.5} y + \frac{26.5}{32.5} T_0 = \frac{T_0}{32.5} (y + 26.5). \quad (35)$$

And, for $-26.5 \leq y \leq 6$,

$$f_c = \frac{-\alpha E T_0}{32.5} (y + 26.5) \quad (36)$$

while, for $6 \leq y \leq 18$,

$$f_c = -\alpha E T_0.$$

Next the net tensile force for each of the five components is calculated by integrating the compressive stresses over each area. The results are

$$(F_t)_1 = 0.121\alpha E T_0,$$

$$(F_t)_2 = 10.2\alpha E T_0,$$

$$(F_t)_3 = 36.8\alpha E T_0,$$

$$(F_t)_4 = 8.25\alpha E T_0,$$

and

$$(F_t)_5 = 14.0\alpha E T_0,$$

and the resultant tensile force is

$$F_t = \sum (F_t)_i = 69.37\alpha E T_0.$$

The area of the cross section is 93 in.²; thus the uniform tensile stress is

$$f_t = \frac{F_t}{A_b} = \frac{69.37\alpha E T_0}{93} = 0.745\alpha E T_0.$$

The resultant bending moment, M_b , is obtained by multiplying the net tensile force at each component by the distance between the centroid of that component and the centroid of the entire cross section. This results in a value for M_b equal to $+392.4\alpha ET_0$ (counterclockwise at the left of the beam). The resulting bending stresses are given as

$$f_b = \frac{M_{by}}{I_b} = \frac{392\alpha ET_0}{17,760}(y) = 0.0221\alpha ET_0 y.$$

The final thermal-stress equations may now be written (not reduced for clarity). For $-26.5 \leq y \leq 6$,

$$f_y = \frac{-\alpha ET_0}{32.5}(y+26.5) + 0.745\alpha ET_0 + 0.0221\alpha ET_0 y, \quad (37)$$

and for $6 \leq y \leq 18$,

$$f_y = -\alpha ET_0 + 0.745\alpha ET_0 + 0.0221\alpha ET_0 y. \quad (38)$$

These stresses are indicated in Fig. D.1. The maximum stress occurs at the top flange of the radial beam and is tensile. Its value is $0.16\alpha ET_0$. For a ΔT of 100°F , the maximum thermal stress is

$$(f_y)_{\max} = 0.16\alpha ET_0 = 0.16 \times 9.5 \times 10^{-6} \times 24 \times 10^6 \times 100 = 3650 \text{ psi.}$$

The value of this maximum thermal stress must be added to the maximum tensile bending stress in the beams at the hangers of 4830 psi. Even for a ΔT of 200°F , the maximum combined stress is less than the allowable value of 14,700 psi.

While the maximum thermal stresses need to be calculated, it is of more importance, of course, to know the restraining effect of the 12-in. stiffeners on the bowing of the radial beams. The amount of restraint afforded by the stiffeners may be calculated from consideration of the thermal-stress distribution for the entire cross section shown in Fig. D.1. There it may be seen that the stress diagram across the radial beam itself and across the stiffener itself appear similar to a stress diagram for an eccentrically loaded column (which, of course, they are, since that is exactly how they were calculated). To find the final shape of the cover for a given ΔT , it is only necessary to consider the radial beam alone, without the stiffeners, subjected to its portion of the stress diagram in Fig. D.1.

The stress diagram for the radial beam as shown in Fig. D.1 is then broken down into two parts, one stress pattern due to a bending moment and the other due to a compressive force acting at the centroid of the unstiffened beam. By analyzing the radial beam for these loads, we can find the deflection curve. The ordinates of this deflection curve (which is concave upwards)

are then subtracted from the ordinates of the bowing curve given by Eq. 30 resulting in the deflection curve for the stiffened beam subjected to a ΔT equal T_0 .

Applying this approach shows that the "free" deflections and rotations of the unrestrained radial beams are reduced by 27% due to the stiffening effects of the 12-in. isothermal beams. It will be seen in Appendix E that a reduction of this order of magnitude was observed in measuring the actual deflections and rotations of the cover at elevated temperatures.

APPENDIX E

Elevated-temperature Tests

Two elevated-temperature tests were performed in order to verify that the method of tank support was reliable in providing uniform radial thermal expansion and also to measure actual deflections and rotations of the cover.

The first test, the "dry-heat test," was run before sodium filling; the second test was performed with sodium in the tank. The first test was somewhat limited, since the air had to be heated and circulated to obtain a reasonably uniform temperature. The maximum temperature attained for the dry heat test was $\sim 350^{\circ}\text{F}$. It was possible, however, during the dry-heat test to gain access to the inside of the tank before and after the test. This made it possible to perform measurements at many locations that became inaccessible after sodium had been introduced to the system.

The second test consisted of measurements made on the cover deflections and rotations while the sodium was being brought up to essentially operating temperature (650°F). Measurements were made at 50°F intervals up to and including 650°F . Since access to the inside of the tank was prohibited, the number of locations at which measurements were made was reduced from the dry-heat test.

1. Dry-heat Test

The dry-heat test had five major objectives:

- a. To determine the temperature distribution throughout the primary tank for the temperature at which sodium will be introduced into the tank ($\sim 250^{\circ}\text{F}$).
- b. To verify calculated temperature drops through the primary-tank cover and resulting deflections and rotations at moderately elevated temperatures.
- c. To check operation of the remote fuel-handling system at a moderately elevated temperature.
- d. To determine the effectiveness of the primary-tank radial beam heaters.
- e. To verify uniform radial thermal expansion of the primary-tank hanger system.

Items a and c will not be covered in this report. Results of the test as they pertain to items b, d, and e will now be discussed.

a. Temperature Drop across the Primary Tank Cover Structure

During the dry-heat test, approximately 180 thermocouples were read and recorded once every hour, 24 hr a day. Although it was not possible (nor was it intended) to reach a true steady-state thermal condition during the dry-heat test, an attempt was made to achieve a temperature condition as near as possible to steady state. The air and the tank were heated with six long electric immersion heaters disposed about the cover. Extrapolating the data recorded during the quasi-steady-state condition to a tank temperature of 700°F and correcting for differences in the existing and normal operating shield-cooling-air temperatures above the tank led to an estimated temperature drop of 180°F across the cover structure (without the radial beam heaters in operation). Calculated value for the temperature drop ranged up to 190°F. Other sets of data, thought to be less representative of a steady-state thermal condition, indicated that the drop might vary between 150 and 200°F.

b. Deflections and Rotations of the Cover

Deflections of the cover at various nozzle locations were measured by tack-welding a steel plate at the bottom of each nozzle and inserting a cold steel rod into the nozzle through a bushing at the top of the nozzle. A bar at the top of the steel rod depressed the plunger of a dial indicator anchored to the heavy concrete in the top support structure. Measuring at the bottom of the nozzle eliminated having to make corrections for differences in temperatures between the hangers and the nozzles. Measuring rods were stored in the room atmosphere to retain a relatively constant length.

Rotations of nozzles were measured by affixing a plumb bob on a 6-ft line to the top of a nozzle and gluing a machinist's steel rule to the top of the nozzle. Since the nozzles are fixed (welded) to the cover at only the bottom 2-in. plate, the rotation of any nozzle is equal to the rotation of the bottom plate at that nozzle. The rotation of the hangers was measured by measuring the distances between blocks fixed to the T-1 radial beam (one at the top flange and one just above the shielding concrete) and the side of the hanger strap. Changes in these distances were then used to determine inward "tilt" of the hangers, which was, in fact, the rotation of the radial beams at the hangers.

In general, the measured rotations and deflections were about 20-40% less than the calculated values. The readings that differed by only 20% were usually those taken near a radial beam, especially the hangers. As an example, for a measured ΔT of 71°F the calculated angle of rotation at the hangers is 3.1×10^{-3} radians, while the average of the six hangers measured was 2.36×10^{-3} radians. The biggest single factor that introduces errors in the calculations is that while the cover assumes, in a

general way, a spherical shape, many local variations can and, undoubtedly, do exist. This is particularly true for the larger nozzles such as the heat exchanger, which no doubt strongly influence the structural behavior of the cover bottom plate locally.

c. Effectiveness of Radial Beam Heaters

In order to evaluate the effectiveness of the radial beam electrical strip heaters in reducing the ΔT across the beams, it was necessary to compare two sets of readings. One set was taken at a quasi-steady condition with the beam heaters off, and the other set with the beam heaters on. The air in the tank was at about 300°F.

With the beam heaters off, the average ΔT was observed at 71°F; with the beam heaters energized, the ΔT was 29°F (~0.5 kW per radial beam and 0.25 kW per hexagonal rib). This observed value of the sizable reduction in ΔT agreed with the predicted effectiveness of the heaters in providing a positive method of controlling cover deflections, if this should ever be necessary. As stated in Section II.A.2.a, the use of these heaters is not contemplated for normal operation.

Upon conclusion of the dry-heat test, the tank temperature was brought back to ambient (~80°F). Matching scribe marks across the tops of the six hangers and T-1 radial beams, made before the dry-heat test, were observed to align again perfectly. Thus the tank had undergone heating to about 350°F, expanded fairly uniformly, and, upon cooling, contracted back to its original position, thus confirming that the design does, in fact, provide uniform radial thermal expansion about the tank centerline.

2. The Second Test

The second test was made after the tank was filled with sodium. The purpose of this test was to obtain measurements on cover deflections and rotations at temperatures much closer to the final operating value of 700°F. Some of the locations measured during the dry-heat test were no longer accessible, but the more dependable locations, such as the hangers, were still in use.

Measurements were made with the bulk sodium at temperatures of 300, 500, and 600°F. In general, the rotations observed were somewhat closer to the calculated values, especially at the hangers, where the differences in observed and calculated rotations were less than 10%. The improved agreement was due, for the most part, to the more uniform temperature of the bottom of the cover with the tank filled with sodium.

Of considerable interest to the operation of the primary system was how much the seal trough for the large rotating plug moved vertically

from one temperature level to another. This movement is important because if the tank temperature does undergo a significant change and the seal alloy is not melted during this change, large forces can be developed in the bond between the "frozen" seal alloy and the "dip-ring" on the plug, causing heavy loading at the center ring of the primary-tank cover. Therefore, the operation manual makes it mandatory to melt the seal whenever the tank temperature or the ΔT across the cover changes by a certain specified amount. In going from room temperature up to 500°F, we observed that the distance between the top of the large rotating plug and the bottom of the seal trough increased by 0.156 in. The calculated value, considering the 12-in. beam stiffeners as fully effective, was 0.181 in. These readings are considered as quite representative of the cover behavior inasmuch as they are obtained by a direct means rather than by indirect means.

In all instances, the thermal-expansion movement of the primary tank radially outward at the hangers occurred in a uniform and predictable fashion, and upon subsequent cooling the hangers all returned to precisely the original position at the lower temperatures. It is concluded that this type of suspension and roller system--although requiring considerable care in fabrication--behaves in a predictable fashion and does, indeed, maintain centerline alignment of the coincident "hot" and "cold" structures.

ACKNOWLEDGMENTS

The design and installation of the primary tank and support structure required the efforts of many people throughout the project. The nozzle layout design of the primary cover was a major task and was performed by E. Hutter, T. E. Sullivan, A. R. Jamrog, and V. Hutter. These people were also responsible for many of the detailed designs such as the hanger layouts and overall fabrication tolerances. The assistance of B. C. Cerutti during field erection was invaluable.

I am indebted to S. H. Fistedis for his work on the hangers and for his consultation on the overall structural design of the system. The advice and helpful criticisms of N. Balai and A. H. Heineman throughout the design and construction are gratefully acknowledged.

R. L. Lenz was responsible for coordination of the cover fabrication. Contributions to the stress analysis were made by H. Zaremba of the H. K. Ferguson Company and E. W. Rothrock of the Chicago Bridge and Iron Company.

T. R. Bump and F. J. Tebo performed the heat-transfer analysis work for the system. K. D. Kuczen did the design work on the tank insulation. In addition, T. R. Bump and H. O. Monson prepared the material for Appendix C.

H. O. Monson had overall responsibility for the primary system, and his consistent encouragement and guidance are gratefully acknowledged. The advice and encouragement of L. J. Koch, Director, are also acknowledged.

REFERENCES

1. L. J. Koch, H. O. Monson, D. Okrent, M. Levenson, W. R. Simmons, J. R. Humphreys, J. Haugsnes, V. Z. Jankus, and W. B. Loewenstein, *Hazard Summary Report: Experimental Breeder Reactor II (EBR-II)*, ANL-5719 (May 1957).
2. L. J. Koch, W. B. Loewenstein, and H. O. Monson, *Addendum to Hazard Summary Report: Experimental Breeder Reactor-II (EBR-II)*, ANL-5719 (Addendum) (June 1962).
3. H. O. Monson and M. M. Sluyter, "Containment of EBR-II," *Proceedings Second International Conference on the Peaceful Uses of Atomic Energy*, Geneva, Switzerland, 11, p. 124 (1958).
4. S. Timoshenko, *Strength of Materials, Part II: Advanced Theory and Problems*, D. Van Nostrand Company, New York, New York (1941).
5. B. E. Gatewood, *Thermal Stresses*, McGraw-Hill, New York, New York (1957).
6. T. R. Bump, discussion of "Studies on Heat Transfer in Packed Beds," by S. Yagi and D. Kunii, *International Developments in Heat Transfer, 1961 International Heat Transfer Conference*, Aug 28-Sept 1, 1961, Boulder, Colo., pp. D-249 and D-250.
7. M. Jakob, *Heat Transfer*, Wiley, New York, V. I (1949), pp. 373-379.
8. T. R. Bump, *Three-dimensional Heat Transfer Program*, Argonne National Laboratory Program No. RE330X.
9. T. R. Bump, *Fins in Series*, AIChE J. 11(1):174-181 (Jan 1965).
10. J. H. Rushton, *Mixing of Liquids in Chemical Processing*, Ind. Eng. Chem. 44, 2931 (1952).
11. H. Fossett and L. E. Prosser, "The Application of Free Jets in the Mixing of Fluids in Bulk," *Proc. Inst. Mech. Eng.* 160; 224-232 (1949). (Also p. 240.)