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FULL SCALE CLOSURE TEST

by

J. F. Patterson, S. P. Wnuk and E. A. Bemis

September 20, 1962

MASTER

Submitted to

**U. S. ATOMIC ENERGY COMMISSION
NORTHERN STATES POWER COMPANY**

and

CENTRAL UTILITIES ATOMIC POWER ASSOCIATES

by

**ALLIS-CHALMERS MANUFACTURING COMPANY
ATOMIC ENERGY DIVISION
Milwaukee 1, Wisconsin**

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ALLIS-CHALMERS MANUFACTURING COMPANY

Under

Agreement dated 2nd Day of May 1957, as Amended
betweenAllis-Chalmers Mfg. Co. & Northern States Power Co.
under

AEC Contract No. AT(11-1)-589

September 20, 1962

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September 20, 1962

PATHFINDER ATOMIC POWER PLANT
FULL SCALE CLOSURE TEST

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FOREWORD

One of a series of reports on research and development in connection with the design of Pathfinder Atomic Power Plant, this particular report deals with tests performed on a full scale prototype of the closure designed for the Pathfinder reactor vessel.

The Pathfinder plant will be located at a site near Sioux Falls, South Dakota and is scheduled for operation in late 1962. Owners and operators of the plant will be the Northern States Power Company of Minneapolis, Minnesota. Allis-Chalmers is performing the research, development and design as well as being responsible for plant construction.

The U.S. Atomic Energy Commission, through Contract No. AT(11-1)-589 with Northern States Power Company, and Central Utilities Atomic Power Associates (CUAPA) are sponsors of the research and development program.

The plant's reactor will be of the Controlled Recirculation Boiling Reactor type with Nuclear Superheater.

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Abstract

This report covers the tests performed on a full scale prototype of the closure designed for the Pathfinder reactor vessel. This head was bolted to another head designed to simulate the reactor vessel. The design pressure of the facility is 700 psig at 500 F.

Strain gages and brittle coatings were used to determine the stresses in the closure. Dial indicator gages were used to measure flange rotation after the head was secured and while it was subjected to hydrostatic pressure.

Tests were performed at reactor operating temperature and pressure to evaluate the closure and gasketing. The vessel was thermal cycled, and temperature gradients in the head measured to establish allowable heating and cooling rates.

Hydraulically operated bolt tensioners were used to seal and open the closure. Special tools and procedures were developed for underwater operation.

1.0 INTRODUCTION

1.1 General Discussion

The general arrangement of the Pathfinder vessel within the containment shell is shown in Fig. 1. The reactor vessel head is located below a 20 ft deep pool of water which serves as a biological shield during reactor operation and also during reactor shutdown. A stainless steel shell and metallic insulation provides a thermal barrier between the hot vessel head and cold pool water. A removable insulated seal ring provides access to 48 three-inch studs and nuts which secure the vessel head.

The shield pool will never be drained except under special circumstances. The 3-inch closure nuts (more than 18 ft below the surface of the pool) will be removed with hydraulically operated bolt tensioners and long-handled tools adapted for underwater operation.

The closure test vessel, Fig. 2, consists of a full scale model of the Pathfinder vessel head bolted to a second head which was designed to simulate the actual vessel.

1.2 Test Program

The test program consisted of a number of extensive tests on the vessel cover, flanges, bolting, gaskets, and tools used in removal and replacement of the cover under water. The tests were performed at the Allis-Chalmers Greendale Laboratories and involved tests of the vessel head and other components under various operating conditions. Underwater tests - concerned with removal and replacement of the head under water - were performed in the underwater handling facility.

Fig. 1...Elevation View - Reactor Vessel and Pump Layout
Within Containment Vessel (43-002-024, Sheet 1)

FULL SCALE CLOSURE TEST

Fig. 2...Full Scale Reactor Vessel Closure Test (203891)

In searching for gaskets to meet Pathfinder requirements it was found that little data or operation experience was available on gaskets of such size, and operating under the design conditions. In view of this lack of experience, and since complete sealing of the vessel is mandatory, it was desirable to test one or more of the most promising types of gaskets.

One of the most important considerations in establishing reactor refueling procedures was minimum plant down time. Hydraulically operated bolt tensioners appeared well suited for underwater removal and replacement of the bolted closure in the minimum amount of time. A set of bolt tensioners, designed for the Pathfinder, was tested both in air and under water along with other tools required for head removal.

Strain gages were used to measure stresses in the head, flange and bolting. Brittle coatings were also used to find points of maximum stress.

Preliminary bolting-down tests provided stress data, gasket seating data, and bolt tensioner experience. Hydrostatic tests were used as a proof-test of the vessel cover, gaskets and bolting and also provided further experimental stress data. Cycling steam tests at design temperature and pressure supplemented the hydrostatic tests and provided further data under actual operating conditions.

Temperature cycling and steady-state elevated temperature tests were run to evaluate the gaskets, bolting and flanges under operating conditions. A thorough analysis of the temperatures on the inside and outside of the closure for various heating and cooling rates would provide a basis for establishing heating and cooling rates at reactor startup and shutdown. Thermocouple probes located in holes inside the flanges were used to

provide information on the temperature gradients for various cooling rates. By running tests with the water level above or below the flanges, the effect of water level on cooling rates could be determined.

A thermal sleeve, similar to that used on the reactor vessel feedwater inlet connection but of much smaller scale, was installed at the cooling water inlet of the test vessel. Thermocouples installed on this sleeve provided information on the temperature distributions in the sleeve.

Finally, underwater tests provided actual experience unbolting and bolting under 18 ft of water along with thorough testing of special tools and equipment designed for that purpose.

2.0 DESCRIPTION OF CLOSURE TEST EQUIPMENT

2.1 Test Vessel (Fig. 2)

The closure test vessel consists of two major parts; the full scale reactor vessel closure and a flanged head designed to simulate the vessel. The two heads are bolted together using identical bolting as designed for the reactor vessel.

2.1.1 Test Closure

The test closure is a full scale duplicate of the Pathfinder reactor vessel closure. It consists of a hemispherical spun head welded to a flange machined from a ring forging. The head is 2-1/2 inches thick; 2-3/8 inches of carbon steel ASTM 212 firebox quality, grade B with a 1/8 inch minimum thickness of bonded 304L stainless steel cladding. The flange is of ASTM A105 Gr II carbon steel and on the inside is clad with a 1/4 inch of stainless weld overlay. The gasket surface is 3/8 inches thick "Inco-A" weld overlay. The surface was machined to a finish better than 63 RMS. The surface hardness, measured three months after deposit of the overlay, indicated a BHN 181. Twenty control rod drive tubes penetrate the hemispherical head. Each tube is a 3-1/2 inch schedule 40 pipe welded with a full penetration weld. Ten of the drive tubes are Inconel and are welded to the carbon steel head with Inco A or BP-85 weld rod. Both rods were tested to establish the most suitable welding procedure. The BP-85 was an experimental rod not available commercially at the time the head was fabricated. The latter rod proved to be more suitable. Carbon steel piping was used for the remaining control rod drive tubes, to save the time and expense, once the

Inconel procedures had been perfected and had been decided that the tubes would have to be relocated or a new head fabricated for the Pathfinder reactor. Control rod drive tubes No. 5, 8, 9, and 12 were flanged to provide access for instruments and safety devices. The rest of the drive tubes were capped.

Forty-eight 3-1/4 inch bolt holes were bored through the flange on a 100 inch diameter bolt circle. The stud holes are numbered as shown in Fig. 3. Studs No. 1, 18, and 33 are bored to 3.375 inch to accommodate three guide stud covers.

2.1.2 Lower Head

The lower head was designed to behave similarly to the actual reactor vessel. It consists of a 3 inch thick, 2:1 elliptical head welded to a ring forging. The head is carbon steel, ASTM 212, grade B. The inside and outside surfaces of the flange and head were metal-sprayed 0.012 inches thick with aluminum, of 99% purity, for a corrosion resistant coating. Forty-eight stud holes on a 100 inch bolt circle were tapped to accept the three inch studs. The gasket surface is of "Inco-A" weld overlay similar to the closure. Three months after weld deposit, a Poldi portable Brinell hardness tester indicated 180 BHN.

A 6 inch 600 lb flanged nozzle welded into the bottom of the head provided a heater mount. Two gasket bleed-off holes are located in the gasket surface exactly between the two gaskets. Four support lugs welded to the ellipsoidal head hold the head in a horizontal position. Four studs welded into the inside of the head are positioned so that the closure lifting rig could be used to lift this head also. Fig. 4 shows the lower head prior to metal spraying.

Fig. 3...Plan View of Full Scale Closure Test (43-025-032)

Fig. 4...Full Scale Closure Test - Lower Head (206770)

2.1.3 Gaskets

The operating conditions require that the gasket be metallic. Since the vessel closure is designed to be opened every 3 or 4 months, an inexpensive, commercially available gasket should be used.

Results of a thorough literature search, followed by a survey of the types of gaskets commercially available in the size required, narrowed the search to a few basic types. A corrugated metallic gasket was selected to be the first type tested. The corrugated metallic gasket has some distinct advantages over other types for this particular application. The corrugated gasket requires less compressive load than a solid metallic type gasket and it is considerably less expensive than an "O" ring.

A set of two Johns-Manville style 929 corrugated metallic gaskets were used to seal the closure. The set consists of an inner and outer gasket; the inner gasket is slightly thicker to compensate for flange rotation and to ensure adequate loading on this inner gasket. A diagram of the gasket cross-section is shown in Fig. 5.

The inner gasket is constructed of a 0.019 inch thick type 316 stainless steel shell and top washer and a 1/32 inch soft iron corrugated core. This gasket is 91-1/2 in. I.D. by 94 in. O.D.

The outer gasket is identical in construction except that the soft iron core is only 0.019 in. thick. The outer gasket is 94-1/2 in. I.D. and 96-1/2 in. O.D.

Although this gasket joint design imposes no limitation on the gasket compression, with the bolt loads involved, the gasket corrugations will not become fully flattened.

Fig. 5...Gasket Cross Section (43-025-065)

The gasket hardness, as measured with a Poldi portable Brinell hardness tester, varied from 90 to 138 SHN.

To obtain a seal with any gasket, it is necessary for the gasket to deform and fill the irregularities and imperfections in the flange. This may be done by direct compression of the gasket material until it yields and completely fills flange imperfections or by attrition whereby the gasket is deformed by shear action, or by a combination of compression and shear. Direct compression of a gasket has several drawbacks: (1) this usually calls for large bolt loads, (2) unless the gasket material is soft enough, brinelling of the flanges (permanent indentation) is likely to occur, and this is undesirable if the joint is to be opened often.

The need for large compressive loads and extremely hard flange surfaces is overcome by employing a corrugated gasket properly designed for the conditions. A seal is obtained by a combination of compression and attrition of the gasket material; and to visualize the action of this gasket, note in Fig. 5, that the crimped shell provides four thicknesses of metal at the inner and outer rim of the gasket. As the two flanges come together, the rim is compressed between the flanges; further compression increases the compressive load on the rims and also tends to straighten out the corrugations. As the corrugations straighten out, the rims are forced outward causing a shearing action between the contact of the gasket rim and the flange. It is this shearing action combined with the direct compression which causes the gasket to fill the imperfections of the flange. The seal is obtained at the rims of the gasket while the corrugations provide the lever required to cause attrition.

The advantages of a corrugated metallic gasket for this application can be summarized as follows: (1) the gasket can be adequately seated without the extremely large bolt loads that would be required for compressive seating for some gasket types, (2) the gaskets need not be plated with a soft material, (3) an extremely hard flange material is not necessary, (4) the surface finish of the flange can be 63 RMS which can be achieved by ordinary machining with no grinding or polishing necessary, and (5) the gaskets are inexpensive and commercially available.

2.1.4 Studs and Nuts

The forty-eight 3-in nominal diameter studs used to bolt down the closure were made from a ferritic stainless steel, ASTM-A437-59T. The material is heat treated to BHN 285/347 to give an ultimate tensile strength of 140,000 psi and a minimum yield strength of 120,000 psi. A 5/16 in. hole was drilled through the center of each stud for measuring stud elongation. A 7/16 in. hardened 440 C stainless steel ball was held in place at the bottom of the 5/16 in. stud hole with a set screw, providing a reference point for measurement of stud elongation. The surface at the top end of the stud provides the other reference surface from which a dial gage extensometer is used to measure stud elongation.

A taper thread of 0.005 inches per inch of thread length was provided on the vessel flange end of the stud to improve the load distribution along the length of the thread and to reduce the stress concentration on the first few load bearing threads.

The threads were a ground Class 2 fit with a modified unified 8N profile, providing a radius at the root of the thread.

Because of the long length of thread on the nut end of the stud it was impractical to taper these threads and, therefore, the taper was provided on the nut.

The nuts were made from the same material as the studs except that it was used in the annealed condition. The annealed material has a minimum yield strength of 85,000 psi.

A gear shape was cut on the periphery of the upper portion of the nut to facilitate bolt tensioner application. The studs and nuts were numbered 1 to 48 and stamped so that there would be no interchange of parts throughout the test.

2.2 Test Loop

The arrangement of equipment, valves and piping is shown in Fig. 6. The lower half of the test vessel was supported on a stand made from wide flange beams, reinforced with angles and pipe. A set of lubrite plates at each of the four support columns permitted the vessel to expand thermally without stressing the support stand.

All control rod drive tubes were vented individually at the top through 1/4 in. valves and piping was vented by means of a similar valve located at the uppermost part of the piping system.

An electric immersion heater, flange mounted to the lower head, was used to generate steam. This heater could be operated at output levels up to 50 kw in approximately 4 kw increments. A set of six switches mounted on the control panel controlled this variable output.

Fig. 6...Schematic of Closure Test (43-024-824)

A 10 gpm reciprocating pump was used to supply makeup water or to inject cooling water into the vessel. The injection of cooling water could be controlled from 0 to 10 gpm by adjusting the by-pass valve R (Fig. 6). Valve P was used to isolate the pump from the loop and a relief valve provided protection for the pump in case it was started with valve P closed.

A 2-1/2 in. blow-off valve (B) was used for rapid cooling. This valve discharged into a 5 in. pipe that extended through the roof of the building. The vessel was drained from a pipe at the bottom of the lower vessel which emptied into the 5 in. stack. Drain valve D controlled drainage. A drain valve was located at the bottom of the 5 in. stack. A steam supply valve controlled the steam supply from the heat transfer loop. This steam entered the vessel through a pipe which discharged at the dome of the closure in the downward direction. For blow-off operations, the steam was exhausted at this point.

The water level in the vessel was visually indicated on a full length magnetic liquid level indicator. An auxiliary electronic liquid level system, based on the conductivity of the water, was used to make an electric circuit between an insulated probe and the vessel. Three probes were used, one located 6 in. above the top of the heater elements, the second at the parting joint of the flanges and the third about 10 in. from the top of the vessel. The low water level probe was connected into the heater control circuit to shut off the heater power should the water level become dangerously low.

Three independent safety devices were used to prevent overheating or overpressurizing the vessel. Two 3/4 in. safety valves, set to open at 720 psig, were flange mounted on the control rod drive tubes. The discharge line from the safety valves emptied into the 5 in. stack to the roof. A mercoird pressure switch installed in the vessel piping would shut off power to the heaters if the pressure exceeded 660 psig. A thermocouple probe, located in the lower vessel and extending 10 in. into the water, measured water temperature. A cam operated controller operating from the first channel of the temperature recorder control maintained the water temperature by controlling power to the heaters. A 220 volt and a 440 volt switchbox mounted on the panel could be used to interrupt power to the heaters or to the reciprocating pump.

All valves, controls, the temperature recorder and pressure gage were mounted on the control panel (Fig. 7).

Before commencing any high temperature tests, the vessel with its associated piping was fully insulated - both upper and lower heads being covered with 2 in. thick calcium silicate blocks.

A 36 in. x 1/4 in. thick ring insulated with calcium silicate sections surrounded both flanges. The ring provided protection in case of gasket rupture, as well as insulating the flanges. The ring was made from four quarter-segments which were bolted together to form a ring. Blanket insulation was used to cover the studs.

2.3 Bolt Tensioners and Accessories

The bolt tensioner shown in Fig. 8 (Dwg. 43-002-454) was specifically

Fig. 7...Control Panel for Closure Test

Fig. 8...Pathfinder Bolt Tensioner Isometri

designed for pre-stressing the studs on the Pathfinder closure. These tensioners are hydraulically operated and use two hydraulic cylinders in series with water as the hydraulic fluid. To operate the tensioner, the puller rod is threaded on to a 3 in. length of stud thread that extends above the nut for this purpose. The housing of the tensioner rests on the flange area between the nut being tightened and the two adjacent nuts. The two hydraulic cylinders are arranged to exert an upward force of up to 336,000 lb on the puller bar when the cylinder chambers are pressurized. This force on the puller bar is transmitted to the stud and the reaction forces are transmitted to the flange through the bolt tensioner housing. With the tensioner in the pressurized condition, the nut is rotated through a gear chain arrangement on the tensioner which engages with the nut. The amount of stud prestress is determined by the amount of nut rotation.

The bolt load is transmitted to the stud and nut assembly upon release of hydraulic pressure. The tensioners are designed to accommodate an additional preloading overload which will be lost to the nut and flange when they take the load from the tensioner. This additional pre-loading may be different with different types of gaskets and is one of the unknowns determined by testing.

The tensioner is held together by a top and bottom plate and four tie rods which also serve to connect the tensioner cylinder to the housing. The cylinders, the inner sleeve, the bases of both stages, and the pistons of both stages are made of heat-treated alloy steel. The base and piston act as supports for the aluminum piston heads which carry the high pressure packings. With this arrangement, any relative motion is always between steel and aluminum.

The inner surfaces of the two cylinders and the outer surface of the inner sleeve are hard chrome plated and polished to resist corrosion by water. All other surfaces have a protective coating of paint.

Spherical washers are located between the top of the moving piston and the holding nut (which is locked to the puller bar) to correct for any mechanical misalignment so that the pull on the stud is in a straight line. These spherical washers are made from case hardened AISI 4140.

The four return springs, built into the top plate, are used to return the pistons to their original positions upon the release of hydraulic pressure.

The tensioner housing is an alloy steel casting and is heat treated to a minimum yield point of 75,000 psi. The contact area is approximately 14.5 sq. in. The housing is shaped to obtain guidance from the two adjacent nuts. The lower edge of the housing is chamfered where it would make contact with an adjacent nut to further assist in locating the tensioner during lowering.

Fig. 8 shows the gear drive arrangement. The gears are made of heat-treated steel and operate in hardened steel bushings with hardened steel thrust washers. The teeth on the driving gear and on the nut are chamfered to ensure engagement as the tensioner is lowered into place.

A spring support, built into the puller bar, prevents any impact between the puller bar socket and the stud. This support contacts the top of the stud as the tensioner is lowered and the spring force is sufficient to support the weight of the puller bar assembly. A slight downward force

is required to engage the puller bar and stud threads. A thrust bearing located in the spring support plate reduces the turning effort to a minimum.

Description of Accessories (Fig. 9)

The air operated pump for the operation of these tensioners is a SC Engineering Corporation's Model No. 10-500-6. A hand pump is also provided to be used as a standby in case of failure of the air supply or air pump. The two valves at the connection manifold may be operated in the open position until such time as one pump is to be removed from the line when its particular valve should be closed.

The gage panel consists of an air filter, air regulator, and air lubricator, manufactured by WATTS Regulator Co., Model No. 616. These units are located on the front left side of the gage panel. The push button valve in the front of the panel is the main air valve and is merely pushed down and released to open and close. Directly behind it is the air gage and the yellow handled valve between the two gages is the release valve. The other gage is the hydraulic gage reading in psi and thousands of lb total pull.

2.4 Stud Galling Test Facility

Special consideration must be given to galling characteristics of the closure bolting. This item is particularly critical since bolting operations are carried out under water.

The galling test assembly (Fig. 10) is a simulated flange arrangement for one stud. The assembly was submerged in a barrel of demineralized

Fig. 9...Bolt Tensioner Accessories (69B-0-7)

Fig. 10...Galling Test Assembly (43-025-069)

water and loaded with a bolt tensioner. Nuts made of several materials, some electroplated with silver or cadmium, were tested for wear and galling characteristics. Various lubricants were also tested.

2.5 Facilities for Testing Underwater Removal Procedures

A "mock up" of the vessel flange and bolting consisted of two galling test assemblies (Fig. 11) spaced on a 100 in. center, equal to the Pathfinder closure bolt circle.

The bolt tensioners described in Section 2.3 were adapted for removal and replacement of the Pathfinder vessel head under approximately 18 feet of water. Appropriate handling rigs and long handled tools were designed for this purpose. These items are described in the following paragraphs.

Guide Stud Covers

The guide stud covers (Fig. 12) provide a method of centering the closure head on the vessel and a method of protecting the stud threads from mechanical damage during head placement and removal. Guide stud holes are number 1, 18 and 33 (Fig. 3) and are held to a closer location tolerance than the other 45 holes and threads. The guide stud covers will fit through only the enlarged diameter guide stud holes in the head. Since the guide hole spacing is not equal the head can only be assembled in one position. The covers are cylindrical tubes with tapered tops. A quick release pin holds each cover in place during head removal.

Bolt Tensioner Handling Rig

For underwater operation the bolt tensioners are suspended from the

Fig. 11...Closure Mockup for Underwater Tests (69B-0-2)

Fig. 12...Guide Stud Covers (69B-0-4)

support structure shown in Fig 13 (43-501-100 Sht. 3). This structure has a cross beam made of 6 in. channels with a bracket for centering the rig radially from the vessel head lifting structure. Each tensioner is suspended from a support leg by three 1/2 in. diameter rods. The rods supply no restraint to the tensioner during the tensioning operations. A 2 in. schedule 40 pipe is attached to the puller bar with a telescoping connection. This pipe is supported by a thrust ball bearing at the top of the leg and is turned by a handwheel mounted on the upper end. A 2 in. schedule 40 pipe is attached to the nut drive shaft and is supported by the shaft during the tensioning operation. A degree wheel is mounted on the upper end of the pipe to measure degrees advance of the nut drive shaft. The pipe terminates in a 2 in. hex bar to be used with a socket wrench.

Nut Handling Tool

The nut handling tool, Fig. 14, is essentially a long handled socket wrench with spring loaded latches for gripping a groove in the geared closure nuts. The top end has a hex head for a torque wrench, a handwheel for turning the nut, and a handle for releasing the nut. The main tube is sealed at the top end and is open at the bottom, to make the assembly float when the socket is slightly above the top of the studs. The water rises to a depth of 77 in. inside the tube. The lower end of the tool is an internal gear assembly to engage the gear on the nuts.

Adapter for Stud Removal

An adapter for stud removal (Fig. 15) has a square socket that matches the square end of the stud and a gear similar to the gear on the nuts. A

quick release pin attaches the adapter to the stud and the stud is then handled with the nut handling tool. A separate long-handled tool is used to install the stud adapter and the guide stud covers.

Gasket Handling Rig

The gasket handling rig (Fig. 16) is an eight point suspension system to hold the metallic gaskets in a horizontal position. An Electro Products Model PS 30 battery eliminator converts 115 volts, 60 cycle power into 12 volt D.C. to activate eight electro magnets that support the gaskets. The magnets are adjustable radially to handle each gasket separately.

Extensometer for Underwater Use (Fig. 17)

The extensometer consists of an Ames dial gage indicator mounted in a cylindrical assembly designed to prevent the dial gage from being exposed to shield pool water. The indicator has a range of 0.25 in. reading in increments of 0.0005 inches. The indicator is mounted on the end of a 20 ft tube along with a 10 power telescope and a sight window.

Stainless steel rods are inserted in the central hole of each stud, eliminating the need for a long extension on the extensometer which can be easily damaged. Readings are taken before and after loading the stud by looking through the telescope. The sight window at pool level prevents ripples at the surface of the pool from interfering with the visual readings.

QTY	IT.	DESCRIPTION	MATERIAL	PART NUMBER	WT.
2	52	HANDWHEEL (2" DIA. X 1/2" THK)	ALUMINUM	THIS	052
2	53	1/2" UNF. 2A HVY. S.F. HEX. NUT	STEEL	THIS	053
2	54	PRINT PROC. SPEC.	ACD 5-5		
2	55	LIGHT GRAY (ASA #6) MIDLAND 9X220	Q0008987		
2	56	WELD ROD	STEEL	THIS	056
2	57	BOLT TENSIONER ACCESSORIES	43-101-312	801	
2	58	WELD ROD (ASTM A198-88T C4E 308)	THIS	058	
2	59	2" SCH. 40 X 21' 4 1/2" LG. PIPE	STEEL	THIS	059
2	60	1/8" BEVEL WASHER	Q0007575		
4	61	RIB. 1/4" THK X 4 1/2" X 6 LG. BAR	STEEL	THIS	061
4	62	MED. DARK GRAY (J.B. SIPS #765)	Q0071761		

QTY	IT.	DESCRIPTION	MATERIAL	PART NUMBER	WT.
1	1	BOLT TENSIONER LIFTING RIG	43-501-100	801	
2	2	6 X 2 (8.25" FT) X 8" LG. CHANNEL	ASTM A-7	THIS	002
4	3	SUPPORT. 1/4" X 4 X 1/2 (12.6" FT) X 21" LG. ANGLE	ASTM A-7	THIS	003
2	4	2" SCH. 40 X 21' 4 1/2" LG. PIPE	ASTM A-7	THIS	004
2	5	SHIFT PLATE. 1/4" THK X 10 X 13 LG. PLATE	ASTM A-7	THIS	005
2	6	TOP BASE. 1/4" THK X 10 X 13 LG. PLATE	ASTM A-7	THIS	006
2	7	SUPPORT END. 1/4" THK X 3 X 6 LG. BAR	ASTM A-7	THIS	007
2	8	SUPPORT END. 1/4" THK X 3 X 6 LG. BAR	ASTM A-7	THIS	008
2	9	SUPPORT END. 1/4" THK X 3 X 6 LG. BAR	ASTM A-7	THIS	009
2	10	SUPPORT END. 1/4" THK X 3 X 6 LG. BAR	ASTM A-7	THIS	010
6	11	DIAL HOLDER. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	011
2	12	DIAL. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	012
2	13	INNER BEARING MOUNT. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	013
2	14	OUTER BEARING MOUNT. 1/4" THK X 5/8" DIA.	ASTM A-7	THIS	014
1	15	PIN. 1/4" DIA. X 3/2 LG.	ASTM A-7	THIS	015
1	16	GUIDE BUSHING. 1/4" DIA. X 3/4 LG. BAR	ASTM A-7	THIS	016
2	17	SUPPORT BRACKET. 1/4" X 3/4 (12.6" FT) ANGLE	ASTM A-7	THIS	017
2	18	SUPPORT BAR. 1/4" THK X 2 X 10 LG. BAR	ASTM A-7	THIS	018
2	19	GUSSET. 1/4" THK X 2 X 10 LG. BAR	ASTM A-7	THIS	019
6	20	CENTERING GUIDE. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	020
6	21	TENSIONER SUPPORT ROD. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	021
4	22	SHIM. 1/4" THK X 2 X 10 LG. STRIP	ASTM A-7	THIS	022
2	23	SHIM. 1/4" THK X 2 X 10 LG. STRIP	ASTM A-7	THIS	023
2	24	HEXAGON HEAD. 1/4" THK X 2 LG. BAR	ASTM A-7	THIS	024
2	25	END PLUG. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	025
2	26	DRIVE BAR SOCKET. 1/4" DIA. X 2 LG. BAR	ASTM A-7	THIS	026
2	27	WHEEL STUD. 1/4" DIA. X 4 1/2 LG. BAR	ASTM A-7	THIS	027
2	28	PULLER BAR SOCKET. 1/4" DIA. X 2 1/2 LG. BAR	ASTM A-7	THIS	028
2	29	GUIDE BAR. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	029
2	30	BASE. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	030
2	31	BASE. 1/4" DIA. X 1/2 LG. BAR	ASTM A-7	THIS	031
2	32	HOLLOW SPLIT PIN. 3/8 DIA. X 3/4 LG.	ASTM A-7	THIS	032
2	33	HOLLOW SPLIT PIN. 3/8 DIA. X 2 LG.	ASTM A-7	THIS	033
2	34	WHEEL STUD. 1/4" DIA. X 4 1/2 LG. BAR	ASTM A-7	THIS	034
2	35	WHEEL STUD. 1/4" DIA. X 4 1/2 LG. BAR	ASTM A-7	THIS	035
2	36	1/2" UNF. 2A X 1/2 LG. HEX. HD. CAP. SCREW	ASTM A-7	THIS	036
2	37	BOTTOM BASE. 1/4" THK X 10 X 13 LG. PLATE	ASTM A-7	THIS	037
12	38	1/2" UNF. 2A X 1/2 LG. SLOTTED FLAT HD. CAP. SCREW	ASTM A-7	THIS	038
8	39	1/2" UNF. 2A X 1/2 LG. SLOTTED FLAT HD. CAP. SCREW	ASTM A-7	THIS	039
12	40	1/2" UNF. 2A X 1/2 LG. HEX. HD. CAP. SCREW	ASTM A-7	THIS	040
44	41	1/2" UNF. 2B HVY. S.F. HEX. NUT	ASTM A-7	THIS	041
6	42	1/2" UNF. 2B HVY. S.F. HEX. NUT	ASTM A-7	THIS	042
43	43	1/2" MEDIUM SPRING LOCKWASHER	ASTM A-7	THIS	043
6	44	1/2" PLAIN ROUND WROUGHT WASHER	ASTM A-7	THIS	044
81	45	1/4" MEDIUM SPRING LOCKWASHER	ASTM A-7	THIS	045
6	46	1/4" UNF. 2A X 1/2 LG. HEX. HD. CAP. SCREW	ASTM A-7	THIS	046
4	47	1/4" UNF. 2A X 1/2 LG. HEX. HD. CAP. SCREW	ASTM A-7	THIS	047
4	48	1/4" UNF. 2A X 1/2 LG. HEX. HD. CAP. SCREW	ASTM A-7	THIS	048
24	49	1/4" UNF. 2A X 1/2 LG. HEX. HD. CAP. SCREW	ASTM A-7	THIS	049
6	50	1/4" UNF. 2A X 1/2 LG. HEX. HD. CAP. SCREW	ASTM A-7	THIS	050
2	51	1/4" UNF. 2A X 1/2 LG. HEX. HD. CAP. SCREW	ASTM A-7	THIS	051

Fig. 14...Nut Wrench (43-501-098)

Fig. 15...Adapter for Stud Removal (69B-0-6)

Fig. 16-B...Gasket Handling Rig (43-501-099, Sheet 2 of 2)

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Fig. 17-B...Extensometer for Underwater Use (43-501-101, Sheet 2 of 2)

3.0 INSTRUMENTATION

3.1 Strain Measurement

Resistance wire strain gages were located on the inside and outside surfaces of the closure as shown in Fig. 3 and Fig. 18. The locations chosen were based on theoretical calculations, and strain gage rosettes were closely spaced in predicted areas of high stress gradient. Two rows of rosettes were installed on the head and flange, one row located on a radial line through bolt No. 1 and the second row on a radial line exactly between the centers of bolts No. 14 and 15. At many locations only two gage elements were used, oriented in the circumferential and meridional directions. In order to justify the use of only two gages at these locations, one must assume that the principal stresses are in circumferential and meridional directions. To satisfy this condition the gages must be attached at an axis symmetry and the bolt load must be sufficiently uniform. To check this assumption, several three-element strain rosettes were bonded to the closure. Graphical analysis indicated that the principal stress directions were within ± 4 degrees of the assumed directions. Single gages, orientated to measure axial strain in the studs, were placed on studs No. 1 and 25 at the mid point of the unthreaded portion on opposite sides.

3.1.1 Strain Gage Location

The four strain rosettes on the head around control rod drive tube No. 14 were placed on at a later date. Two element A-7 strain rosettes were applied to measure circumferential and meridional strains at these

(X) - EQUALLY SPACED BETWEEN ADJACENT GAGES
 GASKET SURFACE IS REFERENCE FOR ALL
 VERTICAL DIMENSIONS.
 INDICATES GAGE LOCATIONS

Fig. 18...Strain Gage Locations (43-024-413)

four locations, shown in Fig. 3. The principal stress directions were determined from stress coat cracks made during a hydrostatic test.

The strain gage locations were marked off on the closure, using the gasket surface as the reference surface. The surface preparation for the strain gages was made and type A-7 strain gages were applied with Duco cement. A-18 gages were used at the fillet radius for gage locations X-17, X-18, X-19, and X-20 as shown in Fig. 18.

Compensating gages were mounted on blocks of flange and head material and these gage blocks were located at each row of strain gages. The blocks were firmly cemented to the parent material with plastic steel putty in a manner which provided intimate contact but did not induce any strain in the blocks. Two compensating gage blocks were similarly installed on the inside of the vessel, one gage block serving a row of gages. In every case, the compensating gage was from the same lot as the active gages. Strain gages located on the outside of the vessel were waterproofed with Tatnall GW-1, covered with liquid plastic steel for mechanical protection, and were connected to the switch box through 14 ft of Belden 8014 lead wire. Lead wires for the strain gages on the inside were brought out of the vessel through a special lead exit connection bolted to one of the flanged control rod drives tubes. This connection was similar to that used by Lawton (Ref. 1). The lead exit connection consisted of a 3-1/2 in. schedule 80 pipe, capped on one end and flanged on the other. Three 6 in. long 1 in. pipes were connected to the 3-1/2 in. pipe through pipe couplings. The free end of the 1 in. pipes were

capped. Twenty holes, the same diameter as the lead wire, were drilled through the cap of the one in. pipe. Once the lead wires were threaded through the holes in the one in. pipe cap, the pipe was filled with molten woodsmetal. This lead exit assembly was used to test waterproofing methods prior to the test. This method of bringing lead wires out of a vessel did not leak at pressures up to 3500 psig.

Belden 8482 wire was used inside the vessel, while Belden 8014 wire was used from the lead exit connection to the switch box. Extreme care had to be exercised in handling Belden 8482 so that it did not become kinked or damaged.

All gages were tested for resistance to ground and for proper bonding prior to waterproofing.

3.1.2 Waterproofing

Tests were conducted on various waterproofing methods prior to the closure test. The lead exit assembly provided with a blind flange was used as the pressure vessel for these tests. The method which was developed during these tests is worth mention because it is reasonably easy to apply and will hold up in water for at least a week at 1050 psig.

Waterproofing Procedure

Materials

1. 3M adhesive backed rubber sandblasting tape.
2. Mystic tape.
3. Devcon B Plastic steel liquid.

4. Devcon A Plastic steel putty.
5. 3M sealer, formula, EC-801 with EC-807 hardener.

Procedure

1. Place 3M sandblasting tape on gage locations extending about 1/16 in. beyond outline of gage.
2. Lightly sandblast the gage area with sand.
3. Remove tape and place gages using standard procedures.
4. Connect lead wires.
5. Place mystic tape over gages.
6. Clean area with acetone (including lead wires)
7. Brush on liquid plastic steel.
8. Allow to dry for 4 hours.
9. Brush on plastic steel putty, thoroughly working it in around lead wires.
10. Allow to dry for 4 hours.
11. Mix EC-801 with hardener and brush it on the plastic steel, extending over the edges by at least 1/2 in. in all directions.

The completed waterproofing installations on the inside of the closure test vessel are shown in Fig. 19. At the same time that preliminary waterproofing tests were made, the effect of pressure on the A-7 gage installation was evaluated and it was found that the effect of 1050 psig hydrostatic pressure was no larger than the reading error of the Baldwin strain indicator employed.

Fig. 19...Waterproofing Installations on Inside of
Closure Test Vessel (211615-B)

3.1.3 Strain Measuring Instrumentation

All strain gage leads were wired into a switch box containing five L & N type 31-3 rotary selector switches. These switches are totally enclosed and employ silver alloy brushes operating on solid silver contacts. Each switch has four poles with eleven positions on each pole. The output of this switch box was connected into a 20 channel Ellis switch and balance unit. The output of the Ellis unit was connected to a Baldwin Model M strain indicator. This switching arrangement made the reading of strains quite efficient. The five L & N switches were rotated to position 1 and twenty strain readings were taken by switching only the Ellis unit. After taking twenty readings, all five L & N switches were advanced to position 2, etc., and readings were again taken. The switches employed were of sufficiently high quality that repeatable readings were always possible, the switching error being less than the reading error of the strain indicator. The instruments were set up on a platform at the top of the head lifting rig in order to keep lead wires out of the way when bolting operations were in progress. A top view of the closure test, staging, and instrumentation is shown in Fig. 20. Four strain gages were placed in position 21-2 and located 90° apart at bolt Nos. 1, 13, 25, and 37. These gages were used to monitor meridional strains in the head during bolting up operations. The gages were connected through Brush universal amplifiers to matching recorders. These recorders were also used to measure stresses in the studs during bolting up and unbolting operations and during the hydrostatic test.

Fig. 20...Top View, Closure Test Setup (211467)

Replacement Gages

Foil type strain gages replaced the A-7 and A-18 strain gages when these gages became damaged or wet. Tatnall type C6-141 gages were used with Eastman 910 cement to replace single A-7 gages, and type C6-121-R2B gages were used to replace rosettes. Many of the gages on the outside of the vessel needed replacement because they were damaged during bolting operations or by water splashing from the steam separator test which was operating nearby. Only a few of the inside gages needed replacement because they had been meticulously waterproofed and were well protected inside of the vessel.

With the various types of strain gages used, the gage factor on the instrument was set on 1.98 and a gage factor correction was made for each gage prior to the calculation of stresses.

3.1.4 High Temperature Strain Measurement

High temperature type C6-630 stripable back foil strain gages were mounted on inside and outside surface of two studs at midpoint. Short leads of nichrome wire, just long enough to reach through the insulation shield, were spot welded to the gage leads. The studs with gages and lead wire were placed in a furnace and cured for 24 hr at 600 F. The studs were then installed in the closure. The closure was bolted up except for the two strain-gaged studs which were left unstressed. The closure was brought up to operating temperature and pressure, held there for one week and then cooled down. The apparent strain curve was plotted for the gage and lead wire circuit during the hot cycle. The studs were

then tightened and strain readings were taken at various times during the rest of the hot tests. Data was taken during rapid cooling to determine the amount of bolt load lost during cooling. Strains were recorded on a Baldwin Model M strain indicator. Resistance of the gage circuit was recorded on a Wheatstone bridge. The gage temperature, resistance to ground, and circuit resistance were taken at the same time as a strain measurement was made.

3.2 Dial Indicator Gages

Dial gages were used to measure flange displacements and rotations during the bolting up and hydrostatic tests. The gages were located as shown in Fig. 21. Two identical rows of gages were employed, one row located between studs No. 6 and No. 7, the second row located 180° from the former. The gages were mounted on a 4 in. dia. vertical pipe which was welded to an 18 in. square base plate and bolted to the floor. Four 1 in. pipe braces were welded to the 4 in. pipe and base plate to further increase rigidity. The pipe was filled with concrete to dampen vibrations which were sometimes induced by the operation of other heavy equipment in the laboratory.

The rotation measuring rods were made from two 1 in. dia. x 54 in. long steel bars which were flanged and bolted to the upper and lower vessel flanges as shown in Fig. 21. A similar bar, bolted to the 4 in. vertical support pipe, was used to support dial gages measuring the movement of the rotation measuring rods. All dial gages were in 0.0001 in. increments except for those on the rotation rods which were in 0.001 in. increments.

DISTANCE FROM DIAL GAGE TO CG. IN INCHES		
	LOCATION #1	LOCATION #2
a	52.64	51.45
b	6.07	7.82
c	51.25	52.13
d	7.13	6.44
e	5.70	
f	5.10	
g	7.00	
h	3.53	
t	5.24	

Fig. 21...Dial Gage Locations (43-024-394)

Flange rotation is computed from dial gage measurement by the relation:

$$\phi_1 = \frac{A - B}{a - b}$$

$$\phi_2 = \frac{C - D}{c - d}$$

where, ϕ_1 equals rotation of upper head, Radians and,

ϕ_2 equals rotation lower head, Radians.

The downward displacements of the center of gravity (C.G.) of the upper flange, Y, is found using the relation:

$$Y = B - \phi_1 b.$$

The outward displacement, X, of the C.G. of the upper flange is given by:

$$X = G + \phi_1 g.$$

3.3 Bolt Load Measurement

An extensometer was used to measure the elongation of the studs from which bolt load could be easily calculated. The extensometer employed a dial gage with 0.0001 inch subdivisions and was mounted on a hollow rod, making contact with the chamfered reference surface inside the bolt hole. The dial indicator button rested on a rod which extended through the hollow base rod to rest on a set screw at the bottom of the stud. The two reference surfaces were located at the mid point of the thread height for the nut and flange. The total length over which this extensometer measured bolt elongation was 19.56 in. Experience with this extensometer indicated that elongation readings could be repeated within ± 0.0002 in., provided that the reference surfaces were clean.

3.4 Pressure Measurement

A 6 in. pressure gage with a 0 to 1500 psig pressure range was used to measure the vessel pressure. The gage was calibrated on a dead weight tester before and after the test. The gage was mounted in the control panel.

3.5 Gasket Leakage Measurement

Two gasket bleed-off ports located between the inner and outer gaskets extended through the vessel flange. A short length of pipe with a valve was threaded into the vessel at the exit of these bleed-off ports so that one or both of the bleed-offs could be closed off. A short manometer tube, connected to the valve outlet with plastic tubing, was located so that the crest of the tube was slightly above the level of the gasket joint. During hydrostatic tests, the channel between the gaskets was filled with water by pumping it through one bleed-off channel and permitting the air to come out of the other bleed-off hole. When all the air was purged, one bleed-off was closed and leakage measurements were made at the other. Leakage was collected in a beaker and weighed on a precision laboratory balance. Time during which the leakage was collected was usually 20 minutes or half-hour periods.

During the hot tests, the manometer was replaced by a small condenser which emptied into a beaker. The condenser consisted of a coil of copper tubing submerged in a 5 gallon bucket of cold water. The condenser tube inlet was connected to the bleed-off while the exit at the bottom of the coil was brought through the bottom of the bucket so that condensate could not be trapped in any part of the system. The beaker was capped with only

a slight air clearance around the inlet tube to prevent condensate evaporation,

3.6 Temperature Measurement

Iron-constantan thermocouples, located near the compensating strain gages and at the back of the switch box, were used to record temperatures during all the cold stress analysis tests. A 10 in. thermocouple probe located in the lower half of the test vessel was used to measure water temperature during the cold hydrostatic tests.

A Bristol 24-point scanner recorder calibrated in the 0 to 600 F range was mounted in the instrument panel and used for recording all temperature measurements. The instrument was internally compensated for cold junction temperature and no ice bath was used.

Since more than twenty-four thermocouples were used, a fifty-point cannon connector was used to connect blocks of thermocouples into the instrument. Three blocks of thermocouples were made up; block A for the thermocouples measuring flange and head temperatures, Fig. 22...block B for the temperatures on the stud and inside the flange stud hole, Fig. 22 and 23...and block C for the cooling water inlet thermal sleeve, Fig. 24. A complete set of temperatures for the thermocouple locations in any block were recorded by plugging the block into the cannon connection at the instrument. The water temperature probe located on channel 1 of the recorder remained connected into the recorder at all times.

The thermocouples located on the closure head and flange (block A) are shown in dwg. 43-024-458, Figure 22. Three thermocouple probes

Fig. 22...Thermocouple Locations for Closure Flange (43-024-458)

Fig. 23...Thermocouple Locations on Flange Stud (43-024-459)

Fig. 24... Thermocouple Locations on Cooling Water Inlet Thermal Sleeve (43-024-460)

were made up to measure temperatures on the inside of the flange. These probes were made from 1/2 in. rods milled on one side to accommodate the wire and with holes drilled through the rod at intervals, as shown in Fig. 21. Thermocouple junctions were made on the surface opposite the milled groove, the thermocouples protruding slightly and brazed in place. The assembly was driven tightly into the 1/2 in. drilled and reamed holes in the flange.

Thermocouples on the inside of the upper head were standard stainless clad assemblies silver-soldered to the vessel. The lead wires were brought out through standard connections at the control rod tube flange.

All other thermocouples were spark welded in place. The thermocouples on the inside of the lower vessel and on the thermal sleeve were brought out through a special connection in the cooling water inlet pipe.

4.0 TEST PROCEDURES

4.1 Preliminary

Prior to any testing the gasket surfaces of the test vessel were inspected for "waviness" and parallelism by the solder compression method. A continuous strip of 1/16 in. dia solder was laid along the vessel gasket surface, 1/2 in. from the inner edge. Another strip was laid 1/2 in. inward from the outer edge of the gasket surface. The closure was gently lowered to rest on the solder. The nuts were threaded down to contact with the flange and then each nut was rotated a sufficient amount to advance it by 0.010 in. The nuts were then removed and the closure taken off. The thickness of the continuous impression made in the solder was measured at each stud location with a micrometer. The overall variation in solder thickness was 0.009 in. and the largest variation from one stud to another was 0.002 in.

Brittle Coatings

Stress coat was used on the upper head, flange, and around penetrations at various stages during testing to determine points of maximum stress and principal stress direction. A suitable lacquer was selected for the existing temperature and humidity conditions. Calibration bars were sprayed along with the head. Results from these tests were used to locate strain gages around one of the head penetrations.

The bolt tensioners were placed on studs No. 20 and 44 and were loaded in steps as shown below.

<u>Step No.</u>	<u>Load - lb</u>	<u>Step No.</u>	<u>Load - lb</u>
1	50,000	4	200,000
2	100,000	5	250,000
3	150,000	6	300,000

The load was reduced to zero after each step for a period of time to allow the coating to creep back to the zero stress condition. One calibration bar showed cracks at the beginning of the test, one after step No. 3 and one at the end of the test. The strain values for each bar were (1) 925, (2) 900, and (3) 940 micro inches/inch. The first cracks appeared on the flange directly under the load, and at 90 degrees to the load, for the 200,000 lb loading. These cracks did not spread with further loading however, as the gaskets directly under the load were then almost fully compressed and further loading did not cause significant bending in the flange. No cracks appeared on the head, the junction of the head and flange or the top surface of the flange at any loading. Therefore, it was safe to proceed with bolting tests without fear of local yielding caused by excessive bolt tensioner loads.

4.2 Stud Loading Procedures

Two methods appeared feasible for loading the studs to a predetermined elongation: (1) use the pressure gage in the bolt tensioner hydraulic system to measure the load applied to each stud, and (2) advance each nut a predetermined number of degrees.

4.2.1 Pressure Measurement Method

The load applied to a stud by the bolt tensioner was measured directly on a pressure gage in the tensioner hydraulic system. The determination of the necessary bolt pre-stress to be applied to each stud is complex owing to the interaction of the following factors: elastic and inelastic

compression of the gaskets...flange rotation...flange bending...the amount of load lost when the nut takes the load from the tensioner. It was therefore necessary to experiment with various loading procedures and make corrections (from one test to the next) which seemed necessary to end up with uniform bolt loading. The loads applied to each set of studs for tests using this method were plotted along with the results.

Since two bolts located 180° apart were tightened simultaneously, twenty-four tightening operations are required to make a complete tightening pass over each stud.

Procedure for Tightening Nuts Using Bolt Tensioners

The following procedure was used for tightening nuts using the bolt tensioners:

1. Center bolt tensioner rig and adjust each tensioner so the it hangs vertical and fits well over each nut. Rotate the handling rig 90° and center tensioner rig again.
2. Set tensioners on studs No. 1 and No. 25.
3. Thread on puller bars.
4. Apply pressure.
5. Tighten nut.
6. Release pressure.
7. Remove puller bar.
8. Move to next set of studs according to sequence in Table 1 and repeat steps 3 through 7 above.

TABLE I

Operation Number	Bolt Numbers (See Fig. 3)
1	1 & 25
2	13 & 37
3	7 & 31
4	43 & 19
5	4 & 28
6	10 & 34
7	46 & 22
8	40 & 16
9	2 & 26
10	14 & 38
11	6 & 30
12	44 & 20
13	48 & 24
14	36 & 12
15	42 & 18
16	32 & 8
17	27 & 3
18	39 & 15
19	33 & 9
20	21 & 45
21	5 & 29
22	11 & 35
23	47 & 23
24	41 & 17

The bolt elongations were measured with the extensometer and additional bolting up passes were made until the bolt stress variations were within acceptable limits.

4.2.2 Degree of Nut Rotation Method

Using this method to tighten the studs, the bolt tensioner was used merely to provide the force required to stretch the stud so that the nut could be freely rotated the calculated number of degrees. A degree wheel, mounted on the bolt tensioner drive gear shaft, was used to accurately measure degrees of nut rotation. With an 8-threads-per-inch stud, 360°

rotation advances the nut 0.125 in. For a given average stud stress, the stud elongation, nut compression and flange rotation should be equal - and the amount of nut rotation needed to compensate for these effects should be constant. Therefore, the key to using this method rests on the ability to determine the nut rotation required to take account of the large variation in the initial thickness of the gaskets.

The initial thickness of the gaskets was measured at four places. The amount of gasket compression due to (1) the weight of the head, (2) 100 ft lb of torque on the bolts and (3) tightening the nuts a certain number of degrees, was measured by taking feeler gage readings of the gap between the two flanges. All nuts were tightened to 100 ft lb of torque to compensate for the initial variation in gasket thickness and to take the slack out of the bolt and nut threads. The nuts were tightened using the sequence of operations shown in Table I.

4.3 Initial Bolting Tests

Dial indicators were mounted and adjusted at this time and the temperature recorder was turned on. The bolt tensioner lifting rig (with tensioners mounted and hoses connected up) was centered on the vessel.

Just prior to bolting up, the following readings were taken:

- a. Dial gage
- b. Bolt extensometer
- c. Strain gage
- d. Strain gage resistance to ground.

The nuts were then tightened (by applying a torque of 100 ft-lb) and the initial bolt elongations were measured. The nuts were then successively retightened, using the bolt tensioner until the desired stud stresses were achieved. The strains in the head and stud were monitored on brush recorders during the bolting operations. The strain gages on the head near stud Nos. 13 and 25 were wired to recorder No. 1. The inside and outside strain gages on stud No. 25 were wired to Brush recorder No. 2. Because there was insufficient chart paper to run the recorders continuously during the test, the following procedure was used:

- a. The bolt numbers to be tightened were marked on the chart paper.
- b. The recorder was turned on just prior to pressurizing the tensioners.
- c. After bolt tensioner pressure was released, a calibration pip was recorded and then the recorders were shut off until just prior to pressurizing on the next set of studs.

The date, time, test run, and bolt load were recorded on the chart paper.

The above instrument readings were taken for the bolted up condition. The nuts were then loosened, using the reverse of the bolting up sequence. Strains were monitored during unbolting with the Brush recorders. With the unbolting complete, a second set of zero-bolt-load readings were taken on all instruments.

Each bolting test was always completed on the same day it was started. The same gaskets were used for three bolting tests. Prior to each test, the studs and nuts and the stud measuring holes were thoroughly cleaned and the threads were relubricated.

Stresses in Studs and Nuts

To determine the bending stresses induced in the stud during tightening ...to determine the distribution of load in the nut...to determine the load lost when the nut takes the load from the tensioner...and to determine the effect on stud-bending of the studs bottomed and backed off one turn - the following test was run.

With studs No. 1 and No. 25 bottomed with 250 ft lb of applied torque. The washer transducer was set under the nut. Bolt tensioners were positioned over these studs. The following loads were applied to the studs: 50,000, 100,000, 150,000, 200,000, and 250,000 lb. The nuts were tightened for each load. Strains in the stud and nut washer transducer were measured on Brush recorders. This procedure was repeated with the stud backed off one full turn.

4.4 Hydrostatic Tests

After completion of the bolting tests the gaskets were replaced and the head was bolted up to an average stud stress of 30,900 psi $\pm$ 6 per cent. The strain gages on stud No. 1 were connected to one Brush recorder and strain gages 1-19-1 and 1-19-2 were connected to the second Brush recorder. The vessel was filled with demineralized water and allowed to sit for a period of several hours until the temperature of the vessel stabilized. All air was vented out of the vessel and piping system. A preliminary test was run to 660 psig and strain gage readings were taken at the most highly stressed areas only. The vessel was then cycled once to 1050 psig to eliminate the effects of plastic strain. Strain readings were taken

for this 1050 psig cycle to ensure that yielding did not occur but these readings were not used for the calculation of stresses. Just prior to pressurizing, a set of zero readings was taken on all instruments, which included: (1) bolt elongation, (2) dial gages, (3) strain gages, (4) strain gage resistance to ground.

The vessel was again pressurized, using the air-operated bolt tensioner pump. The Brush recorders were turned on upon application of pressure and were left on during the duration of pressure increase. At each pressure level, when all other instrument readings had been taken, the recorders were turned off and the pressure level marked on the chart paper. A calibration pip was recorded for each pressure level.

Strain gage, dial gage and bolt elongation readings were taken at pressure levels of 500, 700, 900, and 1050 psig. After taking each set of readings, the pressure was released and a set of zero readings was taken before proceeding to the next pressure level. The vessel was held at 1050 psig for a minimum of two hours. The gasket bleed-off channel was filled with water while at pressure, and bleed-off valve No. 1 was closed. Gasket leakage was measured from bleed-off port No. 2 and the leakage was weighed.

After this test, a pressure line was connected from the inside of the vessel to the No. 2 gasket bleed-off connection. This subjected the outer gasket to a pressure equal to that inside the vessel. The vessel was then pressurized very slowly and a gasket leakage estimate was made by visual inspection. The pressure was increased until 1050 psig was reached.

Another hydrostatic test was run using the above procedures for pressures of 600, 800, 1000, and 1200 psig and all instrument readings were taken. The test vessel was then unbolted, inspected, and prepared for elevated temperature tests.

4.5 Temperature Cycling Tests

4.5.1 Test Preparation

All room temperature strain gages and lead wire exit couplings were removed. The two safety valves were installed and their discharge end connected to the 5 in. exhaust stack. The high temperature strain gages were installed on the two studs. Thermocouples were installed in the flange probe holes, on the inside of the flange stud hole, on the stud, inside the vessel, and on the thermal sleeve. All safety devices were checked out to ensure that they would function correctly. The test vessel and piping were then insulated and a new gasket set installed. The closure was placed on, and bolted up to an average bolt stress of 18,900 psi. The "degree of nut rotation" method was used in tightening the nuts for these tests. Hence, feeler gage readings were taken at six locations measuring the space between the two flanges. The extensometer was used to measure stud elongations. A hydrostatic test to 600 psig was run prior to each temperature cycle to measure the rate of gasket leakage with the vessel cold.

4.5.2 Vessel Heating Procedure

The vent valves on the top of the control rod tubes were opened up and the vessel was filled with de-ionized water.

When the vessel was full the heaters were turned on to 50 kw operation. The vent valves remained open to permit excess water due to thermal expansion to pass freely out of the vessel. When the water temperature reached 212 F the vent valves were closed and the drain valve was cracked open. A slight positive pressure (about 5 psig) was built up and maintained by controlling the setting on the drain valve. By this procedure the water which was forced out of the drain valve was replaced by a head of steam. The water level was lowered to the level desired for the cooling tests and then the drain valve was closed. With the water level 6 in. above the heater elements, the vessel could be heated with the electric heater at a rate of approximately 14 degrees Fahrenheit per hour. The temperature recorder was set at 489 F and the vessel was allowed to heat up to that temperature. The vessel was held at 489 F long enough to stabilize temperatures and to obtain a leakage rate on the gaskets.

For a rapid heating test with a rate in the order of 200 F/hr steam was produced externally in a nearby heat transfer loop. A M-G set supplied current for heat transfer loop heating element. The largest motor-generator set was not available during the closure tests so the smaller m-g unit was used. This unit did not have sufficient capacity to heat the closure test filled with water at a rate of 200 F/hr.

In order to obtain the desired heating rate the closure vessel was cooled to 212 F and allowed to stabilize to a uniform temperature throughout. All water was drained from the vessel and steam was introduced from the heat transfer loop. The heating rate was in excess of 200 F/hr at first and fell to below 100 F/hr at the end of the test, after a considerable amount of condensate had accumulated in the vessel.

4.5.3 Cooling Procedure

The vessel was cooled either by blowing off steam, by injecting cooling water, or by using both methods together. The first two cooling cycles were primarily for familiarization of temperature control with the devices employed - rather than to obtain specific data. During the following three cycles, experiments were run on the effect of water level on the cooling rate and also on the effect of "blowing off steam vs cooling water injection" on the cooling rate. All temperatures were recorded during cooling.

When the vessel had cooled down to approximately 80 F, stud elongations and feeler gage readings were taken after each cycle.

Strain Gage Readings

A unique opportunity was provided, and taken advantage of, in eliminating the effects of temperature on the strain gages and lead wires. For the first temperature cycle the two strain gaged studs were not tightened and only temperature effects on the gage and circuit were recorded. After having established an apparent strain due to temperature, the two strain-gaged studs were tightened and strains measured for the following cycles. Resistance to ground and resistance of the circuit readings were taken with each strain reading.

4.6 Steady State Elevated Temperature Test

Upon completion of the temperature cycling tests the vessel was disassembled and thoroughly inspected. A new gasket set was installed and the vessel was reassembled. The studs were tightened to an average stress of 18,900 psi. The heating procedure using the 50 kw electric heater was the

same as for the cycling tests. The vessel was heated to 489 F and held at this temperature. It was planned to hold the vessel at 489 F continuously for a period of three months, but leaks developed in the piping and three times during this test shutdowns had to be made for repairs. The vessel was at operating temperature for a total of 100 days. Gasket leakage was measured during the entire test. After the final shutdown, stud elongations and feeler gage readings were taken. The vessel was disassembled and thoroughly inspected.

4.7 Galling Tests

Simulated service tests on the studs and nuts were conducted on the mockup of the vessel flange shown in Fig. 10. The simulated flange was placed in a clean 55 gal. barrel and the barrel was filled with de-ionized water to a level several inches above the top of the stud. The nuts were threaded on and off the full length of stud threads and periodically inspected for galling and wear. The materials, lubricants and platings which were tested are tabulated along with the results on Table IX.

Stud and nut combinations which successfully passed this test were tightened to 30,000 psi average stud stress for a minimum of ten times. The bolt tensioners were used to tighten the nuts. After successful completion of this test, a tapered plate was placed under the nut to simulate the effects of flange rotation. The plate was ground to a 0.004 inch/inch taper. The nuts were again tightened to an average stud stress of 30,000 psi for a minimum of 10 cycles.

The two nuts which produced the most satisfactory results during this series of tests were used on the closure test vessel during part of the elevated temperature tests. The nuts were tightened to produce approximately 20,000 psi bolt stress and were operated at 489 F for one month.

4.8 Underwater Tests

Preparation

Upon completion of the galling and wear tests the two simulated flange assemblies were used to make up the reactor vessel flange mockup shown in Fig. 11. The two flange assemblies were separated 100 in. apart and the two angles and the channel were welded to these assemblies.

The mockup was placed into the 40-foot tank and suspended on cables at an elevation approximately twenty ft below the top of the tank. The relative dimensions for the working platform, the water level and the vessel flange simulated actual plant conditions within fractions of an inch.

The bolt tensioners and handling rig were placed into the pool, and positioned over the flange mockup. After the tensioners were properly aligned they were dropped into position over the nuts with the aid of the building crane. The bolting was repeatedly tensioned to 30,000 psi stress, loosened and then the rig removed and repositioned over the nuts. The nuts were also repeatedly removed from the studs and then replaced with the nut handling tool shown in Fig. 14.

5.0 RESULTS OF CLOSURE AND GASKET TESTS

5.1 Stresses

5.1.1 Stress Plots

The stresses in the closure caused by an initial average bolt stress of 20,000 psi for 0 and 600 psig internal pressure are shown in Table II. Stress values are plotted in Fig. 52 and 53. The stresses for a 30,000 psi average initial bolt stress for 0 and 1050 psig internal pressure are shown in Table III. Stresses in the head, near control rod nozzle No. 14, are tabulated for internal pressures of 600 psig and 1050 psig in Table IV.

The stresses were computed from average strains using the formula:

$$\sigma_m = \frac{E}{1 - \mu^2} (\epsilon_m + \mu \epsilon_c)$$

$$\sigma_c = \frac{E}{1 - \mu^2} (\epsilon_c + \mu \epsilon_m)$$

Values for ϵ_m and ϵ_c were taken from a straight line averaging strains for six bolting tests and seven hydrostatic tests.

5.1.2 Bolt Load Increase with Pressure

The increase in bolt stress vs. pressure plotted for two initial bolt loads is shown in Fig. 25. The calculated hydrostatic end force due to pressure is also plotted. Using the basic equation

$$F_b = F_G + H$$

The gasket force F_G may be plotted by subtracting H from F_b .

H = HYDROSTATIC END FORCE
 F_G = FORCE ON GASKET
 F_b = FORCE ON BOLTS

Fig. 25...Effect of Internal Pressure on Bolt and Gasket Forces (43-025-185)

TABLE II

STRESSES IN CLOSURE TEST HEAD

Initial Stress(Ave.)		20,000 psi							
Internal Pressure		0 psig				600 psig			
Gage No. (Fig. 18)	σ_m		σ_c		σ_m		σ_c		
	Row 1	Row 2	Row 1	Row 2	Row 1	Row 2	Row 1	Row 2	Row 2
1	-6200		-8000		-4500		-3700		
2	-11000	-8000	-6300	-5600	-3600	-3000	-2200	-1600	
3	-2900	-4400	-1100	700	2100	2400	3600	4600	
4	-3600	-3800	400	500	4700		5800		
5	-7800	-6800	400	1100	-1000	500	4900	6900	
6	-13100	-12400	-700	0	-5000	-5600	4300	4900	
7	-10600	-10000	1600	2500					
8	-9000	-8400	2000	2400	-2600	-1800	6800	7700	
9	-4900	-4100	2800	3200	600	2800	8500	9800	
10	400	1100	3900	4300	5100		8700		
11	1400	1800	1900	2500					
12	900	800	-4800	-5500	900	900	-1500	-1800	
13	-200	300	-400	-300	-2100	-1000	200	2200	
14	-2300		4100		-3500		5800		
15									
16									
17	16400	13800	9300	8200	15400	11200	10700	10100	
18	13900	12200	8700	8500	13900	11500	10800	10800	
19	15400	12800	9100	8200	16700	13700	11900	11000	
20	16500	15300	10000	8400	20600	19000	13700	12100	
21		12400		8600		18100		12900	
22	9100	8900	7500	7300	13600	13400	11600	11500	
23	5400	5800	6300	6200	10400		10900		
24	800	700	3200	3200	7100	6900	8800	8500	
25	-1400	-1200	1000	1000	4800	5600	6600	7000	

Notes:

 σ_m = Meridional Stresses σ_c = Circumferential Stresses

Gage locations shown in Fig. 18

Row locations shown in Fig. 3

TABLE III
STRESSES IN CLOSURE TEST HEAD

Bolt Stress (Ave.)		30,000 psi							
Internal Pressure		0 psig				1050 psig			
Gage No. (Fig. 18)	σ_m		σ_c		σ_m		σ_c		
	Row 1	Row 2	Row 1	Row 2	Row 1	Row 2	Row 1	Row 2	Row 2
1	-9100		-12000		-6100		-4500		
2	-11700	-12100	-9300	-8400	-3400	-3100	-2200	-1500	
3	-4500	-6600	-1500	-1000	4200	5300	6500	8300	
4	-5200	-5700	800	600	9400		2100		
5	-11600	-10200	600	1900	500	2700	8600	11900	
6	-19900	-18600	-1700	100	-5600	-6700	7700	8700	
7	-15900	-15100	2500	3900					
8	-13300	-12700	3200	3700	-2100	-1100	11600	13000	
9	-7200	-6200	4400	4800	2300	4900	14400	12600	
10	500	1700	5700	6400	8900		14200		
11	2100	2700	2800	3700					
12	1300	1200	-7200	-8100	1400	1300	-1300	-1600	
13	-300	400	-700	-400	-3500	-1800	400	3800	
14	-3500		6200		-5600		9200		
15									
16									
17	24700	20300	13800	12100	19800	16000	16300	15400	
18	20800	18300	13100	12600	20800	17100	16700	16800	
19	23200	19300	13600	12300	25500	21000	18700	17200	
20	24900	23000	15100	12700	31100	29400	21400	19100	
21		18700		12900		28400		20400	
22	13700	13300	11300	10900	21700	21300	18700	18200	
23	8000	8600	9400	9300	16800		14400		
24	1200	1100	4800	5200	12300	11900	14600	14500	
25	-2100	-1900	1400	1500	8700	10200	11300	12000	

TABLE IV
STRESSES AT CONTROL ROD NOZZLE NO. 14

Internal Pressure, psig	600		1050	
Gage Location (Fig. 3)	σ_m	σ_c	σ_m	σ_c
3 x 1	5200	4800	9200	8400
3 x 2	6900	3800	12300	7300
3 x 3	4000	5600	7000	9800
3 x 4	6400	4500	12000	8100

5.1.3 Bending in Studs

The bending strain in the stud vs. applied bolt tensioner load is plotted in Fig. 26. The amount of bolt load lost when the bolt tensioner pressure was released and the load transferred to the nut is indicated by the difference in the "load applied" and "load removed" curves. It was found that there was no difference in bending for the studs solidly bottomed or backed off one turn. The above results were obtained with the remaining studs unbolted. Strains measured on the same stud with the test vessel completely bolted up indicated that the amount of bending was essentially the same for both the bolted up and unbolted conditions.

Compressive strains in the strain gage washer transducer are plotted in Fig. 27 (for the same bolt tensioner loads used in the tests above). They indicated the load distribution on the nut.

X = STRAIN IN STUD WITH NUT SNUGGED UP & BOLT TENSIONER LOAD REMOVED. REMAINING STUDS UNLOADED.

⊙ = STRAIN IN STUD WITH VESSEL COMPLETELY BOLTED UP.

□ = STRAIN ON STUD WITH BOLT TENSIONER LOAD APPLIED.

Fig. 26...Bolt Tensioner Loads vs Strain in Stud No. 1, Closure Test Vessel Unbolted (43-025-187)

Fig. 27...Strain Distribution in Washer Transducer,
Vessel Unbolted, Stud Bottomed (43-025-191)

5.1.4 Stresses Measured During Bolting Operations

The strains in the head and studs were measured during bolting up and unbolting operations. The meridional strain in the head near the juncture of the flange is plotted versus bolting operation number in Fig. 28. The variation in average stud stress for the same test is shown in Fig. 29. It is interesting to note that the stud stress increased for operation Number 2, that is when the two studs 90° away were tightened. After the first few operations, the stress in the first stud tightened...then gradually decreased as more studs were tightened. From this information it is evident that measurements of stud stress (or elongation) should be made only after all the studs have been tightened, and that if one stud is re-tightened it will have an effect on the stresses in the other studs.

During unbolting operations (for which the sequence of operations is exactly reversed) the stress gradually increases in the last studs loosened, as more studs are unbolted. The two curves for bolting up and unbolting resemble each other quite closely.

5.2 Flange Rotation and Displacement

The dial indicator measurements for a number of test runs are given in Tables V thru VII. Refer to Fig. 21 for orientation of the dial indicator gages. Flange rotations calculated from these readings are summarized in Fig. 30. The effect of bolt load and internal pressure is shown separately in this figure. Table VIII gives the measured radial displacements of the upper flange.

Fig. 28...Strain in Vessel Head During Bolting Operations,
Strain Gage 1-21-2 (43-025-186)

Fig. 29...Variation of Stress in Stud No. 1 During Bolting-Up and Unbolting Operations (43-025-188)

- FLANGE ROTATION ~ LOCATION #1.
- FLANGE ROTATION ~ LOCATION #2.
- FLANGE ROTATION DUE TO BOLT LOAD ONLY.
- FLANGE ROTATION DUE TO PRESSURE ONLY.

Fig. 30...Flange Rotation vs Bolt Load and Internal Pressure (43-025-192)

TABLE V
SUMMARY DATA
FOR
DIAL GAGE MOVEMENTS AT LOCATION #1
DURING BOLTING UP TESTS

Test No.	Bolt Load lbs x 10 ⁶	Zero Used	Gage Movement - Inches x 10 ⁻³							
			A	B	C	D	E	F	G	H
1	4.63	Initial	--	--	--	--	--	--	--	--
		Final	67.3	17.0	18.0	1.5	16.6	2.4	-8.6	-2.7
2	5.30	Initial	62.4	19.0	35.7	3.6	17.9	4.0	-10.7	-3.1
		Final	62.4	18.2	34.7	3.3	17.5	3.7	-9.3	-1.7
3	3.27	Initial	47.0	21.9	19.0	1.0	--	-0.6	-5.2	--
		Final	37.0	13.3	19.0	1.0	11.9	0.2	-4.4	--
4	5.39	Initial	69.5	23.8	33.0	3.0	23.7	2.6	-5.5	--
		Final	64.5	19.2	34.0	3.0	18.2	2.7	-4.9	--
5	5.21	Initial	66.0	25.3	36.0	2.5	--	2.5	-10.2	--
		Final	71.0	21.5	36.0	1.0	--	1.4	-17.1	--
6	9.22	Initial	--	--	--	--	--	--	--	--
		Final	129.8	31.0	67.2	2.3	31.1	2.2	-14.3	-2.0

TABLE VI

SUMMARY DATA
FOR
DIAL GAGE MOVEMENTS AT LOCATION #2
DURING BOLTING UP TESTS

Test No.	Bolt Load lbs x 10 ⁶	Zero Used	Gage Movement - Inches x 10 ⁻³							
			A	B	C	D	E	F	G	H
1	4.63	Initial	--	--	--	--	--	--	--	--
		Final	60.6	19.4	25.3	3.0	--	--	--	--
2	5.30	Initial	65.3	19.5	36.7	4.2	--	--	--	--
		Final	64.6	19.5	36.7	4.0	--	--	--	--
3	3.27	Initial	46.0	24.5	17.0	0.0	--	0.2	-6.0	--
		Final	37.0	13.5	14.0	1.0	--	0.9	-1.5	--
4	5.39	Initial	63.0	19.0	38.0	3.5	--	2.6	-1.0	--
		Final	59.0	21.0	37.0	2.5	--	3.1	--	--
5	5.21	Initial	69.5	23.0	36.5	3.0	--	3.7	-4.3	--
		Final	59.5	18.0	35.5	3.0	--	4.6	-4.3	--
6	9.22	Initial	--	--	--	--	--	--	--	--
		Final	117.8	33.3	--	0.1	--	--	--	--

TABLE VII
SUMMARY DATA
FOR
DIAL GAGE MOVEMENTS
DURING HYDROSTATIC TEST

Location	Internal Pressure psig	Gage Movement - Inches $\times 10^{-3}$							
		A	B	C	D	E	F	G	H
1	600	-14.0	-6.7	-14.5	-0.8	-6.9	-1.1	+5.8	+5.8
1	1050	-24.5	-12.0	-26.0	-1.6	-12.0	-2.0	+10.3	+10.3
2	600	-15.0	-7.2	-15.5	-0.7	--	--	--	--
2	1050	-26.0	-12.9	-27.3	-1.3	--	--	--	--

TABLE VIII

FLANGE DISPLACEMENT DUE TO BOLT LOAD

Test No.	Bolt Load lbs x 10 ⁶	Zero Used	Y-inches x 10 ⁻³ Location		X-inches x 10 ⁻³ Location	
			1	2	1	2
1	4.63	Initial	--	--	--	--
		Final	10.4	12.0	-1.0	--
2	5.30	Initial	13.4	11.3	-4.2	--
		Final	12.4	11.4	-2.7	--
3	3.27	Initial	18.6	20.7	-1.4	-2.6
		Final	10.2	9.3	-0.8	+2.3
4	5.39	Initial	17.9	11.1	+1.4	--
		Final	13.3	14.2	+1.9	--
5	5.21	Initial	20.0	14.6	-4.1	-0.2
		Final	15.1	10.6	-9.2	+4.9
6	9.22	Initial	--	--	--	--
		Final	18.1	18.1	-0.9	+1.6

5.3 Analysis of the Gasket Joint5.3.1 Gasket Compression for Hot Tests

A plot of the gasket thickness for various stages of the hot tests is shown in Fig. 31. The initial and final gasket thicknesses were measured with a micrometer and the points plotted were an average of the thicknesses measured. The gasket thickness was determined from feeler gage measurements of the gap between the two flanges for the following:

- after closure placement,
- after the application of 100 ft. lbs. of torque to the nuts,
- after tightening all nuts with the bolt tensioners,

Fig. 31...Gasket Thickness Measurements (43-025-195)

d) after the hot tests, and

e) after loosening all nuts (with closure in place).

Note that after tightening the nuts, the gaskets for both tests were compressed to nearly the same thickness even though the initial gasket thicknesses were quite different.

The reduction in gasket thickness from before tests and after tests indicate the amount of gasket relaxation. The reduction in gasket thickness during the tests causes a reduction in the bolt stress and flange rotations. The amount of gasket relaxation is an important factor in the design of a gasketed joint where the flanges do not make metal to metal contact. The flanges and bolting must be sufficiently flexible to make up for gasket relaxation without excessive loss of bolt load to the point where the gaskets would leak. This gasket relaxation during elevated temperatures and during temperature cycling was perhaps the most questionable unknown factor in the suitability of the corrugated metallic gaskets. For the closure tested, there was sufficient flexibility in the flanges and bolting to make up for gasket relaxation and still have sufficient bolt load to maintain a tight seal. It should be pointed out that overdesign, such as increasing the size of the flanges and bolting, which would stiffen the joint may reduce the number of cycles and the time at temperature which the joint could withstand before gasket leakage occurred.

The percentage loss of bolt load vs. number of temperature cycles is shown in Fig. 32.

Prior to the temperature cycling tests, a cold hydrostatic test was performed to determine initial gasket leakage. The gasket leakage is

Fig. 32...Percentage Loss of Bolt Load vs Number of Temperature Cycles (43-025-196)

plotted for various pressures in Fig. 33. The increase in bolt load due to pressure is also plotted along with the calculated gasket force and hydrostatic end force curves. The curve points out the leakage rates for the gasket force involved. This curve along with Fig. 25 may be used to determine the bolt load required to prevent leakage.

5.3.2 Gasket Leakage

During the cold hydrostatic test with the closure bolted down with a 9.22×10^6 lb total bolt load, (30,900 psi ave. stud stress) there was zero gasket leakage through the inner gasket at 1050 psig internal pressure. The pressure was brought up to 1200 psig before leakage occurred. The leakage rate at this pressure was 0.25 grams of water per hour (this is approximately 6 "drops" per hour).

Pressure was then applied to the bleed-off channel between the gaskets and a visual inspection for leakage was made while the vessel was being pressurized very slowly. The first sign of leakage through the outer gasket occurred at 750 psig. At 800 psig there was a slight drip. At 850 psig the leak was a very thin but steady stream. This leak occurred only between bolts No. 33 and 34. At 950 psig a slight drip developed between bolts No. 10 and 11, and at 1050 psig a slight drip developed between bolts No. 22 and 23. The pressure was increased to 1200 psig; the leakage at the three locations could be described as a thin steady stream.

When the vessel was unbolted the gaskets were inspected in position. It was discovered that the outer gasket was slightly out of round and at

Fig. 33...Gasket Leakage Plotted for Various Pressures (43-025-193)

the three places where leakage occurred the gasket was the closest to the bolt hole. The outer rim of the gasket was exposed by the bolt holes in the closure at these three points. The out-of-roundness of the outer gasket was believed to have been caused by handling and shipping before the tests.

During the temperature cycling tests, leakage rates were measured at operating temperature and pressure and at 600 psig cold hydrostatic run before each cycle. The results are tabulated as follows:

Measurement Made	Leakage Grams/hr.
Before 1st cycle	35.0
During 1st cycle	0.0
Before 2nd cycle	0.0
During 2nd cycle	1.6
Before 3rd cycle	3.1
During 3rd cycle	1.4
Before 4th cycle	0.0
During 4th cycle	0.0
Before 5th cycle	12.0
During 5th cycle	0.0

Prior to the long term elevated temperature test, during the cold 600 psig hydrostatic test on the new set of gaskets, a leakage of 163 grams/hr was measured. But at operating temperature (489 F) there was zero leakage for the duration of the entire test. It should be noted that the two gaskets used during this test had surface imperfections which normally would be unacceptable. The gaskets were used in this case since they were the only set available at the time.

5.3.3 Gasket Surface

The flange gasket surfaces were checked for brinelling every time prior to the installation of a new set of gaskets. Although there appeared to be impressions from the gasket corrugations, the depth of these impressions was difficult to determine, even with a magnifying glass.

An excellent reproduction of the gasket surfaces was made with liquid plastic steel. A thin film of parting compound was rubbed on the gasket surface and a liquid plastic steel casting was made of the surface. The casting was viewed under a 50 power stereo microscope which revealed that the gasket impressions were not as deep as the machining grooves. Plastic steel castings of the gasket surfaces were taken each time the gaskets were removed. There was no evidence of brinelling of the surfaces beyond that described above. These slight impressions did not affect the sealing ability of the gaskets.

5.4 Aluminum Metal Spray Coating

The aluminum metal spray coating on the inside of the lower head withstood the temperature cycling tests quite well. A complete visual inspection was made after the cycling test. There was some rust deposited on the coating which was formed from the carbon steel piping in the system. Thorough cleaning of the coating indicated that the rust was in fact plated out on the aluminum and the coating was good. One blister was found near the gasket surface. The only other place where the coating failed was a small section at the flange-head weld juncture, an area which was rough and perhaps not sand blasted as thoroughly as the rest, prior to metal spraying.

After the long term test during which the vessel was held at reactor operating conditions (600 psig with 489 F steam for 100 days), the aluminum spray coating was again inspected and found to be almost completely oxidized; but for the most part it remained intact except for a number of large blisters. The carbon steel of the lower vessel was not significantly corroded.

In conclusion, it seems that aluminum spray coating provides adequate corrosion protection in 600 psig steam or water for relatively short time periods.

The aluminum spray coating on the outside of the lower head was unaffected after elevated temperature tests. Although this coating was exposed to temperatures up to 500 F, it was not exposed to a water or steam atmosphere while at temperature.

The Pathfinder vessel head has an aluminum spray coating on the outside to protect it from corrosion during reactor shutdown when the head is flooded with water. These results indicate that the coating can withstand elevated temperature operation provided it is in a relatively dry atmosphere, which it is.

5.5 Weld Deposited Stainless Steel Cladding

Weld cladding was applied to the head flange using manual stick electrodes as follows:

First layer - ASTM-A298 E309Cb-15 or 16 electrode...

Subsequent layers - ASTM-A298 E308ELC-15 or 16 electrode.

The total thickness of the weld overlay is 1/4 inch.

After temperature cycling tests and the 100 day long-term temperature tests were complete, the entire cladding surface was inspected using a dye penetrant. No cracks had developed. Several small sections of the cladding were then ground down to base metal, with a dye penetrant check every 1/16th inch. No cracks or inclusions were detected.

5.6 Temperature Distribution in the Flange and Head During Heating and Cooling Cycles

5.6.1 Closure Cooling Tests

Three methods of cooling were tried during the temperature cycling tests. The results are shown in Figs. 34 to 39 inclusive. Fig. 34 shows the effect of blowing off steam with the water level below the gasket surface. The water temperature dropped off rapidly but the temperatures in the closure flange and head remained nearly constant. The thermocouple nearest the water followed the water temperature closely as it was probably splashed by the boiling water. The blow-off valve remained open for about 8 minutes and was then shut off. The cooling water pump remained in operation for the duration of the test.

The effect of injecting cooling water only is shown in Fig. 35. The cooling water was periodically injected for the first 4-1/2 hours. Injecting cooling water alone, without blowing off steam, cooled the test vessel slowly. In comparison, the temperature dropped quickly when the blow-off valve was opened.

The combination of injecting cooling water and controlling steam blow-off demonstrates that the closure could be cooled down rapidly and smoothly,

Fig. 34...Cooling Test, Effect of Blowing Off Steam with
Water Level Below Gasket Surface (43-024-592)

Fig. 35...Effect of Injecting Cooling Water Only (43-024-593)

Fig. 36...Cooling Water Injected and Steam Blow-off Controlled (43-024-591)

Fig. 37...Flange Temperature Distribution During Cooling,
Thermocouples A-4 to A-8 (43-025-194)

Fig. 38...Flange Temperature Distribution During Cooling,
Thermocouples A-9 to A-13 (43-025-197)

Fig. 39...Flange Temperature Distribution During Cooling,
Thermocouples A-14 to A-18 (43-025-199)

as shown in Fig. 36. Figures 37 through 39 show the temperature distribution through the thickness of the flange at various times during a test where the latter method of cooling was used. Thermocouple locations are given in Fig. 22.

5.6.2 Closure Heating Test

The vessel was heated from 212 F to 412 F in 105 minutes as shown in Fig. 40. When the inside of the vessel reached 412 F there was malfunction in the steam production equipment and the test was discontinued. The vessel heating rate at this time had decreased considerably due to the amount of condensed steam in the vessel and any further data would have been of little value anyway. The temperature distributions through the flange during the duration of this test is shown in Figs. 41 through 43. See Fig. 22 for thermocouple locations.

5.7 Temperature Distribution in Thermal Sleeve

Cooling water at 70 F was pumped into the vessel while the steam pressure in the vessel was maintained at 600 psig. The cold water passing through the thermal sleeve connection to the vessel resulted in the temperature distribution shown in Fig. 44. The cold water accumulated at the bottom of the vessel, reducing the vessel water temperature at the bottom to somewhat below the saturation temperature of the steam. The temperature readings shown in Fig. 44 were taken after cold water had been pumped into the vessel for seven minutes and the temperatures had begun to stabilize to some extent. The test was not run until complete equilibrium was reached because accumulation of a large amount of cold water in the bottom of the vessel was not desirable.

Fig. 40...Heating Rate Test (43-025-202)

Fig. 41...Flange Temperature Distribution During Heating,
Thermocouples A-4 to A-8...Test 2 (43-025-198)

Fig. 42...Flange Temperature Distribution During Heating,
Thermocouples A-9 to A-13...Test 2 (43-025-201)

Fig. 43...Flange Temperature Distribution During Heating,
Thermocouples A-14 to A-18...Test 2 (43-025-205)

Fig. 44...Temperature Distribution at Thermal Sleeve (43-025-200)

5.8 Temperature Distribution in the Bolting and Flange During Rapid Cooling

The temperature of the water within the vessel was lowered from 476 F to 121 F at an average rate of 142 F/hr during one test. During a second test the temperature was lowered at a more rapid rate of 216 F/hr from 485 F to 185 F. Figs. 45 and 46 show the resulting average temperatures in the flange and bolting during these tests. The average flange temperature shown is the numerical average of the readings of the thermocouples designated as A-4 through A-18 in Fig. 22. The average stud temperature is the numerical average of thermocouples designated as B-6, B-7, B-10, B-11, B-14 and B-15 in Fig. 23.

The data shows that the average stud temperatures lagged the average flange temperatures by more than 80 F. In the first test, this maximum lag did not occur until 30 minutes after the vessel had reached atmospheric pressure. During the second test the maximum gradient occurred before the pressure in the vessel reached atmospheric.

5.9 Stud Stresses During Thermal Cycling Conditions

The strain readings were corrected for the effects of temperature and desensitization due to the lead wire resistance. Using 27.5×10^6 as the modulus of elasticity for the studs at 489 F, the increase in average stud stress due to temperature and pressure was 6400 psi. This increase was due to the difference in thermal expansion of the stud and flange materials and pressurization of the head.

During the rapid cooling cycle shown in Fig. 46 (pressure dropped from normal operating pressure to almost atmospheric pressure in two hours),

Fig. 45...Temperature of Flange During Cooling Cycle (43-025-208)

Fig. 46...Temperature of Flange and Stud During Rapid Cooling Cycle (43-025-203)

there was a loss in bolt load due to the fact that the flange cooled more rapidly than the studs. The maximum loss of bolt stress was 11,500 psi which occurred when atmospheric pressure was approached. Gasket leakage was evident during the test but the amount of leakage was small. However, due to the transient nature of the test, a leakage rate could not easily be determined.

To summarize, the following stud stresses were measured for the conditions described above:

Initial stud stress (cold)	= 18,900 psi
Stud stress at operating temperature	= 25,300 psi
Minimum stud stress during rapid cooling	= 13,800 psi.

6.0 RESULTS OF DRY CLOSURE BOLTING TESTS

6.1 Gallling and Wear of Studs and Nuts

For the first four bolting up tests the studs and nuts were lubricated with "Fel Pro" anti-seize compound. During the third and fourth tests there was some difficulty in removing the bolt tensioner puller bars. After the fourth bolting up test the studs began to show signs of wear. The puller bar threaded adapters were inspected and found to be worn more than the studs. A new set of puller bar threaded adapters were made from annealed ASTM-A437, 59T, the same material as is used for the nuts.

The studs and nuts were cleaned and this time lubricated with SAE-10 oil. The bolt tensioner puller bar galled solidly to the first stud on which it was threaded. The oil was removed from all studs and never-seize compound applied prior to the fifth bolting tests. The worn puller bar threaded adapters were completely deburred and installed on the bolt tensioners. Two nuts were galled solidly to the studs during this test. Fig. 47 shows one of the nuts after it was removed by slitting it in half on a milling machine.

The stud threads were worn sharp, the worn off metal particles would machine a wear groove in the thread terminating in a burr.

All threads were thoroughly cleaned, dressed with a thread restorer file and finished with crocus cloth. The puller bar threaded adapters and the nuts were similarly treated.

The studs and nuts were reassembled using "Fel Pro" anti-seize compound as the lubricant. There was no galling or wear encountered during the 6th, 7th and 8th bolting tests. The 7th and the 8th bolting up operations

Fig. 47...One-half of Nut Removed after Being Galled Solidly to Stud During Test

were for the hot tests. The "Fel Pro" lubricant worked well during the hot tests; no difficulty was encountered in removing any of the nuts or studs after the exposure to elevated temperatures. "Led Plate" compound was also tested under both hot and cold conditions with similar results.

6.2 Bolt Tensioner Operational Experiences

The hydraulic bolt tensioners proved to be useful in making an accurate and efficient job of bolting up the closure. With an inexperienced operating crew, a complete tightening pass was made in about 2 to 2-1/2 hrs. With the bolting up technique developed during these tests it was possible (on one test) to tighten all the nuts within acceptable tolerances in one pass, taking less than one hour of time. With an experienced crew the closure vessel was unbolted in 55 minutes. These tests were run dry with direct access to the tensioners. Somewhat longer periods of time would be required for underwater operation.

The bolt tensioners were newly developed devices and there were several problems associated with their operation. These problems are discussed in the following paragraphs. All of the problems have now been overcome through appropriate design changes.

The first difficulty was encountered during the first bolting up test. The phenolic pistons had swelled during previous preliminary tests and were expanded to the extent that the tensioners operated very erratically. The tensioners were disassembled and it was found that the cylinders were out of round by about 0.005 inches. The 0.0002 inches of hard chrome plating on the inside of the cylinders was pitted in the area in contact

with the pistons. The cylinders were repaired by grinding and plating on a 0.001-inch-thick chrome plating. The phenolic pistons were replaced with aluminum pistons and the test resumed. During the following five bolting up tests the bolt tensioners were operated without any major difficulty. When the tensioners were disassembled to replace the packings, however, rubbing marks were found on the pistons and cylinders indicating that the pistons apparently became cocked during operation.

The tensioners were tested pulling on a stud in a separate test stand. It was discovered that if the spherical washers under the puller bar holding nut were pushed to one side, the tensioners would pull erratically. Although these washers were clean and thoroughly lubricated with dry Molycote they would not center themselves properly under the puller bar holding nut. The washers were removed and inspected. It was found that the washers were in contact only in a small area along the outer periphery. The washers were lapped together using lapping compound and were then installed again on the tensioners and tested. The results were still unsatisfactory as the washers failed to center themselves under the holding nut. The washers were removed and replaced with a solid flat washer of equivalent height. The tensioners worked better with the flat washers but the pistons still had a tendency to jam from time to time. A new set of spherical washers was made, using a 44 inch radius instead of the original 18 inch radius. The new spherical washers seemed to center themselves and there was no tendency for the pistons to jam during 5 successive bolting up operations of the head.

6.3 Bolting Procedures

The variations in stud elongation using the "pressure measurement method" (see Section 4.2.1) for loading of the studs are shown in Figs. 48 through 50. An attempt was made to properly program the bolt loadings, decreasing it as more bolts were loaded, in order to obtain uniform elongations. In five bolting tests, elongation variations were very large even after the second pass. In one test, shown in Fig. 49, the elongation variance was acceptable after the second pass. The variations in elongation were not always due to improper programming of the stud loading but were sometimes due to variable friction in the mechanism, resulting in a variable load for a given pressure reading.

In summary, it seemed that a minimum of two to three bolting passes were required to bolt up the head uniformly.

Using the "degree of nut rotation" method (see Section 4.2.2) the elongations were fairly uniform even after the first pass, as shown in Fig. 51. The difficulty in this method was determining the required number of degrees to rotate the nuts in order to obtain the proper bolt loading. The initial gasket thickness is quite variable not only from one gasket to the next but also quite variable at different cross sections of the same gasket. Since the flange joint imposes no limitation on gasket compression, variations in gasket thickness result in variations in the number of degrees of nut rotation required.

Some variation in initial gasket thickness can be eliminated by tightening all studs up to some arbitrary torque, say 100 ft lbs. This

Fig. 48...Stud Stress Variation, Bolting Up Test No. 2,
Pressure Gage Method (43-025-206)

Fig. 49... Stud Stress Variation, Bolting Up Test No. 5,
Pressure Gage Method (43-025-207)

Fig. 50...Stud Stress Variation While Bolting Up to 30,000
PSI Stress, Pressure Gage Method (43-024-369)

Fig. 51...Stud Stress Variations, Bolting Up Test No. 7,
Degree of Nut Rotation Method (43-025-209)

operation can take almost as long as one bolting pass and does not completely eliminate the problem of gasket thickness variations.

As a result of variation in gaskets it is not always possible to tighten bolts with the desired range of average stud elongation in one pass. However, the variation of individual stud elongations with one another was usually small, $\pm 10\%$, even with one pass. All gaskets tended to approach the same thickness at relatively large stud elongations (0.008 inches or more). This made it relatively easy to correct to desired average stud elongation with the second pass.

An attempt was made to develop an empirical formula to calculate the required degrees of rotation based on a measured average gasket thickness and the desired stud stress. Such formulas were sometimes used quite successfully for several successive bolting operations and then, during a later operation, would fail to predict the final stud elongations with the desired accuracy.

Finally it was decided that the simplest and quickest bolting procedure would incorporate the following steps:

1. Draw all nuts down snugly against the flange with a hand tool. (The step in which each nut was initially torqued to 100 ft-lbs is eliminated.)
2. The bolt tensioners are engaged and the studs loaded.
3. The nut is rotated a set number of degrees calculated to give the desired final elongation for the average gasket as received from the supplier.
4. The load is released.

5. All studs loaded in the same manner and in the order given in Table I of Section 4.2.1.

This completes the bolting operation in one pass. If the average stud elongation is above or below the range desired a second bolting pass is necessary.

In the written procedures for bolting up of the Pathfinder vessel closure, the following variations in stud elongation are specified:

Desired average elongation = 0.015 inches (equivalent to stud stress of 22,000 psi)

Allowable range in average stud elongation = 0.014 to 0.017 inches

Allowable range in individual stud elongation = 0.013 to 0.018 inches

A rotation of 400° on the index wheel of the bolt tensioners assembly (Fig. 13) is used as a first approach to the above values of elongation.

7.0 RESULTS OF UNDERWATER CLOSURE BOLTING TESTS

7.1 Underwater Galling Tests

Studs used in this series of tests were made of ASTM A437 Grade B4B. Nuts were made of various materials, including ASTM A437 Grade C4C, AISI 4140 and AISI 303. Some were plated with silver, cadmium, chromium, baked "Fel Pro", and baked MoS_2 .

Table IX lists the tests run along with such pertinent information as the nut materials, plating, condition of the stud and nut, number of cycles and the results.

Good results were obtained with the silver plated ASTM-A437, unplated AISI 303 and the unplated AISI 4140 nuts. Other combinations worked well for a limited number of cycles.

Two silver plated nuts were used on the closure during 100 day elevated temperature test (489 F). No galling was experienced and the silver plating seemed unaffected during the test.

Silver plated ASTM-A437 Gr C4C nuts were selected for the Pathfinder vessel. The AISI 303 was not considered, mainly due to its relatively low yield strength at 500 F and AISI 4140 or other grades of carbon steel were not selected because of their poor corrosion resistance. (The bolting is exposed to pool water during reactor shutdown.)

7.2 Underwater Operating of Bolt Tensioners and Associated Tools

During initial tests, considerable difficulty was experienced in getting the bolt tensioners to center properly over the flange nuts. Foot pads with wide tapered leads were attached to the bottom surface of the tensioner

TABLE IX

GALLING TEST RESULTS*

Run No.	Nut Mat'l & Condition	Stud # & Condition	Lubricant or Plating	No. Cycles	Results
1	New ASTM-A437-59T Grade C4C	1 - Used, Fair	None	0	Galled during assembly in air - both nut and stud in very poor condition
2	New 303 S.S.	2 - Used, Fair	None	2	Slight galling. Easily removed - nut good - stud fair
3	New 4140 C.S.	3 - Used, Good	None	6	No galling - high rate of corrosion
4	From Run #2	4 - Used, Excellent	None	6	No galling - excellent results
INSTALLED TAPERER WASHER TO SIMULATE FLANGE ROTATION					
5	Same as Run #4			2	No galling - excellent results
6	New ASTM-A437-59T Grade C4C	4 - Used, Excellent	None	-	Galled during assembly in water - nut fair - stud good
7	New ASTM-A437-59T Grade C4C	5 - New	Baked MoS_2	2	Galled during second assembly - nut fair - stud good
8	New 4140	5 - Good	Cadmium	8	No galling - high wear rates
9	From Run # 7	4 - Good	Chrome	8	Slight galling on first assembly and became progressively worse
10	From Run #6	6 - Good	Baked Fel Pro	3	Galled - Stud good - nut poor
11	New ASTM-A437-59T Grade C4C	6 - Good	Silver	13	No galling - little wear. Bottom thread bare where plating peeled off

*All tests conducted in water except run No. 1.

body. These pads made alignment of the tensioners much easier. If the tensioner spacing and alignment is not correct during underwater operation the tensioner housing binds against the nut, making it difficult or impossible to rotate the nut.

A pin and socket arrangement will be used to align the tensioner rig radially in the field. A socket is provided on the lifting rig, Fig. 13, Item 16. A pin is attached to the upper vessel head lifting structure just above the shield pool water line, Fig. 1. The pin is engaged in the socket while the tensioners are in place on the vessel flange or when the tensioners are being moved from one set of bolting to the next, thus providing radial alignment.

Operation of the nut handling tool (Fig. 14), gasket handling rig (Fig. 16) and underwater extensometer was quite satisfactory under 18 ft of water after the previously-described modifications were made.

For trouble-free operation the bolt tensioners must be properly leveled and there must be proper spacing between the bolt tensioners.

8.0 CONCLUSIONS

8.1 Head and Bolting Design

One of the most satisfying results of the program, besides demonstration of adequate design, was the very close agreement of experimental strain gage data with previous theoretical design calculations. Fig. 52 and 53 show the correlation between theoretical calculations and strain gage data. These figures also compare calculated and measured flange rotations along with predicted and measured increases in stud stress due to pressurization.

The structural design analysis of the Pathfinder vessel including the closure design is fully covered in Reference 2. The theoretical calculations plotted (Figs. 52 and 53) were made using the same methods described in this reference but were based on the actual measured dimensions of the head and not the minimum thicknesses used in Reference 2.

The measured stresses were all below the allowable values. The stresses in the bolting increased with temperature and pressure as theoretically predicted.

8.2 Gasket Performance

Performance of the corrugated soft iron core, stainless steel gaskets was beyond expectations, especially during cycling tests. Bolting provided in the design seems adequate for pressures close to double the reference operating pressure of 600 psig.

Selection of the size and thickness of the first set of gaskets tested was not based on calculation but on the intuition and good judgment

Fig. 52...Stress Distribution in Closure Test Head Bolted Up
with Internal Pressure (43-202-565)

Fig. 53...Stress Distribution in Closure Test Head - 600 PSI
Internal Pressure (43-202-566)

of gasket specialists. Fortunately, their selection was a good one and the gasket exceeded required specifications.

There are gasket modifications which improve the design of such a closure in the future. Since the flange surfaces did not brinell and since the narrower (1 in.) outer gasket could be made to seal under full pressure, an inner gasket thickness of 3/4 to 1 in. looks appropriate. Since the outer gasket is merely a backup, a narrower outer gasket also looks possible. Both gaskets could then be closer to the bolt circle, reducing the stresses in the head and also reducing the flange rotation.

8.3 Allowable Heating and Cooling Rates

Circumferential stresses in the vessel flange base metal due to thermal gradients during heating and cooling cycles may be approximately calculated using the following formula:

$$\sigma_b = \frac{E_b \alpha_b (T_{ave} - T)}{1 - \mu}$$

where, σ_b = circumferential stress in base metal

E_b = Young's modulus of base metal

α_b = coefficient of thermal expansion of base metal

T_{ave} = average temperature of the flange through integration of temperature distribution curves

T = temperature at a given point

μ = Poisson's ratio.

The circumferential stress in the cladding is approximately:

$$\sigma_c = \frac{-E_c [\alpha_c (T_c - T_a) - \alpha_b (T_{ave} - T_a)]}{(1 - \mu)}$$

where, E_c = Young's modulus of cladding

$$E_b \approx E_c$$

α_c = coefficient of thermal expansion of cladding

T_a = ambient temperature = 70 F

T_b = temperature base metal

T_c = temperature cladding

Based upon experimental data, the predicted maximum stresses at the flange region of the closure during heating and cooling cycles are as follows: (Mechanical stresses given are based upon strain gage data given in Section 5.)

8.3.1 Heating Cycle

Rate = 225 F/Hr

Time = as closure reaches full temperature and pressure
(extrapolated from Figs. 40 through 43)

Circumferential in Flange Base Metal	Stress, psi	
	Location X-6 Fig. 18	Location X-14 Fig. 18
Stress due to initial bolt load of 22,000 psi and pressure of 600 psig considering temperature lag in bolting	- 2,000	9,000
Stress due to thermal gradient in Flange	-17,000	8,000
TOTAL STRESS	-19,000	17,000

Circumferential Stress in Stainless Steel Cladding	Stress, psi at Location X-6
Stress due to initial bolt load of 22,000 psi and pressure of 600 psig considering temperature lag in bolting	- 2,000
Stress due to thermal gradient in flange and difference in thermal expansion of materials	-55,000
TOTAL STRESS	-57,000

Meridian Stress in Cladding	Stress, psi Location X-6
Stress due to initial bolt load of 22,000 psi and pressure of 600 psig considering temperature lag in bolting	-22,000
Stress due to differential thermal expansion between base metal and cladding	-38,000
TOTAL STRESS	-60,000

Based upon estimates of stresses given above, the yield point of the base metal should not be exceeded at a heating rate of 225 F/Hr.

The stresses given for the stainless cladding are based upon elastic theory and are much higher than would be encountered during actual operation because plastic flow would take place in the cladding. These values, however, are a good yard-stick in determining total strain range. Based on fatigue design curves given in reference 3, the safe operating life of the vessel well exceeds the design life of 1000 cycles.

8.3.2 Cooling Cycle

Rate = 227 F/Hr.

Time = 97 min. after initiation of cooling cycle, see Fig. 37.

Circumferential Stress in Flange Base Metal	Stress, psi	
	Location X-6 Fig. 18	Location X-14 Fig. 18
Stress due to initial bolt load of 22,000 psi and allowing for temperature lag in bolting	< 1,000	< 3,000
Stress due to thermal gradient through flange	-6,000	20,000
TOTAL STRESS	<-5,000	<23,000

The estimated stresses given indicate that the yield point of the base material will not be exceeded in flange base metal at a cooling rate of 227 F/Hr.

The cladding stresses are not treated herein since their magnitude is not of significance, compared to the magnitude of the stresses encountered in the heating cycle previously described.

8.4 Operation of Tools for Underwater Bolting Procedures

The tests described in this report demonstrate that the Pathfinder vessel closure can be secured and re-opened remotely under 18 ft of water in a reasonable length of time. It is not expected, however, that underwater manipulation of bolt tensioners will be as fast and trouble-free as dry operations with direct access to the bolting. Proper alignment and maintenance of these special tools is necessary for their proper operation.

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