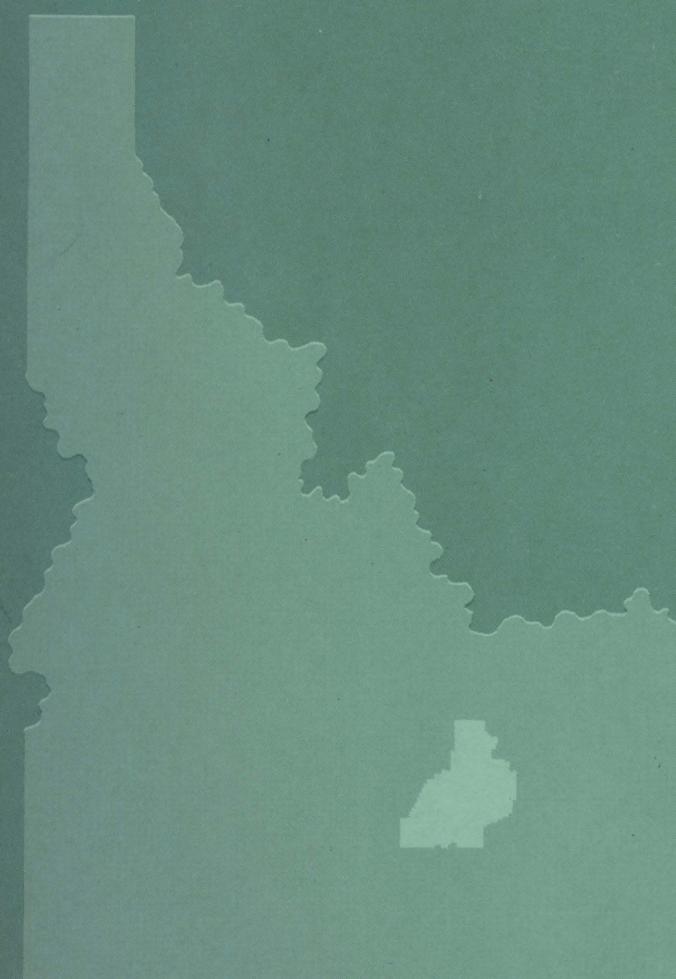
IDO-16743

SPERT III PRESSURIZER VESSEL FAILURE

January 29, 1962



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SPECIAL POWER EXCURSION REACTOR TESTS

SPERT III PRESSURIZER VESSEL FAILURE

by

R. E. Heffner, T. R. Wilson, D. P. Halls,
W. W. Hickman, and K. R. Hoopingarner

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Atomic Energy Division
Contract AT(10-1)-205
Idaho Operations Office

U. S. ATOMIC ENERGY COMMISSION

SPERT III PRESSURIZER VESSEL FAILURE

ABSTRACT

On October 26, 1961, at approximately 8:00 p.m. a failure occurred in the Spert III pressurizer vessel, an integral component of the primary coolant system of the Spert III reactor facility. At the time of failure, the reactor contained a subcritical array of fuel, and non-nuclear tests were being conducted to evaluate experimental procedures and the performance of plant equipment in preparation for a series of cold water accident studies scheduled to follow. Subsequent investigations have led to the conclusion that because of error in the indication of the existing level-control instrumentation on the vessel, it was possible for one or more of the electric heaters to operate above the steam-water interface thus causing superheating of the steam and producing overtemperature in the upper half of the vessel. The failure mechanism was third-stage creep at elevated temperatures.

This report contains the results of the study of the cause of failure. Supporting evidence is presented to enable the reader to make an independent analysis of this and other possible causes of the failure.

SPERT III PRESSURIZER VESSEL FAILURE

SUMMARY

On October 26, 1961, at approximately 8:00 p.m. a failure occurred in the Spert III pressurizer vessel, an integral component of the primary coolant system of the Spert III reactor facility. At the time of failure, the reactor contained a subcritical array of fuel, and non-nuclear tests were being conducted to evaluate experimental procedures and the performance of plant equipment in preparation for a series of cold water accident studies scheduled to follow. The failure was detected when the vessel insulation ripped and began to smoke, and steam emerged from the vessel. The indicated pressurizer operating conditions were 2465 psig and 666°F. The temperature of the water in the primary coolant system was 435 °F. All plant instruments were in service and performing normally prior to and during failure.

Subsequent visual examination of the vessel has revealed yielding of the vessel shell predominantly in the upper half of the vessel, cracking of the center girth weld, and severe cracking of the stainless-steel clad on the interior shell of the vessel. Cladding on the vessel heads was not affected. At least one of the cracks in the girth weld penetrated the weld, permitting steam to escape.

Since no evidence could be found in the operating records or by interviews with operating personnel of any unusual occurrences, or obvious or readily ascertainable causes for failure, various postulated causes were investigated.

The nature of these investigations, the results obtained, and the conclusions drawn are the subject of this report.

Evaluation of the vessel design and the metallurgical studies conducted to date have revealed no defects in the vessel design, materials of construction, or method of fabrication to which the vessel failure can be attributed.

Plant records, such as log books, plant data records, and instrument charts show no evidence of the existence of pressures or temperatures significantly in excess of design and to which the failure might be attributed; however, the available evidence from many sources strongly suggests that the vessel was subjected to overheating which resulted in failure of the vessel at 2660 psig. The evidence indicates the vessel was not subjected to significant overpressure prior to or at the time of failure. Overheating of the vessel appears to have occurred by exposure of one or more of the upper electrical immersion heaters which resulted in superheating of the steam to a temperature of 950°F to 1000°F in the upper portion of the vessel. The principal factors which contributed to the failure are liquid-level instrumentation and the lack of temperature-sensing devices in the pressurizer vessel.

The observed failure of the vessel cladding did not contribute to, and may not be associated with, the vessel failure.

CONTENTS

CONTENTS

CONTENTS

FIGURES

FIGURES

FIGURES

FIGURES

TABLES

SPERT III PRESSURIZER VESSEL FAILURE

I. INTRODUCTION

The Special Power Excursion Reactor Tests (Spert) Project, operated by Phillips Petroleum Co., was established as part of the U. S. Atomic Energy Commission's reactor safety program in 1954, and is directed toward experimental and theoretical investigations of the kinetic behavior and safety of nuclear reactors [1, 2]. The Spert III reactor facility described herein was constructed as a part of this safety program to fulfill the need for a facility in which to conduct reactor behavior and safety studies under operating conditions typical of pressurized-water and boiling-water reactors.

General objectives in the Spert III facility design were: (a) To provide a facility in which reactor power excursion tests could be performed and experimental information gathered on the kinetic behavior of the reactor; (b) To incorporate in the design a complete reactor and coolant system typical of existing and proposed pressurized-water power reactors to permit an investigation of safety problems common to this class of reactors; and (c) To incorporate sufficient flexibility in the over-all design to permit studies on several core designs.

Sustained reactor power operation was not a primary objective in the design of the facility. The majority of the experimental studies are conducted from low initial reactor powers and involve a relatively small total energy release. However, provision for power operation for a limited time has been incorporated in the design to permit limited investigations at conditions which duplicate, as nearly as economically feasible, those significant nuclear and hydraulic conditions normally found in pressurized-water reactor systems. Power operation also is required to obtain experimental information on the effect of high initial powers on reactor behavior, and on the transient response and hydraulic stability of the over-all coolant system.

The major components of the facility include a reactor vessel, a pressurizing vessel, and two primary-coolant loops including pumps and heat exchangers. The reactor vessel and primary-coolant piping system are designed for operation at pressures up to 2500 psig and temperatures up to 650°F. Coolant flow rates up to 20,000 gpm through the reactor core are available. The heat removal capacity of the two evaporative heat exchangers is 60Mw. Auxiliary equipment necessary for operation of a reactor of this type also has been included.

The reactor and major plant equipment are remotely operable, with the controls located in a control center building approximately 1/2 mile from the reactor. The reactor shielding has been limited to that which will permit work on the reactor to proceed following short-time power operation.

---

* Numbers in brackets refer to Section IX, References.

The conceptual design of the facility was prepared by Phillips Petroleum Co. in 1955. Engineering design and inspection were completed by the Stearns-Roger Manufacturing Co. under contract to the U. S. Atomic Energy Commission. The architect-engineering work performed by Stearns-Roger Manufacturing did not include the design of the fuel, the core support structure, the control rods and control rod drives, and the reactor control system. These items were designed and supplied by Phillips Petroleum. Construction was accomplished by a lump sum contract with Paul Hardeman, Inc. as the prime contractor. Construction was completed and the facility accepted for operation by Phillips Petroleum in October, 1958.

This report describes a failure of the pressurizer vessel, the results of the post-failure inspection, and an analysis of the cause of failure.

II. BRIEF DESCRIPTION OF THE SPERT III FACILITY

For orientation purposes, a brief description of the general features of the facility is presented in this section. A more complete description was published previously [3]. The location of the Spert Project with respect to other NRTS projects is shown in Figure 1.

1. GENERAL AREA LAYOUT

The Spert III reactor and major plant equipment are provided with remote control instrumentation and equipment permitting operation from the control center. The building housing the reactor is located about 1/2 mile from the control center as shown in Figure 2.

2. REACTOR VESSEL AND COOLANT SYSTEM

Insofar as is economically feasible, the facility incorporates the principal nuclear and hydraulic features of conventional pressurized-water and boiling-water reactors. The reactor vessel and coolant system are designed for a maximum operating pressure and temperature of 2500 psig and 650°F. The coolant system is comprised of two coolant loops as shown in Figure 3. Each of the primary loops consists of two canned-rotor pumps operating in parallel, a heat exchanger, two flow tubes, and flow-control and check valves. The total flow capability of both loops is 20,000 gpm and the heat removal capacity is 60 Mw. Coolant enters the reactor vessel at the bottom, flows upward through the reactor core, reverses direction and flows downward through the thermal shields, leaving the vessel near the bottom.

The primary-coolant pumps are canned-rotor pumps, each capable of delivering 5000 gpm against a head of 328 ft of water. The pumps require the circulation of cooling water through coils surrounding the motor stator. Heating of the primary coolant to 650°F may be accomplished in about 16 hr independent of nuclear energy input by utilizing the energy transferred to the coolant by the pumps.

The coolant flow is controlled by remotely-operated gate valves. The rate of flow is metered by either an 8-in. or a 16-in. flow tube to achieve an accurate measurement, depending on the rate. The primary piping system was built with spun-cast AISI type 304L stainless steel sized to permit water velocities up to 30 ft/sec.

The reactor vessel is a multi-layer type vessel and the control rod drives are mounted on the bolted top head. Thermal shields comprised of four concentric rings of stainless steel with a combined metal thickness of 5-3/4 in. are provided to minimize the vessel-shell stresses during power operation. Because of the remote operation of the facility, the only shielding provided is a 6-in.-thick lead wall (one thickness of lead brick) around the reactor vessel. Primary-system pressure is maintained by a steam dome in the pressurizing vessel in which the steam pressure is generated by electric immersion heaters.

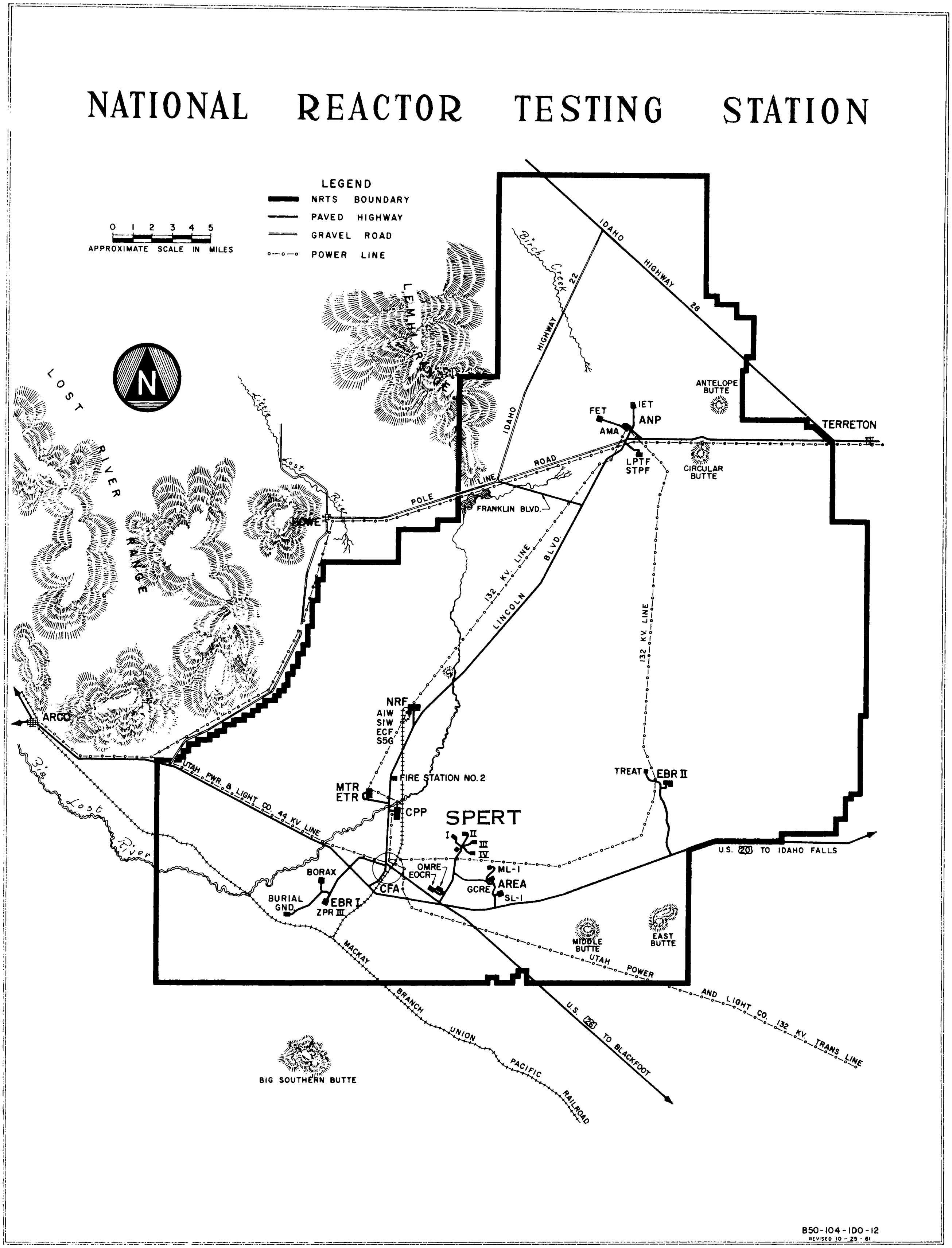

Fig. 1 Location of Spert Project at NRTS.

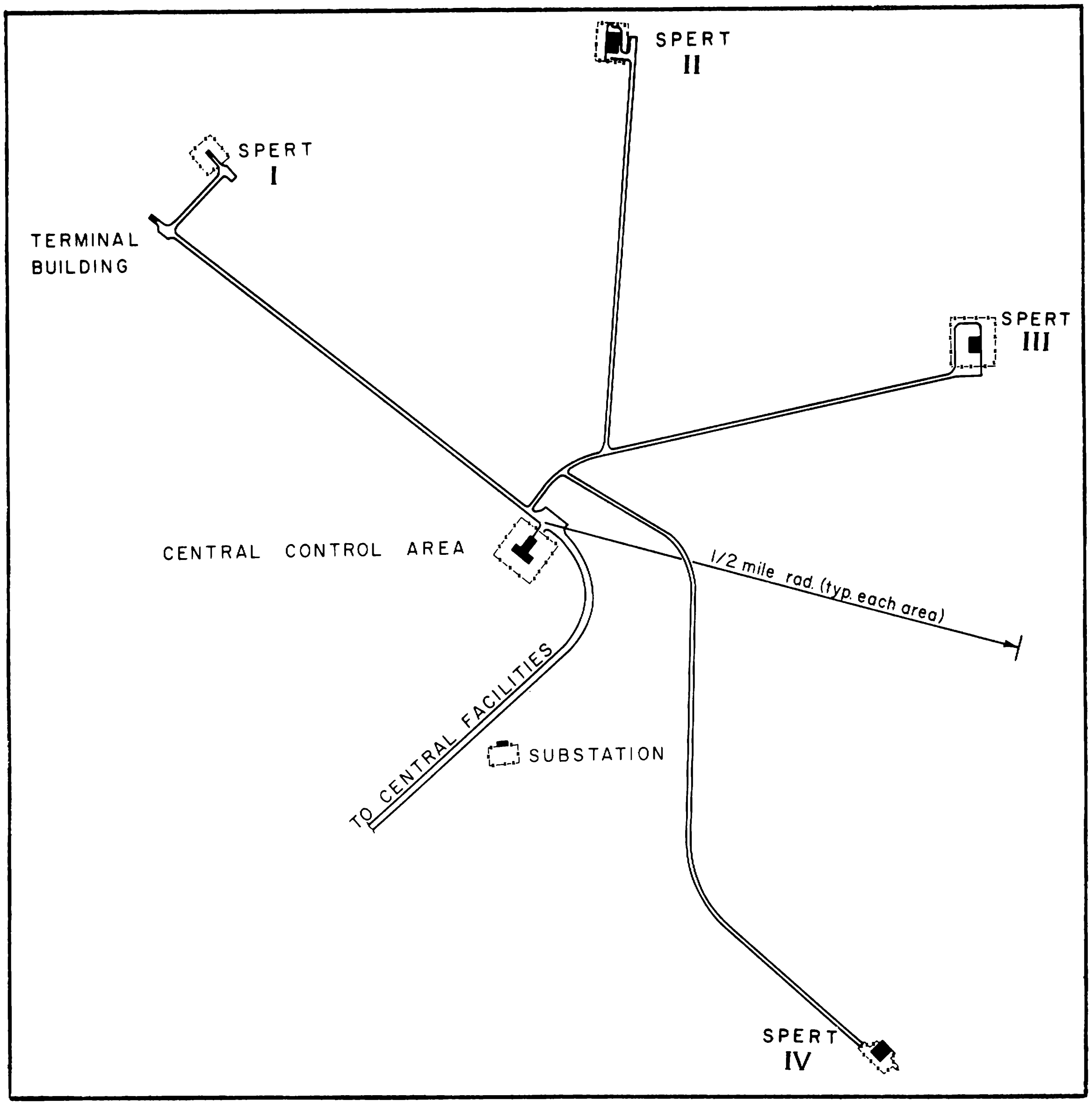

Fig. 2 Spert area layout.

The Spert III facility has been operated at varying conditions of temperature and pressure since December, 1958. The nature of the experimental program has required that the primary coolant system of the Spert III facility be subjected to an unusually large number of pressure and temperature cycles when compared to conventional usage. As a part of the experimental program, a number of reactor transient-power excursions have been conducted resulting in pressure pulses to the coolant system. A summary of the operating history of the plant excluding transient testing is presented in Section IV.

3. REACTOR CONTROL ROD DRIVES

The control rod drives are air-driven with a mechanically positioned lead screw controlling the position and rate of movement during withdrawal.

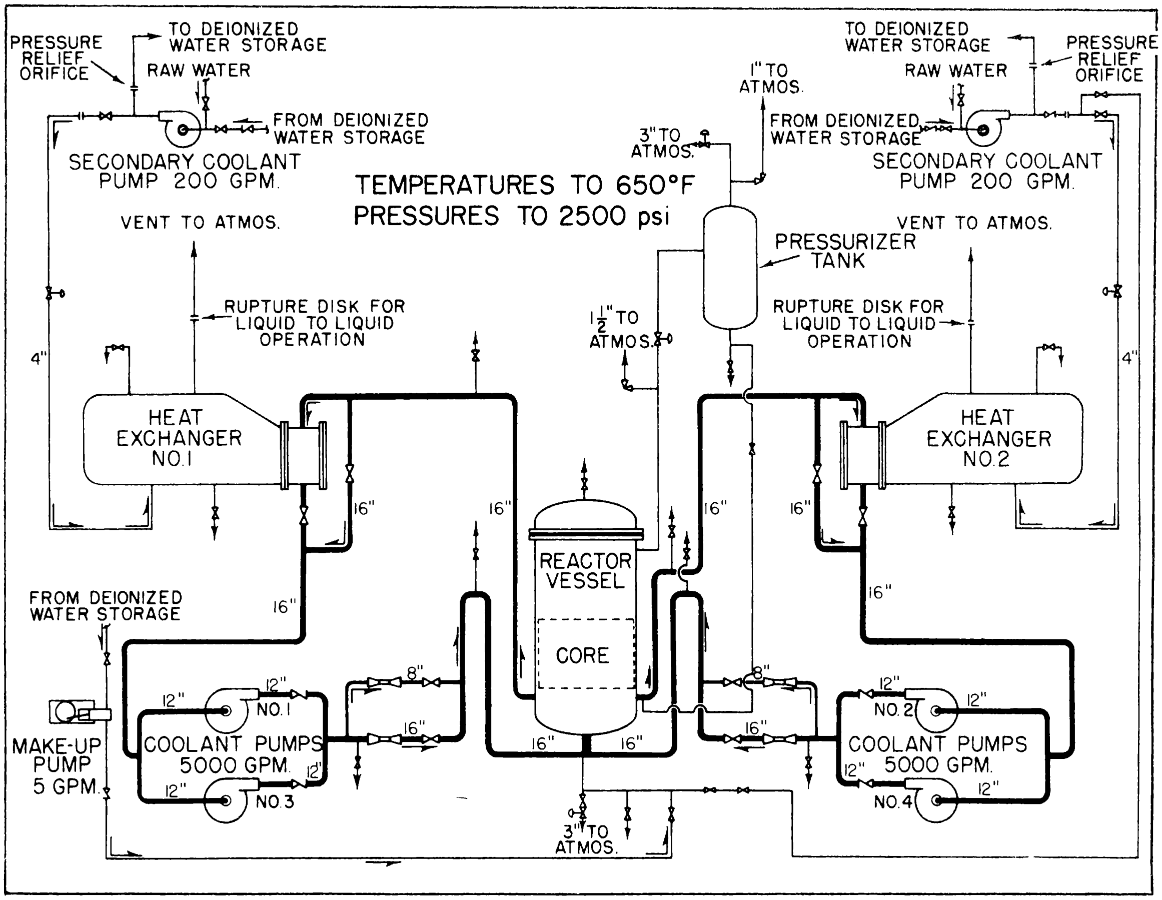

Fig. 3 Spert III piping schematic.

Reactor scram is accomplished by rapidly exhausting the air on one side of the air piston and allowing the compressed air to drive the piston to a seat.

All nuclear operation of the facility is performed remotely from a control room in the control center building approximately 1/2 mile from the reactor building.

4. AUXILIARY EQUIPMENT

All equipment necessary to provide services for an integrated reactor plant has been furnished. Raw water and electric power are furnished from the control center area.

III. DESCRIPTION OF THE PRESSURIZER VESSEL

1. MECHANICAL

The pressurizer is used to produce and transmit a pressure greater than the primary-system saturation pressure, to maintain the required net positive suction head for the primary circulating pumps, and to establish the desired bulk coolant pressure for given experiments. During normal operation the vessel is approximately half-full of water with the electrical heaters in the pressurizer submerged. By heating the water, a steam dome is created in the upper half of the vessel which both maintains the required pressure and serves as a cushion to absorb pressure surges in the primary system.

The Spert III pressurizer vessel specifications were prepared by Stearns-Roger Manufacturing as part of the design of the Spert III reactor facility. Since the vessel was a long-delivery item, Stearns-Roger was directed to purchase the vessel for the AEC and supply the vessel to the construction contractor as a government-furnished item. A copy of the Stearns-Roger specifications is contained in Appendix B.

The pressurizer is a 2-ft, 9-in.-ID x 16-ft, 8-in.-high, all-welded construction vessel of ASTM A-264, grade 3, 0.04% maximum carbon with type 304L stainless-steel fittings and internal cladding. Mill reports for the backing material are shown in Figures 4 and 5. The backing plate is ASTM A-212 grade B firebox quality carbon steel. The vessel was fabricated by the Chicago Bridge and Iron Co. at Birmingham, Ala. Details of fabrication are as shown in Figure 6. Pictorial arrangement of the vessel is shown in Figure 7.

The vessel shell was clad by the Chicago Bridge and Iron Co. by the Hortonclad technique, which is a vacuum-brazing process performed at approximately 2000°F using a high-nickel brazing alloy. Following cladding, the plates were normalized at 1650°F. Samples taken for the test report are shown in Figure 8. The vessel was then hot-formed at 1450° to 1650°F, assembled, welded and stress-relieved at 1150°F for 1 hr for each in. of thickness. The direction of bending of the shell plates was in the direction of the final rolling at the mill.

Shell welds were made as shown in Figure 9 in accordance with the excerpts from the Chicago Bridge and Iron welding sequence listed below:

"I. The outside U-groove will be automatic multipass-welded.

Amps	700-800
Volts	29-32
Speed	17-20 ipm
Wire	7/32"Ø - L70
Flux	Gr. 50 - 8 x 48

"II. The inside U-groove will be back-chipped or arc-air gouged to solid metal.

"III. The inside U-groove will be hand-welded.

Electrode	1/4"Ø E-7020
Amps	250-320
Volts	0.24-28
Polarity	Negative

"IV. The overlay will be hand-welded.

	1st Pass	2nd Pass
Amps	100-145	100-145
Volts	23-26	23-26
Electrode	5/32"Ø 25-12 cb	5/32"Ø 308 ELC
Polarity	Positive	Positive
Technique	Stringers	Stringers

"V. Stress relieve at 1150°F for 1 hour per inch of thickness.

"The vessel was X-rayed completely prior to overlaying the clad seams and was X-rayed at random after clad overlay.

"The 3-in.-thick portion of the vessel was X-rayed as follows:

Machine	1 MEV
Time	4 min, 15 sec
Milliampere Second	0.25, D. C.
Film	Eastman Type AA - Lead Screen"

The vessel walls are 2.95-in. thick including a 1/8-in. cladding thickness. All dimensions shown in this section are nominal dimensions since in accordance with usual fabricating practice, no as-built dimensions were furnished by the fabricator. The vessel top is a 2:1 axis-ratio elliptical head with a minimum thickness of 2.59 in. including the 1/4-in. cladding. The bottom is made up of a 2:1 axis-ratio elliptical dished head welded to a 16-in.-diameter hemispherical head. Minimum thickess of the dished head is 2.59 in. including the 1/4-in. clad thickness. The elliptical dished heads were formed from roll-clad material and furnished by the Lukens Steel Co. A copy of the material test report furnished to Chicago Bridge and Iron by Lukens Steel is shown in Figure 10. The hemispherical head is fabricated of 1-1/4-in.-thick type 304L stainless steel. The theoretical vessel length between the top and bottom head weld lines is 11 ft, 10-1/2 in. Length between the head tangent lines is 12 ft, 7-1/2 in. The manufacturer's report of inspection, Form U-1, is shown in Figure 11. Total vessel weight is approximately 10 tons. The vessel support legs are three 8-in., 13-3/4 -lb channels welded to the straight side section of the vessel at the bottom and braced with 3- x 3- x 1/4-in. angles welded to the legs and bottom head. The vessel is mounted on two 8- x 8-in.,31-lb I-beams running across the top of the pressurizer pit, 3 ft below the process pit floor level. Horizontal movement is prevented by two sway braces connected to a tie band around the vessel near the top.

1520A Rev. 3

TENNESSEE COAL & IRON

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UNITED STATES STEEL CORPORATION

DEPARTMENT OF METALLURGY, INSPECTION & RESEARCH

NO. 44822

DS:3

FAIRFIELD STEEL WORKS

FAIRFIELD, ALA. February 13, 1957

CHEMICAL AND MECHANICAL TEST RECORD ON BASIC OPEN HEARTH High Tensile Firebox STEEL

FURNISHED TO Chicago Bridge and Iron Company

ADDRESS Boyles, Alabama

SPECIFICATION TCI-388 "B" ASTM A-212-54-T

REQUIREMENTS: (FOR PERMISSIBLE EXCEPTIONS SEE FULL COPY OF SPECIFICATION)

CHEMICAL C .35 Mx MN .90 Mx P .035Mx S .04 Mx Si .15/.30 MECHANICAL T.S. 70/85 Y.P. 38000 Min ELONG. 22% Min RED.

TEST NUMBER	MILL ORDER	PUR. ORDER	HEAT NUMBER	ANALYSIS C	MN	P	S	Si			REPRESENTS PIECES	SIZE	DIMENSIONS OF TEST PIECE WIDTH	THICKNESS	AREA	Y.P. PER SQ. INCH	TENSILE PER SQ. INCH	ELONG. % IN 2 INCHES	RED.	HOM. TEST	BEND	INSPECTED BY
D-111	RF-13703	3081 #4	30261	.31	.78	.018	.030	.271			1	2 7/8" x 74 x 123		.507	.2019	41010	81220	27.5		OK	OK	TCI
Top	"	"	"									"					82120					

Plate and tests were heat treated for grain refinement.

SWORN TO AND SUBSCRIBED BEFORE ME THIS 21 DAY OF February 1957

[signature]

NOTARY PUBLIC

CAR NUMBER Sou 316181

[signature], BEING DULY SWORN ACCORDING TO LAW, DEPOSES AND SAYS THAT THE FIGURES SET FORTH ABOVE ARE CORRECT, CONTAINED IN THE RECORDS OF THE COMPANY.

J. W. Cassell

QUALITY CONTROL METALLURGIST

Fig. 4 Backing material mill reports.

1530A Rev. 3

DS:3

TENNESSEE COAL & IRON

DIVISION

UNITED STATES STEEL CORPORATION

DEPARTMENT OF METALLURGY, INSPECTION & RESEARCH

NO. 4446

FAIRFIELD STEEL WORKS

FAIRFIELD, ALA February 13, 1957

CHEMICAL AND MECHANICAL TEST RECORD ON BASIC OPEN HEARTH High Tensile Firebox STEEL

FURNISHED TO Chicago Bridge and Iron Company

ADDRESS Boyles, Alabama

SPECIFICATION TCI-388"B" ASTM A-212-54-T

REQUIREMENTS: (FOR PERMISSIBLE EXCEPTIONS SEE FULL COPY OF SPECIFICATION)

CHEMICAL C .35 Mx MN .90 Mx P .035 Mx S .04 Mx Si.15/.30 MECHANICAL T.S. 70/85 Y.P. 38000 Min ELONG 22% Min RED

TEST NUMBER	MILL ORDER	PUR. ORDER	HEAT NUMBER	C	MN	P	S	Si	PIECES	SIZE	WIDTH	THICKNESS	AREA	Y. P. PER SQ. INCH	TENSILE PER SQ. INCH	% ELONG. IN 2"	RED.	HOLD TEST	BEND	INSPECTED BY
C-8465	HF-13703	3081 #4	26401	.31	.81	.015	.033	.238	1	2 7/8" x 86 x 123		.506	.2011	49230	75180	28.0		OK	OK	TCI
Top	"	"	"							"					75900					

Plates and tests were heat treated for grain refinement.

CAR NUMBER Sou 54788

SWORN TO AND SUBSCRIBED BEFORE ME THIS 21 DAY OF February 1957

NOTARY PUBLIC

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J. W. Cassell

QUALITY CONTROL METALLURGIST

Fig. 5 Backing material mill reports.

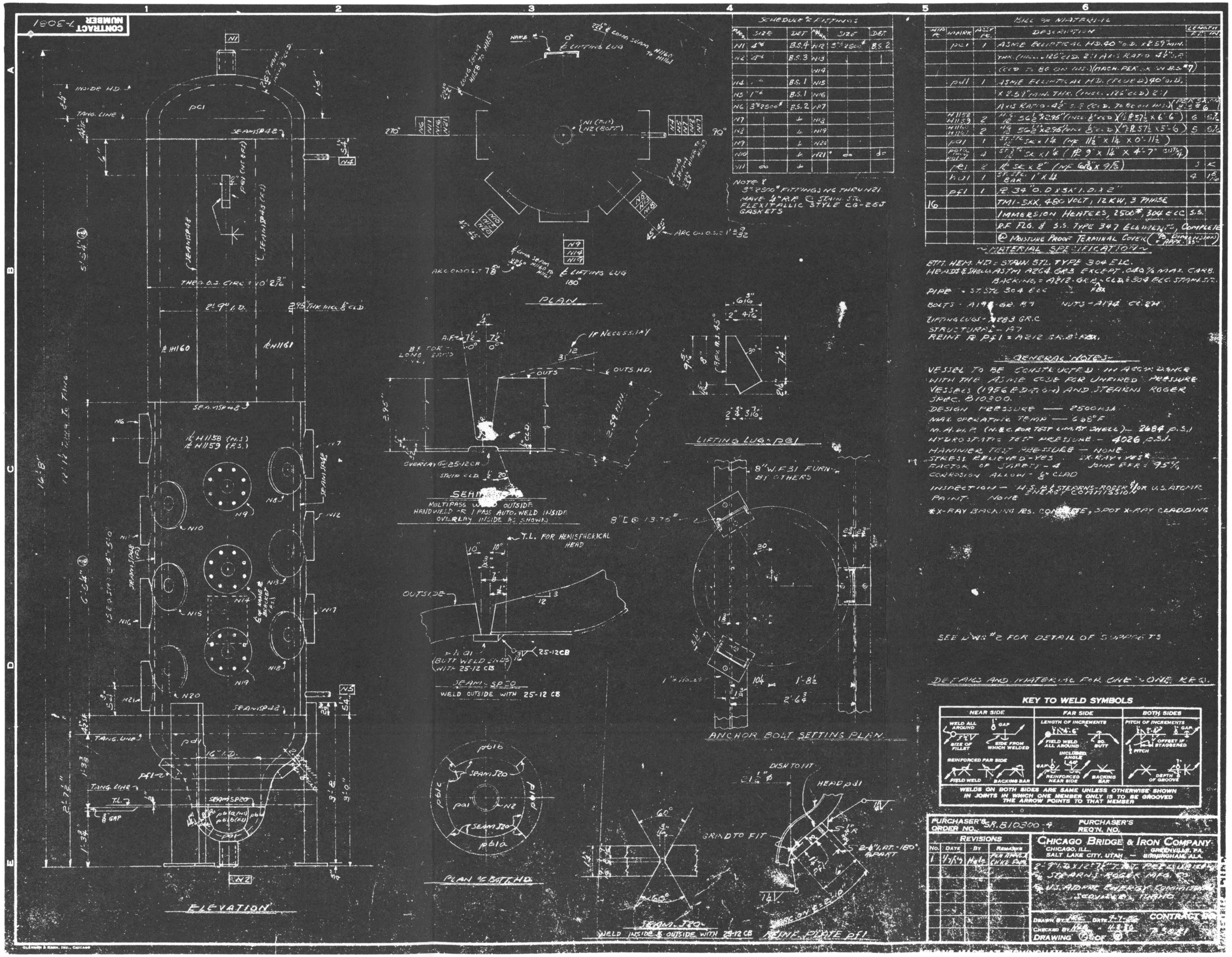


Fig. 6 Details of pressurizer fabrication.

Sixteen 480-volt, stainless-steel sheath, hairpin-type electrical immersion heaters are installed horizontally in the lower half of the pressurizer. The heaters are mounted through 3-in., 2500-psi flanges. Each heater has a capacity of 12 kw, giving a total heating capacity of 192 kw. The heaters are controlled by a pressurizer pressure recorder-controller system and are wired for staged operation so that six heaters turn off automatically at a pressure 100 psi below the control point, four heaters shut off at 50 psi below the control point, and another four shut off at 25 psi below the control point. The remaining two heaters shut off at 5 psi below the control point. The pressure is maintained at the control point by the reverse of the above procedure, ie, two heaters turn on if the pressure drops to 5 psi below the control point, then four additional heaters turn on if the pressure continues to drop to 25 psi below.

Four-inch nozzles are installed in the top and bottom of the pressurizer. The

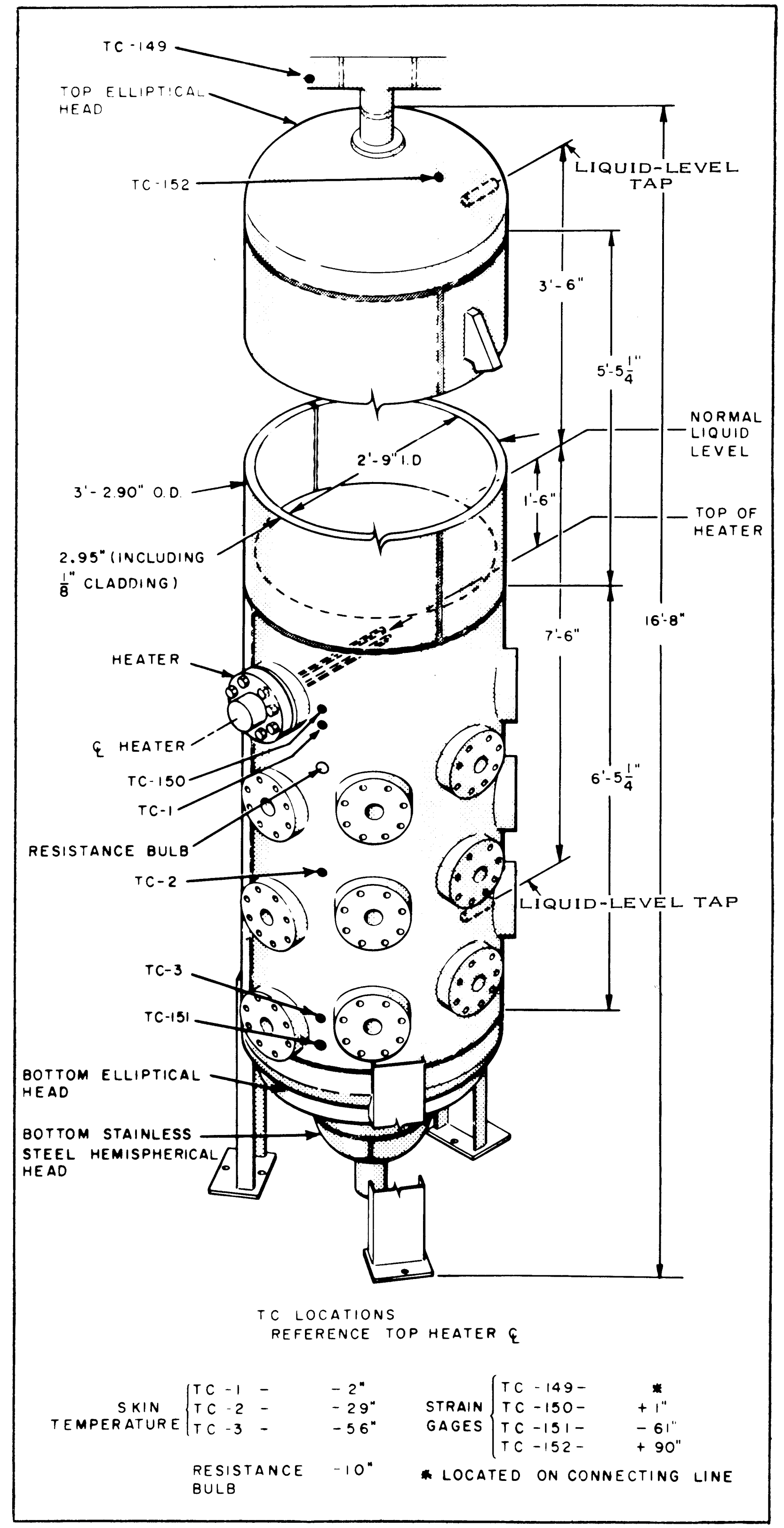
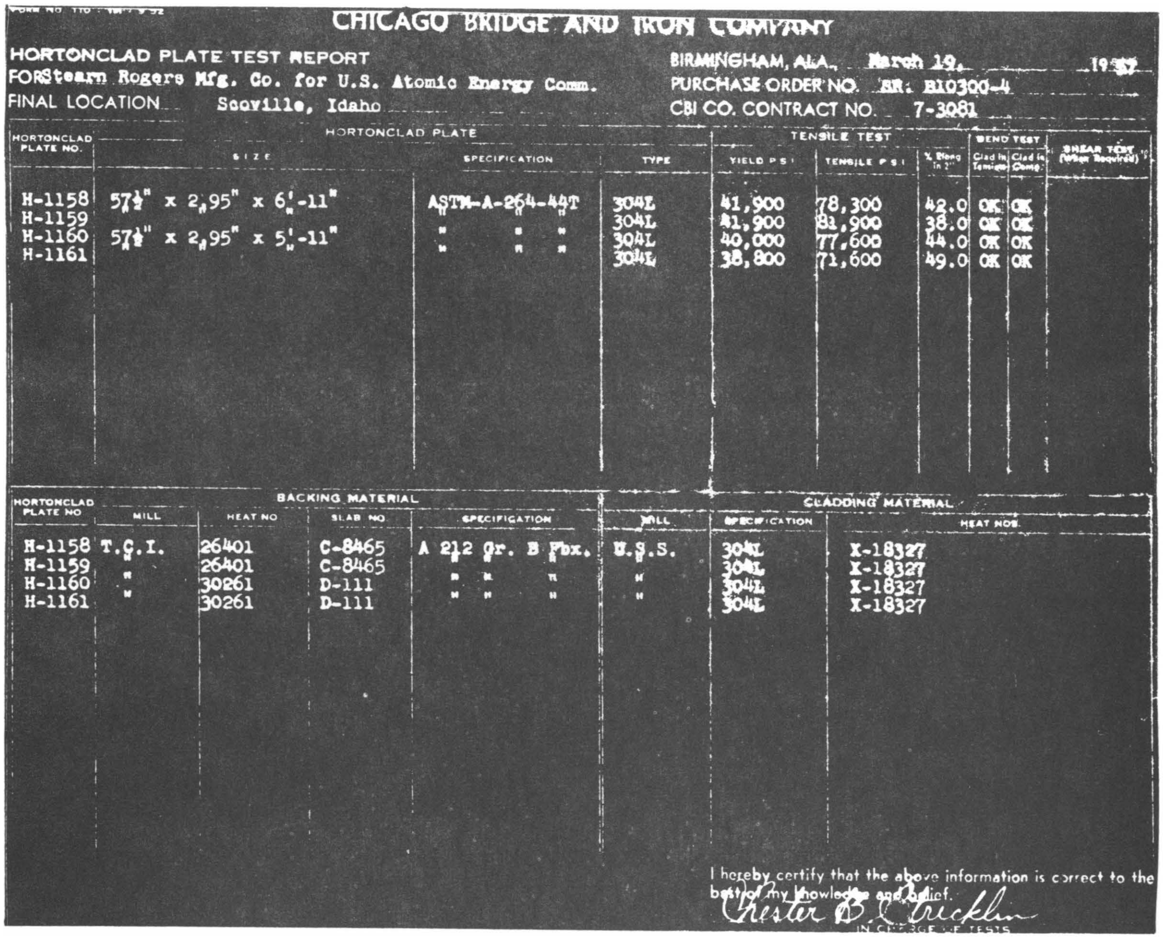


Fig. 7 Pictorial drawing of pressurizer vessel.



CHICAGO BRIDGE AND IRON COMPANY

HORTONCLAD PLATE TEST REPORT
FOR Stearn Rogers Mfg. Co. for U.S. Atomic Energy Comm.
FINAL LOCATION Scoville, Idaho

BIRMINGHAM, ALA. March 19, 1957
PURCHASE ORDER NO. SR: B10300-4
CBI CO. CONTRACT NO. 7-3081

HORTONCLAD PLATE NO.	HORTONCLAD PLATE SIZE	SPECIFICATION	TYPE	TENSILE TEST YIELD P.S.I.	TENSILE P.S.I.	% Elong in 2"	BEND TEST Clad in Tension	Clad in Comp.	SHEAR TEST (When Required)
H-1158	57½" x 2.95" x 6'-11"	ASTM-A-264-44T	304L	41,900	78,300	42.0	OK	OK	
H-1159		" " "	304L	41,900	81,900	38.0	OK	OK	
H-1160	57½" x 2.95" x 5'-11"	" " "	304L	40,000	77,600	44.0	OK	OK	
H-1161			304L	38,800	71,600	49.0	OK	OK	

HORTONCLAD PLATE NO	BACKING MATERIAL MILL	HEAT NO	SLAB NO	SPECIFICATION	CLADDING MATERIAL MILL	SPECIFICATION	HEAT NOS.
H-1158	T.C.I.	26401	C-8465	A 212 Gr. B Fbx.	U.S.S.	304L	X-18327
H-1159	"	26401	C-8465	" " "	"	304L	X-18327
H-1160	"	30261	D-111	" " "	"	304L	X-18327
H-1161	"	30261	D-111			304L	X-18327

I hereby certify that the above information is correct to the best of my knowledge and belief.
Chester B. Tricklin
IN CHARGE OF TESTS

Fig. 8 Shell material test report.

bottom nozzle connects to the reactor outlet piping via a 4-in. line through a 4-in. piston-operated valve which normally is open during operation. The 4-in. top line connects to the top of the reactor vessel through a 2-in. hand-operated globe valve which normally is closed during operation. Prior to September, 1959, the line contained a 4-in. piston-operated valve. Mechanical difficulties and leakage were experienced with the top and bottom pressurizer valves, prior to and following takeover of the Spert III facility by Phillips Petroleum necessitating the change to the hand-operated valve. Also connected to the line from the top of the pressurizer vessel are a 3/4-in. safety relief valve and a 3-in. connection for a 2-in. line to a pressure control valve Both the control and safety relief valves serve to protect the vessel from excessive pressures, and the control valve is also used to bleed non-condensable gases from the vessel during startup and operation, and to reduce the system pressure during operations required to shut the plant down.

Two 1-in. nozzles located 5-1/4 in. below and above the top and bottom head weld lines, respectively, connect to the liquid-level control system. The distance between the nozzles is 11 ft.

The pressurizer is insulated with 3-3/4 in. of Johns-Manville Hi-Temp. insulation. The total calculated heat loss from the pressurizer, including losses through the uninsulated heater flanges and other uninsulated surfaces as well as the insulated surface, is about 50,000 BTU/hr or 15 kw when operating at 668°F. The normal pressurizer heating and cooling rates were limited to 100°F/hr to minimize thermal stresses in the vessel. The capacity of the electric heaters (192 kw) is such that they give about this heat-up rate. The calculated time to raise the pressurizer temperature from 70 to 668°F is 6-1/4 hr for an average heating rate of 96°F/hr. When cooling and depressurizing the system, the pressurizer pressure is bled down in small increments over a sufficient time period so that a 100°F/hr cooling rate is not exceeded.

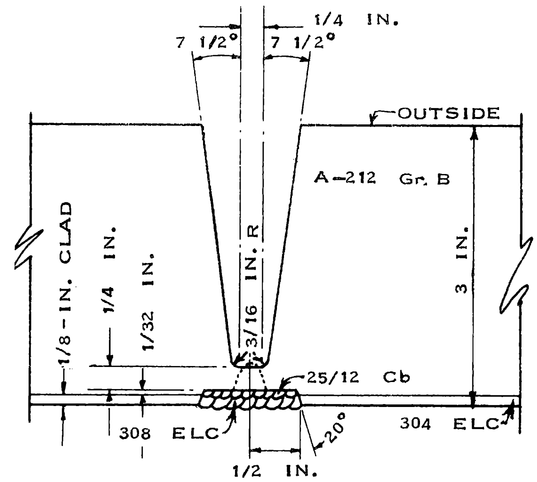


Fig. 9 Shell weld details.

2. INSTRUMENTATION AND CONTROLS

The instrumentation and controls provided on the pressurizer consist of the following systems: pressure control system, liquid-level control system and liquid-level gage, isolation valve controls, and temperature monitors. Since modifications to the instrumentation and controls were made following takeover of the plant by Phillips Petroleum, both the as-built plant and the subsequent changes are described.

2.1 Pressurizer Pressure-Control System

The pressurizer pressure-control system prior to October, 1961, consisted of three pressure transmitters (PT-6aR, PT-6bR, and PT-6cR) located in the reactor building basement; a range selector switch (HIC-6-3R) and a pressure indicator (PI-6R) located on the reactor building control panel; and a pressure recorder-controller (PRC-6C) located on the control-center control panel. A schematic diagram of the system is shown in Figure 12.

Three transmitters were supplied to provide accurate control of the pressure over the range 0 to 3000 psig. The range of each of these transmitters was as follows: PT-6aR, 0 to 1000 psig; PT-6bR, 900 to 1900 psig; and PT-6cR, 1800 to 2800 psig. The transmitters were Manning, Maxwell and Moore, Inc., "microsen" transmitters each having an electrical output signal of 0.5 to 5.0 milliamperes (DC) linearly proportional to the input pressure. At the start of a pressuring cycle the procedure called for all transmitters to be in service. As pressure in the pressurizer approached the upper operating limit of each transmitter, the procedure required removing the transmitter from service by closing block valves at the instrument to protect it from overpressure. The three-position range selector switch (HIC-6-3R) located on the reactor building control panel is engraved "0-1000", "900-1900", and "1800-2800" corresponding to the three available ranges. The range switch is used in

FORM NO. 221

31257 BS

LUKENS STEEL COMPANY AFFI

COATESVILLE, PA

TEST REPORT

DATE	LUKENS OFFICE NO	SPECIFICATIONS
MARCH 18,1957	465-1	A-264 GR. 3,BACKING TO ASTM A-212 GR FBX

CUSTOMER ORDER NO
3081 / /

3 TO · MR. G.H. PUTMAN, PA.
· CHICAGO BRIDGE & IRON CO.
· P.O. BOX 277
· BIRMINGHAM 1, ALA.

REPORT OF CHEMICAL AND PHYSICAL TESTS OF H.T. STL KIL FBX 70000 8.35% STAINLESS CLAD TYPE 304 ELC

ORDER NO. 1543-VR-80062-1,2

CAR NO: HALLS

HOMOGENEITY TESTS: O.K.

BENDING TESTS: O.K.

MELT NO	SLAB NO.	C	MN	P	S	CU	SI	NI	CR	MO		YIELD P.S.I.	TENSILE P.S.I.	% ELONGATION 2"	% RED OF AREA		SIZE OF PLATE
24417	4B	25	87	018	027		19						81500				1-40 OD X 3
66731	2	024	1.25	028	019		64	9.74	18.35			49000	80300	30			MIN GA 2-.59
SHEAR	TEST												32800				1-40 OD X 3
																	MIN GA 2-.59

PLATES AND TESTS ANNEALED AT 1950/2050°F.

I HEREBY CERTIFY THAT THE ABOVE TESTS ARE CORRECT TO THE BEST OF MY KNOWLEDGE AND BELIEF.

LUKENS STEEL COMPANY

ASS'T ENGR OF TESTS

Fig. 10 Top head material test report.

FORM U-1 MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
`equired by the Provisions of the ASME Cod· ·les

1. Manufactured by [illegible] Co., Birmingham, Ala. (Name and address of Manufacturer)

2. Manufactured for Stearns Roger [illegible] Co for U.S.Atomic Energy Comm., [illegible] Ida. (Name and address of Purchaser)

3. Type Vert (Horiz. or Vert.) Kind Tank (Tank, Jacketed, Heat Exch.) Vessel No. (3[illegible]) (Mfrs.' Serial) (..........) (State & State No.) Nat'l Bd. No. .... Yr. Built 1957

Items 4-9 incl. to be completed for single wall vessels (such as air tanks), jackets of jacketed vessels, or shells of heat exchangers.

4. SHELL: Material *A264 Gr.3 (Kind and Spec. No.) T.S. 70,000 (Flg. or F.B. & lowest T.S.) Nominal Thickness 2.95 in. Corrosion Allowance .125 Ins. in. Diam. 2 ft. 9 in. Length 0.A. ft. .... in.

5. SEAMS: Long. Dbl.Butt Fusion Weld (Welded, Dbl., Single, Lap, Butt) S.R. Yes (Yes ~~or No~~) X.R. Spot (Spot or Complete) Clad-(X.R.Backing complete) Sectioned No (Yes or No) Efficiency 95 %

Girth do S.R. Yes X.R. do Sectioned No No. of Courses 2

If riveted describe seams fully on reverse side of form.

6. HEADS: (a) Material .... T.S. .... (b) Material .... T.S. ....

Location (Top, bottom, ends)	Thickness	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a) Top	2.59(inc.11ga.clad)			2:1				Concave
(b) Btm	1-1/4					8"		Concave

If removable, bolts used .... (Material, Spec. No., T.S., Size, Number) Other fastening .... (Describe or Attach Sketch)

7. STAYBOLTS: .... (Material) If hollow .... (Size of Hole) Attachment .... (Threaded, Welded) Pitch .... (Horiz.) × .... (Vert.) Diam. .... (Nominal)

8. JACKET CLOSURE: .... (Describe as ogee & weld, bar, etc. If bar, give dimensions. If bolted, describe or sketch.)

9. Constructed for {Int. ~~Ext.~~} pressure of 2500 psi. Max. Temp. 668 °F. Subzero .... °F. Hydrostatic Test 4025 psi.

Items 10 and 11 to be completed for tube sections.

10. TUBE SHEETS: Stationary. Material .... (Kind & Spec. No.) Diam. .... in. (Subject to Pressure) Thickness .... in. Attachment .... (Welded, Bolted)

Floating. Material .... Diam. .... in. Thickness .... in. Attachment ....

11. TUBES: Material .... (Kind & Spec. No.) O.D. .... in. Thickness .... inches or gage. Number .... Type .... (Straight or U)

Items 12-15 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers.

12. SHELL: Material .... (Kind and Spec. No.) T.S. .... (Flg. or F.B. & lowest T.S.) Nominal Thickness .... in. Corrosion Allowance .... in. Diam. .... ft. .... in. Length .... ft. .... in.

13. SEAMS: Long .... (Welded, Dbl., Single, Lap, Butt) S.R. .... (Yes or No) X.R. .... (Spot or Complete) Sectioned .... (Yes or No) Efficiency .... %

Girth .... S.R. .... X.R. .... Sectioned .... No. of Courses ....

If riveted describe seams fully on reverse side of form.

14. HEADS: (a) Material .... T.S. .... (b) Material .... T.S. .... (c) Material .... T.S. ....

Location	Thickness	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a) Top, bottom, ends								
(b) Channel								
(c) Floating								

If removable, bolts used (a) .... (Material, Spec. No., T.S., Size, Number) (b) ....

(c) .... Other fastening .... (Describe or Attach Sketch)

15. Constructed for {Int. Ext.} pressure of .... psi. Max. Temp. .... °F. Subzero .... °F. Hydrostatic Test .... psi.

Items below to be completed for all vessels where applicable.

16. SAFETY VALVE OUTLETS: Number .... Size .... Location ....

17. NOZZLES:

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Various	4	1"&4"	Pipe stubs	304L	Various	Steel	Welded
Various	16	3"	Pad	A105 Gr.II		steel	Welded

18. INSPECTION OPENINGS: Manholes, No. .... Size .... Location ....
Handholes, No. .... Size .... Location ....
Threaded, No. .... Size .... Location ....

19. SUPPORTS: Skirt .... (Yes or No) Lugs .... (Number) Legs 3 (Number) Other .... (Describe) Attached Btm.Hd.W. (Where & How)

20. REMARKS: 33"IDx16'-8"OA Pressurizer Vessel C.B.&I.Contract 7-3081
*A212 Gr.B.Fbx Backing & type 304 L Clad
(Brief description of purpose of the vessel, as Air Tank, After Cooler, Jacketed Cooler, etc. State contents of each part.)

Form No. 541E VOD/ (Over) Auth.035 expires 12/31/58

Fig. 11a Form U-1 manufacturer data report on pressurizer.

We certify that the statements made in this report are correct and that all details of material, construction, and workmanship of this unfired pressure vessel conform to the ASME Code for Unfired Pressure Vessels.

Date May 8, 1957 Signed Chicago Bridge & Iron (Manufacturer) By Walter M. Floyd

Certificate of Authorization Expires ....

CERTIFICATE OF SHOP INSPECTION

Insurance Company's Serial Number 3596

VESSEL MADE BY .... at ....

I, the undersigned, holding a Certificate of Competency as an Inspector of Boilers and Unfired Pressure Vessels in THE STATE OF .... and employed by THE HARTFORD STEAM BOILER INSPECTION AND INSURANCE COMPANY of HARTFORD, CONN., inspected internally and externally, the vessel described in this report on ...., 19...., and certify that the statements made in this report are correct, corresponding with mill test reports of materials furnished by the builders, and measurements made of the vessel; and that this vessel is constructed in accordance with the ASME Code for Unfired Pressure Vessels.

Date 5-8, 1957

William J. White Commissions NB 1858

Inspector's Signature State or Nat'l Bd. & Number

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a Certificate of Competency as an Inspector of Boilers and Unfired Pressure Vessels in THE STATE OF .... and employed by .... of ...., have compared the statements in this manufacturer's data report with the completed vessel, and certify that parts referred to as data items .... were completed in the field in accordance with the requirements of the ASME Code for Unfired Pressure Vessels. The completed vessel was inspected and subjected to a hydrostatic test of .... psi.

Date ...., 19....

.... Commissions ....

Inspector's Signature State or Nat'l Bd. & Number

Fig. 11b Form U-1 manufacturer data report on pressurizer.

conjunction with the three transmitters. Before a pressure reading can be obtained, the switch must be in a position corresponding to the range of the transmitter in service.

Several weeks prior to the vessel failure, the three pressure transmitters were replaced with a single full range pressure transducer. The transducer is a Fairchild Controls Corp instrument, type 3S-G. The pressure range of the transducer is 0 to 2800 psig and the corresponding output signal is 0 to 150 millivolts. The function of the range selector switch has not been changed and it still is necessary to select the proper range with the selector switch in order to provide the proper signal to the pressure recorder (PR-6R) and pressure recorder-controller (PRC-6C). Since the pressure recorder-controller is constructed to receive a millivolt signal, it was necessary only to remove the load resistors from the instrument. No other revisions were required and the instrument ranges remain unchanged.

The pressure recorder (PR-6R) located on the reactor building control panel is a three-range instrument; the ranges correspond to the ranges of the selector switch. The operating range of the instrument is established by the range selector switch, HIC-6-3R. The instrument has no control function and serves only to record pressurizer pressure at the reactor building.

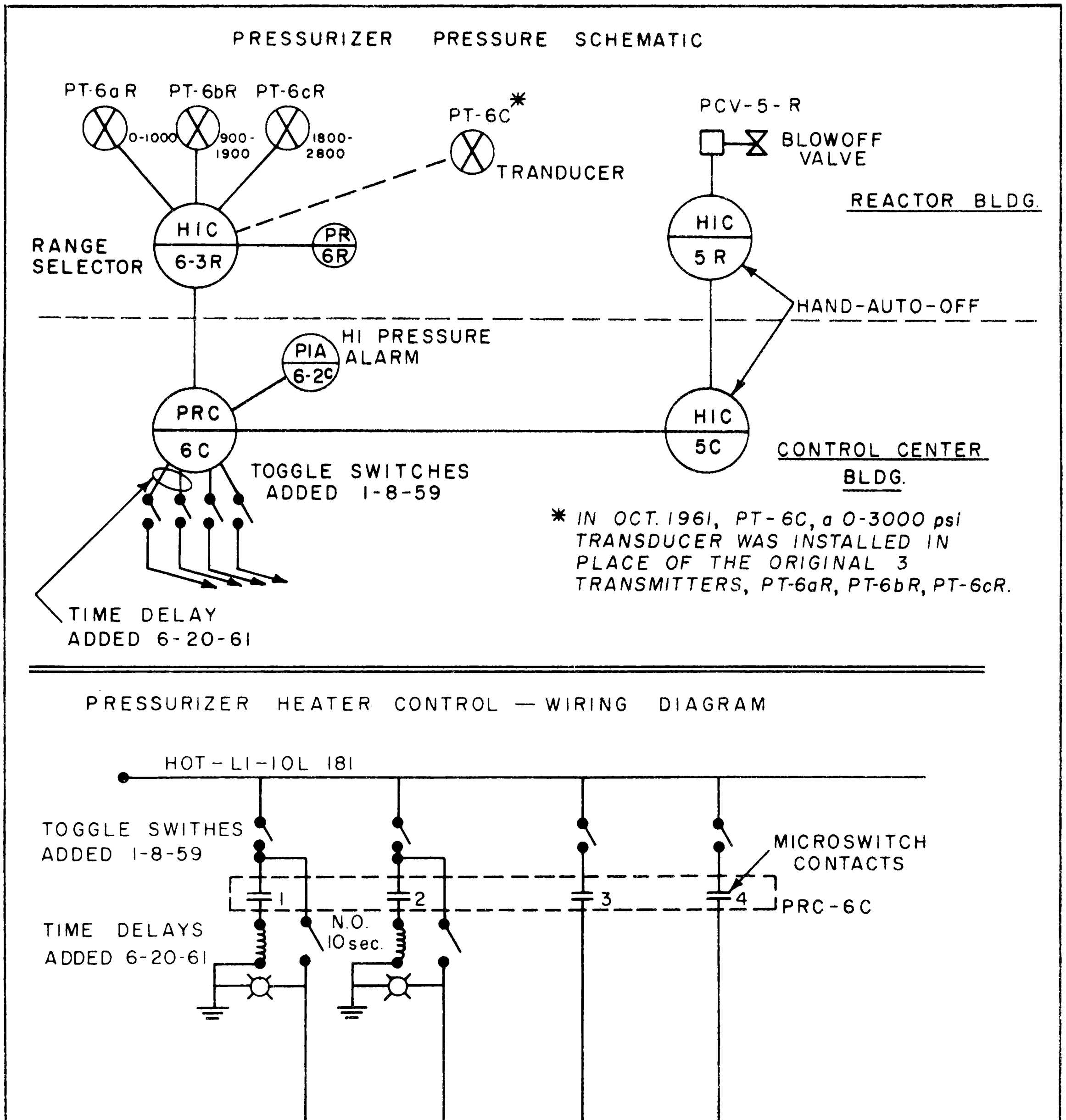


Fig. 12 Pressurizer pressure control schematic.

The pressure recorder-controller (PRC-6C), located on the control center control panel, is also a three-range instrument. The ranges and indicating lights on the case correspond to those on the selector switch. The instrument provides a recording of the pressurizer pressure at the control center, controls the pressurizer pressure by operating electric heaters as required, initiates the high pressure alarm and automatically opens the blowoff valve, PCV-5R, on top of the pressurizer in the event of excessive indicated pressure.

Control of the electric heaters is accomplished by four microswitches located inside the instrument (PRC-6C) case. The switches provide "on-off" control of the heaters by controlling relays in the heater power supplies. The pressure setpoint, adjustable from the front of the recorder-controller, permits selection of the desired operating pressure by simultaneously adjusting all four microswitches. The position of the microswitches relative to each other remains constant. Depending upon the proximity of the pressure-recording pen to the setpoint, the heaters are energized by the microswitches as follows:

Switch No.	Distance of Pen Below Setpoint (psi)	No. of Heaters Energized	Energy Input (kw)	Location of Heaters (from top)
1	5	2	24	1, 2
2	25	4	48	3, 4, 5, 6
3	50	4	48	7, 8, 9, 10
4	100	6	72	11, 12, 13, 14, 15, 16

An additional microswitch in the instrument case energizes the high pressure alarm at both the reactor building and the control center building at either 930, 1830 or 2730 psig, depending upon the operating range of the controller.

Four toggle switches are provided on the control-center control panel, permitting manual permissive control of the power to the four heater banks. The toggle switches are connected in series with the microswitches in the controller, thus the heaters cannot be energized unless the controller is calling for heat, the manual switches are turned on and the heater breakers are on. However, if the pressure controller power supply is turned off when the pressure recording pen is below the setpoint, two or possibly more of the heaters can be energized, depending upon the amount of deviation of the pen from the setpoint.

Because of the present design of the stepwise method of heater control, once the setpoint pressure is reached the two top heaters turn on and off to maintain the pressure in the vessel. (This situation permits the existence of large temperature gradients in the vessel.)

In addition to the above pressurizer pressure indicating and controlling instruments, eight other pressure instruments located on the control panels at both the control center and the reactor building indicate and/or record the pressure in the primary system.

Finally, as a check on all plant pressure instruments, a 12-in.-diameter Heise precision laboratory pressure gage with a range of 0 to 4000 psig is installed directly from the reactor pressure tap. The instrument is located on the operating floor near the pressurizer.

2.2 Liquid-Level System

The pressurizer vessel level system consists of an electronic level transmitter with associated temperature (density) compensation components, a level recorder (LC-6R) located on the reactor building control panel, and a level recorder-controller (LRC-6C) located on the control–center control panel. Associated with the level recorder-controller are hand-operated switches, HIC-6-4R and HIC-6-2C, for control of the system make-up pump and hand-operated switches, HIC-6R and HIC-6-1C, for control of the blowdown valve, LCV-6-1R. Switches are provided on both control panels. A schematic diagram of the level system is shown in Figure 13.

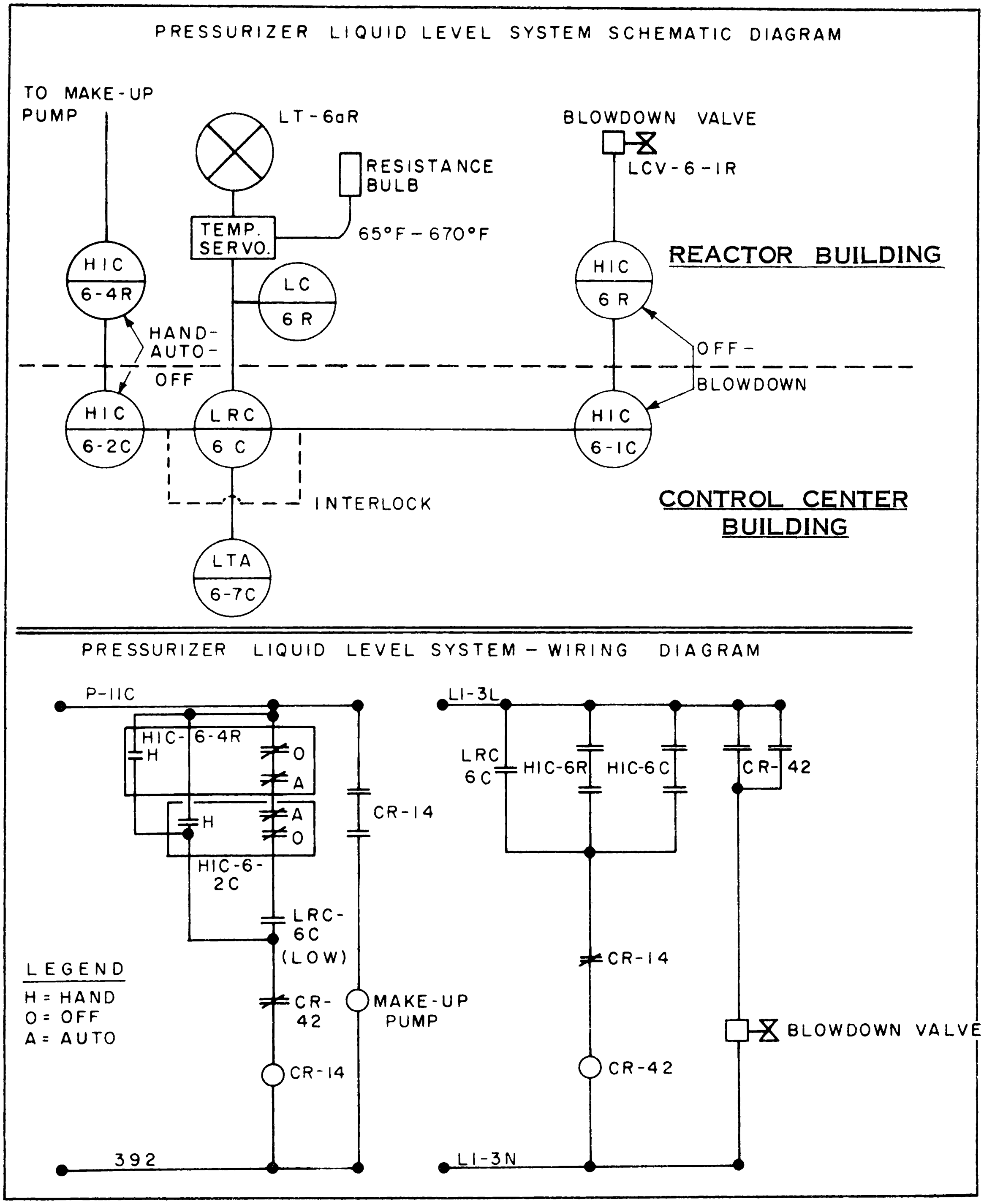


Fig. 13 Pressurizer level control schematic.

The level transmitter is a differential pressure type and has a range of 0 to 12 ft of water. The transmitter and associated temperature compensation components are Manning, Maxwell and Moore, Inc., instruments. The temperature-compensation unit is required to correct for density changes in the pressurizer liquid. The temperature-compensating components consist of a resistance bulb, a servo unit, and a transmitting potentiometer. The transmitter output signal, which is proportional to the height of the water column in the vessel, is corrected for density changes by the temperature compensator and transmitted to the level recorders.

In the original installation, the resistance bulb which senses the liquid temperature was located in the pressurizer outlet line below the pressurizer vessel. The temperature measurement proved to be inadequate, since the water at this location was colder than that in the pressurizer vessel. Since this situation resulted in an erroneously low level reading, the resistance bulb was moved on March 30, 1959, to a location on the shell of the vessel 10 in. below the centerline of the top pressurizer heater.

The level recorder, LC-6R, on the reactor building control panel receives the corrected signal from the level transmitter. The instrument has a range of 0 to 12 ft of water and has no control function, serving only to provide the operator with a recording of the liquid level in the pressurizer.

Level recorder-controller, LRC-6C, located on the control-center control panel, controls the water level in the pressurizer by starting the make-up pump when the level is low or by opening the blowdown valve when the level is high. The level is controlled in this manner between 7 ft, 6 in. and 8 ft, 6 in. These levels correspond to 1 ft, 6 in. and 2 ft, 6 in. above the top heater in the pressurizer. The instrument (LRC-6C) contains four microswitches adjustable by means of a setpoint adjuster on the front of the case. One microswitch starts the make-up pump should the level drop to 7 ft, 6 in. A second microswitch provides a low-level alarm signal at both the reactor building and the control center should the level drop to 7 ft. A third microswitch opens the blowdown valve should the level rise above 8 ft, 6 in. The fourth microswitch, not adjustable by the setpoint adjuster, provides a high-level alarm signal at both control panels should the level rise to a height of 9 ft.

The make-up pump is supplied with three-position switches at both control panels labeled "Hand-Auto-Off". The switches permit remote manual operation of the pump, overriding the controller. The switch is spring-loaded to the "Auto" position requiring the operator to hold the switch in the "Hand" position to operate the pump manually. The "Off" position stops the pump. In the "Off" position the operator is not required to hold the switch. The low-level alarm should warn the operator in the event that the switch is inadvertently left in the "Off" position.

The blowdown valve is also supplied with two-position switches at both control panels labeled "Off-Blowdown". When held in the "Blowdown" position the switches bypass the controller and permit remote manual blowdown of the liquid level by the operator. The switch handle is spring-loaded to the "Off" position. In the "Off" position the valve is controlled by the controller.

A liquid-level gage glass was installed on the pressurizer vessel by Phillips in November, 1960. The purpose of the glass is to provide a second level instrument which can be used to check the cold calibration of the primary level control system.

The accuracy of the level indication is very sensitive to temperature (density) changes; therefore, the location of the resistance bulb associated with the temperature-compensation device is extremely important. As currently installed, the resistance bulb is located 10 in. below the top heater. A post-failure review of the temperature data indicates that for some plant conditions the water near the steam-water interface can be as much as 200°F hotter than the water in the bottom of the vessel. Under these conditions, the indicated liquid level would be approximately 2 to 3 ft higher than the actual level. Further discussion of the liquid-level instrumentation and its contribution to the vessel failure is presented in Section VIII.

2.3 Isolation Valve Controls

As shown in Figure 3, the pressurizer vessel is connected to the primary coolant system by two 4-in. lines; one leading from the bottom of the vessel to the reactor coolant outlet line and one leading from the top of the pressurizer vessel to the top of the reactor vessel. As constructed, the two connecting lines contained air-operated gate valves. Controls for each valve were located on both the control center and reactor building control panels. The valve controls were connected through a key interlock switch such that both valves could not be closed at the same time unless an authorized person, with a key to the interlock switch, turned the switch to the "on" position.

Leakage through the top pressurizer valve necessitated removal of this valve in June, 1959. The air-operated valve was replaced with a hand-operated valve in September, 1959. Since that time the valve controls have been connected such that in order to close the bottom pressurizer valve, the key interlock switch must be used.

2.4 Temperature Devices

The pressurizer, as constructed, contained no sensing devices for detecting either the temperature of the water and steam in the vessel or the temperature of the vessel shell. Therefore in March, 1959, three contact thermocouples were installed on the vessel shell. These thermocouples were located as follows: TC 1, 2 in. below the centerline of the top heater; and TC 2, 29 in. below the centerline of the top heater; and TC 3, 56 in. below the centerline of the top heater. The three thermocouples were inter-connected so that the output signal was proportional to the average temperature detected by the thermocouples. The average temperature of the vessel was recorded on point 8 of temperature recorder TR-3R located in the reactor building.

In November 1960 several strain gages and four thermocouples were attached to the pressurizer vessel shell for use during an experiment which was conducted in April 1961. These thermocouples were intended to provide information regarding the temperature of the strain gages in order that temperature correction could be made to the strain gage readings, and were not a part of the plant instrumentation. The thermocouples were located on the pressurizer vessel as follows:

> TC 149 - On horizontal section of the blow-off line at the top of the pressurizer.

TC 150 - On pressurizer shell, 1 in. above centerline of top heater.

TC 151 - On pressurizer shell, 61 in. below centerline of top heater.

TC 152 - On pressurizer shell, 90 in. above centerline of top heater.

IV. OPERATING HISTORY OF THE PRESSURIZER

1. HYDROSTATIC TESTING OF THE PRIMARY COOLANT SYSTEM PRIOR TO PLANT ACCEPTANCE

Prior to the completion of construction, the construction contractor was required to perform a cold hydrostatic test on the Spert III primary coolant system to demonstrate the integrity of the system at 3750 psig, ie, one and one-half times the system design pressure. Records maintained by AEC inspectors have been reviewed to ascertain the procedures employed during the hydrostatic tests and the status of the relief valves. A tabulation of all hydrostatic tests conducted by the contractor prior to the acceptance test begun on September 9, 1958, are tabulated in Table I.

Information recorded in the mechanical inspector's log book indicates that the relief valves, PSV-2R and PSV-1R, located on the pressurizer and reactor vessel respectively, were welded in the system prior to the initial hydrostatic test conducted July 2, 1958. In order to pressure-test the system, it was necessary to break the seal on the valves and adjust the spring tension

TABLE I

SUMMARY OF COLD HYDROSTATIC TESTING DURING CONSTRUCTION

Date	Pressure (psi)	Remarks
7-2-58	3000	Test terminated because of leaks in top pressurizer valve and 10-in. flow control valve - west loop.
7-7-58	3000	Test terminated because of leaks in west heat exchanger channel cover and reactor head flange.
7-10-58	3000	Test terminated because of leaks past LCV-6R. Several valves failed to operate at 2500 psig.
7-24-58	3750	Test rejected due to leakage past blow-down and blowoff valves.
7-28-58	2500	Test conducted to check valve operation at design pressure.
7-29-58	3750	Hydrostatic test accepted.
8-1-58	2500	Test conducted to check valve operation.

so that the valves would not relieve at 3750 psig. Neither the inspector's log book nor other inspection records contain information to indicate whether the valves were reset and tested following the final hydrostatic test. The relief valves had not been tested since the acceptance date.

2. REVIEW OF OPERATING HISTORY

This section contains a summary of the significant occurrences during the operating history of the Spert III pressurizer vessel as evidenced by a complete review of plant log books and instrument charts. Table II summarizes the number of pressure and thermal cycles to which the pressurizer vessel has been subjected since September, 1958. Although the number of pressure and thermal cycles is unusually large when compared to a conventional power plant, the number also is low compared with the number of cycles required to cause failure.

Total operating time of the pressurizer at pressures in the region of 2500 psi was approximately 700 hr. On 13 occasions instrument charts of the pressurizer vessel pressure and the reactor vessel pressure show indicated

TABLE II

SUMMARY OF SPERT III PRESSURIZER PRESSURE AND TEMPERATURE CYCLES

(9-9-58 to 10-27-61)

Pressure Range (psig)	Number of Cycles [a] at Pressure and Temperature	Number of Cold [b] Hydrostatic Tests
Atmos to 500	46	0
500 to 1500	13	23
1500 to 2500	40	59
Above 2500 [c]	14	1
Total	113	83

(a) A cycle at pressure and temperature is arbitrarily defined here as any cycle in which the pressurizer vessel pressure was increased to 200 psia or above by heating the vessel and creating steam.

(b) A cold hydrostatic test is arbitrarily defined here as any test conducted at 500 psig or above by using the make-up or booster pump.

(c) See Table III.

pressure above 2500 psig. In six of these cases the indicated pressures were less than 2600 psi and resulted from normal cycling of the pressure controls and inherent inaccuracies in the instrumentation which permitted indicated pressures to cycle in this range. Since the over-all accuracy of the pressure-recording systems is quoted by the manufacturer to be no better than 1-1/2%, errors in the recorded pressure of $\pm$ 37 psig can be expected under normal circumstances. Hence, it has been the practice of plant operating personnel to utilize the precision laboratory gage at the reactor building as reference standard in adjusting the pressurizer pressure recorder-controller. Consequently, the recorded pressures must be corrected by reference to the appropriate calibration with the absolute gages, and a portion of the deviations above 2500 psig are due to inherent calibration errors. The date, time, pressure, and reason for each occurrence of pressure above 2600 psig are noted in Table III.

TABLE III

SUMMARY OF SPERT III PRESSURIZER PRESSURES ABOVE 2600 PSIG

Date	Time	Pressure (psig)[a]	Remarks
4-13-59	2112	2600	Pressure overshot during cold hydrostatic test using make-up pump.
12-14-59	0845	2630	Blown fuse in liquid-level instrument caused erroneously low level reading.
12-15-59	1835	2625	Due to temperature rise in primary coolant loops during experiments.
12-15-59	1900	2600	Due to temperature rise in primary coolant loops during experiments.
10-10-60	0955	2795	Not real; instrument turned on during cold hydrostatic test and apparently overshot during warmup. Reactor pressure recorder in service at the time shows no pressure above 2500 psig.
3-28-61	0845	2600	Fluctuation during manual control of pressure; above 2500 psig less than 5 min.
10-26-61	1900	2650-2700	Time of vessel failure. Explanation is contained in Section V.

(a) Pressures obtained from pressurizer pressure-instrument recording chart.

A number of rapid cool–downs have been experienced with the Spert III pressurizer since startup. A summary of these, including the reasons for the occurrences, is contained in Table IV. The principal reasons for the occurrences have been failures of plant equipment, resulting in rapid depressurization of the primary coolant system either directly or indirectly. A number of such depressurizations occurred during initial operation of the plant at design temperature and pressure in March, 1959. The cause of the sudden pressure drops eventually was traced to faulty design of the valve (PCV-3R) installed in the line connecting the top of the pressurizer and the top of the reactor vessel. Leakage of steam through valve PCV-4R permitted a steam void to form in the connecting line and in the top of the reactor vessel. Under certain conditions of flow and temperature in the primary loops the steam void was collapsed resulting in rapid pressure decreases in the system. Similar cases of rapid depressurization have been caused by leaks in system components or failure of fittings associated with in-pile equipment.

TABLE IV

SUMMARY OF RAPID COOL-DOWNS OF SPERT III PRESSURIZER

No.	Date	Time	Level LRC-6C (ft)	Pressure PRC-6C (psig)	ΔP (psig)	Saturation Temperature (°F)	ΔT (°F)	Time Interval (min.)	Temp. Change (°F/min.)	Press. Change (No./min.)	Remarks
*1	3-23-59	1830	8.3 - 9.5	1100-300	800	556 - 417	139	1.25	111	640	Trouble with valve PCV-3R
*2	3-23-59	2038	7.7	920-120	800	535 - 341	194	24	8.1	33.4	Trouble with valve PCV-3R
*3	3-24-59	0028	7.5	965-230	735	540 - 394	146	14	10.4	52.5	Trouble with valve PCV-3R
4	3-24-59	0643	8.0	1120-690	530	559 - 501	58	10	5.8	53.0	Blowdown valve trouble
5	3-31-59	1500	8.2 - 4.9	2520-1875	645	669 - 627	42	1.5	28.0	430	Flow transmitter valve broke off
6	3-31-59	2151	8.2 - 7.3	2480-2255	225	667 - 643	24	3.5	6.9	64.3	Blew out conax fitting
7	4-14-59	1224	7.6 - 5.7	2440-2130	210	665 - 645	20	5	4	42	Reduction of loop temperature
8	4-16-59	0920	7.5	2450-2070	380	665 - 641	24	2.5	9.6	152	Sudden pressure reduction - system shut down
9	6-22-59	1723	7.5	2520-2260	260	669 - 653	16	7	2.3	37	Spurious operation of blow-down valve
10	6-23-59	1615	8.5	2330-1940	390	658 - 631	27	10.5	2.6	37	PCV-3R valve leaking
11	6-23-59	1708	8.0	2185-1930	255	648 - 631	17	0.3	56.7	850	Plant upset due to leakage of valve PCV-3R
12	7-1-59	1316	7.4 - 4.5	1250-1000	250	572 - 545	27	2.75	9.8	91	East loop temperature by-pass control valve leaking
13	7-1-59	1350	6.0 - 2.7	930-730	200	536 - 508	28	1	28	200	East loop temperature by-pass control valve leaking
14	7-15-59	0150	9.0 - 7.2	2420-2135	285	663 - 645	18	1	18	285	Reduced pressurizer level because of high level alarm — Manual operation
15	7-15-59	0157	9.0 - 7.2	2200-1975	225	649 - 634	15	1	15	225	Reduced pressurizer level because of high level alarm — Manual operation
16	10-7-59	1143	7.6 - 1.8	2500-1400	1100	668 - 587	81	1	81	1100	Blew conax fitting on reactor top head
17	8-11-60	0650	7.3---	2360-1960	400	659 - 633	26	1.25	20.8	320	Ruptured gasket on PCV-4R - rapid shutdown
18	8-11-60	0657	---	1760-1410	350	618 - 588	30	2	15	175	Ruptured gasket on PCV-4R - rapid shutdown
19	8-11-60	0703	---5.0	1480-910	685	595 - 533	62	3.5	17.7	163	Ruptured gasket on PCV-4R - rapid shutdown
*20	8-12-60	1405	---	2160-910	1250	647 - 533	114	7.5	15.2	167	Manually opened blow-off valve operator error
21	10-25-60	1455	---	1450-1020	430	592 - 547	45	7.75	5.8	55.5	System shut down rapidly due to operating error
*22	10-28-60	1625	7.6	2130-250	1880	645 - 401	244	104	2.3	18	System shut down rapidly due to operating error
*23	3-20-61	1705	---	2535-30	2505	670 - 250	420	138	3.0	18.2	System shut down because of discovery of sub-standard t-c fittings in reactor access flange
24	4-4-61	1120	---	2450-1980	470	665 - 634	31	0.25	124	1880	Pressure transmitter bourdon tube rupture
*25	10-23-61	1745	7.6 - 2.0	2350-300	2050	659 - 417	242	81	3	25.3	Blew conax fitting on south access flange
26	10-26-61	2000	7.5 - ?	2510-1460	1050	669 - 593	76	28	2.7	37.5	Pressurizer vessel failure

* The occasions marked thus exceed recommendations of the architect-engineer

V. EVENTS PRECEDING FAILURE

At 2000 hours on October 26, 1961, the Spert III plant was shut down because of smoke emanating from the vicinity of the pressurizer vessel. This unusual occurrence was later determined to have been caused by the failure of the pressurizer vessel.

Personnel operating the plant included a shift engineer-in-charge, a reactor technician, and an electronics technician in the reactor building; and a data engineer-in-charge of the experiments, an instrument technician and a shift health physicist in the control center building. All plant instruments were in service, preceding and during the incident, with one exception. The exception was the pressurizer liquid-level gage glass. The gage glass had been removed from service because of steam leakage around the glass.

The test conducted previous to the shutting down of the plant operation was Test No. II-IA-4a-(1) of the Spert III Non-nuclear Cold Water Accident Test Series. (See Appendix "A" for the description of this test series.) The purpose of this test was to determine the effect of the startup of a cold, stagnant loop on the moderator temperature using the east loop as the cold, stagnant loop at a temperature differential of 20°F. The procedure was as follows:

With both loops in operation, the temperature of the system was established at 430°F. The east loop was isolated (time: 1745) by closing the main flow control valve, the temperature control valve, and the heat exchanger by-pass valve. The pumps in the east loop were stopped. The temperature in the west loop and reactor was raised to 445°F using a flow rate of approximately 10,000 gpm. The temperature was then leveled to begin the test. At this time, the west loop was at 445°F and the east loop was at 423°F. The heat exchanger by-pass valve was opened in the east loop to provide suction for the primary pumps. The primary pumps were started in the east loop. The main flow control valve in the east loop was opened (opening time: 13 sec). The east and west loop flows were approximately 9100 gpm each. At this time data were recorded on Data Sheet No. 4 and the recording oscillographs.

Data were recorded for approximately 20 min. During this time the plant system pressure rose approximately 140 psi and the pressurizer level rose 8.3 in. This behavior is normal and is a result of the increase in the bulk water temperature when the pumps are turned on. Calculations verifying that the magnitude of this pressure rise is not anomalous are shown in Appendix D. Actual chart records of the primary system pressure, PR-3-1R; pressurizer pressure, PRC-6C; reactor inlet temperature, TR-3-2R; pressurizer skin temperature, TR-3-8R; and pressurizer liquid level, LRC-6C are shown in Figures 14 through Figures 17, respectively.

Following the period of data aquisition, the west heat exchanger valve setting was increased and the plant conditions were stabilized at 2460 psi for the next test (time: 1920). At about this time, a rip in the pressurizer insulation covering was noted, but this was thought to be due to drying of paint which had been applied to the insulation cover in September, subsequent to the last temperature operation of the plant.

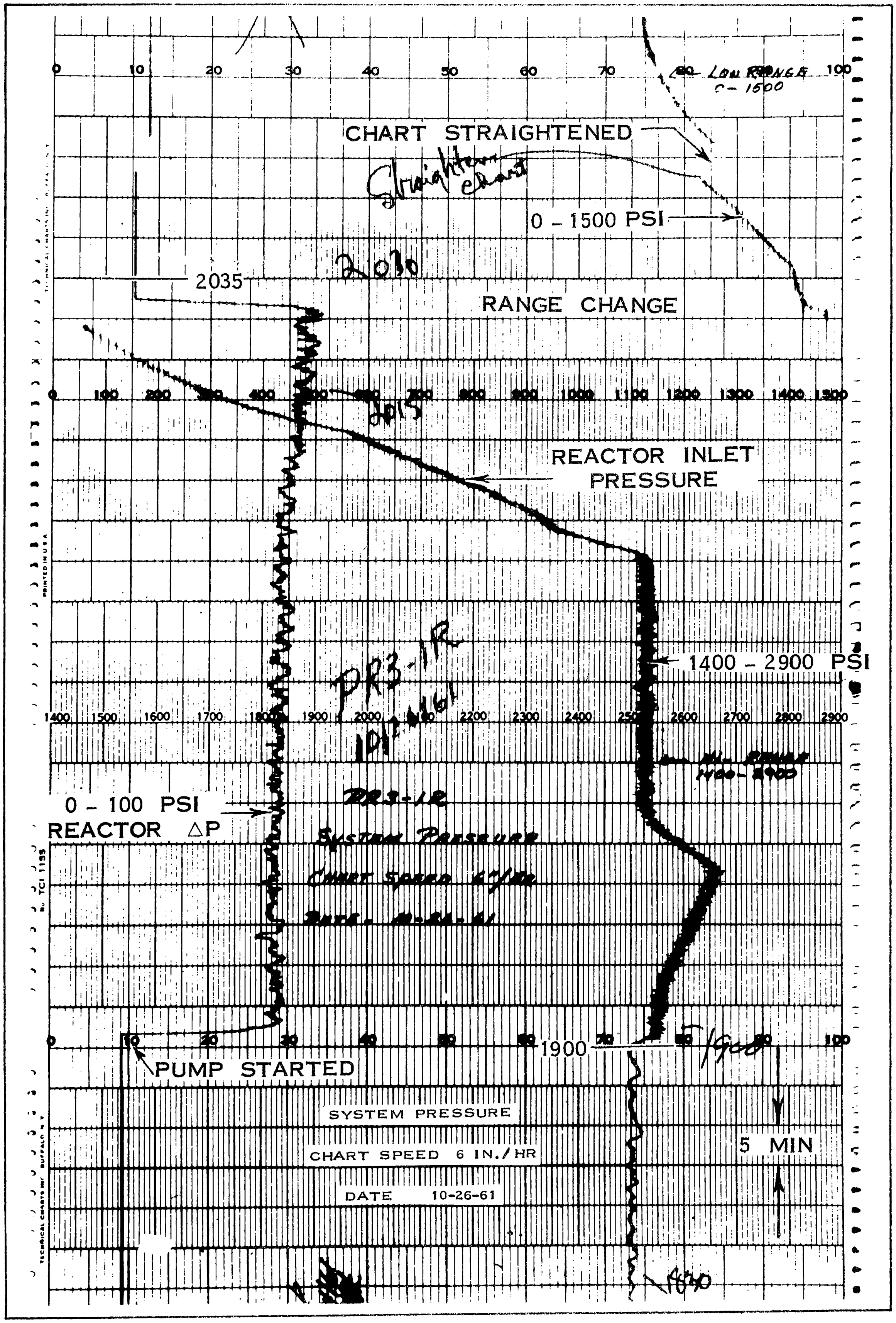


Fig. 14 Plant chart of reactor inlet pressure and pressure differential across reactor.

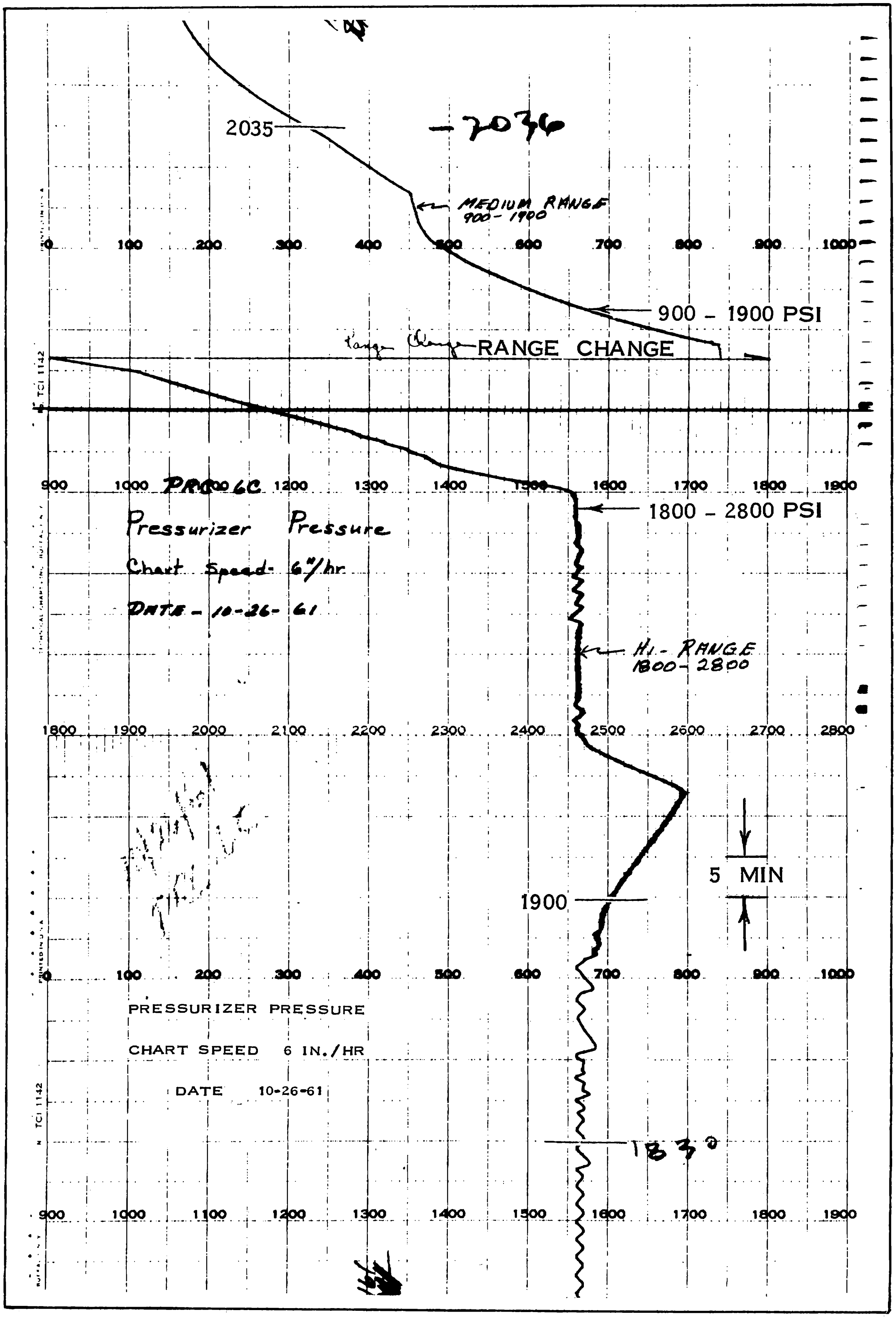


Fig. 15 Plant chart of pressurizer pressure.

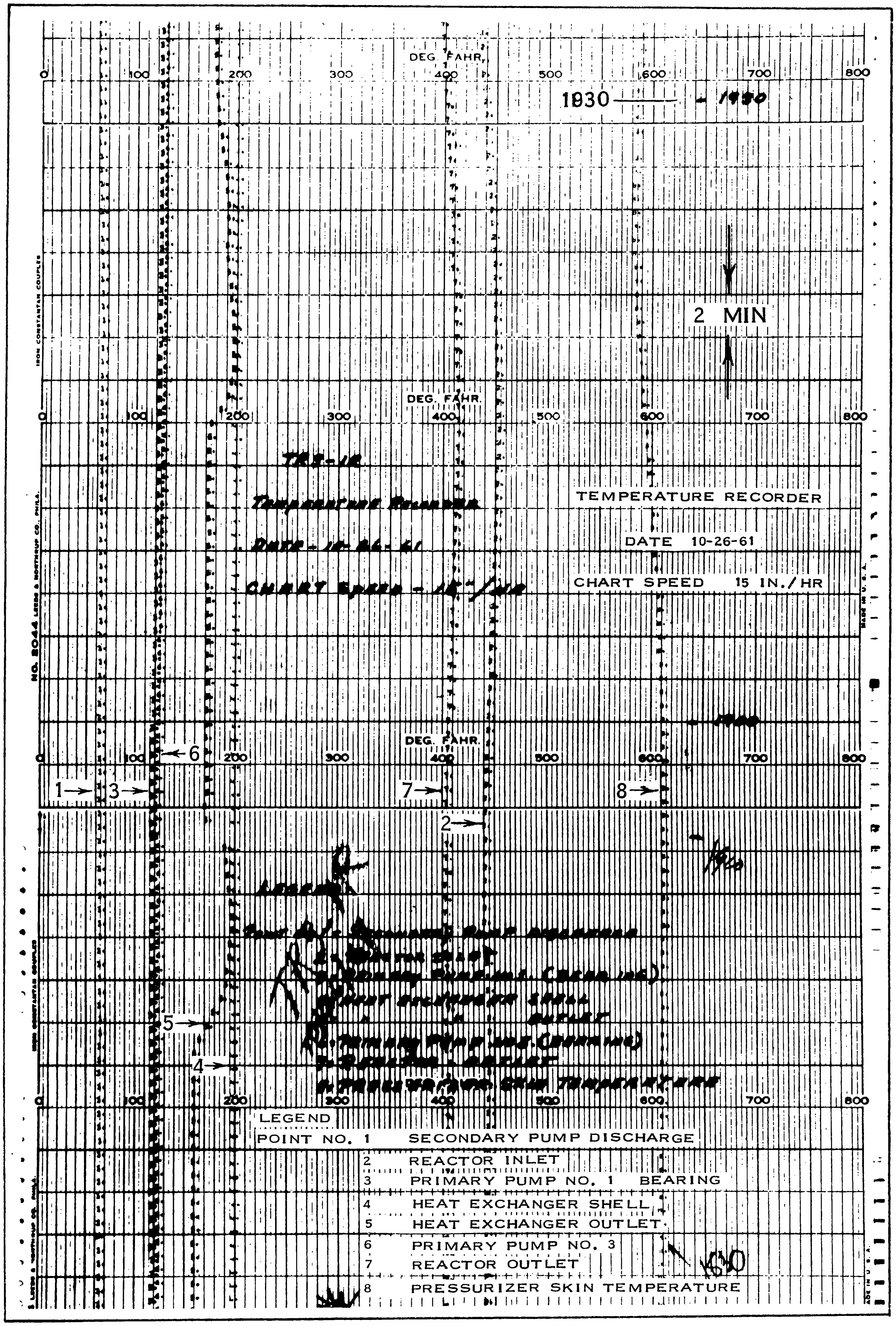


Fig. 16 Plant chart of temperature.

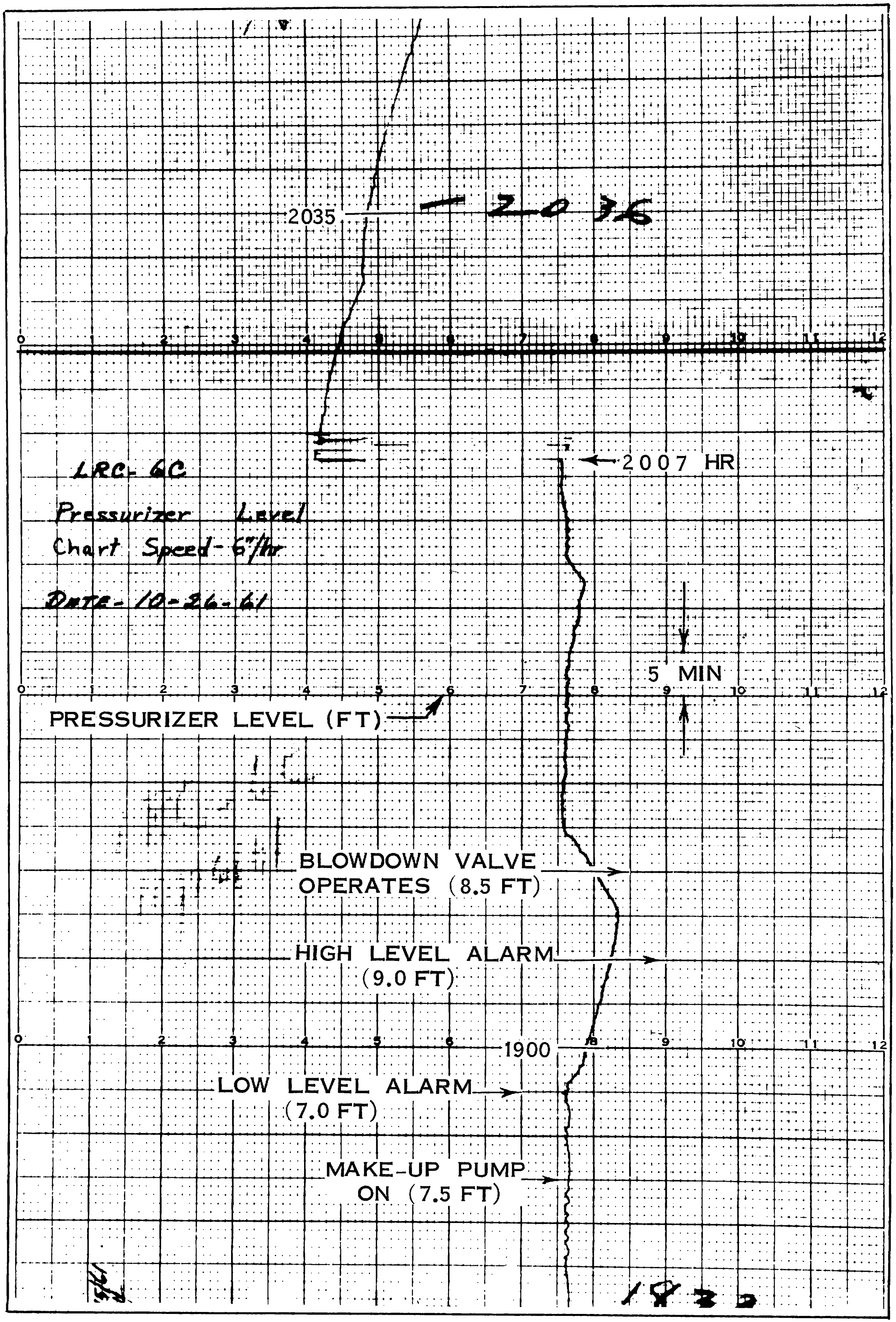


Fig. 17 Plant chart of pressurizer level.

Smoke was then observed at 2000 hours coming from the vicinity of the pressurizer. Normal plant shutdown was started immediately and the fire department called to stand by in case of flames. After the plant shutdown was started a hissing sound was heard, but the source could not be located. The plant shutdown continued at a faster-than-normal rate but within the 100°F/hour limitation on the plant. Plant pressure was maintained for suction head on the primary pumps until 2350 hours, when the plant had cooled to 150°F, the pressure had dropped to 400 psig and the pumps were turned off.

As shown in Figure 17, at approximately 2007 hours, during the cool-down, the pressurizer level instrument indicated an almost instantaneous drop from the 7.5-ft normal level to the 4.2-ft level, sounding the low-level alarm. The level instrument erratically indicated 7.5 ft to 4.5 ft for a short time and then indicated a gradual increase from 4.2 ft.

At 2130, with the temperature at 308°F and a pressure of 820 psig, the pressurizer level recorder indicated 6.1 ft.

At 2132 steam was observed coming from the side of the pressurizer vessel.

At 2140 water, as well as steam, was observed leaking from the pressurizer.

At 2207 the indicated pressurizer level was 6.3 ft.

At 2237 the indicated pressurizer level was 6.7 ft.

At 2327 the indicated pressurizer level was 6.9 ft.

At 2350, when the plant had cooled to 150°F, the pressure had dropped to 400 psig and the pumps were turned off, the indicated pressurizer level was 7.0 ft.

VI. POST-FAILURE INSPECTIONS

1. POST-FAILURE OBSERVATIONS

It was ascertained that the smoke which had been observed originated from the fabric covering of the blow-down line riser which is almost directly in front of the steam leak. The cloth had charred for about 24 in. along the cloth lap. No other charred cloth or evidence of burning was found. The vessel insulation and covering, apart from the rips, appeared normal. The oil base paint on the covering was not blistered or discolored, as shown in Figure 18.

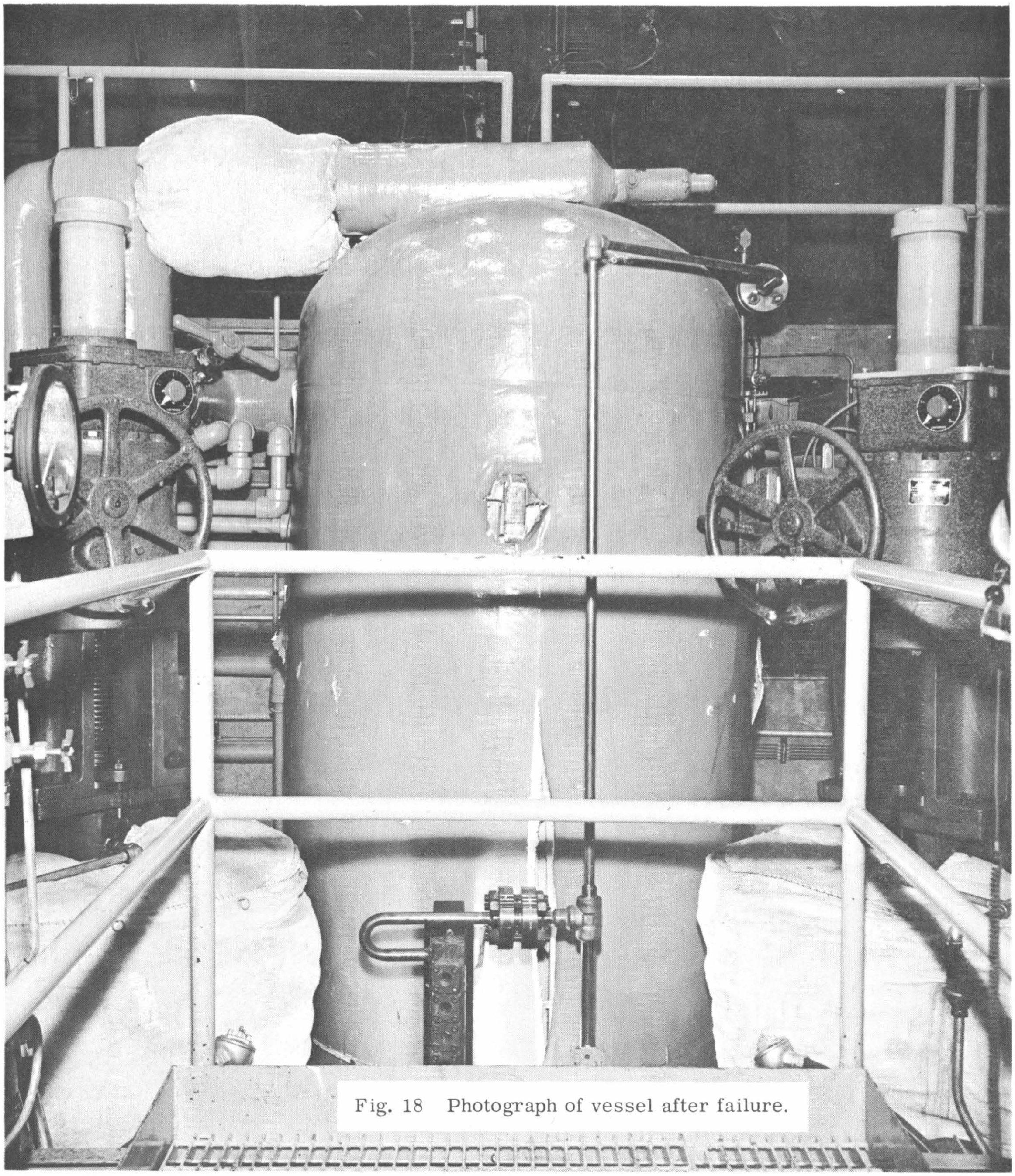


Fig. 18 Photograph of vessel after failure.

A complete checkout of the temperature-compensating device on the pressurizer level instrument on October 30 showed that a wire from the resistance bulb had been broken, apparently by the steam jet from the vessel leak, causing intermittent contact. Therefore, level data following the initiation of the apparent drop in level cannot be considered reliable but the post-failure indication is approximately that which might be expected with removal of temperature compensation. No other indications were noted of any sudden loss of water from the system. The calibration check on the pressurizer level instrument, on October 30, showed the instrument to be recording high by 2.4 in., which is not significant.

On October 27, the insulation on the pressurizer vessel was removed in the area of the steam leak. As shown in Figure 19, a 3/8-in.-wide x 2-1/4-in.-long hole had opened in the weld metal of the central girth seam. Other insulation had torn loose and it was noticed that the 1-in. bolt tying the vertical stabilizing band had broken (Figure 20). The break was covered with corrosion products (Figures 21 and 22), no fresh metal could be seen in the fracture, and the break appeared to be old.

A subsequent close inspection of the vertical stabilizing band on November 1 showed that it had moved in what appeared to be two separate stages, one of which (approximately 1 in.) was old enough to have the vessel coated with corrosion nodules and one of which was new (approximately 5 in.) and showed bright metal in the scratches and gouges (Figure 23). Measurements taken

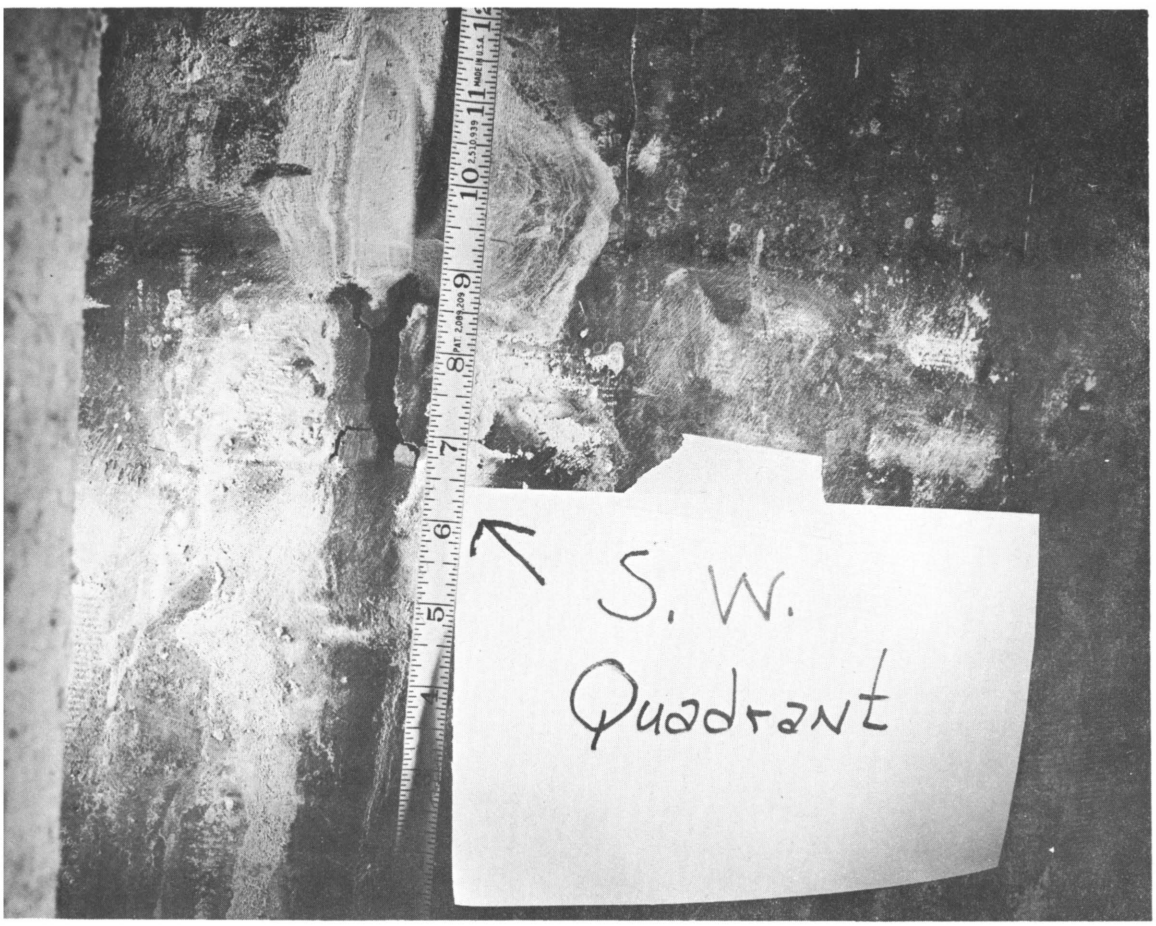


Fig. 19 Photograph of cracked weld at steam leak.

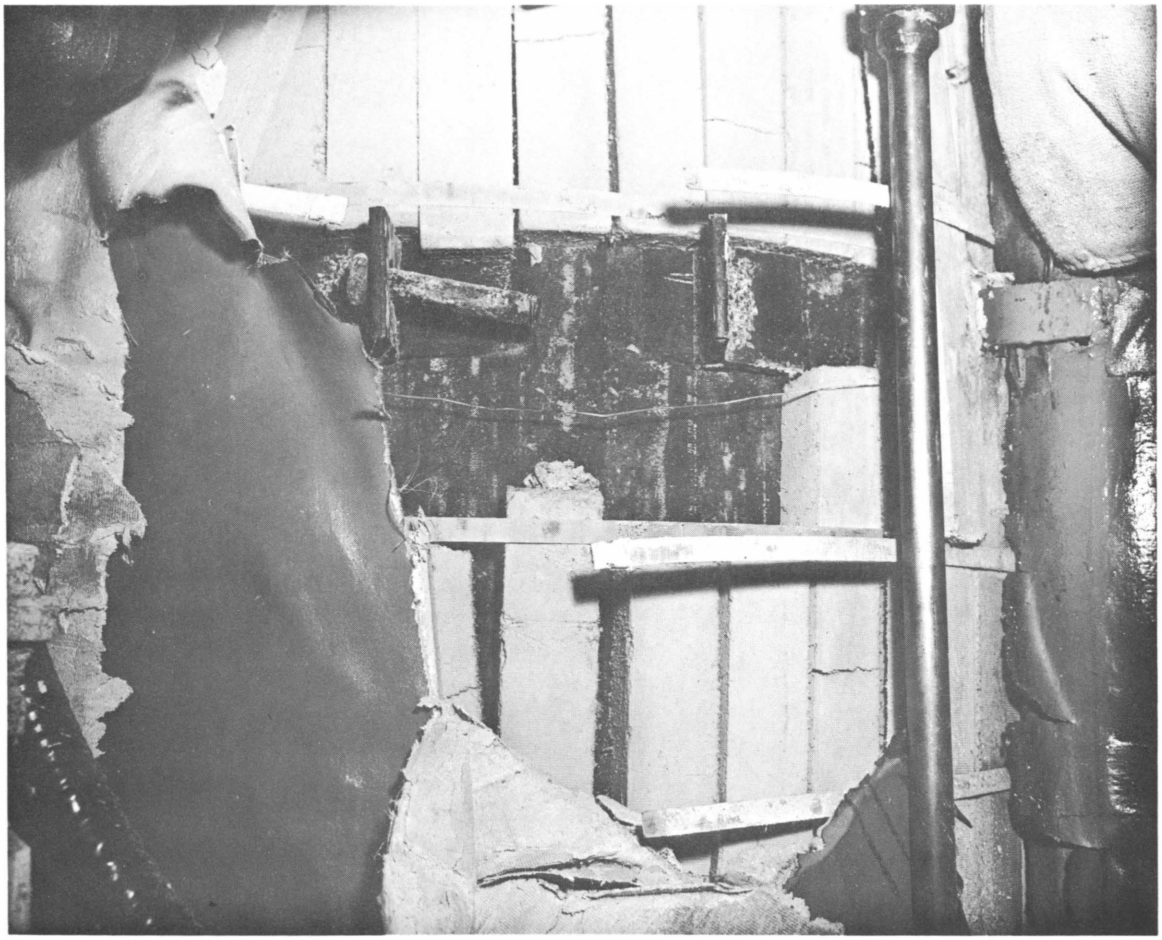


Fig. 20 Photograph showing stabilizing band in place.

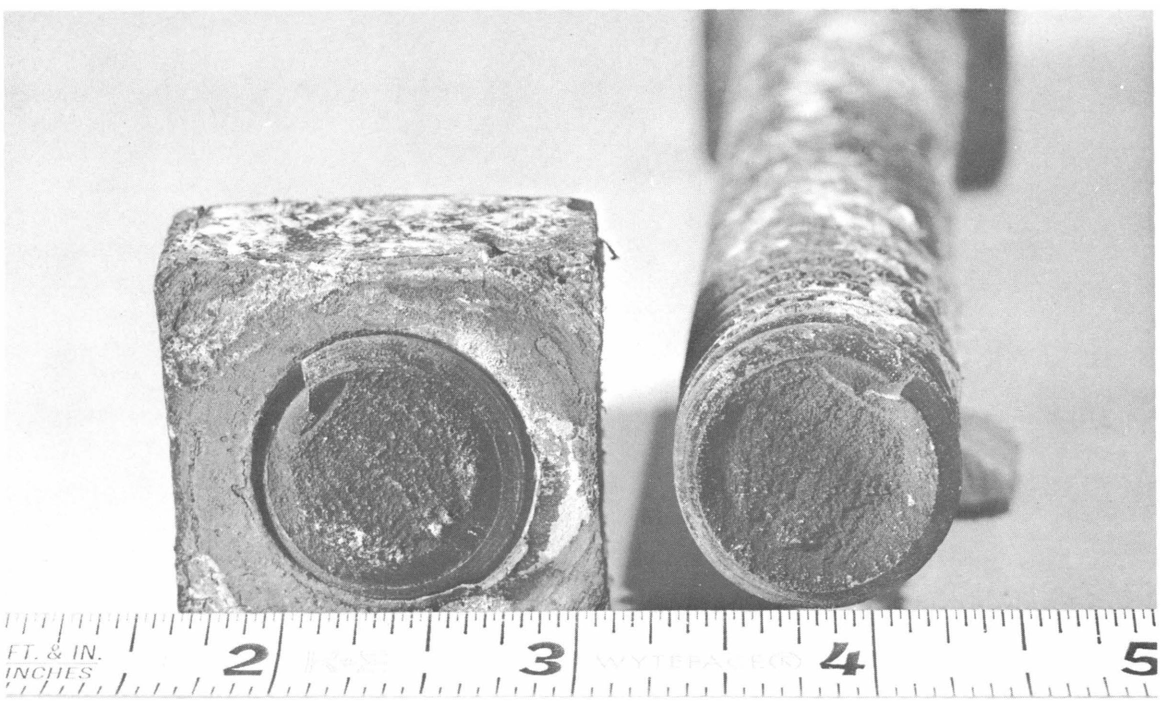


Fig. 21 Photograph of broken stabilizing-band bolt.

Fig. 22 Photograph of broken stabilizing-band bolt.

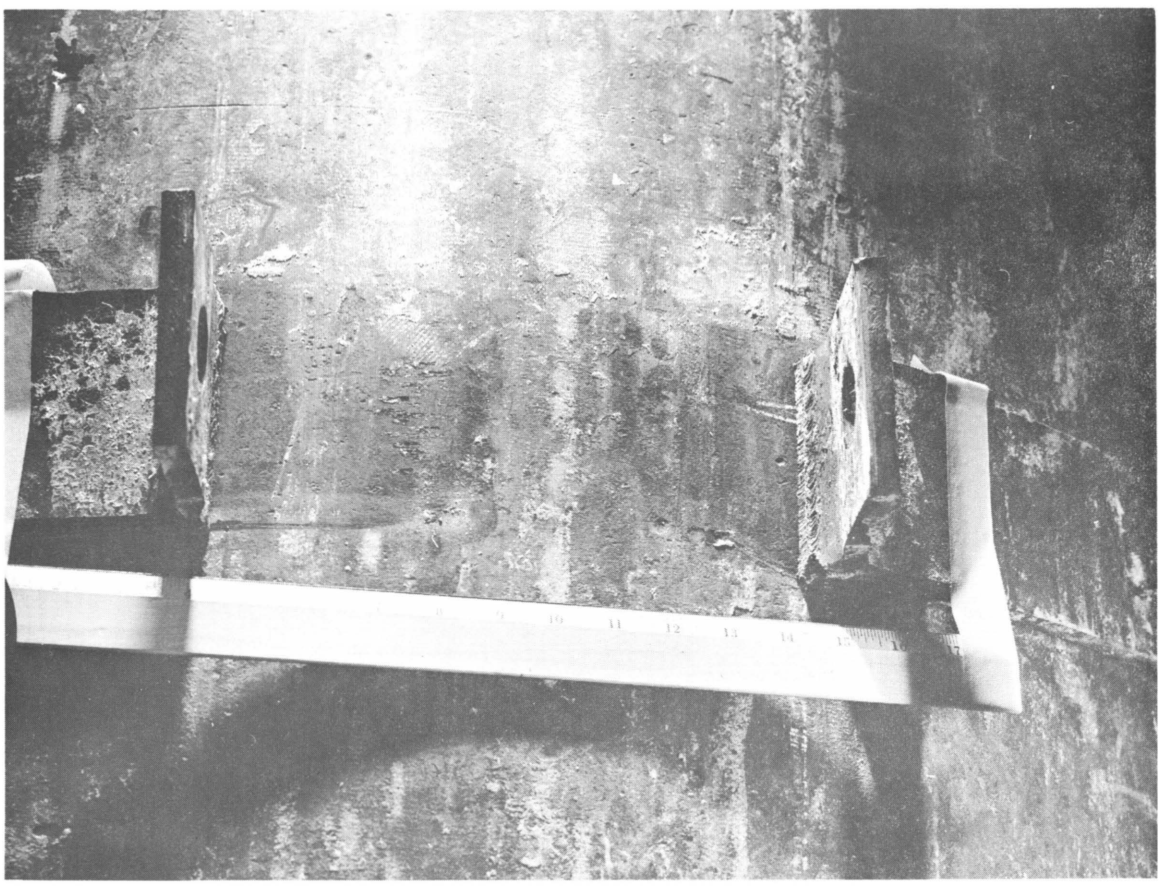


Fig. 23 Photograph showing stabilizing-band movement.

on the bolt and its relative position indicated that the bolt failure probably occurred during the initial stage of movement. The bolt failure appeared to be a brittle fracture and no elongation or "necking" of the body or threads was apparent (Figure 24).

Fig. 24 Photograph of broken bolt on stabilizing band.

The remaining insulation was removed the evening of October 27. Figure 25 shows the vessel after failure with the insulation removed. At the time the insulation was removed, it was noted that all of the steel strapping bands on the vessel insulation were broken except for one band at the bottom of the vessel. The vessel was "strapped" in five places from the top seam to the cracked weld and at the lower end. Results of this strapping are shown in Figure 26. Over-all measurements were taken with the help of a representative of Chicago Bridge and Iron. The greatest deformation was at approximately 2 ft above the central girth weld seam.

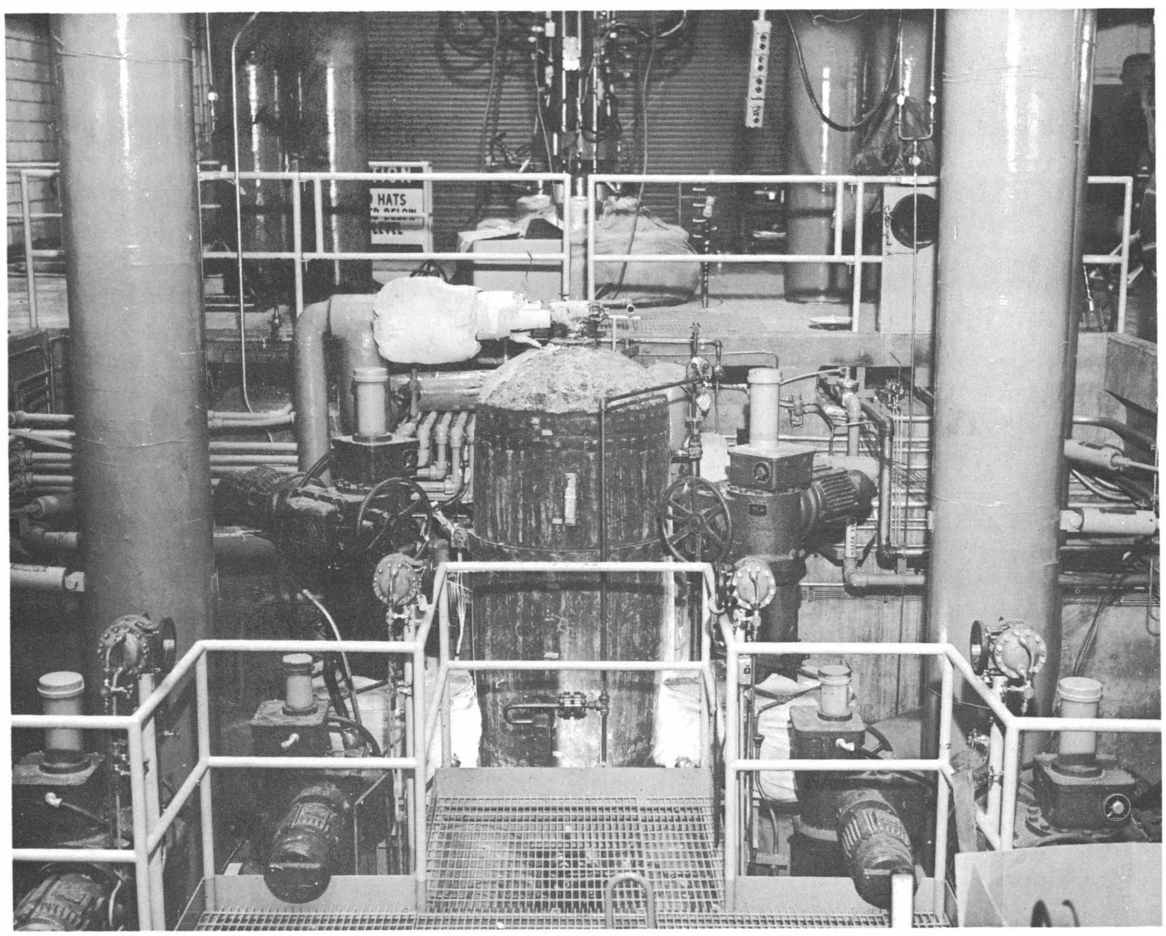


Fig. 25 Photograph of vessel with insulation removed.

A small hairline crack (Figure 27) was found on the top head at the base of the nozzle reinforcing pad. This crack was later shown by dye-penetrant inspection to be in the mill scale only. Approximately 30 vertical cracks were found in the weld metal of the central girth seam, (Figures 28, 29, 30, 31, 32). Some of these were quite wide which allowed visual inspection of the interior of the cracks. The metal in the cracks was dark gray to black and of the coarse-grain structure common in heavy welds. These cracks appeared to be typical hoop stress failures. Two small cracks were found at the base of the reinforcing pad on the upper west heater nozzle.

It was observed that the strain-gage installations on the top head, 12 in. from the bottom head and on the 4-in. top outlet piping, showed temperature colorations on both the stainless-steel foil tie straps and on the Nichrome tabs attached to the strain gages. The strain gages mounted 1 in. above the centerline of the top heater were heat-affected by the sample removal and could not be used for comparison. The coloration was much more pronounced on the parts taken from the upper head and the outlet piping near the top head than it was on the lower part of the vessel.

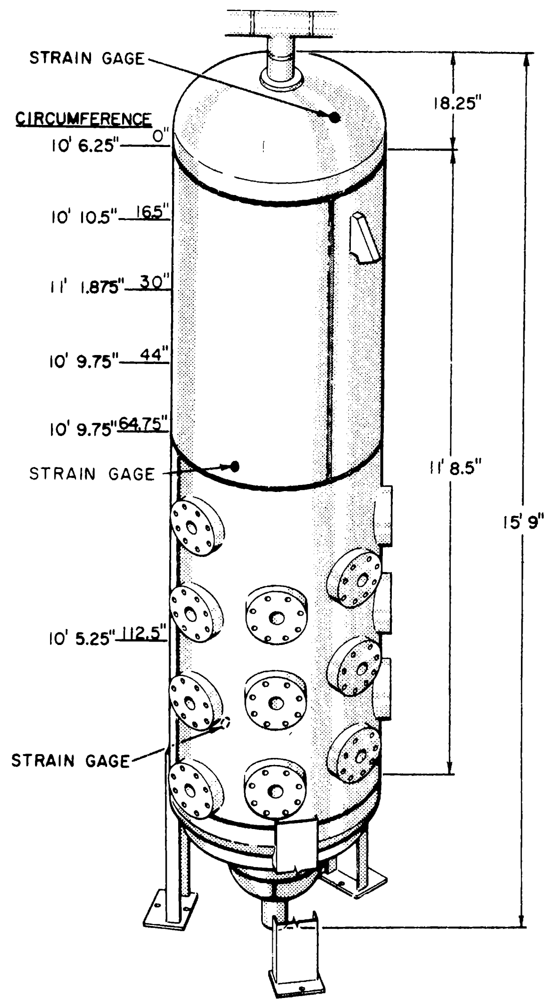


Fig. 26 Post-failure vessel measurements.

Figure 33 is a photograph of a sample removed from the vessel wall. The dark shading at the edges is cutting oil from the drilling operation. The lighter rectangular area is a surface that was ground during fabrication and is only heat-tinted and not covered with black deposit. Visual inspection of the interior of the vessel showed numerous cracks in the cladding. These cracks for the most part were oriented longitudinally with the vessel. Several large blisters were observed, indicating that the clad had become at least partially separated from the shell. The cracking was much more severe in the upper half of the vessel than in the lower half. No cracks were seen in the cladding of the heads. Much of the interior of the vessel was covered with a black deposit.

Inspection of the insulation blocks removed from the vessel showed that some bricks were discolored a light tan for approximately 2/3 the thickness of one brick. The outer layer of the two layers of bricks was a light gray and showed no discoloration.

Inspection of the heater connections showed no damage to the polyvinyl chloride insulation covering which had been installed just prior to the current

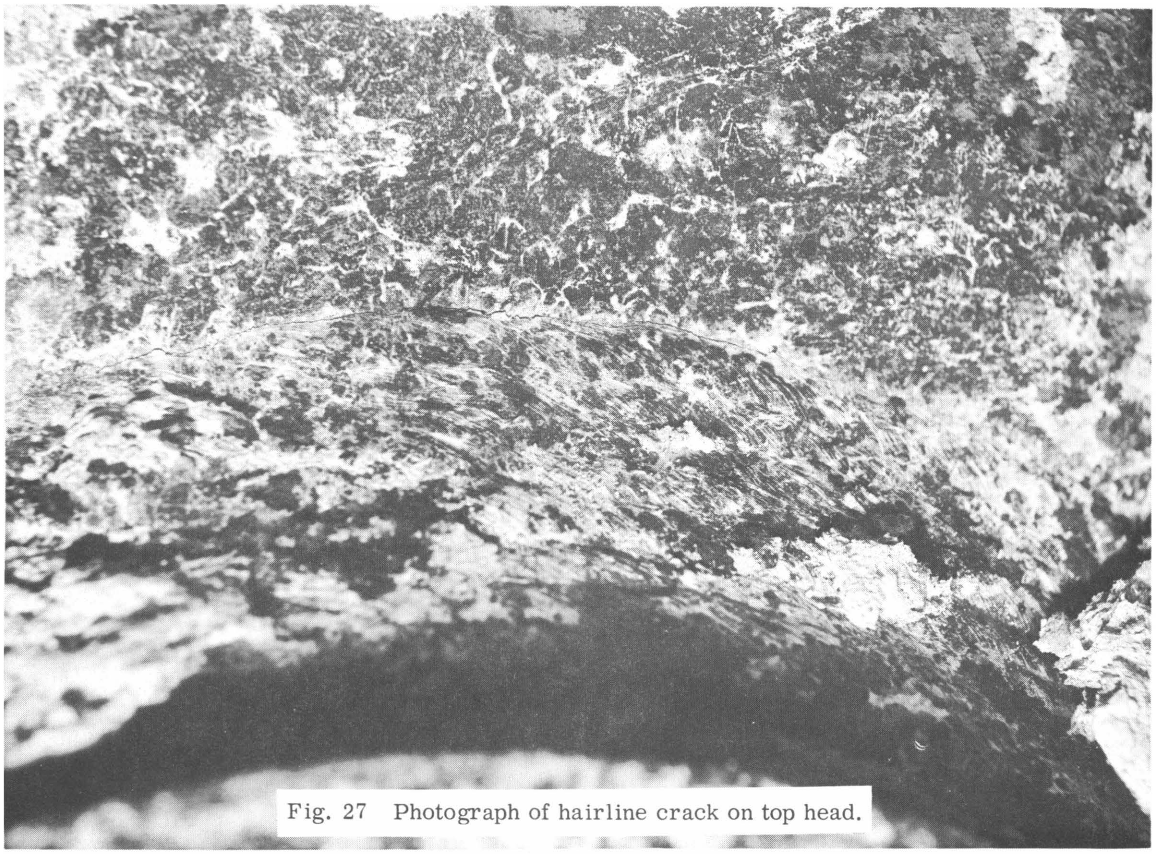


Fig. 27 Photograph of hairline crack on top head.

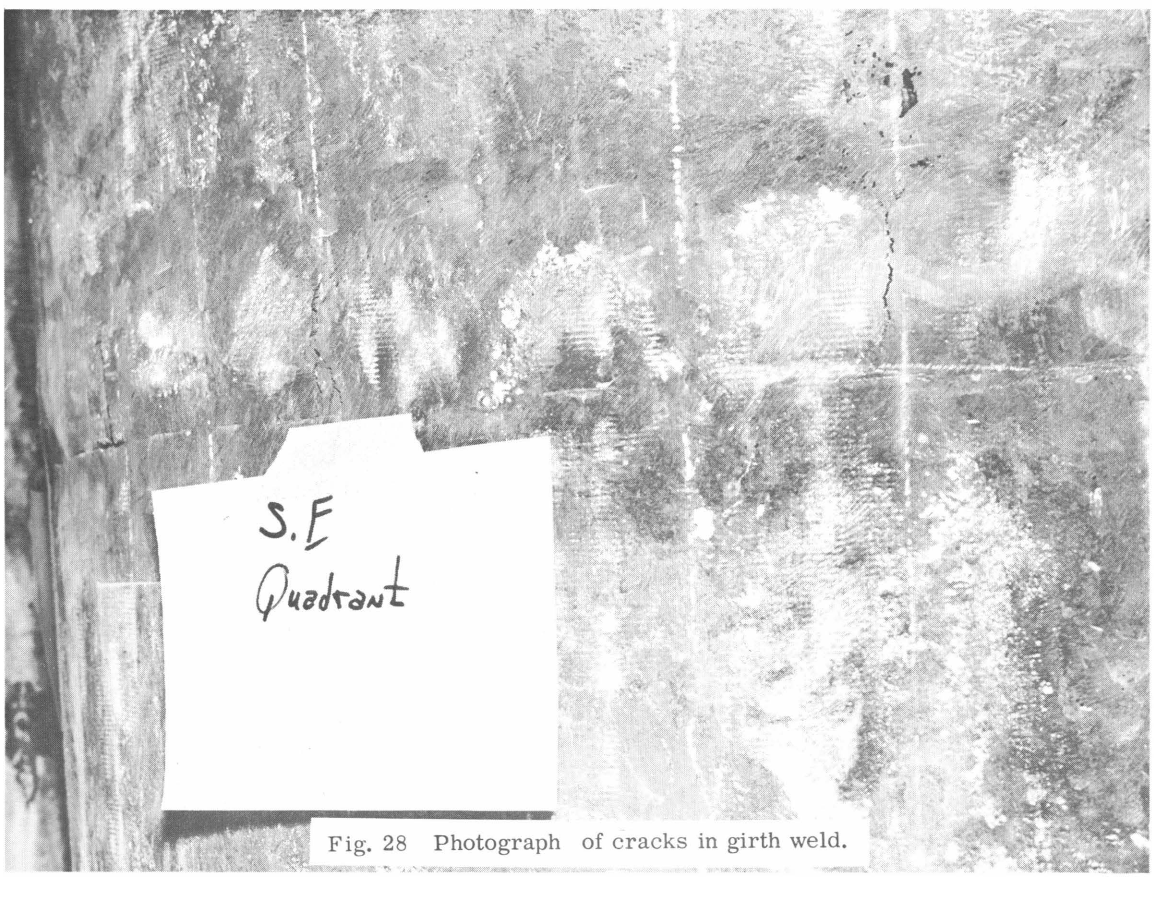


Fig. 28 Photograph of cracks in girth weld.

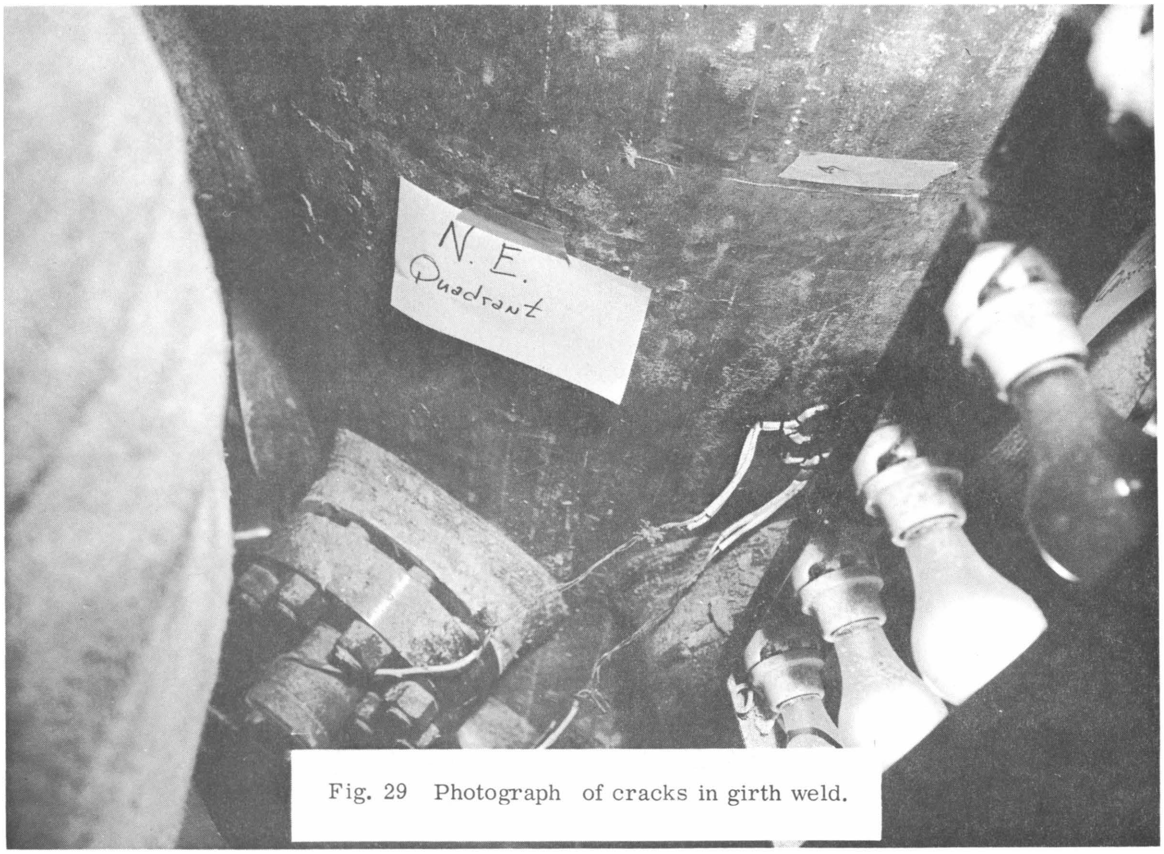


Fig. 29 Photograph of cracks in girth weld.

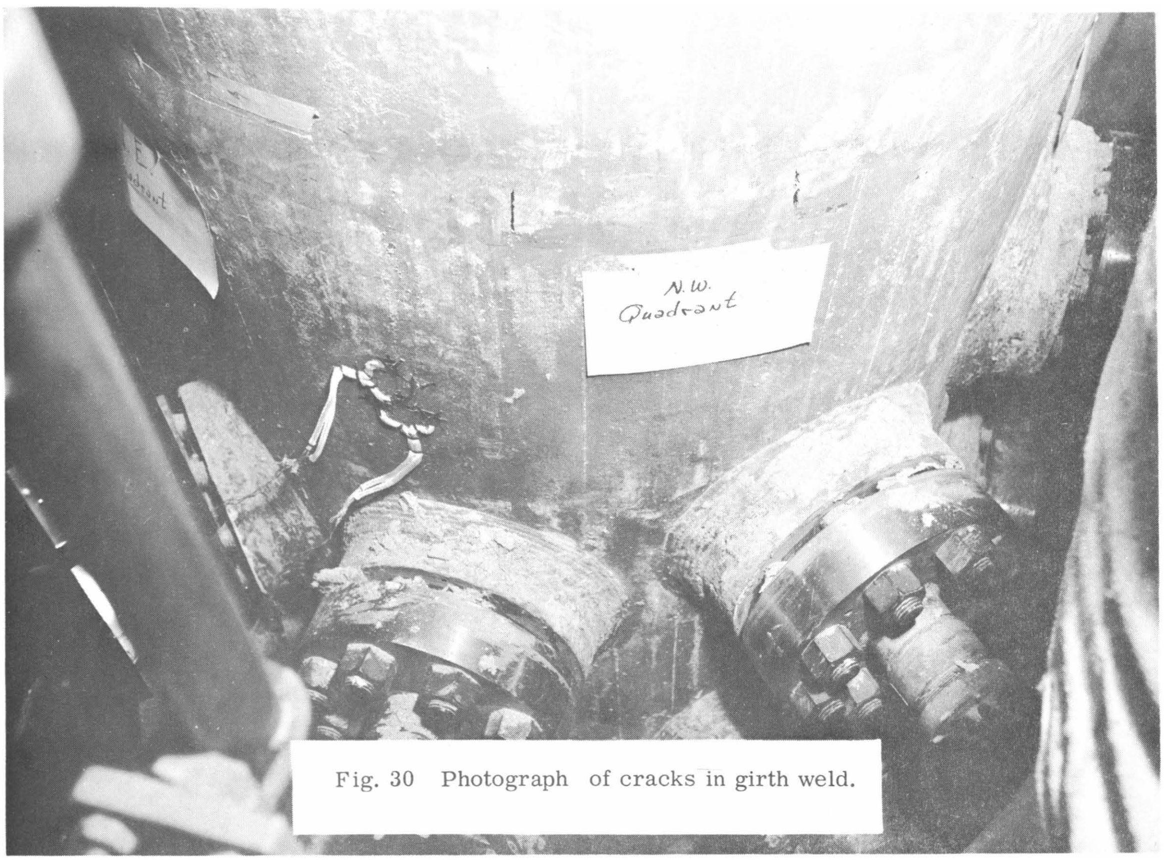


Fig. 30 Photograph of cracks in girth weld.

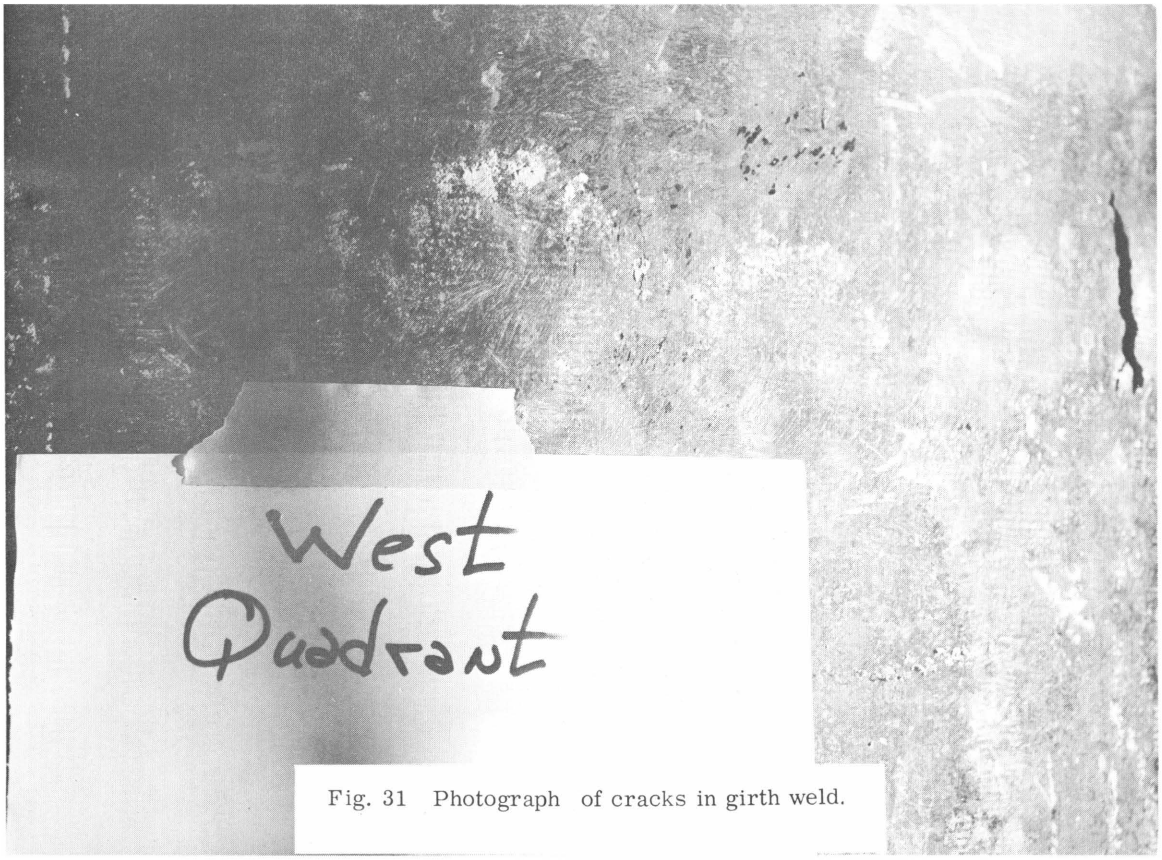


Fig. 31 Photograph of cracks in girth weld.

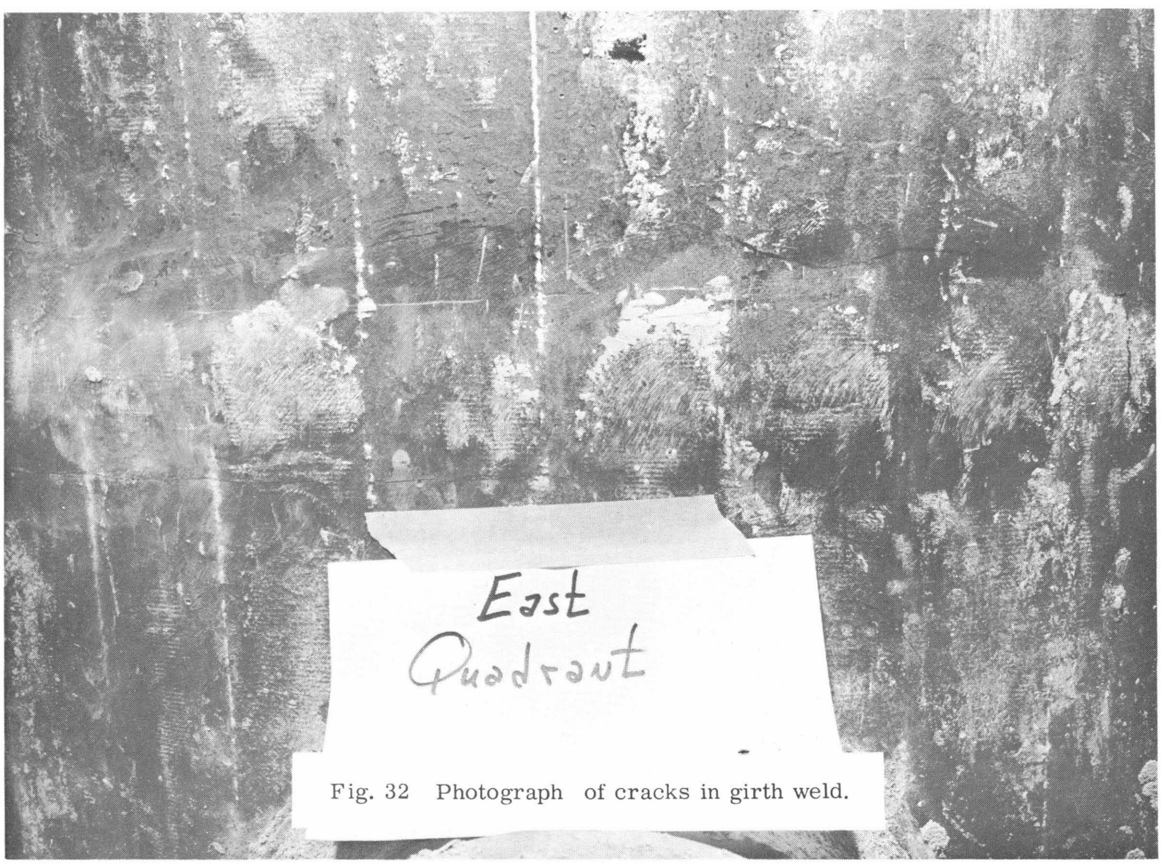


Fig. 32 Photograph of cracks in girth weld.

series of tests. This cable is approximately 8 in. from the vessel and has a maximum service rating of 105°C.

The immersion heaters showed no evidence of any unusual appearance for heaters in this service.

Inspection of the 4-in. top piping showed no obvious evidence of yielding. Micrometer measurements of the piping outside diameter ranged from 4.505 to 4.514 in. which is within the range expected of new pipe. When the piping was cut from the vessel in preparation for vessel removal, little or no movement of the pipe ends was observed.

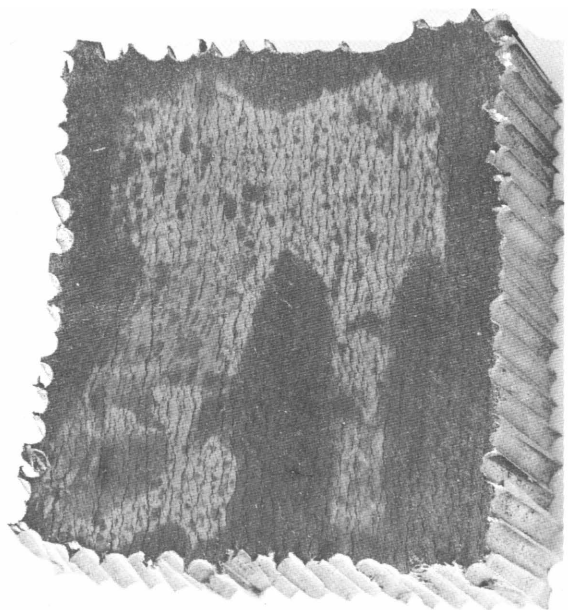


Fig. 33 Photograph of sample removed from vessel wall.

2. POST-FAILURE TESTS PERFORMED

Following the failure, tests were performed on various system components that might have been affected by unusually high pressure or temperature, or which might have affected the validity of the pressure and temperature records. The tests performed are listed below:

2.1 Tests of Pressure Relief Valves

The welded Kieley-Mueller relief valve on the pressurizer has a nameplate rating of 2750 to 3000 psi. This valve, which had been inadvertently disassembled during removal from the primary system, was reassembled without disturbing the setting adjustment. Using water and a Sprague booster pump, a pressure of 4200 psi was required to pop the valve. On each of four successive tests the valve relieved at 3720 psi. (The manufacturer stated that the maximum set pressure could be increased only slightly above the 3000-psi setting.)

The welded relief valve of the same make on the reactor vessel was tested in a like manner and showed a repeatable set point of 4860 $\pm$ 60 psig. This valve also had a nameplate range of 2750 to 3000 psi.

Both valves also were tested to determine the maximum setting. The valve adjustment was screwed to the stop and neither would relieve at 6200 psig, the maximum pressure obtainable with the test equipment. As noted previously in Section IV, these relief valves apparently were adjusted during final stages of construction to permit hydrostatic testing at 3750 psig, and apparently were not reset and tested.

The make-up pump relief valve relieved at 3000 psig. Its nameplate setting was 2740 psig.

2.2 Calibration of Pressure Instrumentation

The calibration of all plant pressure instrumentation was checked. The complete instrument loops, transmitters, compensating units, indicators or recorders and associated wiring were left intact. These loops were indicating or recording exactly as used during plant operation with the exception of the primary medium. The primary medium was replaced by the use of a dead-weight tester, or a mercury column where lower differential pressures were required. It was possible in all cases to apply the calibrating medium without disturbing the instrument systems. The dead-weight tester also was calibrated against a Heise calibration gage and a second dead-weight tester. Results of these tests show that following the failure of the vessel all plant pressure instrumentation was in calibration to better than 2% of full scale reading.

The case on the pressurizer liquid-level differential pressure cell was hydrostatically tested since this case is in the pressure environment of the pressurizer vessel at all times. The case has a nameplate rating of 2500 psig. The case started to leak at 2800 psig and leaked so badly at 3200 psig that the booster pump capacity could not raise the pressure any further. Thus, this instrument case would have failed in the event of any serious overpressure in the pressurizer vessel.

The actuation of the blow-off valve (PCV-5R) was checked during the pressure instrument check. This valve opened at 2750 psi as specified.

Although not a pressure instrument as such, the pressurizer liquid-level device, which consists of a differential pressure cell with a temperature-compensating bridge, was checked for calibration on October 30. The temperature compensation was checked by applying a known voltage to simulate the resistance bulb output at design temperature. Results of this test showed the instrument to be recording high by 2.4 in. at the normal level of 7 ft, 6 in.

2.3 Temperature Tests

In addition to the visual examination for overheating of components near the pressurizer, the following tests were made:

2.31 The pressurizer insulation covering was subjected to a scorching test to determine the temperature at which the observed smoke could have been formed. A piece of the cloth placed in a controlled-temperature furnace scorched badly at 600°F and essentially disintegrated at about 650°F. A photograph of the samples is shown in Figure 34.

2.32 In order to determine the visible effects, if any, to the insulation used on the pressurizer vessel, a brick of the insulation was placed in the controlled-temperature furnace and held at 1000°F for 1 hr. The brick turned from gray to a light tan. Comparison with other insulation that had been removed from the course next to the vessel showed the tan conversion for approximately 1 in. through the brick. Tests at 650°F in the furnace showed no "tanning". These tests were not conclusive because of the difference in time exposure and method of heating but did suggest the presence of temperatures in excess of the design temperature of 670°F on some portion of the vessel.

2.33 The strain-gage rosettes installed on the vessel showed heat color markings on the Nichrome tabs and on the stainless-steel foils which were used

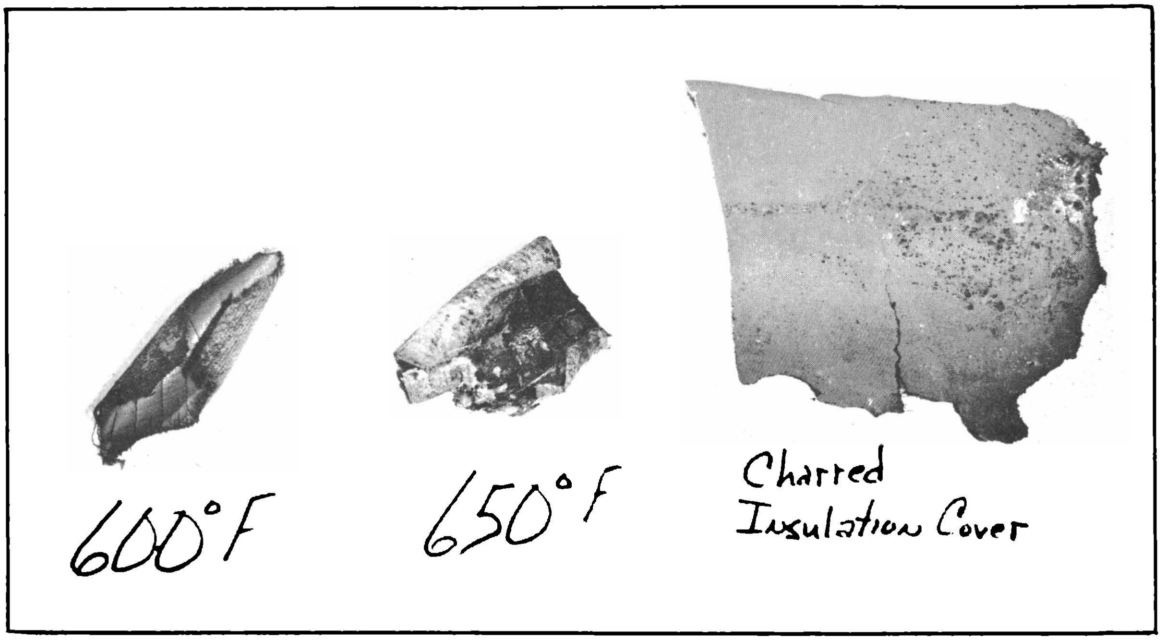


Fig. 34 Photograph of scorching test on insulation covering.

to fasten the leads to the vessel. A typical strain-gage installation is shown in Figure 35. The material used for these parts was still available. Strips were cut of each and a colorimetric chart was made by heating the samples in the controlled-temperature furnace. A photograph of this chart is shown in Figure 36.

Results of the comparison of the Nichrome tabs and stainless-steel foil strips removed from the vessel are shown in Table V.

Again, these are comparison temperatures only and do not necessarily reflect true temperatures. However, all parts in the same location showed essentially identical discoloration.

2.34 Tests were conducted on an electrical immersion heater removed from the pressurizer vessel for the purpose of estimating the life of the heater when exposed to air or steam and the heater temperature. The heater was operated for 4-1/2 hr in air without failure. The surface temperature was estimated on the basis of color to be 1400 to 1500°F. The tests indicate that heaters could have been exposed in the pressurizer vessel for a considerable period of time without failing and temperatures approaching 1200°F may have been attained.

2.4 Measurements on Piping

Dimensional measurements have been performed on the 4-in. nominal-diameter pipe connecting the bottom of the pressurizer vessel to the reactor primary coolant system. The pipe diameter measured 4.505 to 4.515 in., as compared to the theoretical diameter of 4.500 in. These measurements indicate that no significant yielding of the 4-in. piping has occurred.

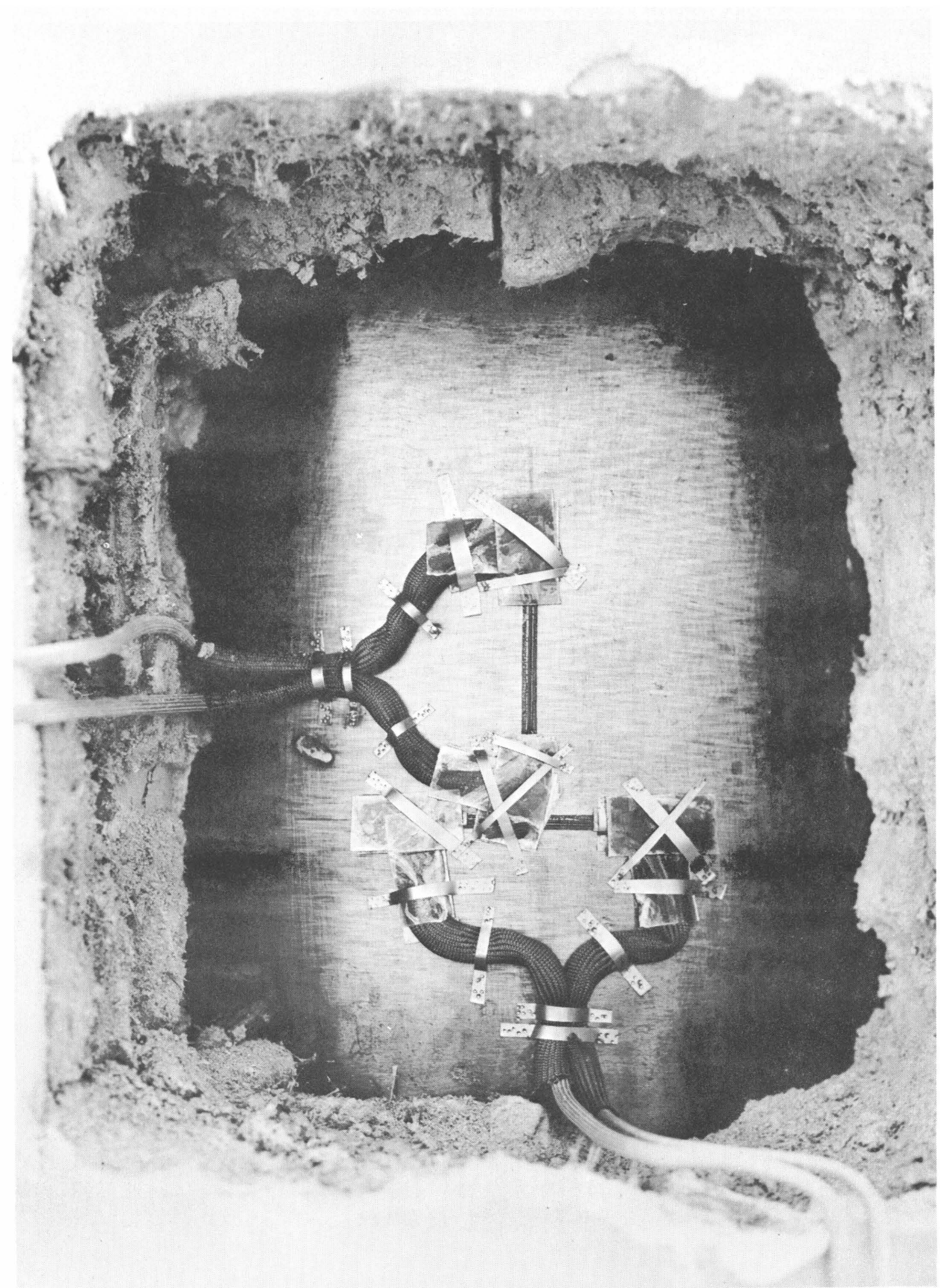


Fig. 35 Photograph of typical strain-gage installation.

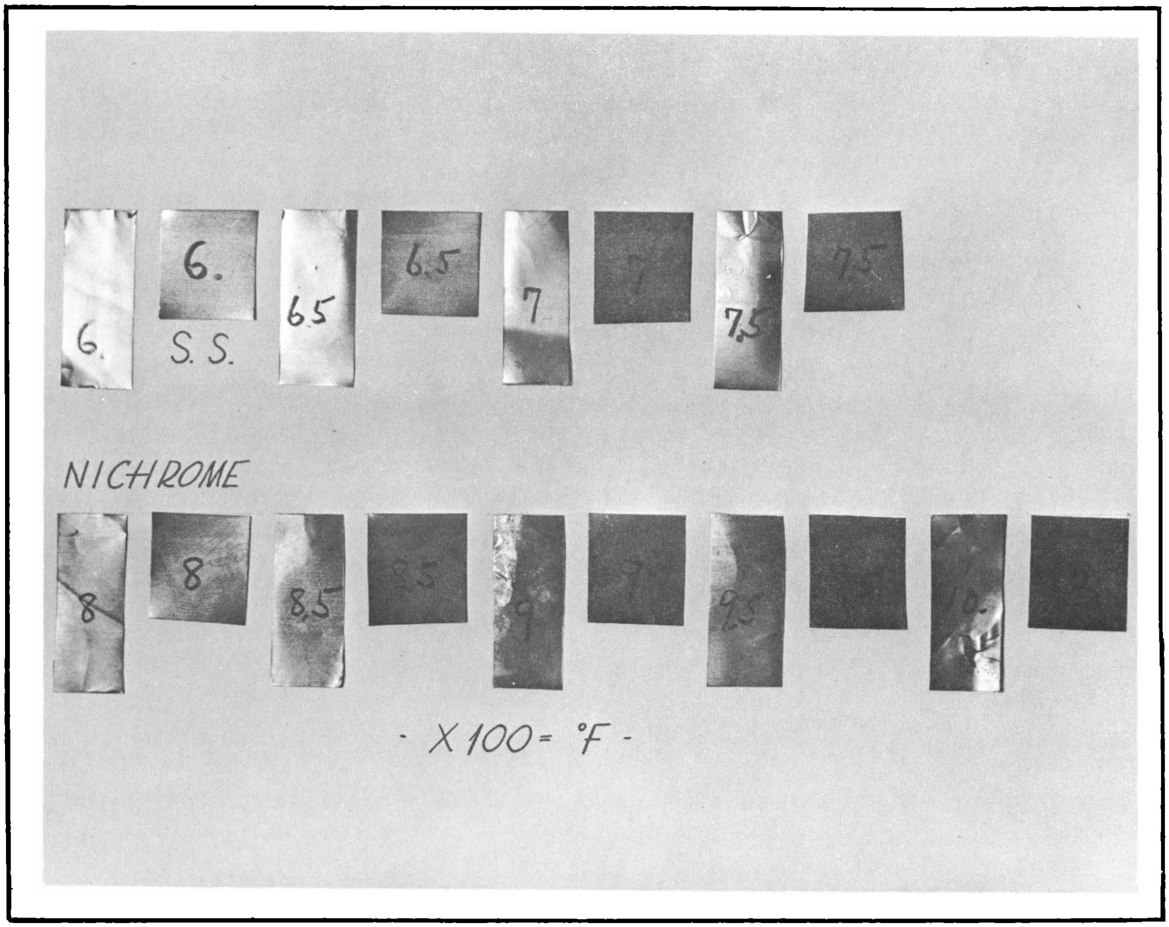


Fig. 36 Photograph of colorimetric chart.

TABLE V

COLORIMETRIC TEMPERATURE COMPARISON

Location on Vessel	Material	Temperature (°F)
Top head	Nichrome	900 - 1000
Top head	Stainless-steel foil	~ 850
4-in. top outlet piping	Nichrome	~ 800 - 850
Near top of vessel	Stainless-steel foil	~ 750
12 in. from bottom	Nichrome	< 650
Seam of vessel	Stainless steel	< 600

3. METALLURGICAL INSPECTIONS

3.1 General

In an effort to determine the post-failure condition of the materials of construction of the vessel, metallurgical examinations of the vessel shell material, the welds and cladding were made by Phillips Petroleum's Atomic Energy Division metallurgists. Concurrent investigations were made by Mr. F. Prange, metallurgist in Phillips Test Division at Bartlesville, Okla., and Mr. M. E. Holmberg, consulting metallurgist. The Chicago Bridge and Iron Co., the vessel fabricator, also is making an investigation. However, to date, no report has been furnished to Phillips. Mr. Holmberg's report and excerpts from the inspection report of Mr. Prange are contained in Appendix C.

The general results of all the metallurgical examinations showed no indication of defective material that would in itself have been a primary cause of the vessel failure. The inspection of the shell cladding did show that the fabrication and/or the materials used in cladding were not suitable for this service.

3.2 Tensile Tests

The initial sample removed from the vessel for evaluation was a 10-in. x 10-in.-square section. It was removed from the yielded portion of the vessel by drilling 1/2-inch-diameter holes in line and severing the connecting segments with a sabre saw. This was done to eliminate any heat effects, by the removal procedure, upon this first sample. Location of this first sample is shown in Figure 37. Subsequent samples were torch-cut using an iron powder oxygen-acetylene cutting torch. These samples were taken from the locations shown in Figure 38. To obtain tensile test specimens, the first sample was sectioned as shown in Figure 39. Standard 1/2-inch-diameter tensile coupons were machined out and pulled at temperatures ranging from room temperature to 800°F. Physical test data for these specimens are shown in Table VI. Additional tensile specimens were pulled on material taken from the top head, shell material removed from the sample labeled "weld sample", weld material from the same sample and the shell material of the bottom course from the sample labeled "bottom sample". These specimens were pulled at temperatures ranging from room temperature to 1000°F at standard pulling rates. Physical test data for these tests are shown in Tables VII and VIII. Figure 40 is a plot of the yield and ultimate strength, as affected by temperature, of the shell material from the upper yielded half of the vessel. Figure 41 is a similar plot for the shell material from the lower half of the vessel, and Figure 42 is the plot for the head material. Original room temperature data of the tensile properties supplied by the vessel fabricator are shown in Figures 43 and 44. As can be readily seen, the room temperature strength of the material removed from the vessel top shell and head is significantly lower than the test results reported by the manufacturer. This possibly can be explained by the fact that the manufacturer's test coupons for the shell material were pulled prior to hot-forming of the vessel and the subsequent stress relief at 1150°F. Thus the test coupons do not necessarily indicate the true condition of the vessel material in its finished form. Annealing of the vessel and top head material in service may account partially for the apparent low room-temperature yield strength. This would be especially true for the material from the upper half of the vessel where it yielded, because the service temperatures

Fig. 37 Photograph of location of first metallurgical sample.

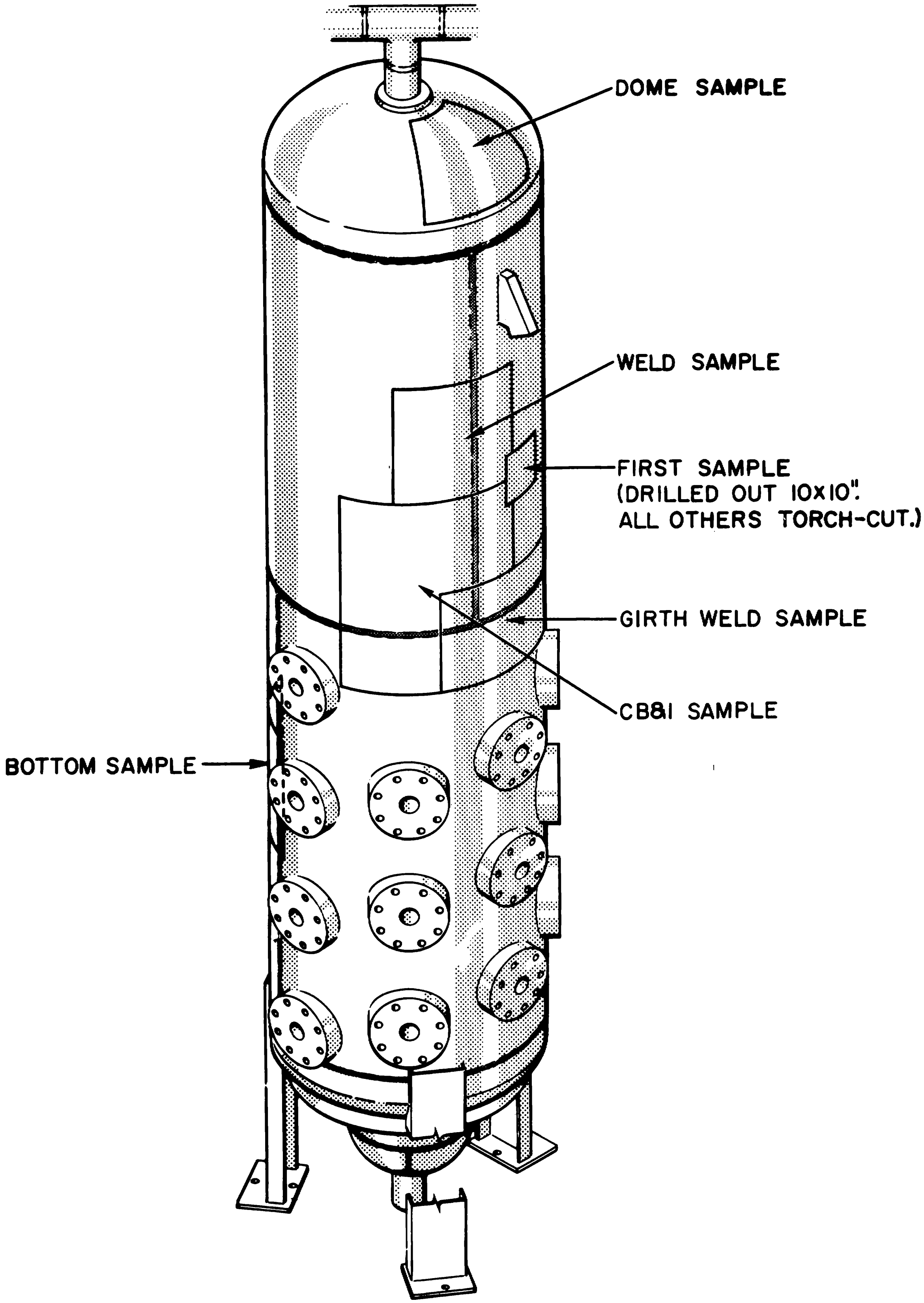


Fig. 38 Location of metallurgical samples.

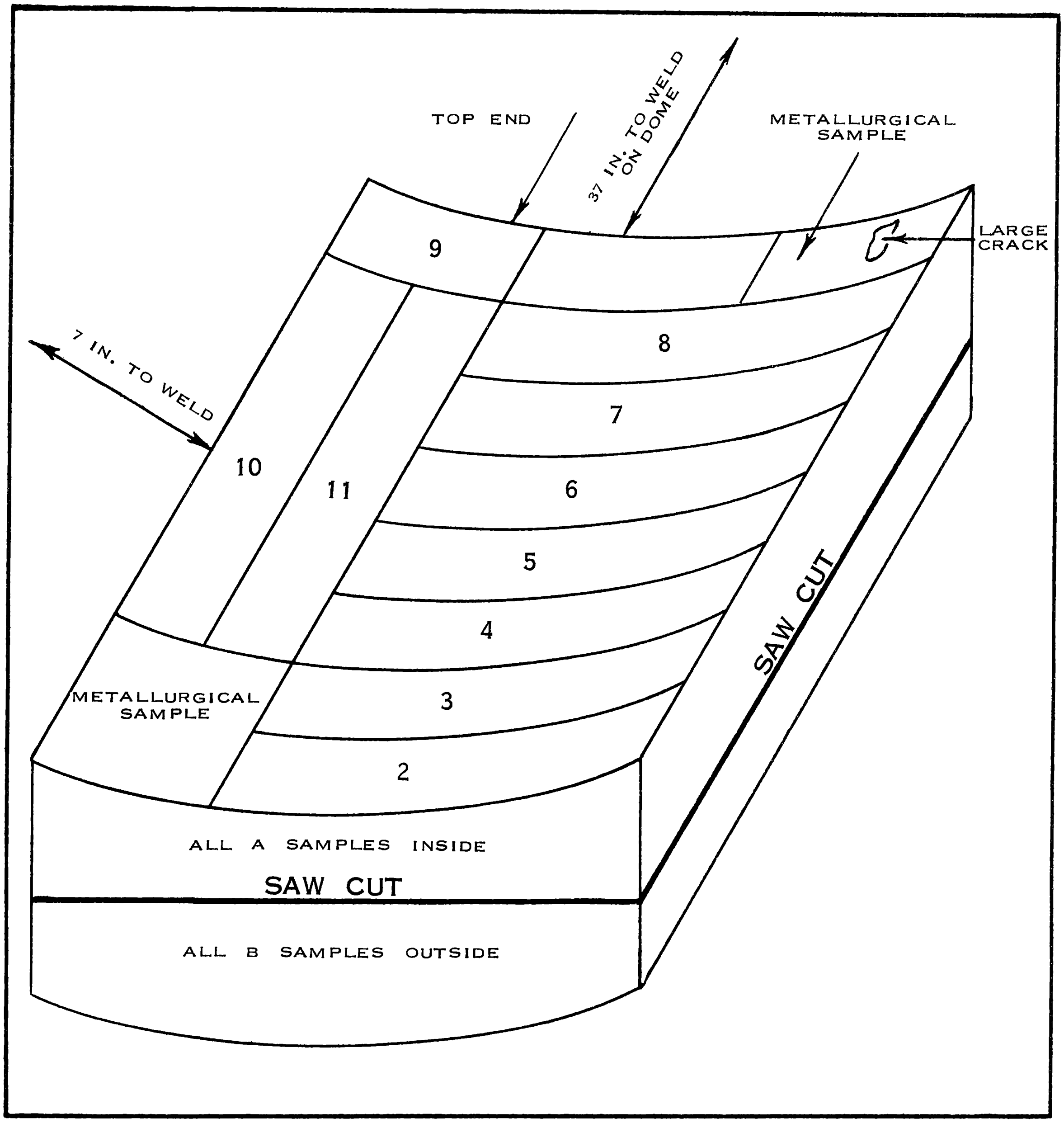


Fig. 39 Method of sectioning first metallurgical sample.

were higher there. The test results show that the material from the head and the lower parts of the pressurizer were within the minimum specifications of the pressurizer design. The plate material from the upper part of the pressurizer did not meet design specifications even after yielding about 10%. At room temperature this would have increased the yield point about 5000 to 7000 psi. However, this part of the pressurizer was being annealed by service conditions in the temperature range of 800° to 1000°F. The stress-strain curve of all the room temperature specimens from the first sample still shows the characteristic dip exhibited by all annealed mild steels. This dip is removed by work hardening. The annealing makes it impossible to fix a numerical value to the original yield strength of this group of specimens. The lower course of the vessel did appear somewhat stronger than the upper

PHYSICAL TEST DATA for

TABLE VI

C2718 ALL SPECIMENS FROM FIRST SAMPLE

DATE 21 November 1961

SAMPLE NO.	SIZE	AREA	LOAD AT AT BREAK	YIELD STRENGTH P.S.I.	TENSION AT BREAK LBS.	TENSILE STRENGTH P.S.I.	ELONGATION ~~INCHES IN IN.~~ %	Rd. ~~%~~	HARDNESS HEAT OR MELT	BRINELL	ROCK	RA	REMARKS
Room Temp.	501	.197	6660	33,810	13,500	68,500	34.5	.328	3A			57.1%	
" "	499	.196	6840	34,900	13,500	68,900	30.0	.327	4A			57.0%	
" "	501	.197	6810	34,600	13,650	69,300	35.0	.346	11A			52.4%	
" "	500	.196	6900	35,200	12,900	65,900	34.5	.318	4B			59.5%	
" "	502	.198	6660	33,610	13,600	68,700	32.5	.348	10B			51.9%	
400°F.	501	.197	NO CURVE		12,040	61,120	28.5	.325	8A			57.6%	
"	501	.197	7040	35,780	11,820	60,000	25.5	.346	7B			52.4%	
600°F.	502	.198	8080	40,750	13,120	66,400	29.0	.365	9A			46.9%	
"	500	.196	6480	33,100	13,180	67,200	26.5	.358	3B			48.6%	
"	498	.195	7040	36,100	12,380	63,480	28.5	.344	9B			52.3%	
668°F.	500	.196	6400	32,620	12,820	65,400	31.0	.347	2A			51.6%	
"	501	.197	6400	32,500	12,580	63,800	31.0	.327	6A			57.1%	
"	501	.197	6380	32,400	12,820	65,080	28.5	.348	10A			51.8%	
"	501	.197	6240	31,620	12,400	62,950	29.0	.335	8B			55.0%	
"	501	.197	6720	34,100	12,700	64,460	27.0	.349	11B			51.2%	
700°F.	502	.198	6760	34,130	12,260	61,900	30.0	.316	7A			60.4%	
"	501	.197	6240	31,700	12,260	62,200	29.0	.320	5B			59.1%	
"	501	.197	6280	31,900	12,100	61,400	31.0	.329	6B			56.6%	
800°F.	501	.197	5960	30,220	9,620	48,800	26.5	.295	5A			65.5%	
"	501	.197	5680	28,800	10,430	53,000	30.5	.293	2B			65.0%	

OPERATOR

INSPECTOR

PHYSICAL TEST DATA
C2718 for
PRESSURIZER SAMPLES

TABLE VII

DATE 6 December 1961

SAMPLE NO.	SIZE	AREA	LOAD AT AT BREAK	YIELD STRENGTH P.S.I.	TENSION AT BREAK LBS.	TENSILE STRENGTH P.S.I.	ELONGATION INCHES IN-IN.	ELONGATION %	Final Dia. (~~HEAT OR MELT~~)	Percent Reduction in Area (~~HARDNESS~~ ~~BRINELL~~)	~~ROCK~~	REMARKS
A-1	.501	.197	8400	42,600	14,100	71,500		32.	.324	58.1	%	Room Temperature
A-7	.502	.198	7600	38,400	14,850	75,000		29.	.359	48.85	%	Room Temperature
B-1	.502	.198	7200	36,350	14,000	70,700		35.	.318	59.9	%	Room Temperature
B-7	.501	.197	7480	37,940	14,400	73,100		29.	.355	49.75	%	Room Temperature
A-2	.501	.197	7190	36,450	15,560	78,900		27.	.375	44.2	%	600°F.(lower-plate bottom)
A-8	.501	.197	7200	36,590	13,580	68,600		24.	.382	41.9	%	600°F.(dome)
B-2	.502	.198	7200	36,390	14,330	72,475		21.5	.370	45.6	%	600°F.(lower-plate bottom)
B-8	.501	.197	6800	34,600	13,140	66,750		25.5	.375	43.6	%	600°F.(dome)
W-1	.500	.196	9400	48,000	13,420	68,500		18.0	.415	30.8	%	700°F.(weld outside)
W-4	.500	.196	9800	50,000	13,740	70,000		22.0	.371	45.1	%	700°F.(weld inside)
A-3	.500	.196	6800	34,700	11,660	59,500		29.0	.295	65.	%	800°F.(lower-plate bottom)
A-9	.499	.195	6400	32,800	11,510	59,100		26.0	.321	59.0	%	800°F.(dome)
B-3	.501	.197	6600	33,500	11,025	56,100		31.0	.283	68.1	%	800°F.(lower-plate bottom)
B-9	.501	.197	6200	31,450	11,510	58,500		28.0	.320	59.1	%	800°F.(dome)
A-4	.501	.197	6100	30,990	9325	47,300		34.5	.230	79.8	%	900°F.(lower-plate bottom)
A-10	.502	.198	5900	39,250	9370	47,250		27.0	.328	57.4	%	900°F.(dome)
B-4	.502	.198	6200	31,300	8715	44,000		33.0	.215	81.6	%	900°F.(lower-plate bottom)
B-10	.502	.198	6160	31,100	9020	46,000		25.0	.318	59.9	%	900°F.(dome)
W-2	.500	.196	8800	44,900	11,350	58,000		16.0	.398	36.6	%	900°F.(weld outside)

OPERATOR

INSPECTOR R. E. Nace Met.

R. E. Nace M.E.

PHYSICAL TEST DATA for
C2718 WELD SAMPLES MARKED "W"
(1 & 2 are outside — 3 & 4 are inside)

TABLE VIII

DATE 6 December 1961

SAMPLE NO.	SIZE	AREA	LOAD AT AT BREAK	YIELD STRENGTH P.S.I.	TENSION AT BREAK LBS.	TENSILE STRENGTH P.S.I.	ELONGATION INCHES IN-IN.	ELONGATION %	Final Dia. (~~HARDNESS~~ ~~HEAT OR MELT~~)	Percent Reduction in Area (~~BRINELL~~)	~~ROCK~~	REMARKS
A-16	.501	.197	5520	28,000	8260	41,800		26.5	.260	72.6 %		900°F. (from weld sample)
W-3	.500	.196	9140	46,600	11,540	59,100		20.0	.312	61.0 %		900°F. (weld inside)
B-17	.501	.197	5520	28,000	8180	41,500		28.5	.263	72.5 %		900°F. (from weld sample)
A-5	.501	.197	5420	27,500	6760	34,300		33.	.189	85.9 %		1000°F. (lower-plate bottom)
A-11	.501	.197	5240	26,590	7520	38,200		23.	.353	50.3 %		1000°F. (dome)
B-5	.501	.197	5440	27,600	6740	34,200		34.	.192	85.4 %		1000°F. (lower-plate bottom)
B-11	.501	.197	5040	27,400	6900	35,000		27.5	.337	54.7 %		1000°F. (dome)
B-16	.501	.197	4880	24,750	6540	33,200		47.5	.240	77. %		1000°F. (from weld sample)
A-17	.501	.197	5570	28,300	8370	42,500		29.0	.255	74.6 %		900°F. (from weld sample)
A-18	.500	.196	5080	25,900	6850	34,950		44.5	.234	78.0 %		1000°F. (from weld sample)

OPERATOR

INSPECTOR R. E. Nace

R. E. Nace

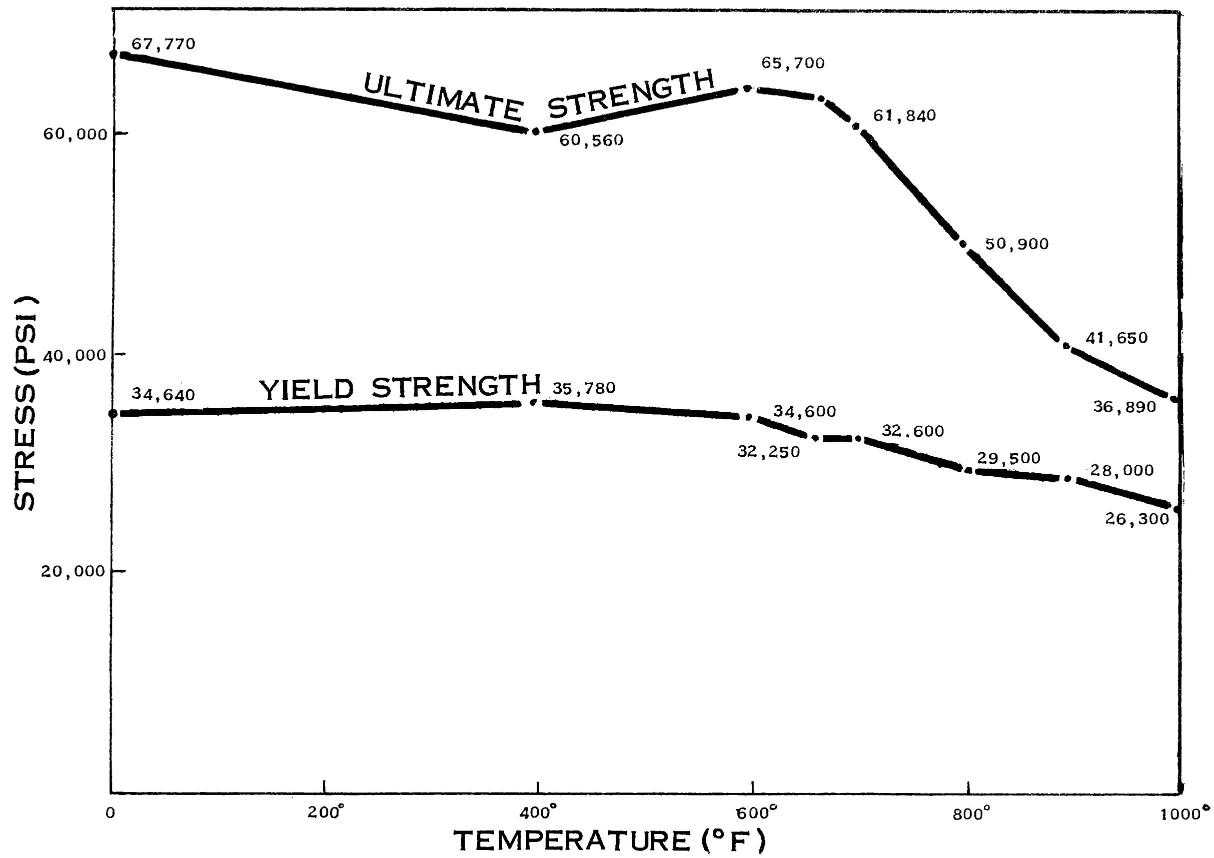


Fig. 40 Plot of yield and ultimate strength of upper half material.

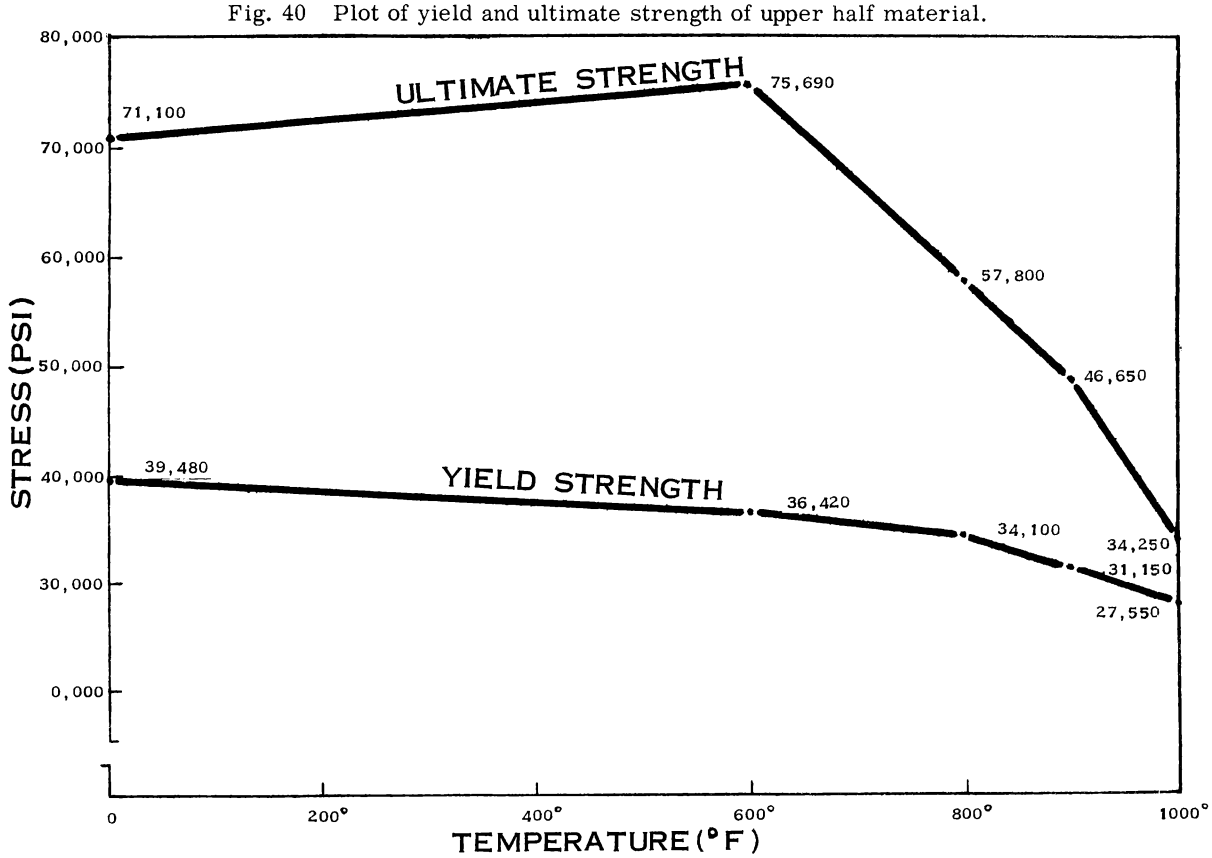


Fig. 41 Plot of yield and ultimate strength of lower half material.

SEE ERRATA AT END OF ITEM

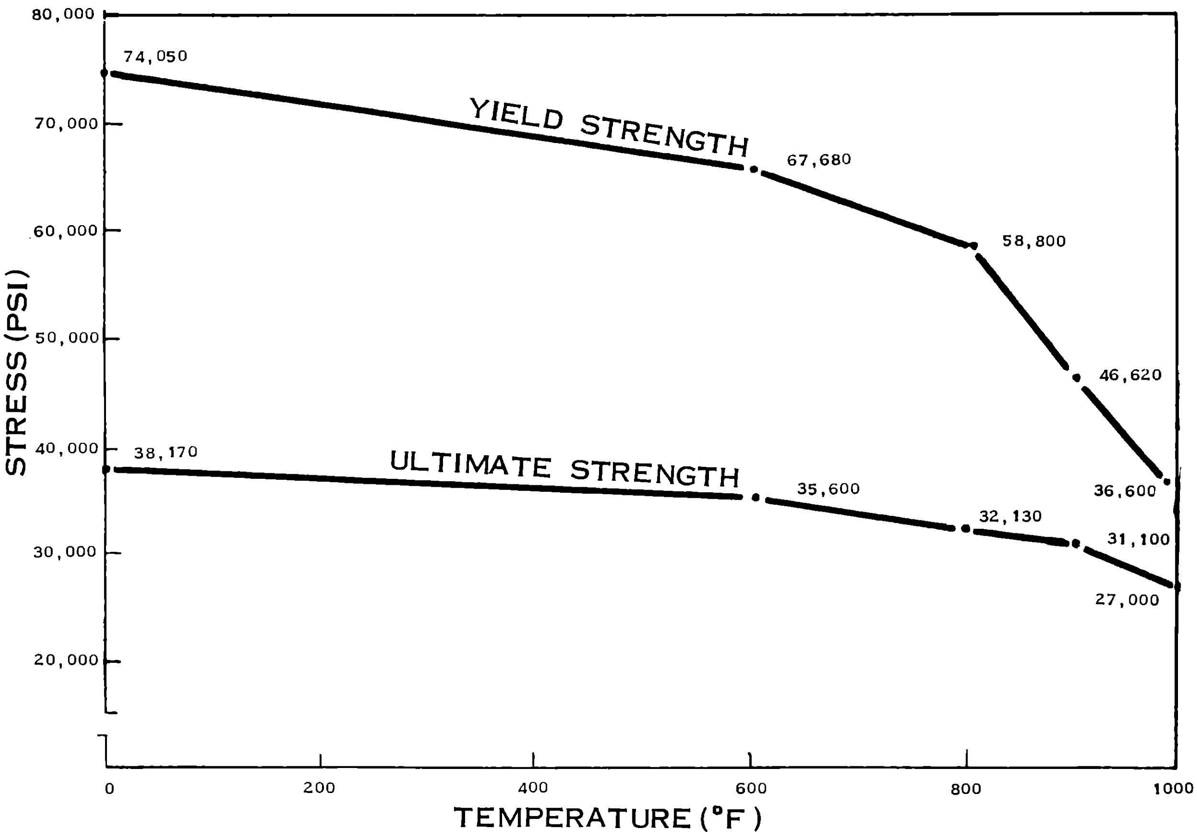
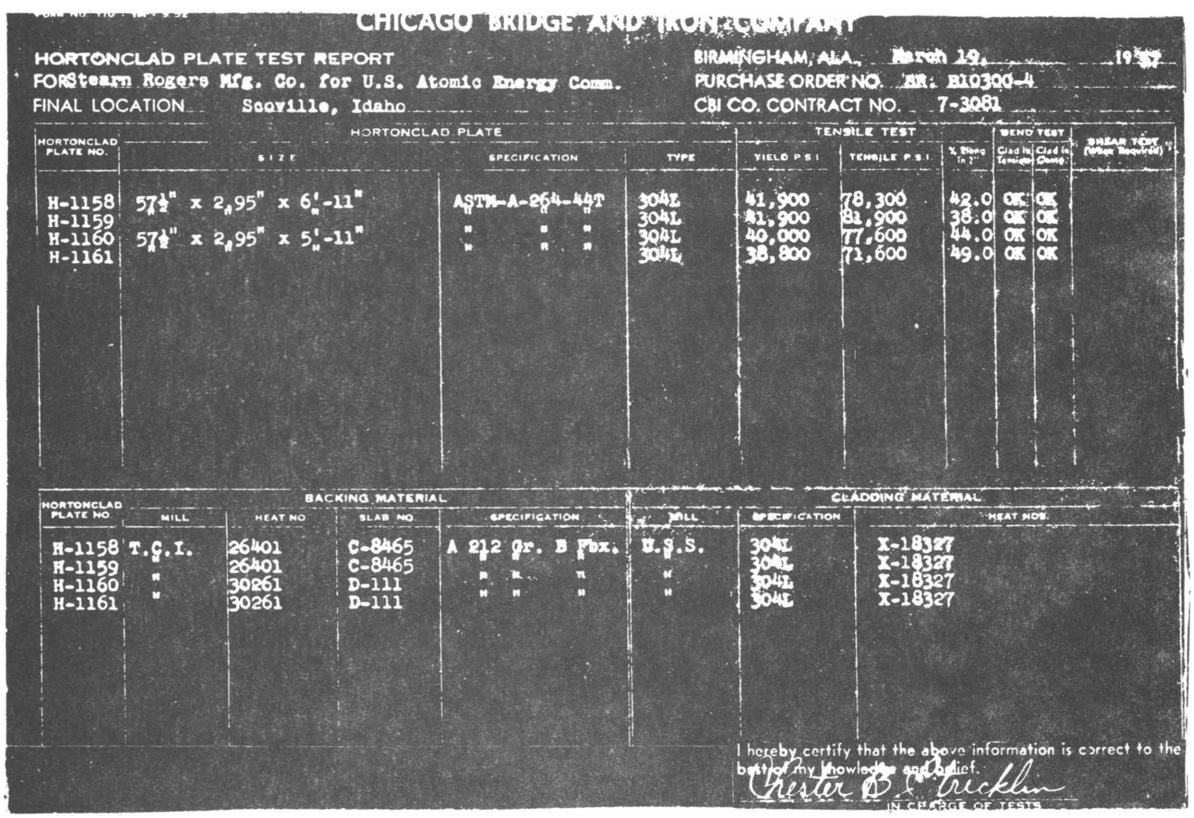


Fig. 42 Plot of yield and ultimate strength of head material.



CHICAGO BRIDGE AND IRON COMPANY

HORTONCLAD PLATE TEST REPORT — BIRMINGHAM, ALA. March 19, 1957
FOR Stearn Rogers Mfg. Co. for U.S. Atomic Energy Comm. — PURCHASE ORDER NO. BR: B10300-4
FINAL LOCATION Scoville, Idaho — CBI CO. CONTRACT NO. 7-3081

HORTONCLAD PLATE NO.	HORTONCLAD PLATE SIZE	SPECIFICATION	TYPE	TENSILE TEST YIELD P.S.I.	TENSILE P.S.I.	% Elong in 2"	BEND TEST Clad in Tension	Clad in Comp.	SHEAR TEST (When Required)
H-1158	57½" x 2,95" x 6'-11"	ASTM-A-264-44T	304L	41,900	78,300	42.0	OK	OK	
H-1159	" " "	" " "	304L	41,900	81,900	38.0	OK	OK	
H-1160	57½" x 2,95" x 5'-11"	" " "	304L	40,000	77,600	44.0	OK	OK	
H-1161	" " "	" " "	304L	38,800	71,600	49.0	OK	OK	

HORTONCLAD PLATE NO.	BACKING MATERIAL MILL	HEAT NO.	SLAB NO.	SPECIFICATION	CLADDING MATERIAL MILL	SPECIFICATION	HEAT NOS.
H-1158	T.C.I.	26401	C-8465	A 212 Gr. B Fbx.	U.S.S.	304L	X-18327
H-1159	"	26401	C-8465	" " "	"	304L	X-18327
H-1160	"	30261	D-111	" " "	"	304L	X-18327
H-1161	"	30261	D-111	" " "	"	304L	X-18327

I hereby certify that the above information is correct to the best of my knowledge and belief.
Chester B. Stricklin
IN CHARGE OF TESTS

Fig. 43 Room temperature shell material data supplied by fabricator.

31257 65

LUKENS STEEL COMPANY AFFI

COATESVILLE PA

TEST REPORT

DATE MARCH 18,1957 | LUKENS OFFICE NO 465-1 | SPECIFICATIONS A-264 GR. 3,BACKING TO ASTM A-212 GR FBX

CUSTOMER ORDER NO 3081

3 TO · MR. G.H. PUTMAN, PA.
· CHICAGO BRIDGE & IRON CO.
· P.O. BOX 277
· BIRMINGHAM 1, ALA.

REPORT OF CHEMICAL AND PHYSICAL TESTS OF H.T. STL KIL FBX 70000 8.33%STAINLESS CLAD TYPE 304 ELC | ORDER NO 1543-VR-80062-1,2

CAR NO HALLS | HOMOGENEITY TESTS O.K. | BENDING TESTS O.K.

MELT NO	SLAB NO	CHEMICAL ANALYSIS C	MN	P	S	CU	SI	NI	CR	MO		YIELD PSI	TENSILE PSI	% ELONGATION 2"	% RED OF AREA		SIZE OF PLATE
24417	48	25	87	018	027		19						81500				1-40 OD X 3 MIN GA 2-.59
66731	2	024	1.25	028	019		64	9.74	18.35			49000	80300	30			1-40 OD X 3 MIN GA 2-.59
SHEAR	TEST												32800				

PLATES AND TESTS ANNEALED AT 1950/2050°F.

I HEREBY CERTIFY THAT THE ABOVE TESTS ARE CORRECT TO THE BEST OF MY KNOWLEDGE AND BELIEF.

LUKENS STEEL COMPANY

ASS'T ENGR OF TESTS

Fig. 44 Room temperature head material data supplied by fabricator.

course, which corresponds to the manufacturer's data. The lower course also did not show as significant a decrease in room temperature strength from the data reported by the manufacturer.

3.3 Chemical Tests

Because the yield strength at elevated temperatures was higher than expected, a spectro-analysis was taken to determine the extent of significant tramp elements. Results of the analysis are shown in Table IX. None of the tramp elements individually is in sufficient quantity to account for the elevated-temperature properties. Collectively those tramp elements would improve the high temperature strength.

The chemical analysis furnished by the supplier of the A-212 grade B material is shown in Figures 45 and 46.

3.4 Bend Tests

Several samples were removed from the upper portion of the failed vessel shell and subjected to a 180° bend test to roughly determine the ductility and the general condition of the metal. The metal showed excellent ductility (Figure 47) and no cracking was visible.

3.5 Metallographic Examination

3.51 Shell Material. Metallographic examination of the shell material, from both the upper and lower sections of the vessel and the inner and outer sides of the shell at these locations, showed no significant structural defects or differences due to location. The ASTM A-212 grade B material showed a normal amount of spherodizing of the carbon platelets, considering the annealing and service temperatures, as shown in Figure 48. Some ferrite banding was observed (Figure 49), which again is normal for this type of material. Slag inclusions and carbon content (Figure 50) appeared to be of usual amounts for A-212 grade B. No cracks were found in any of the ASTM A-212 material.

TABLE IX

SPECTRO-ANALYSIS FOR TRAMP ELEMENTS

Element Symbol	Weight %
Cr	0.04
Cu	0.012
Mo	0.005
Ni	0.014
U	< 0.005 (a)
W	< 0.040 (a)

(a) No trace found. Limits shown are minimum sensitivity values.

1520A Rev. 3

DS:3

TENNESSEE COAL & IRON

DIVISION

UNITED STATES STEEL CORPORATION

DEPARTMENT OF METALLURGY, INSPECTION & RESEARCH

FAIRFIELD STEEL WORKS

NO. 44822

FAIRFIELD, ALA. February 13, 1957

CHEMICAL AND MECHANICAL TEST RECORD ON BASIC OPEN HEARTH High Tensile Firebox STEEL

FURNISHED TO Chicago Bridge and Iron Company

ADDRESS Boyles, Alabama

SPECIFICATION TCI-388 "B" ASTM A-212-54-T

REQUIREMENTS (FOR PERMISSIBLE EXCEPTIONS SEE FULL COPY OF SPECIFICATION)

CHEMICAL C .35 Mx MN .90 Mx P .035Mx S .04 Mx Si.15/.30 MECHANICAL T.S. 70/85 Y.P. 38000 Min ELONG. 22% Min

TEST NUMBER	MILL ORDER	PUR ORDER	HEAT NUMBER	C	MN	P	S	Si	PIECES	SIZE	WIDTH	THICKNESS	AREA	Y.P. PER SQ. INCH	TENSILE STR. PER SQ. INCH	ELONG. IN 2"	BEND TEST	BEND	INSPECTED BY
D-111	RF-13703	3081 #4	30261	.31	.78	.018	.030	.271	1	2 7/8" x 74 x 123		.507	.2019	41010	81220	27.5	OK	OK	TCI
Top	"	"	"							"					82120				

Plate and tests were heat treated for grain refinement.

CAR NUMBER Sou 316181

SWORN TO AND SUBSCRIBED BEFORE ME THIS 21 DAY OF February 1957

NOTARY PUBLIC

, BEING DULY SWORN ACCORDING TO LAW, DEPOSES AND SAYS THAT THE FIGURES SET FORTH ABOVE ARE CORRECT, AS CONTAINED IN THE RECORDS OF THE COMPANY.

J. W. Cassell

QUALITY CONTROL METALLURGIST

Fig. 45 Chemical analysis of A-212 grade B material.

1530A Rev. 2

TENNESSEE COAL & IRON

DIVISION

UNITED STATES STEEL CORPORATION

DEPARTMENT OF METALLURGY, INSPECTION & RESEARCH

DS:3

NO. 44846

FAIRFIELD STEEL WORKS

FAIRFIELD, ALA February 13, 1957

CHEMICAL AND MECHANICAL TEST RECORD ON BASIC OPEN HEARTH High Tensile Firebox STEEL

FURNISHED TO Chicago Bridge and Iron Company ADDRESS Boyles, Alabama

SPECIFICATION TCI-388"B" ASTM A-212-54-T

REQUIREMENTS: (FOR PERMISSIBLE EXCEPTIONS SEE FULL COPY OF SPECIFICATION)

CHEMICAL C .35 Mx MN .90 Mx P .035 Mx S .04 Mx Si.15/.30 MECHANICAL T.S. 70/85 Y.P. 38000 Min ELONG 22% Min RED

TEST NUMBER	MILL ORDER	PUR. ORDER	HEAT NUMBER	C	MN	P	S	Si	PIECES	SIZE	WIDTH	THICKNESS	AREA	Y. P. PER SQ. INCH	TENSILE PER SQ. INCH	% ELONG. IN 2 INCHES	RED.	HOT TEST	BEND	INSPECTED BY
C-8465	MF-13703	3081 #4	26401	.31	.81	.015	.033	.238	1	2 7/8" x 86 x 123		.506	.2011	49230	75[illegible]	28.0		OK	OK	TCI
Top	"	"	"							"					75[illegible]0					

Plates and tests were heat treated for grain refinement.

SWORN TO AND SUBSCRIBED BEFORE ME THIS 21 DAY OF February 1957

NOTARY PUBLIC

CAR NUMBER Sou 54788

BEING DULY SWORN ACCORDING TO LAW DEPOSES AND SAYS THAT THE FIGURES SET FORTH ABOVE ARE CORRECT, AS CONTAINED IN THE RECORDS OF THE COMPANY.

J. W. Cassell

QUALITY CONTROL METALLURGIST

Fig. 46 Chemical analysis of A-212 grade B material.

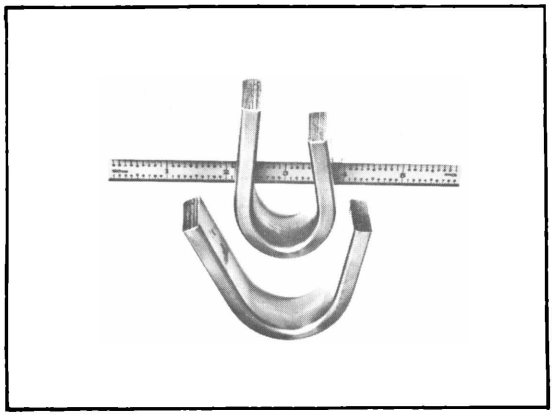


Fig. 47 Photograph of shell material bend test.

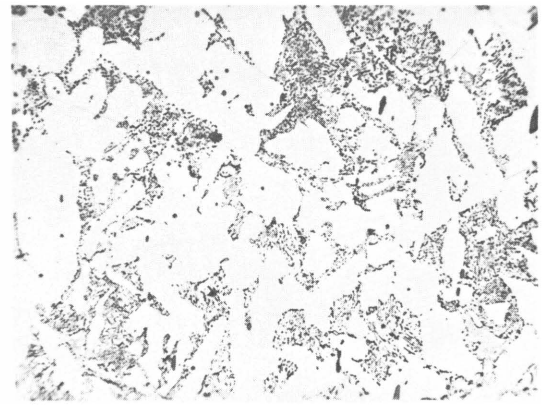


Fig. 48 A-212-B shell material, 750X, nital etch, showing the carbide spheroidization from the original pearlitic structure.

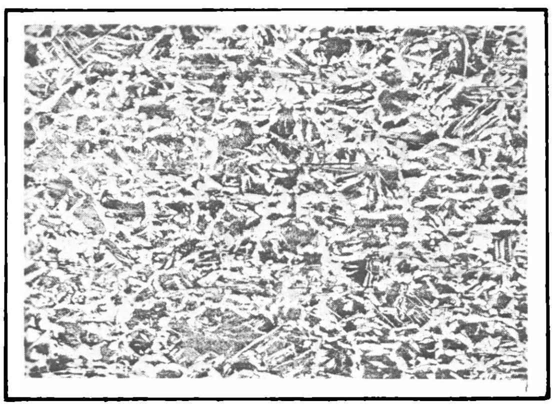


Fig. 49 A-212-B shell material, 50X, nital etch. This shows the general structure, ferrite banding, and carbon content of the material.

Fig. 50 A-212-B shell material, 250X, showing the small slag inclusions and general structure.

3.52 Weld Material. Sections from the longitudinal as well as from the girth weld were taken for study. Figure 51 is a view of the interior surface of the longitudinal section of the girth weld showing the typical weld edge cracks. As Figure 52 shows, these cracks are in the 304L cladding at the edge of the weld bead but not in the stainless-steel weld itself. These cracks do not penetrate the A-212 material to any extent but stop within 1/8-in. of the stainless-carbon-steel interface. The stainless-steel weld material had very few cracks of the type seen in the 304 cladding as shown in Figure 53. However, these cracks did appear in the welds (Figure 54) and were about half way through the weld clad. Figure 51 is a section of the girth weld taken parallel to the center line of the weld. This photograph shows the typical major cracking similar to the one that indicated the failure by leaking. These cracks (Figure 55) were caused by stress concentration due to the expansion of the A-212 plates above and below the weld. The smaller cracks shown in Figure 56 on the inside surface did not contribute to the major cracks or failure. These cracks appear to be the result of thermal stress and fatigue.

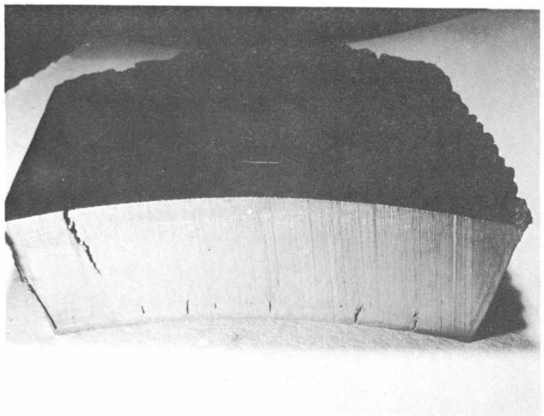


Fig. 51 Photograph of the longitudinal section of the girth weld.

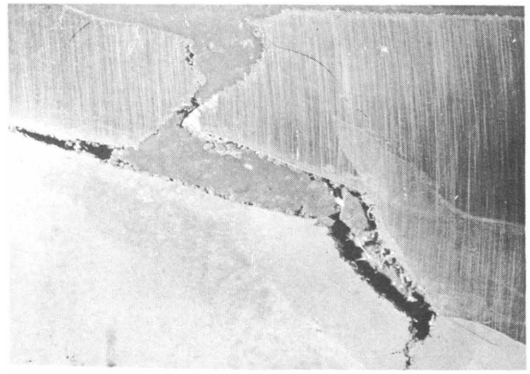


Fig. 52 304L cladding crack at the edge of the weld, 14X, electrolytic etch. The weld is the thicker material.

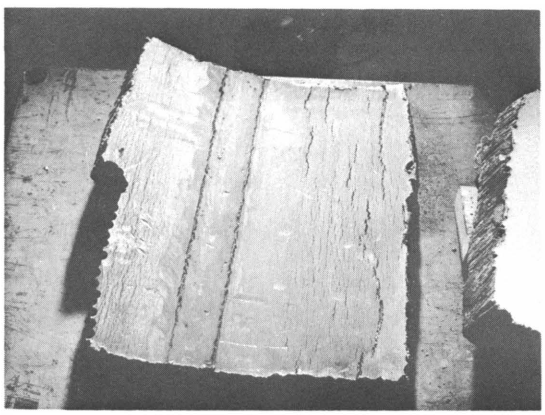


Fig. 53 **Photograph** of the stainless-steel clad **weld, showing the crack at the edge of the weld.**

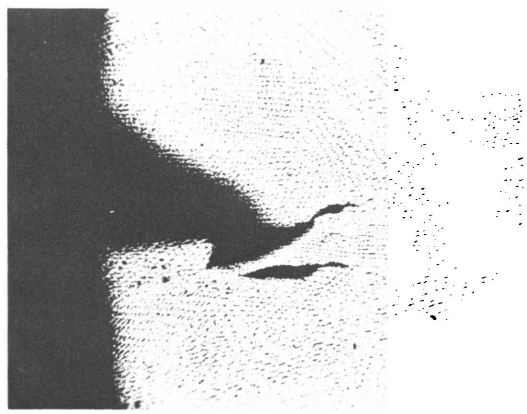


Fig. 54 Crack in the weld surface, 40X, electrolytic etch.

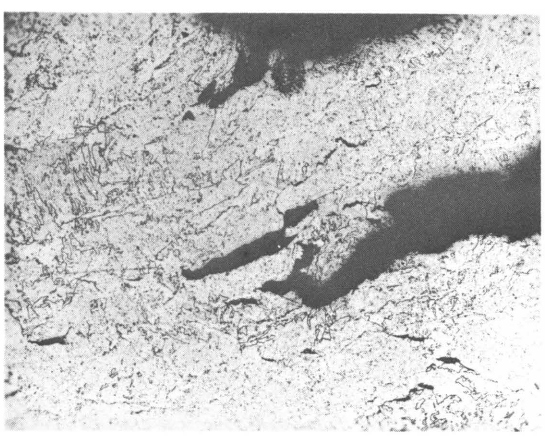


Fig. 55 Girth weld material, 50X, nital etch. This shows the major crack root.

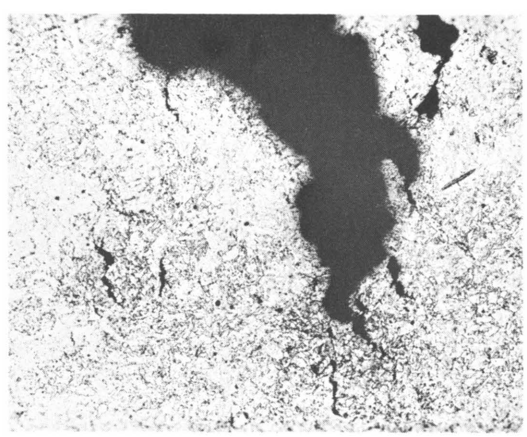


Fig. 56 Girth weld material, 50X, nital etch, showing the root of the clad side crack in the carbon steel.

The mean coefficient of thermal expansion of the steel shell was about 7.8 x 10^{-6} in./in./°F in the service temperature range, while that of the austenitic stainless was 10.1 x 10^{-6} in./in./°F. As the vessel was heated, this condition would tend to put a compressive stress on the stainless cladding, and a tensile stress on the shell material adjacent to it. During any cooling cycle this condition would reverse. No other significant defects, such as blowholes, cracks or porosity were noted. The largest porous area noted in cutting of the welds and machining of tensile specimens was about 1/2 in. in diameter and seemed to be quite thin.

3.53 Cladding Failure. The significant results of the metallurgical examination of the plate cladding showed that the Hortonclad material had failed completely while the roll-clad material in the head had not (Figures 57, 58, and 59). Sectioning of the A-212 plate from the head showed roll cladding 1/4-in. thick, and the Hortonclad side plates showed cladding 1/8-in. thick. Refer to Figure 60 for the cracking pattern and appearance of the cladding from the upper side plate material. The largest cracks are vertical, or normal to the hoop stress. The failed cladding was brazed to the A-212 plate using a 92% nickel braze material and a brazing temperature of 2050°F. The cladding would expand at this temperature due to its higher coefficient of thermal expansion as compared to the steel shell. After fusion of the braze material the 304L stainless-steel cladding would then have a tensile stress as the assembly cooled down. This is indicated in Figure 61 where a sample of the shell and clad was machined and surface-ground, then hack-sawed to release the strain. These strains were then measured so as to estimate the stress in the body material and the cladding.

Fig. 57 304L roll-clad material, 100X, electrolytic etch (90 sec), showing structure and lack of cracking.

Fig. 58 304L - Carbon-steel bond from the head, 100X, 10-sec electrolytic etch. Carbon steel is the darkened area.

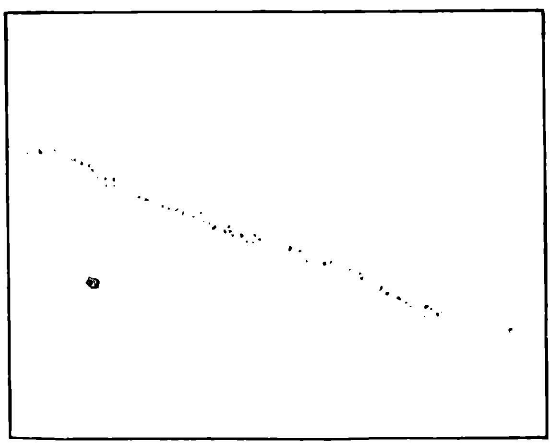


Fig. 59 304L- Carbon-steel bond from the head, 100X, unetched. The pin holes appear in the cladding side of the bond.

SEE ERRATA AT END OF ITEM

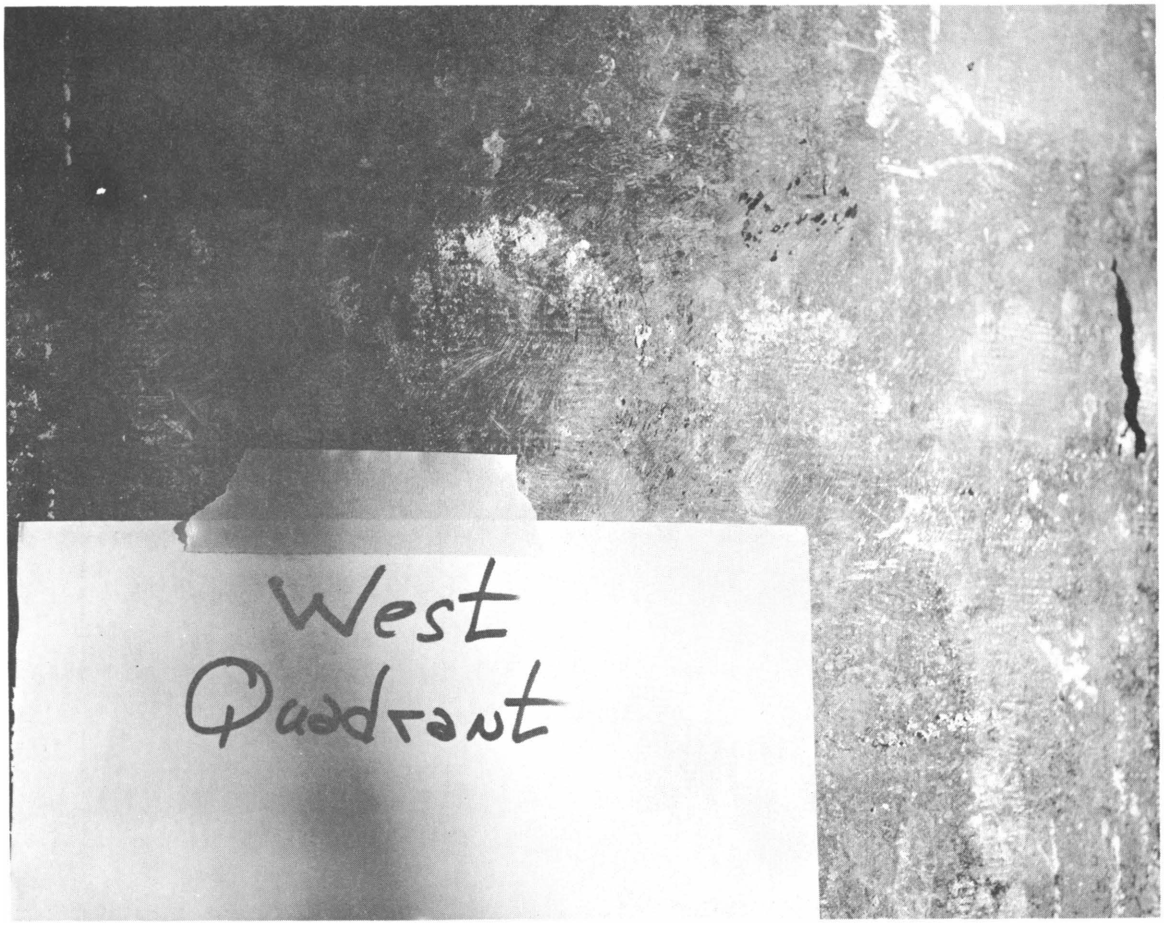


Fig. 60 Photograph of cracks in girth weld.

The cladding cracks were all of an intergranular type as shown in Figures 62 and 63. A Strauss copper sulfate test was made on the stainless-steel liner to determine if the material was susceptible to intergranular corrosion. The test revealed the intergranular cracks caused by the precipitation of intergranular carbides. This test consists of boiling a test piece in a solution of copper sulfate and sulfuric acid. Unsatisfactory metal will show intergranular cracks after a period varying from a few hours to hundreds of hours. The cladding in the pressurizer developed intergranular cracking throughout in a 72-hr test period. Test coupons were not checked to see if shorter exposure time would develop these cracks. Possible failure mechanisms of the intergranular type are stress corrosion, hot shortness, creep, and thermal fatigue.

Hot shortness seems to be the least likely mechanism of failure, due to the maximum temperature indications of approximately 1000°F. Thermal fatigue also is discounted from playing a major role due to the relatively low number of thermal cycles. Thermal fatigue would accelerate the stress corrosion mechanism. The major cause of the cladding failure is thought to be stress corrosion and creep. Stress, temperature and other service conditions are right for these failure mechanisms to occur. There is considerable evidence to show that a non-service condition also contributed substantially to the cladding failure.

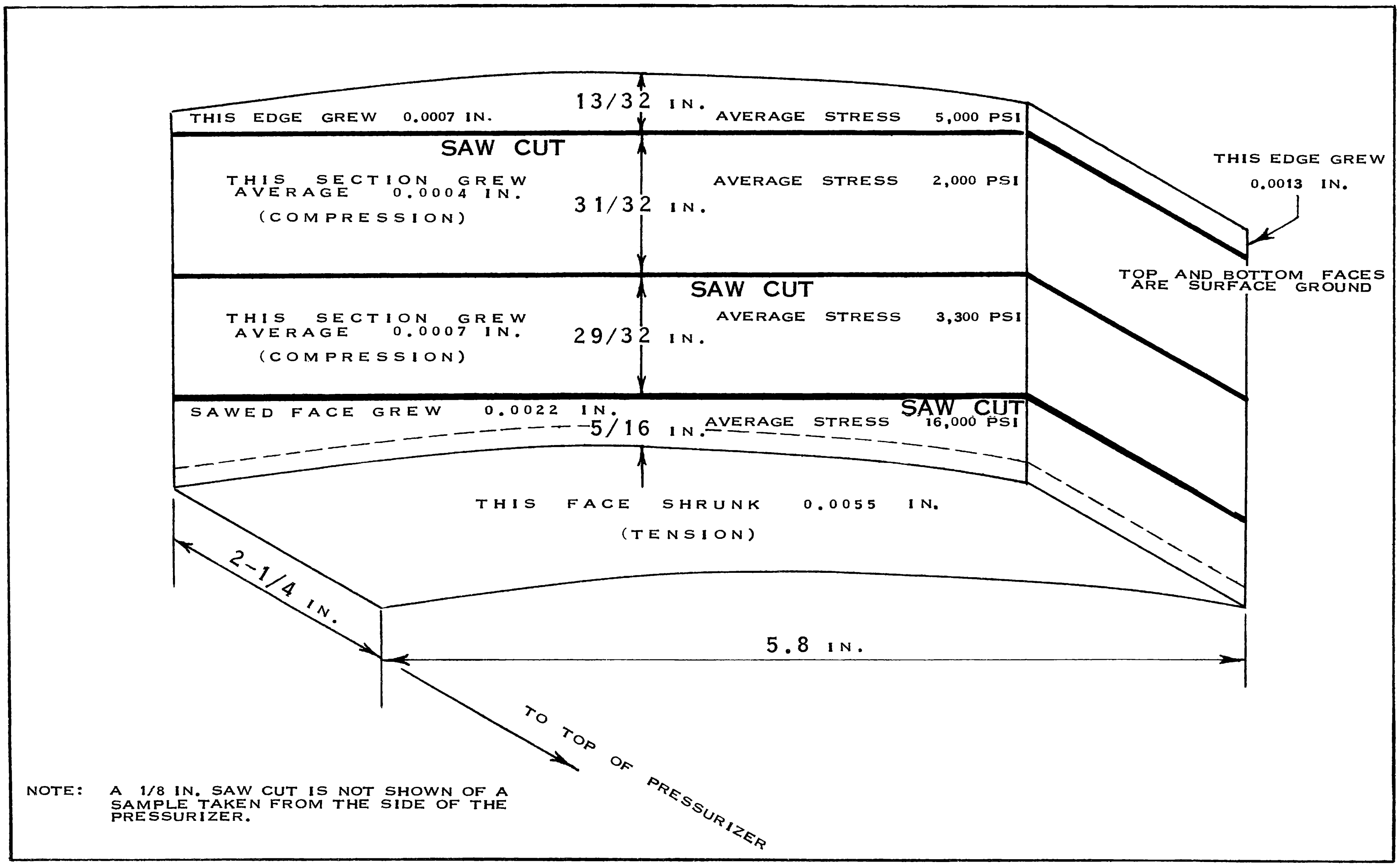


Fig. 61 Sample taken from the shell material to measure the residual strain released by sawing a ground test block as shown.

Fig. 62 304L – Cladding, 250X, eletrolytic etch (10 sec), showing the intergranular corrosion from the upper sensitized area.

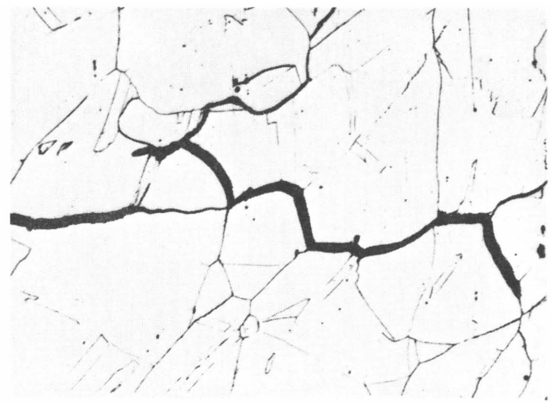


Fig. 63 304L – Cladding, 250X, electrolytic etch (10 sec), showing the intergranular corrosion from the upper sensitized area.

It was observed that a series of cracks on the upper side shell cladding were in a straight line for about 3 ft. Some of these cracks were about 10 in. long. This was about in the center of the plate to the left of the first sample taken. It also was observed that some areas, including one from the first sample taken, were surface-ground. This probably was done with a disc grinder. This occurred after the brazing operation and hot working of the plate, as indicated by the light tan oxide as compared to the black scale on the rest of the cladding. In the light tan areas the cracking was much less severe and in some spots none was evident. This indicates that some notch or type of scratch was removed in the grinding operation. These notches, or scratches, would serve to concentrate stress and also serve to nucleate certain corrosion mechanisms. Austenitic stainless is the most corrosion resistant in the annealed condition with the surface polished and passivated. The cladding of the pressurizer was not in the annealed state or in this surface condition. Thus, the failure of the cladding was greatly accelerated by these non-service conditions.

VII. FAILURE ANALYSIS

1. GENERAL

Since the failure of the Spert III pressurizer vessel occurred following a period of continuous service during which there was no apparent indication of operation outside the design limitations, and since an examination of the operating records since the acceptance of the plant revealed no immediately obvious explanation for the failure, a number of possible mechanisms which could have contributed to or caused the failure were postulated, and each of these was evaluated in the light of all the information available. The mechanisms considered were:

(1) Failure due to deficient design, materials or fabrication of vessel.

(2) Failure due to thermal cycling in normal operation.

(3) Failure due to corrosion of the vessel walls.

(4) Failure due to thermal shocks during plant upsets caused by malfunction of other plant equipment.

(5) Failure due to transient pressure rises during nuclear experiments.

(6) Failure due to pressures in excess of design at or below design temperature.

(7) Failure due to temperature in excess of design at or near design pressure.

The following section discusses the evaluation of each of these mechanisms as possible contributors to the failure.

2. DISCUSSION OF VARIOUS FAILURE MECHANISMS

2.1 Vessel Design and Materials

The design calculations for the vessel have been checked using various methods including Section VIII of the ASME Code for Unfired Pressure Vessels and the dimensions shown on the Chicago Bridge and Iron drawing. In all respects this vessel met code requirements. Vessel design calculations are shown in Appendix D-2, Calculations.

The information available to date has disclosed nothing in the vessel materials or fabrication to which failure can be attributed. Although data from tensile specimens indicate that the material used in the pressurizer did not conform to the ASME specifications for room temperature strength, the strength at temperature was greater than normally found in A-212, grade B. It cannot be said therefore that the plate was not adequate for the design service. The welds, in spite of slight porosity, behaved better than might have been expected. Failure of the cladding did not contribute to vessel failure.

2.2 Fatigue Through Thermal Cycling in Normal Operation

Stresses incurred during normal start-up and cool-down were calculated using methods previously developed by others [4, 5]. The maximum possible heat-up rate of 100°F/hr (limited by available heating) induces no stresses significantly above design allowables and no stresses in the range of the yield strength. Normal cool-down is less severe because the pressure stresses are simultaneously being reduced during cooling.

The metallurgical examination presented in Section VI also showed no significant evidence of fatigue.

Calculations of stresses induced by normal thermal cycling are shown in Appendix D-3. Failure by normal thermal cycling is not indicated.

2.3 Corrosion

Although the AISI type 304 cladding of the vessel shell was covered with numerous cracks and was unbonded in many areas, there was no evidence of severe corrosion of the ASTM A-212 material. Micrometer measurement of the total shell wall, including cladding, was 2.850 in. This slightly thinner wall can be attributed to the deformation of the wall and in any case is not enough to make any significant change in the allowable working pressure. No failure of the vessel by corrosion is indicated.

2.4 Vessel Failure Due to Thermal Shock

On several occasions, equipment difficulties have necessitated or resulted in rapid pressure and temperature reductions in the Spert III pressurizer vessel. As may be noted in Table IV, seven such pressure reductions have resulted in apparent temperature drops in excess of 100°F at a rate exceeding 100°F/hr. A temperature change of 100°F/hr exceeds that recommended by the architect-engineer, and results in large thermal stresses in the vessel shell. Calculations have been made to determine the magnitude of the thermal stress and to evaluate the contribution of thermal stress to vessel failure (see Appendix D-3, Calculations).

The seven occasions which exceed the rate of 100°F/hr are noted by an asterisk in column 1 of Table IV. Since no actual temperature measurements were obtained, it has been necessary to assume that the temperature of the water existing at any time was the saturation temperature corresponding to the recorded pressure. The pressure variation as a function of time was obtained from chart records.

The most severe drop occurred on March 23, 1959 (occasion 1 of Table IV). The transient temperature distribution in the vessel wall has been calculated using an IBM 650. Preliminary calculations using these data (see Appendix D-4) indicate the stress in the base metal did not exceed about 29,000 psi, which is insufficient to yield the base metal. Thermal stress in the inner stainless-steel clad approached 33,000 psi, which may have been sufficient to yield the clad and perhaps cause the blistering observed. (The calculated stresses may be higher than actual stresses since an infinite heat transfer rate to the vessel wall was assumed.)

Further refinements in the calculations would be required before firm conclusions could be drawn. However, in the absence of more reliable temperature information on the vessel shell, further calculations are not justified.

If thermal shock had been a primary cause of the failure, stresses above the ultimate strength of the material would have been locally induced to start cracks and checking or ratcheting of the vessel wall of sufficient frequency above the yield strength to produce progressive yield. The cracks would be expected on the interior surface of the vessel. Although the AISI type 304 cladding was severely cracked, no cracks were found in the base metal. As reported in Section VI, a bend test on a coupon cut from the vessel wall showed excellent ductility and no stress risers such as would be found if the metal were cracked. While thermal shocks may have been a contributor to the cracking of the cladding, no evidence has been found that would indicate a weakening of the vessel shell material. The maximum "ratcheting" effect for a cold shock would be the temperature expansion of the circumference from ambient to 650°F. Assuming a complete expansion deformation for each thermal shock, this would amount to approximately 0.45 in. per shock as shown by the calculations below. Even this unrealistic extreme case would require approximately 20 "cold shocks" to attain the deformation noted. This is far in excess of the number of severe cold shocks experienced by the vessel.

Ratcheting effect by cold shock maximum movement possible of circumference by thermal expansion is given by

$$\Delta L = L_o(\Delta T)C$$

where ΔL = change in length

L_o = circumference at 50°F

ΔT = change in temperature

$C = 6.5 \times 10^{-6}$ coefficient of expansion of steel in./°F in.

$$\Delta L = 120 \times 600 \times 6.5 \times 10^{-6}$$

$$\Delta L = 0.45 \text{ in.}$$

Thermal shock damage as a primary cause of the vessel failure is not indicated.

2.5 Damage by Transient Pressures During Nuclear Testing

Although there have been some 200 transient nuclear experiments performed in the Spert III facility, no pressure rises above the design operating pressure have been observed using either the plant instrumentation or the transient experiment pressure transducers.

The time of rise of the pressure pulses observed during the transient experiments is about 5 msec which is too long to permit the formation of shock wave fronts during transmission of the pressure pulse to the pressurizer vessel. Further, the total energy available in these pressure pulses is far too low to produce damage of the type observed. Calculations relative to the credibility of damage by transient pressure testing are shown in Appendix D-5.

2.6 Vessel Failure Due to Pressure in Excess of Design at or Below Design Temperature

Vessel failure due to overpressure has been carefully examined. Several potential conditions existed whereby pressures in excess of design might have been obtained by operational error. These are as follows:

(1) The pressurizer control system, as installed, permits energizing the electric immersion heaters under certain conditions with the pressure controller out-of-service.

(2) With the pressure controller out-of-service, the blow-down valve would not function; therefore, the pressurizer relief valve was thought to be the only available safety to prevent excessive prssure. However, the instrument case test (Section VI) demonstrated that other plant equipment would have failed, preventing gross overpressure, even if the relief valve failed to function.

Although overpressurizing of the vessel was operationally credible, the following considerations indicate that excessive overpressure within the design temperature range did not occur:

(1) Tensile specimens pulled at 668°F indicate the yield strength of the material was 32,250 psi. Calculation of the internal pressure sufficient to yield the vessel at 668°F shows a pressure of about 5023 psi is required. It is reasonable, therefore, to conclude that the pressurizer safety relief valve, which opened at 4200 psi under test, would have prevented reaching pressures of this magnitude.

(2) Since all instruments were in service at the apparent time of failure and their accuracy verified following failure, it can be concluded that excessive overpressure did not occur. Excessive overpressure prior to the apparent time of failure, assuming all instruments were out-of-service, would have resulted in vessel failure at the time of that overpressure or would have required an equally excessive overpressure at the actual time of failure. The latter conclusion is apparent from the stress-strain curve for materials similar to that used in the Spert III pressurizer vessel.

(3) Plant instrument records do not show any evidence of excessive overpressure at any time. Nor is there any evidence of major calibration shift or other mechanical damage which would be expected to result from such overpressure. Further the pressure test conducted on the pressurizer vessel liquid-level transmitter case shows that excessive pressures would have damaged this unit.

(4) Measurements of the diameter of the pressurizer connecting piping do not indicate the pipe has been subjected to pressure sufficient to yield.

In consideration of the above there is no evidence that pressure in excess of design was the principal cause of vessel failure. In the absence of other mechanisms, the 2660 psig to which the vessel was subjected at the time of failure would not have resulted in any damage to the vessel.

2.7 Vessel Failure Due to Temperature in Excess of Design At or Near Design Pressure

Results of metallurgical examination, colorimetric tests conducted on strain-gage materials, and limited temperature data obtained during strain measurements in March, 1961, indicate that heating of the upper half of the vessel beyond the design temperature was the probable cause of vessel failure. It remains, therefore, to discuss how overheating may have occurred, the available temperature data and other evidence supporting this mode of failure.

Since no source of heat existed in the pressurizer vessel other than the electrical immersion heaters, if it is assumed the vessel was heated to temperatures in excess of design (668°F), the heat must have been supplied from this source. Provided the heaters are always submerged in water, the temperature of the water and steam in the vessel cannot exceed the saturation temperature for a given pressure. As previously discussed, no evidence of pressures in excess of about 2700 psig exist, therefore, with the heaters submerged, the vessel temperature could not have exceeded about 678°F. Tensile test data again demonstrate that an internal pressure of about 5000 psig is required to yield the vessel at this temperature. It is, therefore, inferred that only by removing the electric immersion heaters from the water could temperatures above 678°F be obtained.

The pressurizer liquid-level system, as discussed in Section IV, has been studied carefully in conjunction with temperature data available from the strain-gage test program. These temperature data were hand-recorded as millivolt readings during strain-gage tests on the reactor vessel and primary piping to determine the strains to the system induced by transient testing. Eighty-one strain gages were installed on the reactor vessel and 58 were installed on the primary piping. These 58 included the four strain gages on the pressurizer vessel. Although data were recorded from all locations, the dynamic effects on the reactor vessel were of prime interest. Data of 72 excursion tests initiated at varying pressure conditions of 0 to 2500 psi and varying flow conditions of 0 to 20,000 gpm were examined for the reactor vessel.

The greatest strain observed was 58.2 microinches/in. which occurred during a ramp excursion initiated at 200 psi and 20,000-gpm flow. The stress calculated from this strain was only 2.6% of the yield stress.

Since the recording oscillograph strain traces for the piping were of the same magnitude as the trace variations caused by electronic "noise", no attempt was made at that time to reduce the data further. When it became apparent that temperature data on the pressurizer were important to an analysis of the failure, the temperature data were reduced. These temperature data are presented in Table X. If it is recalled that the resistance bulb associated with the temperature compensator for the liquid-level system was located 10 in. below the centerline of the top electric heater, the data in Table X show that whereas the temperature in the vicinity of the top heater was 668°F or above, the temperature near the bottom of the vessel was as low as 359°F. By assuming a temperature distribution of the water in the vessel based on these data, it can be shown by calculations presented in Appendix D-6 that the indicated level may be as much as 20% higher than the actual level. Although a normal liquid level (7.5 ft) was indicated, one or possibly more of the top heaters could have been above the water level. Once the two top heaters, which

TABLE X

PLANT CONDITIONS AND PRESSURIZER VESSEL THERMOCOUPLE READINGS DURING THE STRAIN-GAGE CALIBRATION TEST OF MARCH 1961

Run No.	Date	Time	Pressure	TE-36R	Thermocouple Reading (°F)				TR-3-8R (°F)	LRC-6C (feet)
					149	150	152	151		
1	3-20-61	0100-0155	0	Amb.	73	67	69	66	66	11.6
2	3-20-61	0621-0700	500	Amb.	472	470	472	450	388	7.9
3	3-20-61	0805-0857	1000	63°	549	545	546	518	457	7.7
4	3-20-61	0955-1045	1500	64°	598	507	598	540	500	7.6
5	3-20-61	1155-1225	2000	63°	639	638	638	538	532	7.5
6	3-20-61	1307-1350	2500	63°	672	670	669	539	555	7.5
7	3-20-61	2025-2039	Atmos.	---	433	241	588	120	191	7.6
8	3-21-61	0005-0045	500	Amb.	493	472	545	411	393	8.3
9	3-21-61	0207-0255	1500	Amb.	601	599	584	556	501	8.2
10	3-21-61	0355-0427	2500	Amb.	671	670	671	607	564	7.7
11	3-21-61	0627-0659	2500	150°	677	676	670	451	532	7.75
12	3-21-61	1000-1030	2500	258.8°	685	682	683	359	468	7.6
13	3-21-61	1315-1355	2500	359.1°	687	681	690	369	448	7.5
14	3-21-61	1615-1626	2500	450°	676	672	678	396	456	7.8
15	3-21-61	Void	----	---	---	---	---	---	---	-----
16	3-21-61	2215-2220	2500	550°	671	670	672	519	503	8.2
17	3-22-61	0825-0855	2500	625°	676	674	678	530	510	7.6
18	3-22-61	1145-1210	2500	552°	688	675	694	555	527	7.5
19	3-22-61	1430-1505	2500	448.4°	689	679	695	526	515	7.6
20	3-22-61	1627-1640	2500	350°	690	677	697	500	505	7.3
21	3-22-61	1840-1851	2500	250°	712	672	723	451	480	7.4
22	3-22-61	2103-2115	2500	150°	713	681	724	410	467	7.7
23	3-22-61	2310-2321	2500	100°	717	703	726	382	473	7.5
24	3-23-61	0028-0100	2000	95°	694	660	716	352	435	7.5
25	3-23-61	0243-0311	1500	95°	658	596	692	244	376	7.6
26	3-23-61	0410-0440	1000	95°	590	515	640	210	349	7.6
27	3-23-61	0515-0545	500	95°	566	473	660	177	310	7.6
28	3-23-61	0755-0820	0	90°	394	229	605	132	202	7.5

Location of Thermocouples

149 - On horizontal section of blow-off, upper reactor pressurizer connecting line.

150 - Pressurizer vessel - one inch above centerline of the top heater.

151 - Pressurizer vessel - 61 inches below centerline of the top heater.

152 - Pressurizer dome - 90 inches above centerline of the top heater.

TR-3-8R - Pressurizer vessel - the average of three thermocouples located 27 inches apart with the top thermocouple 6 inches above the top heater.

normally were employed for pressure control, were exposed to the steam, pressure control would be accomplished by adding heat to the steam only. Stratification and general cooling of the remaining water would then increase, thus increasing the error between actual and apparent level. The level control would then reduce the actual water level, possibly uncovering more heaters and compounding the error. Figure 64, is a plot of the data shown in Table X.

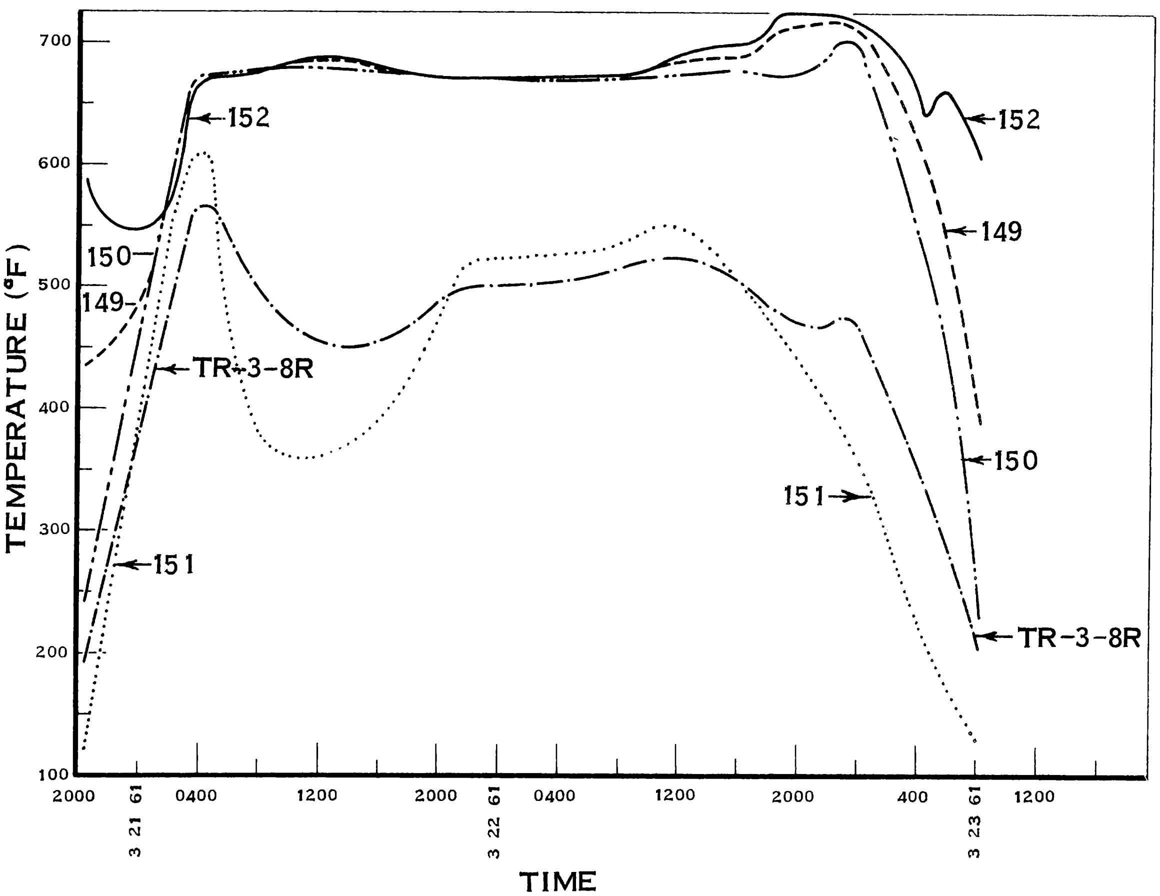


Fig. 64 Plot of pressurizer temperature data taken during strain-gage tests.

Once the liquid level in the vessel has dropped below one or more of the electric heaters, superheating of the steam will occur. Tests on the heaters have shown that the heaters will continue to operate for at least 4-1/2 hr at temperatures from 1400 to 1500°F in air. Calculations (see Appendix D-7) have been performed to establish the length of time required to heat the steam and vessel wall above the water level to 1000°F. The calculations show that if one heater is exposed about 13.3 hr are required, if two heaters are exposed about 6.9 hr are required, etc. Pressure increases which might be expected to accompany superheating of the steam will not be observed due to condensation at the surface of the liquid, contraction of the water in the lower portion of the vessel which is cooling, and pressure-control sensing of the superheated steam pressure.

Examination of the temperature data in Table X further shows that temperatures as high as 726°F actually were recorded in the top of the pressurizer vessel at about 2300 hr on March 22, 1961. At this same time, the average temperature of the vessel skin as recorded on TR-3-8R was 473°F. Thus a temperature difference of 253°F existed.

On the evening of vessel failure, the maximum temperature of the vessel skin as recorded by TR-3-8R was 630°F. Using a simple ratio method to approximate the temperature of the upper portion of the vessel on the evening of the failure, a temperature of about 965°F is obtained. Although undoubtedly this method of approximating the vessel temperature is in error, it does illustrate that exceedingly high temperatures could have been obtained, not only on the night of failure but throughout the operational life of the vessel, without any of the existing instrumentation showing unusual conditions.

Review of the literature regarding high-temperature creep failure of materials similar to that used in the Spert III pressurizer vessel indicates that failure of the vessel at 2500 psig and 1000°F can be expected. The failure occurs in progressive stages, usually referred to in the literature as three stages. The initial two stages usually result in small deformation, whereas the third stage is a sudden large deformation. The three stages of creep failure can occur in times as short as 10 hr at 1000°F.

Based on the evidence available at this time it is concluded that the Spert III pressurizer vessel failed because of progressive creep failure. The first two stages of creep failure probably occurred at an earlier date as evidenced by the stabilizing–band bolt failure. The third stage of failure occurred on the evening of October 26, 1961.

VIII. CONCLUSIONS

1. CONCLUSIONS

Information developed during the review reported herein has led to the conclusion that failure of the Spert III pressurizer vessel was due to progressive high temperature creep failure. Heating of the upper portion of the vessel shell in excess of the design temperature (670 to ~1000°F) probably has occurred on numerous occasions; however, from the data available the total accumulative length of time at the excessive temperatures cannot be determined. Weakening of the vessel due to prior damage resulted in failure of the vessel at 2660 psig and an estimated temperature of about 950°F.

Lack of a reliable liquid-level control system and adequate temperature-sensing devices on the pressurizer shell are directly responsible for the overheating. The fact that the two heaters controlling the pressure were the top heaters and located in the center of the vessel rather than the bottom also was a major contributor.

Failure of the vessel cladding appears to be due to the method of fabrication of the vessel; however, thermal stresses caused by rapid depressurization may have contributed to blistering of the clad and weakening of the center girth weld.

The investigation of the Spert III pressurizer vessel failure has brought to light several deficiencies in the design and specifications of the vessel and the plant instrumentation, which warrant further consideration.

2. DESIGN CHANGES FOR THE PRESSURIZER VESSEL

(1) The pressurizer specification was sufficiently complete and detailed to have obtained a satisfactory pressure vessel. However, specifications for the new vessel will require the fabricator to provide tensile specimens of each material and of identical materials from different heats used in fabricating the vessel. The specimens shall be subjected to identical heat treatment as the vessel and shall be representative of the materials in the finished vessel. The tensile specimens shall be pulled at the vessel design temperature and at room temperature, and certified results reported.

(2) Future specifications will require the fabricator to supply design calculations and drawings with as-built dimensions of the finished vessel. The design calculations shall reflect all design considerations, ie, pressure, temperature, thermal cycles, fatigue, etc.

(3) The specifications also will require that the vessel fabricator specify the recommended maximum heat-up and cool-down rate for the vessel, and the rates which may result in damage to the vessel. Supporting calculations will be required.

(4) Results of the metallurgical examinations indicate that stainless-steel cladding applied by the present Horton clad process (vacuum-braze) is not suitable for the service conditions existing in the Spert III pressurizer vessel. Therefore, if a stainless-steel-clad pressurizer vessel is specified, the specification will require that the clad be applied by the roll-clad process. However, serious consideration is being given to the use of a pressurizer vessel without an internal cladding or to a vessel with nickel-clad steel.

(5) Consideration is being given to an A-302, A-387, or A-204B steel for fabrication of a replacement pressurizer vessel. This would provide better properties in the 700 to 800°F range, although this range will be above the pressurizer service range.

(6) Inclusion of inspection ports to permit periodic inspection of the vessel interior will be included in the vessel design.

3. DESIGN CHANGES IN THE PRESSURIZER INSTRUMENTATION

(1) The present pressure control system is subject to operator error. As presently installed, when the power supply to the pressurizer pressure recorder-controller is turned off with the recording pen below the pressure setpoint, two or more heaters may be energized provided the toggle switches on the control center panel are left in the "on" position and the heater breakers are closed. The number of heaters energized depends upon the amount of deviation of the pen from the setpoint. Under these conditions, the pressure recorder-controller will not be recording or controlling the pressurizer pressure.

The pressure control circuit will be altered such that the heaters cannot be energized unless the pressure recorder-controller is in service.

(2) Probably the prime contributor to the vessel failure was the stepwise method of the pressurizer heater control in which the two top heaters were turned on and off to maintain the pressure in the vessel. This causes large temperature gradients in the bottom of the vessel leading to erroneous liquid–level indication. These temperature gradients may also have caused thermal stresses in the vessel.

Consideration is being given to redesigning the pressurizer heater control system so that the energy required to achieve and maintain pressurizer pressure be uniformly applied by all heaters. A controller employing a saturable core reactor for control of the electrical input to all heaters is a method of providing the desired results. Relocation of the control heaters to the bottom of the vessel may be adequate.

(3) Because of the seriousness of an erroneous liquid-level indication in a pressurizer vessel, a more reliable liquid-level sensing device will be installed on the replacement pressurizer vessel and sufficient temperature-sensing devices will be provided in the vessel to give an accurate measurement of the average temperature of the liquid in the vessel.

(4) Several temperature-sensing devices will be installed in the pressurizer vessel skin. The temperature devices will be located to detect the temperature of the outside of the shell in several locations, the temperature distribution through the shell, and the temperature of the liquid and steam. A high temperature alarm will be provided.

(5) The replacement pressurizer vessel will be protected from overpressure by two relief valves in parallel. These valves will be flanged to permit removal for testing and maintenance.

(6) Limit switches will be installed on the pressurizer isolation valve, PCV-4R, which would actuate "open" or "closed" position lights on both control boards.

(7) The make-up pump control circuit will be redesigned such that the make-up pump cannot be started at any time unless the reactor pressure recorder is in service.

(8) In order to provide a continuous record of the plant pressure, one slow-speed pressure recorder will be installed on the reactor primary system. A speed change limit switch in the instrument will increase the chart speed at any time the recording pen reaches a preset level. This instrument will be left on at all times.

(9) A recording ammeter will be installed on the pressurizer heater power supply to maintain a complete record of the heater usage.

IX. REFERENCES

1. W. E. Nyer and S. G. Forbes, Spert Program Review, IDO-16415 (September 27, 1957).
2. W. E. Nyer and S. G. Forbes, Spert Program Review, IDO-16634 (October 19, 1960).
3. R. E. Heffner and T. R. Wilson, Spert III Reactor Facility, IDO-16721 (October 25, 1961).
4. B. F. Langer, Design Basis for Reactor Vessels, WAPD-CE-43 (October 18, 1955).
5. M. P. Heisler, "Transient Thermal Stresses in Slabs and Circular Pressure Vessels", J. Appl. Mech, 20: No. 2, p 261 (June 1953).

X. APPENDIX A

SPERT III NON-NUCLEAR COLD WATER ACCIDENT TEST

J. A. Norberg
September 27, 1961

SPERT III NON-NUCLEAR COLD WATER ACCIDENT TEST

Page No.

Page No.

Page No.

Page No.

Page No.

Page No.

Page No.

SPERT III NON-NUCLEAR COLD WATER ACCIDENT TEST

I. INTRODUCTION

Knowledge of the Spert III plant operating parameters which affect the moderator temperature are of importance for the evaluation of a nuclear cold water accident test program and for general nuclear safety aspects associated with plant operations.

The nonnuclear plant operations which most seriously affect the moderator temperature are: (1) Cold loop start-up, (2) heat exchanger load, (3) make-up pump operation, (4) blow-off valve operation, (5) blow-down valve operation and (6) primary circulating pump load.

A. Objectives

A test series has been planned whose primary objective is to obtain the moderator temperature response in the active core region as a function of transient changes in the above mentioned plant operating parameters. Of secondary importance is the obtainment of the primary system loop response to these plant operating changes. Associated with this second objective is an evaluation of a ΔT scram circuit which is designed to protect the reactor from moderator temperature drops caused by rapid heat exchanger load changes.

B. Instrumentation

The moderator temperature change in the active core region will be measured by fast response thermocouples connected to CEC oscillograph recorders. These thermocouples are for two purposes: (1) Sixteen units will measure the moderator temperature as it passes through the upper grid and will give the spacial-time-temperature distribution of moderator, (2) three sets of 2 each will measure the moderator temperature above and below the fuel plates to determine the temperature rise time through the core.

The core instrumentation will be located in the following lattice positions:

Single Thermocouples: E-11, S-12, N&S-21, N&S-41, S&E-31, N&E-32, E-51, S-23, S-24, S-42, S-14 and S-33.

Thermocouple Pairs : S-11, E-24 and N-31.

Key plant instrumentation and value position indicators will be parallel circuited such that during the actual response test their signal can be recorded on CEC oscillograph records. The following list presents the data points to be recorded:

TEMPERATURE	LOCATION
TR-5-1C	Heat Exchanger Inlet - West Loop
TR-6-1C	Heat Exchanger Inlet - East Loop
TR-31-5C	Reactor Vessel - Top Bulk
TR-31-6C	Reactor Vessel - Bottom Bulk
TR-1-1C	Reactor Outlet - West Loop
TR-2-1C	Reactor Outlet - East Loop
TR-3-2R	Reactor Inlet - West Loop
TR-4-2R	Reactor Inlet - East Loop
TR-3-5R	Heat Exchanger Outlet - West Loop
TR-4-5R	Heat Exchanger Outlet - East Loop
TRC-3C	Heat Exchanger Mixed Temp. - West Loop
TRC-4C	Heat Exchanger Mixed Temp. - East Loop
ΔT Scram	West Loop Primary Pump Suction
ΔT Scram	East Loop Primary Pump Suction

MISCELLANEOUS	LOCATION
PRC-6C	Pressurizer Pressure
LRC-6C	Pressurizer Level
PR-5C	Reactor Vessel Pressure

MISCELLANEOUS	LOCATION
LCV-6-1R	Blow-down Valve
PCV-5-1R	Blow-off Valve
HIC-6-2C	Make-up Pump
TRC-3C	10" Heat Exchanger Valve - West Loop
TRC-4C	10" Heat Exchanger Valve - East Loop
FRC-1C	16" Flow Control Vavle - West Loop
FRC-2C	16" Flow Control Valve - East Loop

Two 26 channel CEC oscillograph recorders will be used for the response test data records. Fifty-two data channels are thus available of which 46 to 48 will be utilized leaving 4 to 6 channels for spares. Existing plant and transient instrumentation amplifiers will be used as required. Previous experience has indicated that power to the plant recorders may have to be shut-off during the response tests due to noise pick-up by the CEC recorders. These details will be worked out during the actual hook-up.

C. Initial Plant Conditions

The reactor core will have dummy fuel assemblies in lattice positions S-11, E-24, N-31, E-21, W-21, N-33, W-33, S-32 and E-41. With dummy fuel assemblies in these positions, the reactor has been demonstrated to be subcritical at room temperature with all control rod poison out of the core. Since the reactor is subcritical at room temperature and since it has a negative temperature coefficient of reactivity, it will therefore be subcritical at all higher temperatures thus eliminating criticality hazards.

The initial tests will begin with the system conditions of 450°F and 2500 psig. At this point the reactivity insertion rate due to negative temperature coefficient begins to increase rapidly. In addition, the possibility of rapid moderator temperature changes caused by plant operating parameters is greatly enhanced as the system temperature reaches 450°F and above.

II. TEST PROCEDURE

The following procedure will be followed as closely as possible; however, it is not to be construed as an absolute schedule since operation experience may cause some changes.

1. STEP ONE - 450°F AND 2500 PSIG

Raise the system temperature to 440°F and system pressure to 2500 psi. Establish equilibrium.

A. Cold Loop Start-up Accident

This test will determine the system response to start-up of one loop which is at a lower temperature than the operating loop. The parameters to be investigated are: (1) Initial temperature difference between loops, (2) initial flow rate of operating loop, (3) type of cold loop start-up, i.e., either start-up against a closed or open flow control valve, and (4) effect, if any, of the heat exchangers on automatic or manual temperature control.

1. Calibrate the flow control valves in the following manner:

a. Close flow control valves FCV-2aR and FCV-2bR and stop the pumps in the east loop. Close the flow control valves, FCV-laR and FCV-lbR in the west loop. (HIC-laR and HIC-lbR are used to close these valves.) Open the flow control valve FCV-laR with HIC-laR for 3 sec and allow the flow to become steady. On data sheet No. 1 record the resulting primary flow shown by the Barton ΔP gage, FI-laR and FRC-lc and the valve position shown by the position indicator on the valve, the position indicator at the control center and the position recorded by the CEC. (NOTE: Operate the CEC at 1/2" per sec.) Repeat the determination by opening the valve for 4.5 secs., 6 secs., 7.5 secs., and 9 secs. to get flow rates ranging from 2000 gpm to 10,000 gpm. Plot this data as a curve of opening time versus resulting flow. This information will be required for later tests. (NOTE: Do not allow the pumps to operate against the closed valves for more than 30 seconds.)

b. Close flow control valve FCV-1aR and stop the pumps in the west loop. Start the pumps in the east loop and repeat step (a) above recording the data on data sheet No. 2. Plot the opening time versus flow for the east loop.

2. Determine the effect of the start-up of a cold, stagnant loop on the moderator temperature using the <u>west loop</u> as the cold, stagnant loop at a temperature differential of 10°F.

a. East Loop Heat Exchanger on Manual Proportional Control

(1) With the pumps operating in both the east and west loops, bring the temperature of the system to 440°F. Close flow control valve FCV-1aR with HIC-1aR and temperature control valve TCV-3-2R with HIC-3-2R (west loop). The temperature control valves are to be in proportional control so the by-pass opens when the heat exchanger valve closes. Stop the west loop primary pumps. Raise the temperature in the reactor and east loop to 450°F using a flow rate of 10,000 gpm. (The west loop becomes a stagnant loop.) Level the temperature at 450°F for 15 minutes using the east loop heat exchanger on manual proportional control. Switch the key plant instrumentation to CEC record, and turn on the CECs set at a speed of 10 inches per second.

Approximately a half second later start the pumps (one at a time) in the west loop and open the west loop flow control valve, FCV-1aR with HIC-1aR for approximately 9 sec. (The exact length of time required to give a flow rate of 10,000 gpm can be determined from the opening time versus flow chart obtained in Section II-1A-1, pg. 4.) Allow the CECs to run for 30 seconds before switching them off. This would give approximately 2 loop times of operation. Switch the key plant instrumentation back to plant record. Record the information required by data sheet No. 3.

(2) Repeat the procedure described in Section II-1A-2a-(1) for flow rates of 7500 gpm. The length of time required to open FCV-1aR to give 7500 gpm can be determined from the curve which was developed in Section II-1A-1 (pg. 4). Increase the running time of the CECs to 45 secs., and decrease the speed to 5 inches per sec.

(3) Repeat Section II-1A-2a-(1) (pg. 5) for flow rates of 5000 gpm. Increase the running time of the CECs to 60 secs. and decrease the speed to 5 inches per sec.

(4) Repeat Section II-1A-2a-(1) (pg. 5) for flow rates of 2500 gpm. Increase the running time of the CECs to 120 secs., and decrease the speed to 2 inches per sec.

b. East Loop Heat Exchanger on Automatic Control

(1) Lower the system temperature to 440°F. Repeat the procedure described in Section II-1A-2a (pg. 5) with the exception that the east heat exchanger is set on automatic control instead of manual control. The controller (TRC-4C) settings should be as listed below:

gain 5.5-6.5, proportional band 15-50%, reset .3-1%, rate .1-.9%.

Record the information required by data sheet No. 3 and record the controller settings.

(2) Repeat Section II-1A-2b-(1) (pg. 6) for flow rates of 7500 gpm. Increase the running time of the CECs to 45 secs., and decrease the speed to 5 inches per sec.

(3) Repeat Section II-1A-2b-(1) (pg. 6) for flow rates of 5000 gpm. Increase the running time of the CECs to 60 secs., and decrease the speed to 5 inches per sec.

(4) Repeat Section II-1A-2b-(1) (pg. 6) for flow rates 2500 gpm. Increase the running time of the CECs to 120 secs., and decrease the speed to 5 inches per sec.

3. Determine the effect of the start-up of a cold stagnant loop on the moderator temperature using the west loop as the cold, stagnant loop at a temperature differential of 20°F.

The procedure to follow is the same as described in Section II-1A-2 (pg. 5) with the exception that the temperature of the cold loop is to be 430°F instead of 440°F.

4. Determine the effect of the start-up of a cold, stagnant loop on the moderator temperature using the east loop as the cold, stagnant loop at a temperature differential of 20°F.

a. West Loop Heat Exchanger on Manual Proportional Control

(1) With both loops in operation, bring the temperature of the system to 430°F. Close flow control valve FCV-2aR with HIC-2aR and temperature control valve TCV-4-2R with HIC-4-2R (east loop). Stop the east loop primary pumps. Raise the temperature in the reactor and west loop to 450°F using a flow rate of 10,000 gpm. Level the temperature at 450°F for 15 minutes using manual proportional heat exchanger control. Switch the key plant instrumentation to CEC record and turn on the CECs set at a speed of 10 inches per sec. One-half second later start the pumps (one at a time) in the east loop and open the east loop flow control valve, FCV-2aR with HIC-2aR for approximately 9 secs. (The exact length of time which would give a flow of 10,000 gpm can be determined from the curves obtained in Section II-1A-1 (pg. 4).) Allow the CECs to run for 30 seconds before switching them off. Record the information required by data sheet No. 4. Switch the key plant instrumentation back to plant record.

(2) Repeat the procedure described in Section II-1A-4a-(1) (pg. 7) for flow rates of 5000 gpm. Increase the running time of the CECs to 60 seconds, and decrease the speed to 5 inches per sec.

b. West Loop Heat Exchanger on Automatic Control

(1) Lower the system temperature to 430°F. Repeat the above procedure (II-1A-4a) with the exception that the west heat exchanger is set on automatic control instead of manual control. The controller (TRC-3C) settings should be as listed below:

gain 5.5-6.5, proportional band 5-30%, reset .1-1%, rate .05-.1%.

Record the necessary information as required by data sheet No. 4 and record the controller settings.

(2) Repeat Section II-1A-4b-(1) (pg. 8) for flow rates of 5000 gpm. Increase the running time of the CECs to 60 secs. and decrease the speed to 5 inches per sec.

5. Determine the effect of the start-up of a cold, stagnant loop on the moderator temperature using the west loop as the cold, stagnant loop at a temperature differential of 10°F. The pumps will be started against an open valve; therefore, extreme caution will have to be used to prevent the pumps from overloading during start-up (do not overload pumps for more than 5 secs.) and destructive water hammer effects.

a. With both loops in operation, bring the temperature of the system to 440°F. Close flow control valve FCV-1aR with HIC-1aR and temperature control valve TCV-3-2R with HIC-3-2R (west loop). Stop the west loop primary pumps. Raise the temperature in the reactor and east loop to 450°F, and establish equilibrium at 450°F for 15 minutes using manual heat exchanger control. Reduce the flow in the east loop to 1000 gpm, and open the west loop flow control valve FCV-1aR to a position to give a flow rate of 1000 gpm. Switch the key plant instrumentation to CEC record and turn on the CECs set at a speed of 2 inches per sec. One-half second later start the pumps (one at a time) in the west loop taking care to prevent the pumps from overloading. Allow the CECs to run for 240 secs. before switching them off, and record the necessary information required by data sheet No. 3. Switch the key plant instrumentation back to plant record.

b. Repeat the procedure described in Section II-1A-5a (pg. 8) for flow rates of 2500 gpm. Decrease the running time of the CECs to 120 secs. and increase the speed to 5 inches per sec.

c. Repeat the procedure described in Section II-1A-5a (pg. 8) for flow rates of 5000 gpm. Decrease the running time of the CECs to 60 secs. and increase the speed to 5 inches per sec.

d. Repeat the procedure described in Section II-1A-5a (pg. 8) for flow rates of 7500 gpm. Decrease the running time of the CECs to 45 secs. and increase the speed to 5 inches per sec.

e. Repeat the procedure described in Section II-1A-5a (pg. 8) for flow rates of 10,000 gpm. Decrease the running time of the CECs to 30 seconds and increase the speed to 10 inches per sec. (NOTE: In all probability, it will not be possible to extend these tests to flow rates of 10,000 gpm; however, they have been included for completeness.)

6. Determine the effect of the start-up of a cold, stagnant loop on the moderator temperature using the <u>west loop</u> as the cold, stagnant loop at a temperature differential of 20°F. The pumps will be started against an open valve; therefore, extreme caution will have to be used to prevent the pumps from overloading during start-up (do not overload pumps for more than 5 secs.) and destructive water hammer effects.

The procedure to follow is the same as described in Section II-1A-5 (pg. 8) with the exception that the temperature of the cold loop is to be 430°F instead of 440°F.

7. Determine the effect of the start-up of a cold, stagnant loop on the moderator temperature using the east loop as the cold, stagnant loop at a temperature differential of 20°F. The pumps will be started against

an open valve; therefore, similar precautions described in Sections II-1A-5,6 (pg. 8&9) will have to be taken.

a. With both loops in operation, bring the temperature of the systems to 430°F. Close flow control valve FCV-2aR with HIC-2aR and temperature control valve TCV-4-2R with HIC-4-2R (east loop). Stop the east loop primary pumps. Raise the temperature in the reactor and west loop to 450°F and establish equilibrium at 450°F for 15 minutes using manual heat exchanger control. Reduce the flow in the west loop to 2500 gpm and open the east loop flow control valve FCV-2aR to a position to give a flow rate of 2500 gpm. Turn on the CECs set at a speed of 5 inches per sec. One-half second later start the pumps in the east loop taking care to prevent the pumps from overloading. Allow the CECs to run for 120 secs. before switching them off, and record the information required by data sheet No. 4.

b. Repeat the procedure described in Section II-1A-7a (pg. 10) for flow rates of 7500 gpm, and decrease the running time of the CECs to 45 secs. (NOTE: It may not be possible to use a flow rate of 2500 and 7500 gpm; however, the test is to be performed with the east loop as the cold loop for applicable flow rates of some magnitude.)

B. Heat Exchanger Load Changes

This test will determine the system response to rapid changes in heat exchanger load. A sudden opening of the heat exchanger 10 inch temperature control valve causing a larger flow of primary system water to be passed through the heat exchanger will result in an lowering of the primary system water temperature. The drop in primary system water temperature will affect both the nuclear and physical parameters of the plant. The nuclear effect will be that of adding reactivity with its resultant power increase. The physical effect will be that of water volume shrinkage and associated system depressurization.

Test and evaluation of the ΔT scram system will also be accomplished during the heat exchanger load change tests.

1. West Loop Heat Exchanger - Manual Control

Determine the effect of the heat exchanger load on the moderator temperature using the west loop heat exchanger in manual control for system temperature control. The affect that the position of the temperature control valve has on the temperature change is unknown, and will have to be "played by ear" in order to prevent a temperature drop greater than 20°F.

a. With both loops in operation at a total flow rate 20,000 gpm, establish equilibrium at a system temperature of 450°F for 15 minutes using the west loop heat exchanger for system temperature control. The east loop heat exchanger will be valved off by closing TCV-4-2R with HIC-4-2R (with the valves in proportional control, the by-pass valve TCV-4-1R will be opened). Switch the key plant instrumentation to CEC record and start the CECs at a speed of 10 inches per second. One-half second later, with the west loop temperature control valves in proportional control, adjust the west heat exchanger temperature control valve, TCV-3-2R, manually by holding HIC-3-2R open for ___ secs. Allow the CECs to run for 30 seconds before switching them off and record the necessary information as required by data sheet No. 5. Switch the key plant instrumentation back to plant record. Close temperature control valve TCV-3-2R and bring the system temperature back to 450°F. Repeat the above procedure for various valve settings so that the resulting west loop temperature drop is in approximately 10 degree increments up to a maximum ΔT of 20°F. Monitor TR-3-2R for this temperature.

b. Repeat Section II-1B-1a (pg. 11) for a total flow rate of 15,000 gpm. Decrease the speed of the CECs to 5 inches per second and increase the running time to 45 seconds.

c. Repeat Section II-1B-1a (pg. 11) for a total flow rate of 10,000 gpm. Decrease the speed of the CECs to 5 inches per second, and increase the running time to 60 seconds.

d. Repeat Section II-1B-1a (pg. 11) for a total flow rate of 5000 gpm. Decrease the speed of the CECs to 5 inches per second and increase the running time to 120 secs.

2. East Loop Heat Exchanger - Manual Control

Determine the effect of the heat exchanger load on the moderator temperature using the east loop heat exchanger in manual control for system temperature control. The affect that the position of the temperature control valve has on the temperature change is unknown and will have to be "played by ear".

a. With both loops in operation at a total flow rate of 20,000 gpm, establish equilibrium at a system temperature of $450^{\circ}F$ for 15 minutes using the east loop heat exchanger for system temperature control. The west loop heat exchanger will be valved off by closing TCV-3-2R with HIC-3-2R. Switch the key plant instrumentation to CEC record and start the CECs at a speed of 10 inches per second. One-half second later, with the east loop temperature control valves in proportional control, adjust the east heat exchanger temperature control valve, TCV-4-2R, manually by holding HIC-4-2R open for ___ secs. Allow the CECs to run for 30 seconds before switching them off, and record the necessary information as required by data sheet No. 5. Switch the key plant instrumentation back to plant record. Close temperature control TCV-4-2R and bring the system temperature back to $450^{\circ}F$. Repeat the above procedure for various valve settings so that the resulting east loop temperature drop is in approximately 10 degree increments up to a maximum ΔT of $20^{\circ}F$. Monitor TR-4-2R for this temperature.

b. Repeat Section II-1B-2a (pg. 12) for a total flow rate of 10,000 gpm. Decrease the speed of the CECs to 5 inches per second, and increase the running time to 60 seconds.

3. West Loop Heat Exchanger - Automatic Control

Determine the effect of the heat exchanger load on the moderator temperature using the west loop heat exchanger for system temperature control. This test will be similar to II-1B-1 (pg. 11) except the temperature will be controlled automatically instead of manually.

a. With both loops in operation at a total flow rate of 20,000 gpm, establish equilibrium at a system temperature of 450°F for 15 minutes. Use the west loop heat exchanger in automatic control for system temperature control with the east loop heat exchanger valved off. The control settings should be as noted in Section II-1A-4b (pg. 8). Switch the key plant instrumentation to CEC records and start the CECs at a speed of 10 inches per second. One-half second later lower the west loop temperature set point 5 degrees. Allow the CECs to run for 30 seconds before switching them off, and record the information required by data sheet No. 5 noting any changes in controller settings. Switch the key plant instrumentation back to plant records. Reset the set point to control at 450°F and repeat the above procedure for 5° increment setting up to a maximum west loop ΔT of 20°F. Monitor TR-3-2R for this temperature.

b. Repeat II-1B-3a (pg. 13) for a total flow rate of 15,000 gpm. Decrease the speed of the CECs to 5 inches per second and increase the running time to 45 seconds.

c. Repeat II-1B-3a (pg. 13) for a total flow rate of 10,000 gpm. Decrease the speed of the CECs to 5 inches per second and increase the running time to 60 seconds.

d. Repeat Section II-1B-3a (pg. 13) for a total flow rate of 5000 gpm. Decrease the speed of the CECs to 5 inches per second and increase the running time to 120 seconds.

4. East Loop Heat Exchanger - Automatic Control

Determine the effect of the heat exchanger load on the moderator temperature using the east loop heat exchanger for system temperature control. This test will be similar to II-1B-2 (pg. 12) except the temperature will be controlled automatically instead of manually.

a. Use the procedure described in II-1B-3a (pg. 13) except use the east loop heat exchanger automatic control with the west loop heat exchanger valved off.

b. Use the procedure described in II-1B-3c (pg. 13) except use the east loop heat exchanger in automatic control.

2. STEP TWO - 500°F AND 2500 PSIG

Raise the system temperature to 490°F and establish equilibrium for 15 minutes.

A. Cold Loop Start-up Accident

1. Repeat the procedure outlined in Section II-1A-1 (pg. 4) with the system temperature at 490°F.

2. Repeat the procedure outlined in Section II-1A-2 (pg. 5) with the cold loop at 490°F and the system temperature at 500°F. Reverse the order of the loops using the east loop as the cold loop.

3. Repeat the procedure outlined in Section II-1A-3 (pg. 7) with the system temperature at 500°F. Use the east loop as the cold loop.

4. Repeat the procedure outlined in Section II-1A-4 (pg. 7) with the system temperature at 500°F. Use the west loop as the cold loop.

5. Repeat Section II-1A-5 (pg. 8) at a system temperature of

500°F. Use the east loop as the cold loop.

6. Repeat Section II-1A-6 (pg. 9) at a system temperature of 500°F. Use the east loop as the cold loop.

7. Repeat Section II-1A-7 (pg. 9) at a system temperature of 500°F. Use the west loop as the cold loop.

B. Heat Exchanger Load Change

1. Repeat Section II-1B-1 (pg. 11) at a system temperature of 500°F. Use the east loop heat exchanger for control.

2. Repeat Section II-1B-2 (pg. 12) at a system temperature of 500°F. Use the west loop heat exchanger for control.

3. Repeat Section II-1B-3 (pg. 13) at a system temperature of 500°F. Use the east loop heat exchanger for control.

4. Repeat Section II-1B-4 (pg. 14) at a system temperature of 500°F. Use the west loop heat exchanger for control.

C. Make-up Water Addition

Operation of the make-up pump lowers the reactor core moderator temperature since the make-up water is added to the reactor vessel inlet at essentially room temperature. The following test will determine this affect for various plant flow rates:

1. Level the system temperature at 500°F and adjust the total flow rate to 500 gpm. Switch the key plant instrumentation to CEC record and start the CECs at a speed of 1 inch per second. One-half second later start the make-up pump and allow to run for 480 seconds. Stop the CECs after 480 seconds. Switch the key plant instrumentation back to plant record.

2. Repeat Section II-2C-1 (pg. 15) for a flow rate of 1000 gpm. Stop the CECs and make-up pump after 360 seconds.

3. Repeat Section II-2C-1 (pg. 15) fow a flow rate of 2000 gpm. Increase the speed of the CECs to 2 inches per second and decrease the running

time of the CECs and make-up pump to 240 seconds.

4. Repeat Section II-2C-1 (pg. 15) for a flow rate of 5000 gpm. Increase the speed of the CECs to 5 inches per second and decrease the running time of the CECs and make-up pump to 120 seconds.

5. Repeat Section II-2C-1 (pg. 15) for a flow rate of 10,000 gpm. Increase the speed of the CECs to 5 inches per second and decrease the running time of the CECs and make-up pump to 60 seconds.

6. Repeat Section II-2C-1 (pg. 15) for a flow rate of 20,000 gpm. Increase the speed of the CECs to 10 inches per second and decrease the running time of the CECs and make-up pump to 30 seconds.

3. STEP THREE - 550°F and 2500 PSIG

Raise the system temperature to 540°F and establish equilibrium for 15 minutes.

A. Cold Loop Start-up Accident

1. Repeat Section II-1A-1 (pg. 4) at 540°F.
2. Repeat Section II-1A-2 (pg. 5) with the system temperature at 550°F.
3. Repeat Section II-1A-3 (pg. 7) with the system temperature at 550°F.
4. Repeat Section II-1A-4 (pg. 7) with the system temperature at 550°F.
5. Repeat Section II-1A-5 (pg. 8) with the system temperature at 550°F.
6. Repeat Section II-1A-6 (pg. 9) with the system temperature at 550°F.
7. Repeat Section II-1A-7 (pg. 9) with the system temperature at 550°F.

B. Heat Exchanger Load Changes

1. Repeat Section II-1B-1 (pg. 11) with the system temperature at $550^{\circ}F$.

2. Repeat Section II-1B-2 (pg. 12) with the system temperature at $550^{\circ}F$.

3. Repeat Section II-1B-3 (pg. 13) with the system temperature at $550^{\circ}F$.

4. Repeat Section II-1B-4 (pg. 14) with the system temperature at $550^{\circ}F$.

C. Make-up Water Addition

Repeat Section II-2C (pg. 15) with the system temperature at $550^{\circ}F$.

4. STEP FOUR - $600^{\circ}F$ and 2500 PSIG

Raise the system temperature to $590^{\circ}F$ and establish equilibrium for 15 minutes.

A. Cold Loop Start-up Accident

1. Repeat Section II-1A-1 (pg. 4) at $590^{\circ}F$.

2. Repeat Section II-2A-2 (pg. 14) (See II-1A-2, pg. 5) with the system temperature at $600^{\circ}F$. (NOTE: Calculations indicate a limitation of $14^{\circ}F$ ΔT in system temperature due to NPSH requirement at $600^{\circ}F$ (1560 psig). Approach the $14^{\circ}F$ ΔT with caution and do not allow the system pressure to drop below the NPSH at $600^{\circ}F$.)

3. Repeat Section II-2A-3 (pg. 14) (See II-1A-3, pg. 7) with the system temperature at $600^{\circ}F$.

4. Repeat Section II-2A-4 (pg. 14) (See II-1A-4, pg. 7) with the system temperature at $600^{\circ}F$.

5. Repeat Section II-2A-5 (pg. 14) (See II-1A-5, pg. 8) with the system temperature at $600^{\circ}F$.

6. Repeat Section II-2A-6 (pg. 15) (See II-1A-6, pg. 9) with the system temperature at 600°F.

7. Repeat Section II-2A-7 (pg. 15) (See II-1A-7, pg. 9) with the system temperature at 600°F.

B. Heat Exchanger Load Changes

1. Repeat Section II-2B-1 (pg. 15) (See II-1B-1, pg. 11) with the system temperature at 600°F.

2. Repeat Section II-2B-2 (pg. 15) (See II-1B-2, pg. 12) with the system temperature at 600°F.

3. Repeat Section II-2B-3 (pg. 15) (See II-1B-3, pg. 13) with the system temperature at 600°F.

4. Repeat Section II-2B-4 (pg. 15) (See II-1B-4, pg. 14) with the system temperature at 600°F.

C. Make-up Water Addition

1. Repeat Section II-2C (pg. 15) with the system temperature at 600°F.

D. Blow-off Valve Operation

Operation of the blow-off valve results in a depressurization of the primary system with associated cooling caused by vaporization in the pressurizer. In general, it is felt that this should have little affect on the moderator temperature unless the level in the pressurizer drops to the extent that the make-up pump turns on. However, it is possible for a malfunction of the blow-off valve to occur whereby it remains open longer than intended by its automatic control. It is therefore of importance that the system response be studied for conditions simulating this occurrence.

1. Determine the effect of the blow-off valve on the moderator temperature with the make-up pump in automatic control.

a. Establish equilibrium at 600°F and a total flow rate of 20,000 gpm. Switch the key plant instrumentation to CEC record and start the CECs at a speed of 5 inches per second. One-half second later open blow-off valve PCV-5 with HIC-5R for ___ seconds. Stop the CECs after 60 seconds and record the data required by data sheet No. 6. Switch the key plant instrumentation back to plant record. A number of runs should be made with increasing time lengths until the pressure approaches the NPSH pressure. (NOTE: The valve should not be held open long enough to allow the system pressure to drop below NPSH requirements (1560 psig at 600°F).)

b. Repeat Section II-4D-1 (pg. 18) for a flow rate of 15,000 gpm.

c. Repeat Section II-4D-1 (pg. 18) for a flow rate of 10,000 gpm.

d. Repeat Section II-4D-1 (pg. 18) for a flow rate of 5,000 gpm. Increase running time of the CECs to 120 seconds.

2. Determine the effect of the blow-off valve on the moderator temperature with the make-up pump off. Repeat Section II-4D-1 (pg. 18) except turn the make-up pump off.

E. Blow-down Valve Operation

The affect of blow-down valve operation is important since it lowers the pressurizer level causing operation of the make-up pump. Controlled tests to study the system response will be conducted to evaluate the affect of a malfunction of this valve.

1. Determine the effect of the blow-down valve on the moderator temperature with the make-up pump in automatic control.

a. Establish equilibrium at 600°F and a total flow rate of 20,000 gpm. Switch the key plant instrumentation to CEC record and start the CECs at a speed of 5 inches per second. <u>One-half second</u> later open the blow-down valve LCV-6R with HIC-6R for ___ seconds. Stop the CECs after 60 seconds, and record the data required by data sheet No. 6. Switch the key plant instrumentation back to plant record. A number or runs should be made with increasing time length until the pressure approaches the NPSH. (See Note Section II-4A-2, pg. 17).

b. Repeat Section II-4E-1 (pg. 19) for a flow rate of 15,000 gpm.

c. Repeat Section II-4E-1 (pg. 19) for a flow rate of 10,000 gpm.

d. Repeat Section II-4E-1 (pg. 19) for a flow rate of 5,000 gpm. Increase the running time of the CECs to 120 seconds.

2. Determine the effect of the blow-down valve on the moderator temperature with the make-up pump off.

5. STEP FIVE - 625°F AND 2500 PSIG

Raise the system temperature to 625°F, and establish equilibrium for 15 minutes. (NOTE: Calculations indicate that a 4°F ΔT will lower the pressure to NPSH requirements ($\sim$1870 psig). Approach the 4°F ΔT with caution and do not allow the system pressure to drop below the NPSH at 625°F.)

A. <u>Cold Loop Start-up Accident</u>

1. Repeat Section II-1A-1 (pg. 4) at 625°F.

2. Repeat Section II-1A-2 (pg. 5) with the system temperature at 625°F. (NOTE: Calculations indicate the minimum temperature of the cold loop to be 615°F.)

3. Repeat Section II-1A-3 (pg. 7) with the system temperature at 625°F.

4. Repeat Section II-1A-4 (pg. 7) with the system temperature at 625°F.

5. Repeat Section II-1A-5 (pg. 8) with the system temperature at 625°F.

6. Repeat Section II-1A-6 (pg. 9) with the system temperature at 625°F.

7. Repeat Section II-1A-7 (pg. 9) with the system temperature at 625°F.

B. Heat Exchanger Load Changes

1. Repeat Section II-2B-1 (pg. 15) with the system temperature at 625°F.

2. Repeat Section II-2B-2 (pg. 15) with the system temperature at 625°F.

3. Repeat Section II-2B-3 (pg. 15) with the system temperature at 625°F.

4. Repeat Section II-2B-4 (pg. 15) with the system temperature at 625°F.

C. Make-up Water Addition

1. Repeat Section II-2C (pg. 15) with the system temperature at 625°F.

D. Blow-off Valve Operation

1. Repeat Section II-4D-1 (pg. 18) with the system temperature at 625°F.

2. Repeat Section II-4D-2 (pg. 19) with the system temperature at 625°F.

E. Blow-down Valve Operation

1. Repeat Section II-4E-1 (pg. 19) with the system temperature at 625°F.

2. Repeat Section II-4E-2 (pg. 20) with the system temperature at 625°F.

6. STEP SIX - 650°F AND 2500 PSIG

Raise the system temperature to 650°F and establish equilibrium for 15 minutes. (NOTE: Calculations indicate that a 1°F ΔT will lower the pressure to NPSH requirements ($\sim$2230 psig). Do not allow the system pressure to drop below the NPSH at 650°F.)

A. Cold Loop Start-up Accident

1. Repeat Section II-1A-1 (pg. 4) at 650°F.

2. Repeat Section II-2A-2 (pg. 14) (See II-1A-2, pg.5) with the system temperature at 650°F. (NOTE: Calculations indicate the minimum temperature of the cold loop to be 647°F.)

3. Repeat Section II-2A-3 (pg. 14) (See II-1A-3, pg. 7) with the system temperature at 650°F.

4. Repeat Section II-2A-4 (pg. 14) (See II-1A-4, pg. 7) with the system temperature at 650°F.

5. Repeat Section II-2A-5 (pg. 14) (See II-1A-5, pg. 8) with the system temperature at 650°F.

6. Repeat Section II-2A-6 (pg. 15) (See II-1A-6, pg. 9) with the system temperature at 650°F.

7. Repeat Section II-2A-7 (pg. 15) (See II-1A-7, pg. 9) with the system temperature at 650°F.

B. Heat Exchanger Load Changes

1. Repeat Section II-2B-1 (pg. 15) with the system temperature at 650°F.

2. Repeat Section II-2B-2 (pg. 15) (See II-1B-2, pg. 12) with the system temperature at 650°F.

3. Repeat Section II-2B-3 (pg. 15) (See II-1B-3, pg. 13) with the system temperature at 650°F.

4. Repeat Section II-2B-4 (pg. 15) (See II-1B-4, pg. 14) with the system temperature at 650°F.

C. Make-up Water Addition

1. Repeat Section II-2C (pg. 15) with the system temperature at 650°F.

D. Blow-off Valve Operation

1. Repeat Section II-4D-1 (pg. 18) with the system temperature at 650°F.

2. Repeat Section II-4D-2 (pg. 19) with the system temperature at 650°F.

E. Blow-down Valve Operation

1. Repeat Section II-4E-1 (pg. 19) with the system temperature at 650°F.

2. Repeat Section II-4E-2 (pg. 20) with the system temperature at 650°F.

7. STEP 7 - ATMOSPHERIC CONDITIONS

Lower temperature to 150°F, depressurize and shutdown.

XI. APPENDIX B

PRESSURIZER SPECIFICATIONS

THE STEARNS-ROGER MFG. CO.

DENVER, COLORADO

for

THE UNITED STATES
ATOMIC ENERGY COMMISSION
IDAHO OPERATIONS OFFICE

IDAHO FALLS, IDAHO

PRIME CONTRACT
AT (10-1)-809

S-R ORDER NO.
B-10300

SPECIFICATIONS FOR

PRESSURIZER VESSEL

1.0 SCOPE

A. These Specifications are intended to provide for the design, material, construction and delivery to the project site of one (1) pressurizer vessel as further described hereinafter. This vessel will be installed in the SPERT-III reactor facility at the National Reactor Test Station in Idaho.

B. All exposed unfinished work shall be thoroughly cleaned and smoothed before leaving the factory. All parts shall be carefully boxed or otherwise suitably prepared for shipment to insure against damage or accumulations of dirt during shipment.

C. The apparatus shall be furnished free from all defects in design, workmanship and materials and shall be guaranteed to give successful service under the operating conditions hereinafter specified.

2.0 DETAILED DESCRIPTION

A. The pressurizer will operate in the primary cooling circuit of a pressurized water type nuclear reactor. The

2.0 DETAILED DESCRIPTION, Con't.

function of the pressurizer is to de-gas circulating water, provide surge space, create, maintain and control the pressure of the water in the primary cooling system. This is accomplished by the use of electric heaters submerged in the water in the vessel which are controlled to keep the water and steam above it at the saturation temperature of the desired operating pressure of the system. The pressurizer also acts as a surge tank on the system to absorb variations in specific volume of primary coolant due to changes in operating temperatures. The pressurizer will have equalizing connections to the reactor vessel so that a steam dome and water level may be maintained in the reactor vessel with controls on the pressurizer.

B. The pressurizer vessel shall be designed and constructed in accordance with the latest revision of the ASME Code for Unfired Pressure Vessels. The vessel shall measure 2'-9" inside diameter by 14'-0" inside length. The vessel shall be constructed of high tensile strength carbon-silicon steel plate and clad on the inside with Type 304 ELC stainless steel of .100 inch minimum thickness. The following openings of the nominal pipe size indicated shall be provided:

(1) One - 4" surge line connection in bottom.

(2) One - 6" relief and pressure control line in the top.

(3) One - 4" reactor dome supply line in side near top.

(4) Five - 5 1/4"I.D. flanged heating element receptacles on sides encircling lower section.

(5) Two - 1" level controller equalizing lines in the side near the top.

Refer to Drawing No. 809-PER-609-M-6 for general arrangement and dimension of the vessel.

C. The pressurizer shall be furnished with the electric heating elements installed. The heaters shall be of the nickel-chrome resistance wire type packed in a Type 304

2.0 DETAILED DESCRIPTION, Con't.

ELC stainless steel tube (316 S.S. will be acceptable if delivery can be improved and no welding is involved) and mounted in the lower area of the straight vessel wall.

The total rating of the heaters shall be not less than 200 Kw. They shall be arranged for wiring of five equal step control while maintaining as near as possible equal phase loading of the 3 phase, 240 volt, 60 cycle supply.

The five heating elements will have a capacity of 24 Kw each. They will be wired so that they may be energized singly or in any combination up to five operating at the same time.

Vapor-tight junction boxes mounted on the vessel wall where the leads from each element are brought through shall be furnished. The normal operating water level will be the indicated water level approximately three feet from the top of the vessel. Low water level is five feet above the bottom of the vessel at a point where all heating elements are covered. It is not planned that the water will ever reach this level except for a very short period. Control and indication of this level to be similar to conventional boiler drum level controllers.

Heat requirements are under minimum conditions to maintain 2500 PSIA against a water column at 250°F. and at maximum condition to maintain 2500 PSIA against a water column at 660°F. when normal test conditions are occurring. Quantity of steam generated for pressurizing the reactor dome is greater than normal, but the test period is of short duration.

D. The pressurizer will be spring-mounted to take the weight load off the piping, but allow freedom of movement in all directions to compensate for pipe expansion due to temperature changes. Refer to Drawing No. 809-PER-609-M-6 for piping expansion information. The vessel manufacturer shall design and furnish the spring supports.

The vessel and all associated piping and equipment will be designed to withstand 2500 PSIG at the maximum operating temperature of 668°F. All inside surfaces of vessel, piping, connecting flanges and nipples wherever

2.0 DETAILED DESCRIPTION, Con't.

in contact with the pressurized water will be stainless steel of a minimum of 0.100" and where welding is involved will be 304 ELC.

3.0 DESIGN CONDITIONS

A. Pressure and Temperatures

Pressure and Temperatures	Shell and Piping
Design Pressure	2500 PSIG
Design Temperatures	668° F.

Test pressure to be at least one and one-half times design pressure in accordance with the ASME Code for Unfired Pressure Vessels.

B. Materials

The shell shall be of high tensile strength carbon-silicon steel similar to ASTM-A-201-52, Grade A, or ASTM-A-301-52, Grade A or B, and USS Type T-1 clad with 0.100" minimum stainless steel Type 304 ELC.

C. Connections

Weld end connections shall be provided suitable for welding to Type 304 ELC stainless steel pipe, as shown on Drawing No. 809-PER-609-M-6.

4.0 INFORMATION TO BE SUPPLIED BY BIDDER

A. Design and Construction

Each bidder shall furnish a detailed description of the proposed pressurizer vessel including a dimensional outline drawing. Specific items of information shall include:

Shell: I.D.______In., Thickness_______In.

Material_______Length__________Ft.

Weight_________Lbs.

4.0 INFORMATION TO BE SUPPLIED BY BIDDER, Con't.

Number and Size of Openings:

Welding procedure with regard to carbon steel and stainless clad materials:

Overall length_________Ft.

Overall diameter_______Ft. (including nipples)

B. Any variance in materials or procedure in fabrication which the builder believes will expedite delivery or in any way improve performance or aid construction, may be offered in the Proposal and the reasons so stated.

5.0 TESTS AND INSPECTION

A. Each proposal shall contain a description of the bidder's quality control procedures before, during and after fabrication of the pressurizer vessel. Each bidder shall identify the inspection agency which furnishes the qualified ASME Code Pressure Vessel Inspector. Ten copies of the following tests and inspections, certified by the inspector shall be furnished to the Purchaser:

1. Reports of chemical analysis and heat treatment of all raw materials, in accordance with applicable ASTM Specifications for each material.

2. Reports of radiograph and/or sonic inspections of the pressurizer vessel, in accordance with (ASTM E94-52T and ASTM A388-55T).

3. Report of final shop hydrostatic pressure test by authorized agency inspector.

4. Data reports by qualified code pressure vessel inspector, ASME Forms U-1, U-2, and U-3.

B. Inspection in the vendor's shops by the Purchaser, the government or their agents will be required at various stages of completion of the pressurizers. The successful vendor shall give prior notice to the Purchaser of such tests as the Purchaser may wish to witness, as mutually agreed upon between vendor and Purchaser.

5.0 TESTS AND INSPECTION, Con't.

C. Release of the equipment for shipment shall be authorized by the Purchaser after final shop inspection and approval for Code compliance by the qualified inspector.

6.0 DELIVERY

A. Delivery of equipment covered by these Specifications is desired in a minimum of time. Should use of premium space in the vendor's shop, extra shift work or other factors dictate a price differential in order to gain shorter delivery time, the bidder's proposal shall so state.

7.0 SPARE PARTS

A. The Proposal shall include a Spare Parts List recommending initial provisioning of spare parts, unit prices and agreement to hold these prices firm for concurrent delivery of spare parts with the Pressurizer Vessel. The A.E.C. will have the right to delete, increase, or decrease the recommended selection.

ADDENDUM NO. 1

TO

SPECIFICATIONS

FOR

PRESSURIZER VESSEL

The Stearns-Roger Mfg. Company

P. O. Box 5370 Terminal Annex

Denver 17, Colorado

for

The U. S. Atomic Energy Commission

Idaho Operations Office

Idaho Falls, Idaho

Prime Contract No. AT(10-1)-809

Stearns-Roger Order No. B-10300

ADDENDUM NO. 1
to
PRESSURIZER VESSEL

The following corrections, additions, and deletions are hereby made a part of the referenced specification. Bidders are reminded that receipt of Addendums must be acknowledged on the Invitation, Bid, and Award form:

1. Bidders are advised that the Pressurizer Vessel will be thermally cycled from completely cold to approximately 670°F an estimated 20 to 40 times per year.

2. Section 2.0, Part B, Item (4)
Change this item to read: "16- 3" flanged heating element receptacles on the sides encircling the lower section of the vessel".

3. Section 2.0, Part C, Second Paragraph
This paragraph should read: "The total rating of the heaters shall be not less than 200 KW, and shall be arranged for wiring of four-step control in increments of:

   Step One, 24 KW
   Step Two, 72 KW
   Step Three, 120 KW
   Step Four, 192 KW

   They shall also be wired such that it shall be possible to maintain approximately equal phase loading of the 3 phase, 240 volt, 60 cycle supply.

4. Section 2.0, Part C, Third Paragraph
Delete this paragraph in its entirety. The deleted paragraph reads: "The five heating elements will have a capacity of 24 KW each. They will be ....".

5. Section 2.0, Part C, Fourth Paragraph
Change this paragraph to read as follows: "Vapor-tight junction boxes shall be furnished and mounted at each point where electrical leads from the elements are brought through the vessel wall. The normal operating water level will be between the

ADDENDUM NO. 1
to
PRESSURIZER VESSEL, Con't.

5. high and low water levels indicated on Drawing 809-PER-609-M-6. Low water level shall be about 6'-9" above the bottom of the vessel and such that all heating elements shall always be covered. It is not planned that the water level will ever reach the low level except for momentary short periods. Control and indication of liquid level will be similar to conventional boiler drum level control.

6. Section 2.0, Part C, Fifth Paragraph
Delete this paragraph in its entirety. This paragraph reads: "Heat requirements are under minimum conditions ....".

XII. APPENDIX C
METALLURGICAL REPORTS

Phone JAckson 6-2079

M. E. HOLMBERG
METALLURGICAL CONSULTANT
4101 San Jacinto
HOUSTON 4, TEXAS

November 27, 1961

File: 381-61
Subj: Investigation of Failure in Pressurizer of Spert III Reactor at the National Reactor Testing Station, Idaho Falls, Idaho

Mr. C. E. Rawlins (5)
Phillips Petroleum Company
Engineering Department - Test Division
Bartlesville, Oklahoma

Dear Mr. Rawlins:

At the request of Paul Ogden, I visited the National Reactor Testing Station near Idaho Falls to investigate the failure in the pressurizer of the Spert III reactor.

After the investigation, Mr. Lyons asked that I furnish you a letter report to be forwarded to him. This is the requested report.

****

After reviewing the files on the pressurizer, examining the vessel, heating elements and associated piping; discussions with R.I. Kuehl and R.E. Heffner; and examination of preliminary metallographic specimens; the failure was discussed Monday afternoon, November 20th, at a meeting attended by:

J. Lyons	R. E. Heffner	J. H. Ronsick
T. R. Wilson	F. Schroeder	W. E. Nyer
R. I. Kuehl	M. H. Bartz	M. E. Holmberg

A summary of this meeting follows:

1. My opinions and observations at the time of the meeting were:

   A. The failure was a high temperature-high pressure failure in which the base plate in the shell section was in the process of failing by creep. As the plate yielded, the higher strength girth weld was over-stressed and cracked. The girth weld, in effect, failed by high temperature stress rupture.

   B. Failure had developed in the temperature range of 1000° - 1200°F. This was based on the discoloration of ground surfaces where thermocouples had been attached to the outside of the vessel, the black color of the fractures, and what I consider a hot short crack pattern in the Type 304 cladding.

   C. The spheroidized structure observed in the metallographic samples of the base plate might have slightly lowered the mechanical properties of the plate, but not enough to account for the failure. Indications were that the base plate was not defective.

D. Most of the cracks in the linings were surprisingly shallow considering their width. However, some of the cracks definitely extended through the lining. The cracks were not of the type that I would have expected from thermal shock or thermal fatigue. If thermal shock or fatigue were responsible, it seemed probable that they were formed by rapid heating that superficially overheated the inside surface for short periods of time.

2. It was explained the system operated under conditions that made it impossible to develop rapid heating, but that rapid cooling was possible.

3. It was stated that a review of the pressure and temperature charts showed no cases where the vessel had overheated or overpressured.

4. I questioned the accuracy of the pressure and temperature recording practices and the calibration of instruments. This doubt was prompted in part by a change several months earlier in the technique for checking pressures. It is understood this is to be investigated further.

5. As it is unlikely the instrumentation is off sufficiently to account for the high temperature-high pressure conditions believed responsible for the failure, the possibilities were then discussed of the pressurizer being heated at times when the recording instruments were not in service, and heated under conditions whereby extremely high pressure and temperature might develop. It is understood this is also being investigated further.

****

During the next two days, November 21st and 22nd, additional tests and investigations were made with Fred Prange. Results of tests previously started also became available. These developments are summarized below:

1. <u>Bend tests</u> made on specimens machined from the plate showed good ductility.

2. <u>Metallographic examinations</u> showed the brazing metal, attaching the Type 304 cladding to the base plate, was badly cracked. The brazing metal had also diffused slightly into the Type 304. --- The specimens were not examined before etching, but after nital etching, some grain boundaries were evident at the inside surface of the cladding. I believe these were tight intergranular cracks. There was slight evidence of surface oxidation, which was also present in some of the cracks. --- The root of some of the cracks in the cladding was broad, indicating strain had stretched them apart. --- Several very shallow cracks were noted extending from the bonding into the cladding. --- The grain size of the base plate was large and the carbides appeared spheroidized, but were typical of structures often observed in samples removed from pressure vessels.

3. <u>Room and high temperature tensile tests</u> with .505 specimens were made by an outside laboratory. Multiple tests were made with most of the specimens orientated to check the hoop strength of the shell section. A summary of the minimum yield and their corresponding tensile strengths is as follows:

	Yield Strength (psi)	Tensile Strength (psi)
Specification for A-212 Grade "B" plate	38,000 min.	70,000 min.
Test Results:		
Room Temperature	33,610	68,700
High Temperature - 400°F	35,780	60,000
- 600°F	33,100	67,200
- 668°F	31,620	62,950
- 700°F	31,700	62,200
- 800°F	28,800	53,430

The ductility as shown by elongation and reduction of area, was satisfactory for all specimens.

The room temperature yield and tensile strengths were below the minimum value specified for ASTM A-212, Grade "B" plate, but not enough to account for the failure. --- The high temperature yield strengths were about average for those on page 10 of the ASTM STP-100 "Report on The Strength of Wrought Steels at Elevated Temperatures" and well above the curves showing the lower limit for the yield strengths. --- The high temperature properties were surprisingly high and failed to offer an explanation for the failure up to 800°F.

4. Vessel Interior and Samples. Additional pieces were cut from the shell sections and from the top head. This permitted examinations of the inside of the vessel, and of additional sections. The results of these examinations are summarized below:

Base Plate - Except for stretching, no cracks or other evidence of failure was observed.

Center Girth Weld

a. Root of weld contained considerable porosity, which is not consistent with x-ray quality work; but evidently did not contribute towards the failure.

b. The largest cracks were on the outside surface and some had opened up about 3/16". The cracks did not extend appreciably into the base metal, which stretched and "necked" where the cracks had opened up.

c. A cut along the centerline of the center girth weld showed transverse root cracks (longitudinal to the vessel) which were perpendicular to the overlay. Some of these cracks had opened up with the result that the overlay had stretched and necked down, and in some cases cracked.

Cladding

a. Examination of the inside of the vessel showed the cladding contained several large longitudinal cracks and innumerable others.

b. The cladding had blistered at several areas. One of these blisters was about 18" in diameter.

Mr. C. E. Rawlins
Phillips Petroleum Co.
Bartlesville, Oklahoma

File: 381-61
Date: 11/27/61
Page: No. 4

Cladding (cont.)

c. Where blistered, the cladding showed few (if any) visible cracks.

d. In the lower half of the vessel, where creep had been less severe, the cladding showed fewer visible cracks. However, when a sample removed from this area was Zyglo inspected, the inside surface of the cladding was crazed with cracks. The most pronounced were several in the circumferential direction.

e. When the cladding from the top head (made from Lukens-clad plate) was Zyglo checked, no cracks were found.

5. Operating Time. It was reported the pressurizer had been subjected to 2500 psi and 668°F 39 times and to 2500 psi pressure a total of 60 times. In addition, it had been subjected to varying temperatures and pressures below the above approximately 100 times. The records indicate the pressurizer had been subjected to 600°F or above 600 or more hours.

6. Conditions at Time of Failure. When the 4" crack was observed in the insulation (shortly before the leak was discovered), the pressurizer was operating at 2500 psi and 668°F. The reactor was at 430°F and 2500 psi. About 20 minutes later, it was noted the crack in the insulation was increasing in length. About 2 hours after the crack was first noted, smoke was observed (blieved to have been caused from hot steam burning paper and cloth in the insulation) and the pressurizer was brought down in about 4 hours to 90°F. When the pressure had been reduced to about 1600 psi, a hissing sound was suddenly heard as though the vessel "gave" on cooling.

7. Information from Chicago Bridge & Iron Co. There were several questions concerning information available from CB&I. These were discussed by long distance with M. Smith at Birmingham. Don Bertosi, Metallurgist and Henry Wailes, Design Engineer, later called back with the information requested. The information developed from CB&I is as follows:

A. The Type 304-L cladding was brazed to the plate by heating the entire plate to 2050°F using a brazing material containing 92% nickel.

B. After brazing, the plates were cooled to atmospheric temperature and then normalized by heating to 1650°F and air cooled.

C. The tensile tests reported had Horton-clad plate Nos. H-1158 - 1161. CB&I reported these were tests on composite clad plate after normalizing and were as follows:

Size Plates	Horton-clad Plate No.	Yield (psi)	Tensile (psi)	% Elongation in 2"
6'11"	H-1158	41,900	78,300	42.0
	H-1159	41,900	81,900	38.0
5'11"	H-1160	40,000	77,600	44.0
	H-1161	38,800	71,600	49.0

Mr. C. E. Rawlins
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File: 301-61
Date: 11/27/61
Page: No. 5

D. The clad plates were hot formed by heating to 1650°F. Forming operations extended down to probably 1400°F.

E. The longitudinal direction of rolling the plates was orientated in the circumferential direction in the shell section.

F. CB&I would appreciate an opportunity to make an examination of material from the pressurizer. For their examination they would like to obtain approximately a 2' square section. This should be sent in care of Don Bertosi, Horton-clad Laboratory, Chicago Bridge & Iron Company, P.O. Box 277, Birmingham, Alabama. --- Correspondence concerning the failure and investigation by CB&I should be addressed to Mr. Henry Wailes at the same address.

8. Miscellaneous.

A. The check valve was tested after being disassembled. Upon reassembly and setting as removed from the vessel, it popped at 4300 psi on the first test and 3600 psi on the next test.

B. Discussions indicate that valve leakage would not likely result in circulation, or other conditions which would develop high pressures and temperatures.

C. There is a difference of approximately 3' between the minimum and maximum water level in the vessel.

D. Visual examination of the inside bore of the Type 304 flange on the outlet line from the top of the vessel failed to reveal any evidence of cracking.

TENTATIVE CONCLUSIONS AND COMMENTS

As a result of the above further investigations, my present opinions are:

1. Material.

A. The ASTM A-212, Grade "B" plate was not defective or responsible for the failure, even though tests made on the plate after failure showed the yield strength to be somewhat below minimum specifications.

B. The welds contain porosity and defects not consistent with x-ray quality, but were not responsible for the failure.

C. Type 304 Horton-clad plate is not suitable for the high temperature service encountered in the pressurizer. --- The Type 304 cladding cracked, crazed, and blistered. In addition, the brazed bonding was badly cracked. --- The exact reason for failure of the Horton-clad is not clear at this time. There is a possibility that it failed by thermal fatigue and/or shock on heating and cooling. Because of the low yield strength of Type 304 stainless and its high coefficient of expansion, it is possible that the cladding was slightly upset on each heating and developed surface cracks on cooling from high temperatures, especially during rapid cooling. --- There is also a possibility that the Type 304 cladding was damaged

Mr. C. E. Rawlins
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File: 381-61
Date: 11/27/61
Page: No. 6

during manufacturing operations. Further examination and study will be required to determine the exact cause of failure. --- It is characteristic of failures in high temperature service that carbon steel will stretch and deform; whereas austenitic steels crack with very little deformation. Indications are that shallow cracks developed at the inside surface of the Type 304-L cladding during some phase of manufacturing or service. When the carbon steel crept and the circumference increased, the bonding cracked. The austenitic steel apparently had sufficient ductility to stretch at the bottom of many of the cracks, with the result that many of the longitudinal cracks in the cladding opened up. However, it appears that the austenitic steel cladding did not have sufficient high temperature ductility to "follow" the shell plate completely and developed additional intergranular cracks when further strained. --- The Horton-clad shows evidence of slight surface corrosion and some intergranular cracking and corrosion. This requires further checking to verify and determine its seriousness.

D. Type 304 Lukens-clad appears to be satisfactory for the service; but the bottom head, which is apparently subjected to more severe thermal shock on cooling, should be Zyglo checked and examined before reaching a final conclusion.

2. The failure was a typical progressive type of high temperature failure with the shell section failing by creep while the weld failed by stress rupture. The failure was developing at a rapid rate when the leak occurred. It is possible that prior damage had weakened the vessel sufficiently that the maximum temperature and pressure to which it had been subjected were not required for failure to progress at the time leakage developed. I now believe failure developed within a temperature range of 800° - 1200°F instead of 1000° - 1200°F as originally estimated.

****

RECOMMENDATIONS

1. "Heat Tint" tests should be made in an effort to determine the temperature to which the shell of the pressurizer had been subjected.

2. Additional metallographic examinations should be made to more definitely determine the cause of failure of the Horton-clad liner and its suitability for high temperature service, not only with respect to the pressurizer, but for other high temperature applications. --- This should include a study of a sample from the bottom shell section where creep has been a minimum, but the cladding is crazed.

3. Make a Zyglo inspection of the bottom head, which was made from Lukens-clad Type 304-L and has been subjected to rapid cooling in service.

4. Make a metallographic examination of a sample from the top head to check the suitability of Type 304-L Lukens-clad plate for the high temperature service.

5. Consider the use of .5 chrome- .5 molybdenum Lukens-clad Type 304-L plate for the replacement vessel. This requires further evaluation and study. Pending the results of the above tests and investigations, the use of nickel clad plate may be preferred to Type 304-L.

Mr. C. E. Rawlins
Phillips Petroleum Co.
Bartlesville, Oklahoma

File: 331-61
Date: 11/27/61
Page: No. 7

6. Submit a sample of the Horton-clad plate and girth weld to CB&I for their study.

7. Make Zyglo checks at the inside bore of the pipe flanges for thermal stress cracks. It is believed the flange sections will be the most susceptible to thermal cracks.

8. Make a metallographic examination of the Type 304 pipe from the outlet at the top of the pressurizer to check for evidence of intergranular corrosion and thermal stress cracking. Operating conditions have been such that this piping has been highly stressed and its performance should be compared with that of the Horton-clad and Lukens-clad liners in the pressurizer. This affords an opportunity to evaluate the condition of the piping in the system.

9. If, when the new replacement vessel is obtained, it is still believed that abnormally high temperatures and pressures cannot be developed, controlled tests should be made in an effort to determine the conditions under which such pressures and temperatures can occur.

****

I believe the additional laboratory work and studies should be by the Test Division. However, if I can be of any assistance, I would welcome the opportunity to follow this problem and will appreciate being kept advised concerning further developments.

If there are any questions on this report, or I can be of further assistance, do not hesitate to contact me.

Very truly yours,

M E Holmberg

M. E. Holmberg

MEH:cr

cc: R. I. Kuehl

PHILLIPS PETROLEUM COMPANY

Bartlesville, Oklahoma

December 1961

Engineering Department

Failed Pressurizer
SPERT III Unit
Idaho Falls

Pra-191-61

Mr. C. E. Rawlins
O f f i c e

On November 21 and 22 I visited the Phillips Petroleum Company unit at Idaho Falls to investigate the cause of failure of the SPERT III pressurizer. This vessel is 2' 9" ID x 12' 7½" T to T. It was constructed of 2.83" thick A-212B steel clad with 1/8" of 304L (0.04% max. C). The vessel was built by and the shell plates clad by Chicago Bridge & Iron at Birmingham. The heads of 2.59" t A-212B clad with 304L were made by Lukens. The vessel was furnished with 16 immersion heaters each of 12 KWH capacity. The intended operating pressure was 2500 psia; the temperature, 668°F. The immediate cause of shutdown was leaking at the center girth weld. In the upper course of the vessel there was a maximum swelling of about 11%. The girth weld showed a swelling of about 3%, and the weld contained numerous deep cracks on the exterior. The base plate on each side of the girth weld was swelled more than the weld.

From the examination of the vessel, from the tests made by the Idaho Falls personnel, and from our laboratory work, I conclude that the vessel failed because of overheating or overpressurizing, or a combination of overheating and overpressurizing. Overheating was the most probable.

Some deficiencies were present in the vessel, but these did not materially affect the vessel in the mode in which it failed. These points are covered later in this letter. Recommendations covering a replacement vessel are also given.

MACRO EXAMINATION

Photographs of the vessel and measurements were made by the Phillips personnel at Idaho Falls. Only the most pertinent points will be mentioned here.

The upper course of the vessel was swelled about 11%; the girth weld joining the two courses, about 3%; and the upper part of the lower course slightly more than the girth weld. The girth weld showed many deep cracks, one of which allowed the vessel to leak. Typically, carbon steel weld metal has a high yield strength for a given tensile strength. Generally, the yield-to-ultimate ratio is above 0.8. Consequently, it would be more resistant to permanent expansion than base metal. At the same time, it is of lower ductility and would thus crack more readily than base metal. Thus, the performance of the girth weld was typical.

The cladding of the shell was pulled loose in several places on the top course of the shell. One area was about 2 ft. by 3 ft. in extent. The larger cracks were mainly longitudinal. Many showed a two-stage development--the crack near the surfaces covered with a heavy black scale. Extending somewhat deeper was a bright shiny appearance. Examination with Zyglo showed a mud-flat cracking pattern in the lower shell section, where the gross cracks were not present. In the upper part of the shell the cracks were generally further opened in the longitudinal direction than in the transverse direction. Significantly, the heads of Lukens-clad material showed no cracking.

At one location near the girth weld there was an area that indicates the sequence in which the cracking of the cladding occurred (Figure 1). The clue lies in the grinding at several points, two of which are shown. The cladding was touched by the grinding wheel when the inside girth seam was smoothed off. Some of the grinding was deep enough to remove the cracks. The black heavy scale from the previous operations was replaced by a dark tan transparent oxide film formed during stress-relieving at 1150F. This indicates that the main cracking resulting in the black oxide in the cracks occurred either during forming of the shell or during the brazing of the cladding (at 2050F).

METALLOGRAPHIC EXAMINATION

Metallographic examination of the A-212B base plate and the weld metal was unenlightening for explaining the failure of the vessel. The structure showed a coarse grain with distinct spheroidization of the pearlite (Figure 2). Some banding, not at all untypical of A-212 steel, was present (Figure 3). Small cracks were found originating near the interface of stainless weld metal and the first bead of automatic ferritic weld deposit (Figure 4). These could have been caused by differential expansion of the austenitic and ferritic materials during service of the pressurizer. Cross-sections examined in Bartlesville, as well as in the reactor machine shops, did not indicate these cracks to be the source of the major cracks in the weld metal. Instead the major cracks seemed to originate on the outside surface.

While metallographic examinations on the base material did not demonstrate major deficiencies, the metallographic examination of the cladding was more discouraging. A great deal of intergranular cracking was present, extending from the inner surface toward the brazing alloy joining the 304L to the base plate. The typical crack depth was about 0.02" (Figures 5 and 6). The black oxide noted in the macro examinations was present to this depth. Intense twinning of the surface grains--noted when severe thermal shock occurs--was not found. An intergranular precipitate, probably carbides from precipitation during the stress-relieving operation on the vessel, was present everywhere in the 304L.

The brazing alloy joining the austenitic cladding to the A-212B base plate also showed defects. CB&I had reported that this alloy was 92% nickel, with the other constituents undisclosed. On one micro specimen there was considerable porosity in the brazing alloy. On a second specimen cracking predominated, with the cracks emanating from an intermetallic phase (Figure 7).

 December 7, 1961

OTHER TESTS

A section of plate from the upper shell was belt sanded and then heated to determine the temperature at which a black shiny oxide formed. This color had been found on the shell of the pressurizer near the lower head. Here a spot had been polished for application of strain gages. Overnight exposure at 700F gave too shiny a scale, while overnight at 750F caused greater oxidation than observed on the pressurizer. This indicates that the lower shell of the pressurizer had not been overheated—at least for any appreciable length of time.

The Idaho Falls group had hot tensile tests made on steel removed from the pressurizer. Since the strengths measured are significant in analysis of the failure, they are repeated below:

Temperature	Yield Strength, psi	Ultimate Tensile Strength, psi
Room	33,610	68,700
669	31,620	62,950
700	31,700	62,200
800	28,800	53,430

For comparison, data on killed carbon steel are taken from "Timken Digest of Steels," Sixth Edition:

Temperature °F	Y.S. psi	U.T.S. psi	Stress to Produce Rupture in		
			10 hr.	100 hr.	1000 hr.
80	42,000	62,400	—	—	—
750	24,600	58,000	—	42,000	35,000
900	23,500	45,500	—	28,000	20,000
1000	20,000	36,500	25,000	18,000	13,000
1100	14,250	27,200	—	—	—
1200	10,200	20,000	8,100	5,000	—

To demonstrate the character of the stress-strain curve, we tested a bar of Bethlehem X2 sucker rod made of AISI 1036 steel. The results are shown in the illustration used to analyze the failure (Figure 8).

Hardness tests were made on a cross section $4\frac{1}{2}$" long, extending from the center of the weld. The weld itself had a hardness varying from Rockwell B 83 to B 87. The base plate varied in hardness from B 71 to B 73. Thus the weld metal was about 15,000 psi stronger than the base metal.

DISCUSSION OF THE FAILED PRESSURIZER

While the vessel did spring a leak, it performed admirably as a pressure container under the conditions to which it was subjected. That is, when it did finally fail, there was no brittle fracture, fragmentation, or large openings. Instead it stretched a great deal and gave warning of distress. In this respect, it performed better than most vessels that fail. As a pressure container there was no crippling deficiency.

Consequently, we must look further for explanation of the deformation. On the conservative formula for stress $\frac{PD}{2t}$, (D = outside diameter) the net stress on base material was 17,000 psi. Thus, there was almost a safety factor of 2 against yielding, since the base plate showed a yield strength of 31,000 at 668F. At 668F, the vessel could have withstood at least 5000 psi without visual evidence of overstress in the upper shell section. At 2500 psi pressure, we would also expect the vessel to withstand 1000°F momentarily without noticeable deformation and withstand rupturing for a period of 10 to 1000 hours.

One logical explanation of failure is overheating of the upper half of the vessel. One must conclude this from studying the deformation of carbon steels in the plastic range. The steel of Figure 8 illustrates the reasoning involved. Actually, because the yield strength is closer to the ultimate strength than was true for the pressurizer steel at 700F, this curve somewhat understates the relationship for the A-212B steel.

Let us assume, for the moment, that the pressurizer was at a temperature of less than 700F in both top and bottom halves. The 10-11% expansion experienced on the pressurizer would indicate stresses close to the ultimate strength of the material. This should also have resulted in swelling of the lower half of the vessel. That the lower half did not swell would indicate that the material of the lower half was stressed to a much lower stress or that it was not similar to the material of the upper half in its stress-strain characteristics. Lower stress could come only from greatly increased wall thickness. This can be checked readily, but it is not all likely.

Let us now examine the second postulate--that the material of the lower shell did not have similar stress-strain characteristics. The material as reported on pressure vessel test certificates indicated very similar analysis and properties. If this was true then the only way for the material of the lower shell to be appreciably stronger than the material of the upper shell was for the lower portion to be at lower temperature. If the analysis of the metal of the lower shell was such to make a much higher strength material, this could be checked by 700°F tensile tests of the metal from the lower portion of the pressurizer. If the test results indicated lower yield at 700°F than the material of the upper half, the demonstration of overheating would be overwhelming. If material was considerably stronger, there would be the possibility of overheating of the entire vessel. However, alloying agents would have to be present to account for the heat tinting displayed on the area polished for strain gages.

From the facts and deductions discussed above, I believe the failure of the vessel was caused by overheating. Because the weld metal was stronger than base metal, it did not allow so great a swelling at the girth seam as in base metal. Because the weld metal was less ductile (as is typical), it opened up before the base metal.

Some other points also need attention, although they had no significant part in the leaking of the vessel.

The room temperature tensile tests made for the Idaho Falls office showed a yield strength of 33,610 and a tensile strength of 68,700. Low strength was also confirmed by our hardness tests. The steel mills have always had some difficulty in maintaining the strengths of A-212 steel in thicker sections. Also it is difficult to evaluate the strength of a 3" thick plate by measuring the properties of a small part of the cross section. In addition, the forming of the shell and the stress relieving change the properties somewhat. This has been considered by Code-making bodies, but it was never deemed serious. In the instance of the SPERT III pressurizer, the 668°F properties were unexpectedly high. While the material used in the pressurizer does not fully conform to the specifications for room temperature strength, the strength at temperature was greater than expected. Consequently, we cannot say that the plate was unsuited for the design service conditions.

As was mentioned earlier, the cracks in the cladding were the result of the vessel manufacturer's operations--not the result of service at Idaho Falls. Hortonclad material should not be considered for further application on heavy wall pressurizers until Chicago Bridge and Iron Company understands the nature of the cracks found in the present vessel and uses necessary steps to prevent such damage in the future.

DISCUSSION OF REPLACEMENT VESSEL

Selection of material for a replacement vessel requires attention from several viewpoints. Since some minor cracking seemed to originate in the ferritic weld metal, caused by the differential expansion of the austenitic and ferritic materials, avoiding austenitic material may be desirable. Mr. Kuehl had suggested a nickel-clad steel. The mean coefficient of thermal expansion for steel in the range of 70-800F is 7.8×10^{-6} in./in./°F; for nickel about 8.2×10^{-6}; and for 18-8 stainless 10.1×10^{-6}. The combination of nickel and steel thus minimizes stresses and deformations resulting from temperature changes on dissimilar materials. Nickel cladding seems more desirable for cyclic temperature operation than does use of 18-8 stainless clad steel.

The base metal under the cladding is also in question. Appendix P of Section VIII of the Pressure Vessel Code gives the basis of design stresses for steels. A great deal of favorable performance has demonstrated the adequacy of this basis. Consequently, we believe that A-212B steel is satisfactory. However, in order to get the same or better high temperature properties than found in the steel of the failed pressurizer, it will be necessary to use some low-alloy steel. In the next A-212B steel we use we cannot rely on getting 700°F strengths that are as good as those found in the present vessel. In order to assure ourselves of better properties in the 700-800F range we will have to consider A-302 or A-387 steel. The first specification covers manganese molybdenum steel with 75,000 psi minimum TS in Grade A and 80,000 psi minimum TS for Grade B. The ½% Cr-½% Mo steel of A-387 has a tensile strength of only 65,000 psi. The use of the chrome-moly steel will require greater thickness than A-212 and appreciably increase cost of a replacement vessel. A third alternative is use of A-204B or C. Of these I would favor the B grade for best combination of fabrication and elevated temperature properties in the 700-800F range. For temperatures above

800°F, the chrome-moly material is recommended. This would give a good safeguard against runaway temperatures.

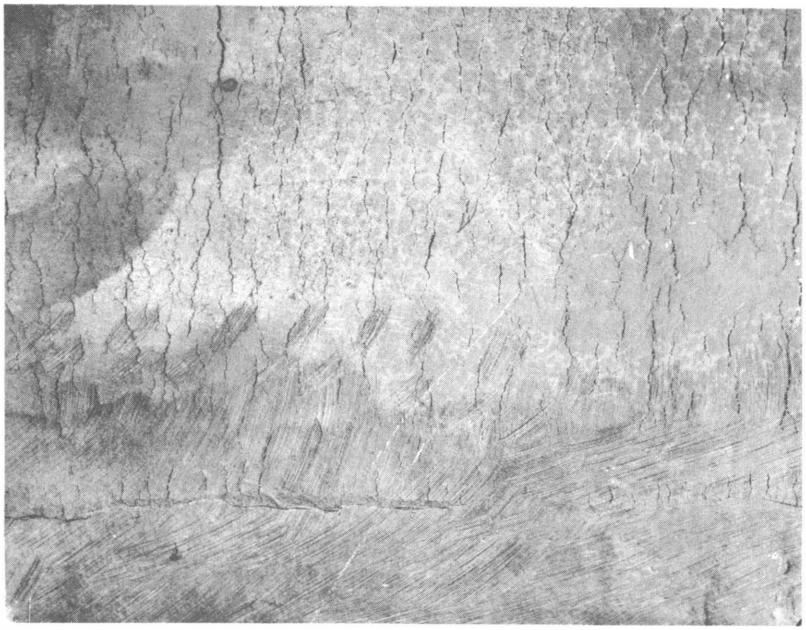
CHECKING PIPING

Mr. Holmberg in his letter suggested that the austenitic piping be checked. He has reported to me that the section from the pressurizer outlet that he examined by flattening showed no cracks or other distress. Consequently, there is no strong argument for checking the remainder of the piping system. My recommendation is that the piping near the pressurizer be checked for swelling at a few points. This would be a measure of overheating near the pressurizer. I see no need of thorough metallographic examination. Short-time overheating up to 1000°F will be of no consequence on the 18-8. Neither would any overpressuring that did not result in swelling the pipe cause damage to it.

F. Prange

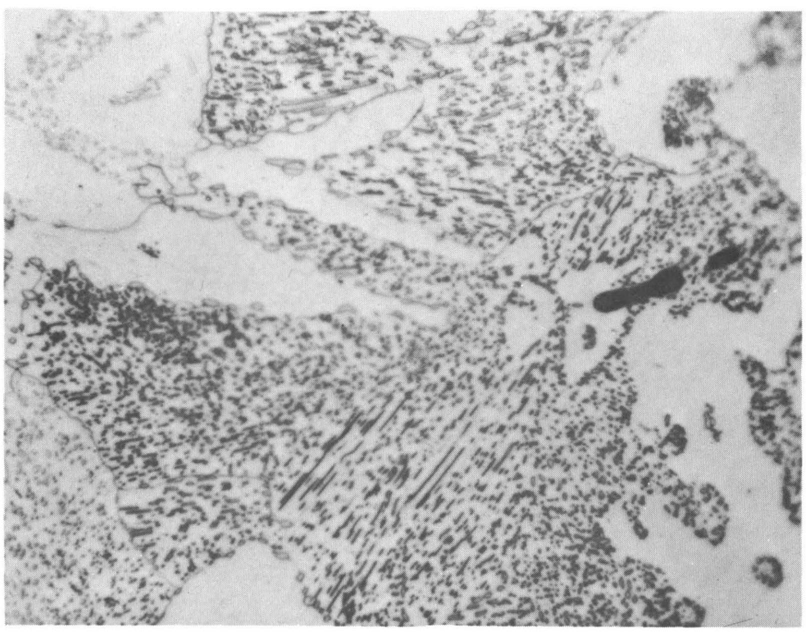
F. Prange

mjs
Enclosures



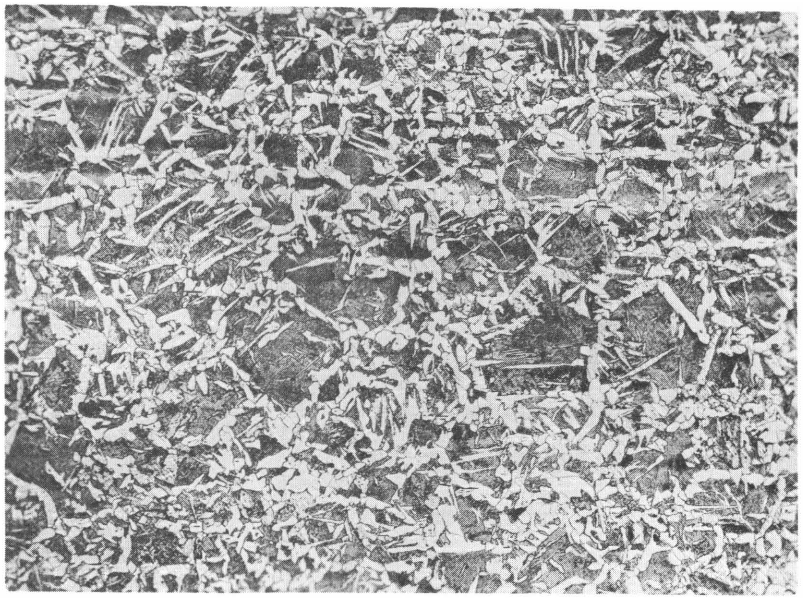
7/8X Mag. A-9144

Fig. 1 Areas where grinding removed cracks from surface of cladding.



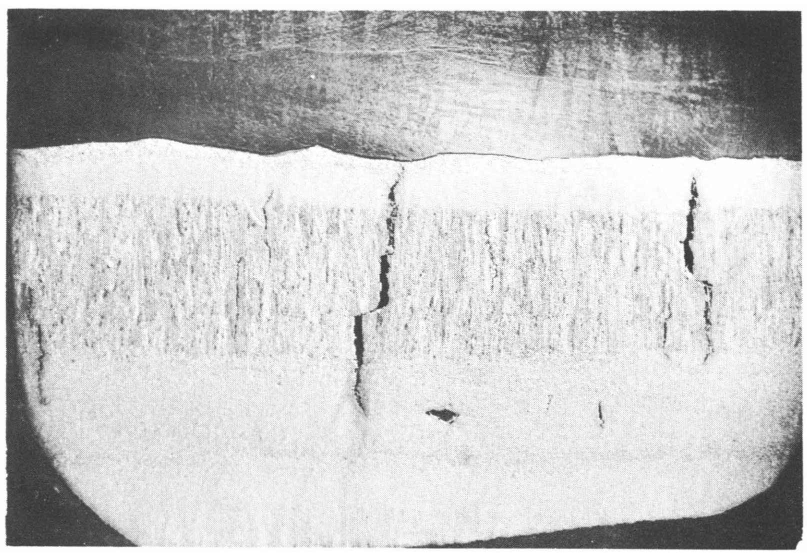
1000X Mag. Nital Etch E-4084

Fig. 2 Structure of the A-212B steel.



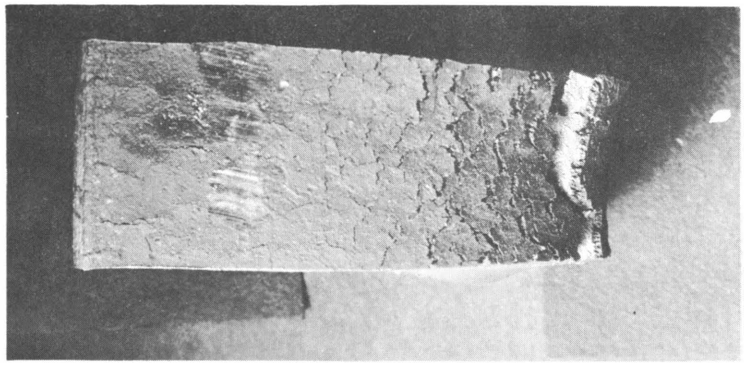
50X Mag. Nital Etch E-4085

Fig. 3 The steel was coarse-grained with some banding.



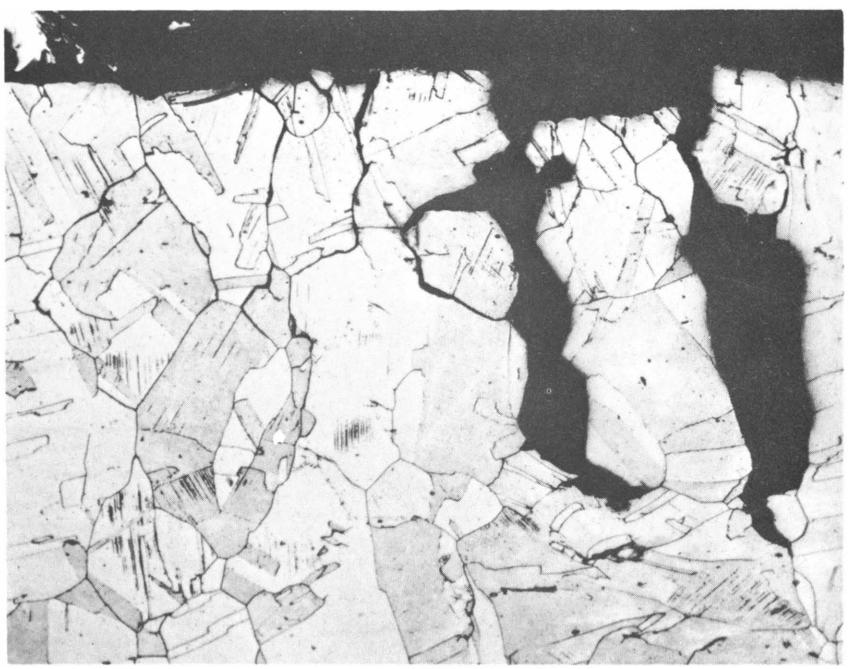
4-7/8X Mag. A-9145

Fig. 4 Cracks in ferritic weld metal starting at interface with austenitic weld metal.



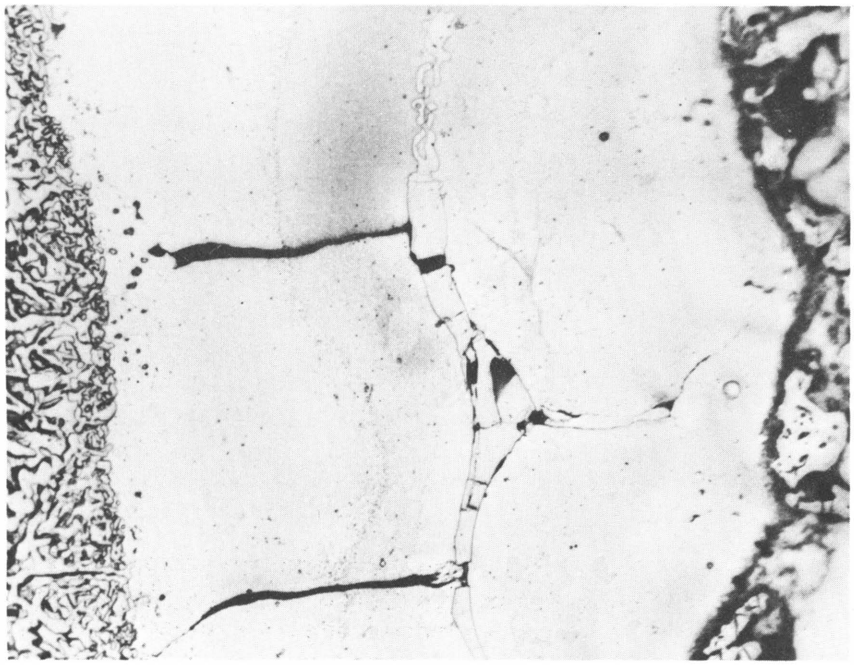
Full Size A-9146

Fig. 5 Cracks in the cladding. This bar was deformed to open up the cracks and show the black scale in them.



100X Oxalic Electrolytic Etch E-4086

Fig. 6 Cracking and structure of the cladding alloy.

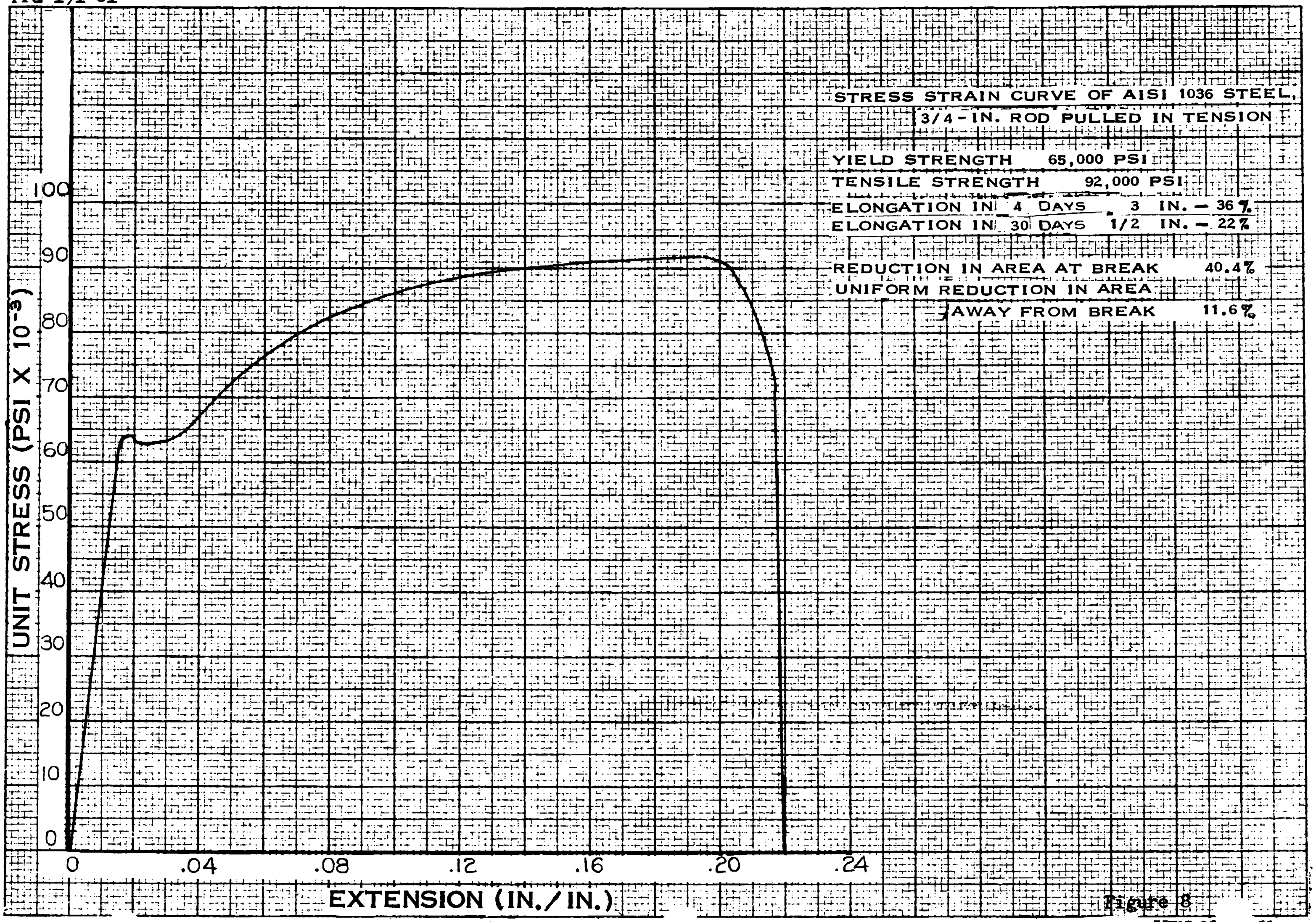


500X Mag. Glyceregia Etch E-4087

Fig. 7 Cracking in the brazing alloy. Carbon steel base metal at extreme right; brazing alloy diffusing into cladding at extreme left.

Mr. C. E. Rawlins
Pra-191-61

December 7, 1961



HWS 12 ?-61

Fig. 8 Stress-strain curve of AISI 1036 steel, 3/4-in. rod pulled in tension.

XIII. APPENDIX D

CALCULATIONS

1. CALCULATION OF PRESSURE RISE DURING RUN II-1A-4a(1)

(J. Dugone)

CALCULATION OF PRESSURE RISE DURING RUN II-1A-4a(1)

The pressure, level, and temperature which were recorded at the control center during Run II 1-A-4a(1) of the Spert III non-nuclear cold water accident tests are as follows:

Pressurizer pressure, PRC-6C, increased from 2485 to 2595 psig.

Pressurizer level, LRC-6C, increased from 7.64 to 8.33 feet.

Temperature, TRC-3C, increased from 444 to 454°F.

Temperature, TRC-4C, increased from 424 to 455°F.

The temperature increase recorded by TRC-4C was very large; however, this temperature was measured in the cold loop and should show a large increase.

The following calculations indicate that raising the bulk water temperature of the system from 444°F to 454°F should increase the water level in the pressurizer 10.9 inches. The actual level increase according to the strip charts, LCV–6R, was 8.3 inches. Assuming adiabatic expansion a level increase of 8.3 inches in the pressurizer would give a corresponding increase of pressure from 2500 psia to 2790 psia. This is larger than the pressure increase of 2485 psig to 2595 psig which was recorded during the test. The calculated pressure increase is larger than the measured because the heat lost to the pressure vessel and the possibility of leaks in the system are not taken into account in the calculations.

CALCULATIONS

Data from Strip Charts

Pressurizer pressure and level rise occurred during 20-min data-recording period.

Pressurizer pressure, PRC-6C, increased from 2485 to 2595 = 110 psi. Pressure level, LRC-6C, increased from 7.64 ft to 8.33 ft = 0.69 ft or 8.3 in. Temperature, TRC-3C, increased from 444°F to 454°F = 10°F. Temperature, TRC-4C, increased from 424°F to 455°F = 31°F. However, this was the cold loop and should show a large temperature increase.

Calculation of Pressurizer Level Rise at 450°F and 2500 psi *

Temperature °F	v_f ft^3/lb	v_1-v_f ft^3/lb	v_1 ft^3/lb
440	0.01926	-0.00029	0.01897
445	0.01935	-0.00030	0.01905
450	0.0194	-0.000305	0.019095
455	0.0195	-0.00031	0.01919

System volume:

Primary piping - 2904 gal.
Reactor vessel - 1605 gal.
4509 gal. or 602.76 ft.^{3} of water

Pressurizer at 7-1/2 ft - 380 gal.

Weight of water at 444°F:

The specific volume at 444°F = 0.01903 ft^3/lb.

$$\frac{602.76 \text{ ft}^3}{} \left| \frac{\text{lb}}{0.01903 \text{ ft}^3} \right. = 31{,}674 \text{ lb}$$

Volume of water at 454°F:

The specific volume at 454°F = 0.0192 ft^3/lb

$$\frac{0.0192 \text{ ft}^3}{\text{lb}} \left| \frac{31{,}674 \text{ lb}}{} \right. = 608.14 \text{ ft}^3$$

Increase in volume of water with temperature increase of 444°F to 454°F:

608.14 ft^3 of water at 454°F
602.76 ft^3 of water at 444°F
5.38 ft^3 increase in volume

Pressurizer volume as a function of height:

$$\pi r^2 h = \text{volume}$$

$$(3.14)\left(\frac{16.5}{12}\right)^2 \text{ft}^2 \ (1) \text{ ft} = 5.93 \text{ ft}^3/\text{ft of height}$$

*Symbols are defined on page 149.

Increase in pressurizer level:

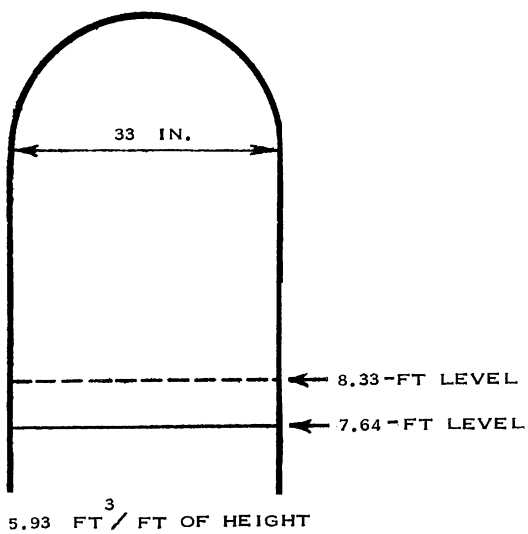
$$\frac{5.38\ \text{ft}^3}{} \Big| \frac{\text{ft of height}}{5.93\ \text{ft}^3} \Big| \frac{12\ \text{in.}}{\text{ft}} = \underline{10.9\ \text{in.}}$$

From J. F. Koenig's unpublished calculations:

The change in pressurizer level at 450°F = 1.1 in./°F. A 10°F ΔT = 11-in. increase in level which agrees with above calculation. The actual level increase according to the strip charts was 8.3 in.

Calculation of Pressure Increase with a Level Increase

As the bulk water of the system increases in temperature, it expands and level in the pressurizer rises. The volume of the steam decreases and the pressure increases. As the pressure increases, some steam is condensed and the latent heat will increase the temperature of the water which will increase the pressure. A schematic of pressurizer levels is shown in Figure 1.



Total volume of pressurizer = 74.9 ft^3

Water volume at 7-1/2-ft level= 49.7 ft^3

Water volume at 7.64-ft level = 50.5 ft^3

Steam volume at 7.64-ft level = 24.4 ft^3

Water volume at 8.33-ft level = 54.6 ft^3

Steam volume at 8.33-ft level = 20.3 ft^3

Fig. 1 Schematic of pressurizer levels.

Assume:

At start - Pressurizer level was 7.64 ft at 2485 psig or 2500 psia.

At end - Pressurizer level was 8.33 ft at ? pressure.

Adiabatic Process

Datum plane at 7.64-ft level

Water that rises the 0.69 ft to be at a temperature of 668°F.

Enthalpy Balance:

$$\left[(V_g\,\text{ft}^3)\left(\frac{1}{v_g}\,\frac{\text{lb}}{\text{ft}^3}\right)\left(H_g\,\frac{\text{Btu}}{\text{lb}}\right) + (V_l\,\text{ft}^3)\left(\frac{1}{v_l}\,\frac{\text{lb}}{\text{ft}^3}\right)\left(H_l\,\frac{\text{Btu}}{\text{lb}}\right) \right] \text{ at 2500 psia} =$$

$$\left[(V'_g\,ft^3)\left(\frac{1}{v_g}\frac{lb}{ft^3}\right)\left(H_g\frac{Btu}{lb}\right) + (V_1\,ft^3)\left(\frac{1}{v_1}\frac{lb}{ft^3}\right)\left(H_1\frac{Btu}{lb}\right)\right] \text{ at new pressure}$$

<u>7.64-ft level</u> - 2500 psia and 668.13°F

Steam

$$\frac{24.4\ ft^3}{} \Big| \frac{lb}{0.1307\ ft^3} \Big| \frac{1091.1\ Btu}{lb} = 203{,}700\ Btu$$

Water to be added

$$\frac{4.1\ ft^3}{} \Big| \frac{lb}{0.0287\ ft^3} \Big| \frac{730.6\ Btu}{lb} = 103{,}600\ Btu$$

Total enthalpy = 203,700 + 103,600 = 307,300 Btu

<u>8.33-ft level</u>

Steam

$$\left[20.3\ ft^3\,\frac{H_g}{v_g} + 4.1\ ft^3\,\frac{H_1}{v_1}\right]_{x\ \text{pressure}} = 307{,}300\ Btu$$

As shown in Figure 2, the pressure = 2802 psia

Material Balance:

<u>8.33-ft level at 2800 psia</u>

$$\frac{20.3\ ft^3}{} \Big| \frac{lb}{0.1035\ ft^3} + \frac{4.1\ ft^3}{} \Big| \frac{lb}{0.0315\ ft^3} = 326.293\ lb$$

<u>7.64-ft level at 2500 psia</u>

$$\frac{24.4\ ft^3}{} \Big| \frac{lb}{0.1307\ ft^3} + X = 326.293\ lb$$

$X = 326.293 - 186.687 = 139.606$ lb

Enthalpy Balance:

<u>7.64-ft level at 2500 psia</u>

Steam = 203,700 Btu

Water = (139.606 lb) (730.6 Btu/lb) = 101,996

Total = 203,700 + 102,000 = 305,700 Btu

<u>8.33-ft level</u>

As shown in Figure 2, the pressure = <u>2790 psia</u>

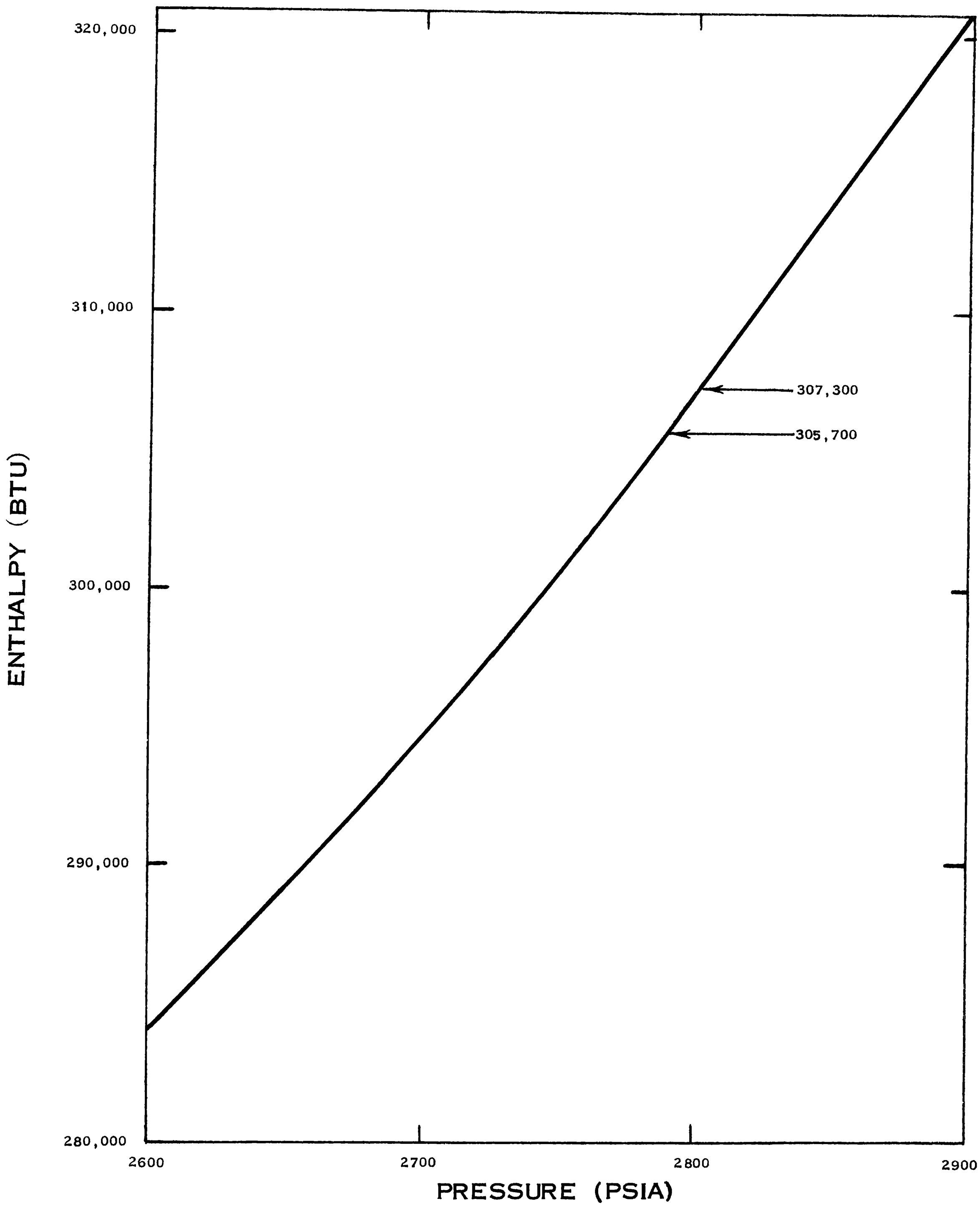


Fig. 2 Enthalpy balance vs pressure for pressurizer.

The calculations indicate that a level increase of 0.69 ft in the pressurizer would give a corresponding increase of pressure from 2500 psia to 2790 psia. This is assuming adiabatic expansion. Some heat would be lost to the pressurizer vessel which would result in a lower pressure. This would account for the fact that the pressure increase noted by the pressurizer pressure recorder (PRC-6C) was from 2485 to 2595 psig.

SYMBOL DEFINITIONS

v_f = Specific volume of saturated liquid (ft^3/lb)

v_l = Specific volume of liquid (ft^3/lb)

r = Radius of vessel (ft)

h = Height of vessel (ft)

$\triangle T$ = Change in temperature (°F)

V_g = Volume of gas (steam) (ft^3)

v_g = Specific volume of gas (steam) (ft^3/lb)

H_g = Enthalpy of gas (steam) (Btu/lb)

V_l = Volume of liquid (ft^3 at 2500 psig)

H_l = Enthalpy of liquid (Btu/lb)

XIII. APPENDIX D

CALCULATIONS

2. PRESSURIZER VESSEL DESIGN CALCULATIONS.

(J. A. Norberg)

SPERT III PRESSURIZER VESSEL DESIGN CALCULATIONS

Maximum Stress due to Internal Pressure.

Normal pressure = 2500 psig

Membrane stress due to internal pressure (*) in a thin wall vessel

$$S = \frac{P(D+a)}{2a}$$

where
S = membrane stress (psi)
P = internal pressure (psi)
D = inside diameter of vessel (in.)
a = vessel wall thickness

Neglecting the 1/8-in. clad:

$$S = \frac{(2500)\ (33.25 + 2.825)}{(2)\ (2.825)} = 15{,}960 \text{ psi}$$

Peak stress at inside vessel wall due to internal pressure: (*)

$$S = \frac{2P \left[\frac{D}{2} + a\right]^2}{a\ (D + a)}$$

Using same symbol definitions as above and again neglecting the 1/8-in. clad:

$$S = \frac{(2)\ (2500)\ (16.625 + 2.825)^2}{(2.825)\ (33.25 + 2.825)} = 18{,}560 \text{ psi}$$

Maximum stress for a thick wall vessel as calculated in Roark.(**)

$$Smax = P \left[\frac{(b^2 + a^2)}{(b^2 - a^2)}\right]$$

where:

P = internal pressure (psi)
b = outside radius of vessel (in.)
a = inside radius of vessel (in.)

(*) B. F. Langer, Design Basis for Reactor Vessels, WAPD-CE-43 (October 18, 1955).

(**) R. J. Roark, Formulas for Stress and Strain, 3rd Ed., p 268, New York, McGraw-Hill (1954).

Neglecting the 1/8-in. clad:

$$Smax = (2500) \left[\frac{(19.45)^2 + (16.625)^2}{(19.45)^2 - (16.625)^2} \right] = 16{,}100 \text{ psi}$$

From ASME Unfired Pressure Vessel Code (***)

$$S = \frac{P(R + 0.6t)}{E\ t}$$

where: S = max allowable stress (psi)

P = internal pressure (psi)

t = thickness of shell (in.)

R = inside radius of vessel (in.)

E = joint efficiency

Neglecting the 1/8-in. clad:

$$S = \frac{(2500)\ [16.625 + 0.6\ (2.825)]}{(0.95)\ (2.825)} = 17{,}100 \text{ psi}$$

Note: The maximum allowable stress for SA-212-grade B steel is 17,500 psi at 650°F and 16,600 psi at 700°F.

Stress for yield at 2650 psig (from Roark**)

Neglecting the 1/8-in. clad:

$$Smax = (2650) \left[\frac{(19.45)^2 + (16.625)^2}{(19.45)^2 - (16.625)^2} \right] = 17{,}020 \text{ psi}$$

(***) "Unfired Pressure Vessels, Section VIII", ASME Boiler and Pressure Vessel Code, The American Society of Mechanical Engineers, Secs. UCS-23, UW-12, UG-27, New York (1959).

(**) R. J. Roark, Formulas for Stress and Strain, 3rd Ed., p 268, New York, McGraw-Hill (1954).

XIII. APPENDIX D

CALCULATIONS

3. PRESSURIZER VESSEL THERMAL STRESS - PRELIMINARY CALCULATIONS

(J. A. Norberg)

SPERT III PRESSURIZER VESSEL
THERMAL STRESS - PRELIMINARY CALCULATIONS

Normal Operating Conditions:

Basis: Cool down rate = 100°F/hr

Heat up rate = 100°F/hr

Starting temperature = 70°F

Final pressure = 2500 psig

For the maximum thermal stress consideration, assume the inside wall to have an infinite heat transfer film coefficient, h, and that the outside wall is perfectly insulated.

From References A and B(*)

Physical properties of pressurizer

Material: SA - 212 - grade B

Coefficient of linear expansion: 7.45×10^{-6} in./in.°F (assume same as SAE-1035)

Density: 7.85 gm/cm^3 = 0.283 $lbs/in.^3$ = 490 lbs/ft^3

Modulus of elasticity: 30×10^6 psi (conservative)

Poisson's ratio: 0.3 (conservative)

Minimum tensile stress: 70,000 psi

Allowable Stress

Temperature (°F)	-20 to 650	700	750	800	850
Stress (psi)	17,500	16,600	14,750	12,000	9,250

Thermal Conductivity

Temperature (°F)	32	212	392	572	752
Thermal Conductivity $\left(\frac{Btu}{Hr\ Ft°F}\right)$	30	29.4	28.1	26.5	24.4

Specific Heat

Temperature (°C)	50-100	150-200	200-250	250-300	300-350	350-400
Heat Capacity $\left(\frac{Btu}{Lb°F}\right)$	0.116	0.1235	0.1265	0.132	0.1365	0.1415

(*) References A through E are listed on page 162.

Using the method shown in a report by Heisler, [C] the maximum cooling rate is calculated as follows:

The maximum stress will occur when the inner surface temperature is cooled to 70°F.

$$t(1,N) = 668°F - 70°F = 598°F$$

where $t(1,N)$ = wall temperature change

$$\frac{(1-\mu)\sigma(1,N)}{\alpha E\ t(1,N)} = \frac{(1-0.3)\ (16{,}600\ \text{psi})}{(7.45 \times 10^{-6}/°F\ \ (30 \times 10^{6}\ \text{psi})\ \ (598°F)} = 0.087$$

where $\sigma(1,N)$ = allowable stress (psi)

α = linear coef. of thermal expansion (in./in.°F)

μ = Poisson's ratio

E = modulus of elasticity (psi)

Assuming an infinite heat transfer coefficient

$$\frac{1}{m} = \frac{hL}{k} = \infty \quad \text{(most conservative case)}$$

Assuming $h = 1000\ \text{Btu/hr-ft}^2\text{°F}$

$$\frac{1}{m} = \frac{(1000)\ (0.246)}{28} = 8.9 \quad \text{(most probable case)}$$

where h = film coefficient (Btu/hr-ft^2 - °F)

L = vessel wall thickness (ft)

k = thermal conductivity (Btu/hr-ft^2 - °F/ft)

From Figure 8 of Reference C:

$$N \cong 4.0 \text{ for } \frac{1}{m} = \infty \text{ and } \frac{(1-\mu)\ \sigma\ (1,N)}{\alpha E\ t\ (1,N)} = 0.087$$

and

$$N \cong 4.1 \text{ for } \frac{1}{m} = 8.9$$

From Figure 5 of Reference C:

$$T_2\ (1,N) = \frac{t\ (1,N)}{cL^2/d} = 4.1 \text{ for } \frac{1}{m} = \infty$$

$$T_2\ (1,N) = \frac{t\ (1,N)}{c\ L^2/d} = 3.9 \text{ for } \frac{1}{m} = 8.9$$

where c = rate of change of temperature (°F/hr)

d = thermal diffusivity (ft^2/hr)

$$\frac{1}{m} = \infty, \; c = \frac{(598)}{(0.246)^2 \left[\frac{(0.125)\;(490)}{(28)}\right] (4.1)} \cong \frac{(598)}{(0.246)^2\;(2.19)\;(4.1)} \cong 1100°F/hr$$

$$\frac{1}{m} = 8.9, \; c \cong (1100)\;\frac{(4.1)}{(3.9)} \cong 1160°F/hr$$

The 100°F/hr for cooling is very conservative.

Maximum Heating Rate:

For the maximum heating rate, the maximum thermal stress will occur when the inner surface temperature reaches 668°F. At this time the internal pressure is also a maximum and must be accounted for.

To account for the internal pressure stress, the allowable stress, $\sigma(1,N)$, will be lowered by an amount equal to the stress caused by the internal pressure as given by the following formula from Reference D.

$$Smax = P\left[\frac{b^2 + a^2}{b^2 - a^2}\right]$$

where P = internal pressure (psi)

b = outside radius of vessel (in.)

a = inside radius of vessel (in.)

$$Smax = (2500)\left[\frac{(19.45)^2 + (16.625)^2}{(19.45)^2 - (16.625)^2}\right] = 16{,}100 \text{ psi}$$

16,600 - 16,100 = 500 psi

where 16,600 psi is the allowable stress at 700°F

$$\frac{(1-\mu)\;\sigma(1,N)}{\alpha E\; t(1,N)} = \frac{(0.7)\;(500)}{(7.45 \times 10^{-6})\;(30 \times 10^{6})\;(598)} = 0.0036$$

Then for:

$\frac{1}{m} = \infty$, from Figure 8 of Reference C, N $\cong$ 90 (estimation)

And from Figure 5 of Reference C, $T_2(1,N) \cong 90$ (extrapolating)

$$c = \frac{598}{(0.246)^2\;(2.19)} = 50°F/hr$$

This method of calculation shows a heat-up rate of 50°F/hr would be required to keep the thermal stress below 500 psi which when added to the maximum stress due to internal pressure results in a combined stress of 16,600 psi (allowable at 700°F). However, the maximum pressure stress occurs only when the system pressure is 2500 psig. For any pressure less than this, also corresponding to a lower temperature, the combined stress is less than the 16,600 psi allowable.

To estimate the effect of increasing the heating rate to 100°F/hr the following calculation is made:

$$\frac{1}{m} = \infty$$

$$T_2(1,N) = \frac{t(1,N)}{cL^2/d} = \frac{598}{(100)\ (0.246)^2\ (2.19)} \cong 45$$

Assuming that Figure 5 of Reference C extrapolates to this region, N $\cong$ 45.

From Figure 8 of Reference C, again extrapolating:

$$\frac{(1-\mu)\sigma\ (1,N)}{\alpha\ E\ t\ (1,N)} \cong 0.0072$$

$$\sigma(1,N) = \frac{(0.0072)\ (7.45 \times 10^{-6})\ (30 \times 10^{6})\ (598)}{0.7} = 1375 \text{ psi}$$

Adding the thermal stress to the circumferential stress results in a combined stress of 17,475 psi which is slightly above the allowable of 16,600 psi for 700°F and slightly below the allowable of 17,500 psi for 650°F. However, it is considerably below the yield stress of 35,000 psi. Plant operating experience and data have shown that the heating rate of the pressurizer between 635°F (~2000 psig) and 668°F (~2500 psig) is ~60°F/hr. This is only 10°F/hr above the calculated heating rate to hold the combined maximum stresses below the allowable of 16,600 psi at 700°F.

Another method of calculating the thermal stress is shown in the paper by Langer (E). Using the cooling rate calculated by the previous method (C), ie, 1100°F/hr., the stress is calculated as follows:

Thermal stress produced by temperature transients:(E)

$$S = \frac{E\alpha\theta}{1-\mu} \text{ x(factor)},\ \text{lbs/in.}^2$$

(value of factor found from Figure 2 of Reference E)

where:

S = equivalent intensity of combined stress (psi)

E = modulus of elasticity (psi)

α = linear coefficient of thermal expansion (in./in.°F)

μ = Poisson's ratio (dimensionless)

θ = temperature decrease at inner wall (°F)

For a cooling rate of 1100 °F/hr

$$S = \frac{(30 \times 10^{6})\ (7.45 \times 10^{-6})\ (598)}{(1 - 0.3)} \times (\text{factor})$$

$$S = 19.1 \times 10^{4}\ (\text{factor})$$

From Figure 2 of Reference E:

$$\tau = \frac{\epsilon t}{a^2} = \frac{(0.456)\ (598/1100)}{(0.246)^2} = 4.09$$

where ϵ = diffusivity (ft^2/hr)

t = time for temperature change (hr)

a = vessel wall thickness (ft)

Factor $\cong$ 0.083 for a $\frac{k}{ha}$ of 0 ($h = \infty$)

$$S = (19.1 \times 10^{4})\ (0.083) = 15{,}850 \text{ psi}$$

This stress is very close to the 16,600-psi stress used in the previous calculations where a cooling rate of 1100°F/hr was obtained.

In order to obtain a thermal stress sufficient to cause the pressurizer to yield, a very rapid temperature change must occur. Operation of the Spert III reactor plant to date has shown that it is impossible to heat the pressurizer with its heaters at a sufficient rate to cause yielding from thermal stress. However, it may be possible to obtain a sufficient thermal stress to yield the vessel by very rapid cooling.

Limiting Thermal Stress Conditions

Some limits can be set as to the conditions required for such a thermal stress. These calculations are shown below:

Basis: yield stress = 35,000 psi.

Minimum temperature drop: (Reference E)

$$S = \frac{E\alpha\ \theta}{(1 - \mu)} \times (\text{factor})$$

$$\theta = \frac{(S)\ (1 - \mu)}{E\alpha\ (\text{factor})}$$

Assume the drop to be instantaneous with the film coefficient = ∞. From Figure 2 of Reference E, the factor = 1.0

$$\theta = \frac{(35{,}000)\ (0.7)}{(30 \times 10^6)\ (7.45 \times 15^6)\ (1)} = 109.5°F$$

In order to yield the vessel, the instantaneous temperature drop must be ~109.5°F. Say ~100°F. This is a very conservative estimate. To show how much the film coefficient, h, and the time for temperature change, t, affect the required temperature change to yield the vessel, consider the following:

h, $\frac{\text{Btu}}{\text{hr-ft}^2\text{-°F}}$	Temperature Change $\Delta T(t = 0)$, °F	$\Delta T(t = 30\ \text{sec})$ °F	$\Delta T(t = 60\ \text{sec})$,°F
10,000	118	140	155
5,000	128	144	157
1,000	193	196	203

The cooling time required to obtain vessel yield through thermal stress using the maximum temperature drop of 598°F can be calculated as a function of the heat transfer coefficient, h, as follows:

$$\text{Factor} = \frac{(S)\ (1 - \mu)}{E\ \alpha\ \theta} = \frac{(35{,}000)\ (0.7)}{(30 \times 10^6)\ (598)\ (7.45 \times 10^{-6})} = 0.183$$

For a factor of 0.183, from Figure 2 of Reference E,

where $\tau = \frac{\epsilon t}{a^2}$, $t = \frac{(\tau)\ (a2)}{\epsilon}$

Then the time for temperature change, t, = $\frac{(\tau)\ (0.25)^2}{0.456}$

and the maximum cooling time as a function of the heat transfer coefficient, h, is related as shown below:

h, $\frac{\text{Btu}}{\text{hr-ft}^2\text{-°F}}$	τ	Maximum Cooling Time
∞	1.58	0.216 hr = 12.95 min
10,000	1.58	0.216 hr = 12.95 min
5,000	1.56	0.214 hr = 12.85 min
1,000	1.55	0.212 hr = 12.7 min

It can be concluded, from the above calculations, that a 100°F temperature drop must occur in less than 13 min to yield the vessel from thermal stress.

REFERENCES

A. "Unfired Pressure Vessels, Section VIII", ASME Boiler and Pressure Vessel Code, The American Society of Mechanical Engineers, Secs. UCS-23, UW-12, UG-27, New York (1959).

B. Taylor Lyman, Ed., Metals Handbook, ASM, p 307 (April 1960).

C. M. P. Heisler, "Transient Thermal Stresses in Slabs and Circular Pressure Vessels", J. Appl. Mech., 20: No. 2, p 261 (June 1953).

D. R. J. Roark, Formulas for Stress and Strain, 3rd Ed., p 268, New York, McGraw-Hill (1954).

E. B. F. Langer, Design Basis for Reactor Vessels, WAPD-CE-43 (October 18, 1955).

XIII. APPENDIX D

CALCULATIONS

4. THERMAL SHOCK CALCULATIONS FOR PLANT UPSET

(J. A. Norberg)

THERMAL SHOCK CALCULATIONS FOR PLANT UPSET

A review of the operating history of the Spert III pressurizer has shown that on several occasions very rapid depressurizations occurred (see Table IV of text).

The following calculations were made on what is believed to have been the most severe plant upset from the thermal shock aspect.

Calculational Method:

The time-temperature relationship of the inner surface of the pressurizer vessel wall was assumed to follow the saturation temperature for the corresponding depressurization. This information was obtained from plant instrumentation records and used as input data to an IBM 650 transient heat conduction code to determine the time-temperature distribution through the vessel wall. This code solves the one-dimensional heat diffusion equation through multiregion materials.

Thirty-four mesh points were used: three for the stainless-steel clad, ten for the carbon steel shell and ten for the insulation. The inner surface boundary condition was supplied as input data (explained previously) and the outer surface boundary condition (insulation to air) was calculated from standard engineering equations. No contact resistance was assumed between the clad-shell interface and the shell-insulation interface. Each region used the physical properties of its respective material and allowed for the temperature dependence of these properties.

The thermal stress was estimated by solving the following equation from Timoshenko. (*)

$$\sigma_\theta = \frac{\alpha E}{1 - \nu} \frac{1}{r^2} \left[\frac{r^2 + a^2}{b^2 - a^2} \int_a^b Trdr + \int_a^r Trdr - Tr^2 \right]$$

where

σ_θ = tangential stress (psi)

α = coefficient of thermal expansion (1/°F)

E = modulus of elasticity (psi)

ν = Poisson's ratio

r = radius (in.)

a = inner vessel wall radius (in.)

b = outer vessel wall radius (in.)

T = temperature (°F)

(*) S. Timoshenko and J. N. Goodier, Theory of Elasticity, 2nd Ed., p 412, New York, McGraw-Hill (1951).

For this estimation, the physical properties (α, E and ν) were considered to be that of the carbon steel. More refined stress calculations are in progress utilizing the IBM 650 to account for the two metal regions of the vessel wall and their individual physical properties.

The time of maximum temperature differential across the vessel wall was chosen for the stress calculation. Figure 1 shows the vessel wall temperature distribution and its associated stress for the time of maximum temperature difference. The solution of the stress equation was accomplished by hand calculation using the trapezoidal rule for integration. As can be seen from Figure 1, the maximum stress (at the inner surface) due to thermal shock is about 30,500 psi. Another calculation was made on just the carbon steel, (neglecting the stainless-steel clad) using the same temperature distribution. The maximum thermal stress for this condition was about 26,300 psi which corresponds closely to the stress at the stainless-steel clad-carbon steel interface shown in Figure 1. These calculations are considered to be conservative, that is, the calculated stress is probably higher than was actually incurred since the heat transfer rate from the saturated steam to the vessel wall was neglected (considered ∞). Any heat transfer rate less than ∞, which was undoubtedly the actual case, would reduce the thermal shock effect.

Adding the stress at the inner surface due to the system pressure results in a total stress of about 33,000 psi for the case including the clad, and about 29,000 psi if the clad is neglected.

The conclusions from these calculations are (1) It is unlikely that the pressurizer has undergone thermal shocks of sufficient severity to yield the carbon steel shell, and (2) the stainless-steel clad may have been damaged from the most severe plant upsets.

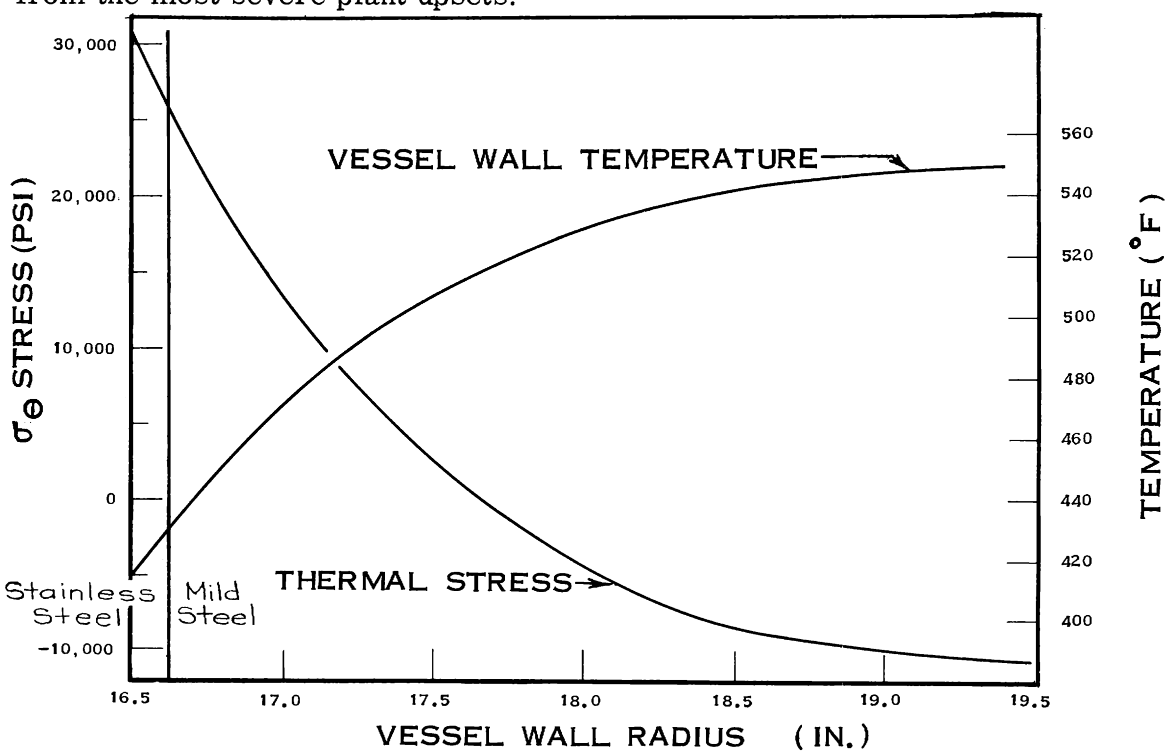


Fig. 1 Thermal stress on pressurizer due to sudden loss of pressure which occurred at 1825 on 3-23-59.

XIII. APPENDIX D

CALCULATIONS

5. CREDIBILITY OF DAMAGE TO SPERT III PRESSURIZER BY A TRANSIENT PRESSURE PULSE

(S. O. Johnson)

The following is an examination of the credibility of damaging the Spert III pressurizer to the observed extent by a pressure pulse generated during a transient test. The approach used is to first estimate the net work done on the pressurizer in distorting it to its present shape. If the observed distortion of the pressurizer was caused by a pressure pulse, the pulse must have contained at least an amount of energy equivalent to the work done on the pressurizer.

To determine the work done on the pressurizer, a minimum value for the average circumferential unit strain was obtained from measurements of the vessel. It was assumed that no strain occurred above 11.25 in. below the top or below 76.5 in. below the top of the pressurizer, giving a strained length of 65.25 in. It was also assumed that the initial circumference of the strained portion of the vessel was 10 ft, 5.75 in. (as measured on the lower part of the vessel). This value for the initial circumference is somewhat lower than that indicated on the manufacturer's drawing, and hence its use may underestimate the strain. The measured circumferences, elongations, and unit strains are summarized in Table 1.

TABLE 1

Distance from Top of Pressurizer (in.)	Measured Circumference	Elongation	Unit Strain ϵ
11 1/4		0 (assumed)	0
16 1/2	10' 10 1/2"	4 3/4"	0.0378
30	11' 1 7/8"	8 1/8"	0.0646
44	11' 2 3/4"	9"	0.0716
64 3/4	10' 9 3/4"	4"	0.0318
76 1/2	10' 5 3/4"	0 (assumed)	0

The average unit strain was determined by

$$\bar{\epsilon} = \frac{1}{10'\ 5\ 3/4''} \int_{11\ 1/4''}^{76\ 1/2''} \epsilon(x)\, dx \quad ,$$

where the integral was evaluated by the trapezoidal rule. This procedure tends to underestimate the average strain, and yields a value of $\bar{\epsilon} = 0.031$.

Tests of material taken from the pressurizer wall have indicated a yield point of ~ 30,000 psi at temperatures up to 800°F. For the purpose of estimating the work done, the stress-strain curve was assumed to follow Young's modulus ($30\ (10^6)$ psi) to the yield point and then remain constantat 30,000 psi for all strains. Since the strain at the yield point is only 0.001, the area under the stress-strain curve was taken to be simply the yield point stress multiplied by the average strain, or 30,000 x 0.031 psi. The cross-sectional area perpendicular to the direction of strain was taken as 184.5 sq in., assuming a wall thickness of 2.825 in. The initial mean circumference was assumed to be 9.75 in. Then, the work done

$$W = (\text{area under stress-strain curve}) \cdot (\text{cross-section}) \cdot (\text{mean circumference}).$$

This procedure yielded an estimated work of 1.7 (10^6) ft-lb or about 0.6 kwh.

If a triangular pressure pulse of base τ and amplitude P_{max} is assumed, the total energy contained in it is

$$E = \frac{1}{2} P_{max} \tau V A \quad ,$$

where V is the velocity of propagation, and A is the area of the wave front. The pressure pulse must be transmitted through the pipe connecting the pressurizer to the pressure vessel. Hence the area was taken as that of a 4-in. pipe, or 12-1/2 sq in. The propagation velocity was assumed to be 5000 ft/sec, the velocity of sound in cold water. Setting the energy of the pulse equal to the work done on the pressurizer yields

$$P_{max} \tau \simeq 50 \text{ psi-sec} \quad .$$

This means that, for example, a pressure pulse having at least a maximum pressure of 5000 psi and a width of 10 msec would be required to produce the observed damage.

Since no pressure pulses having a P_{max} τ approaching 50 psi-sec have been observed in any Spert tests, it appears highly improbable that a simple pressure pulse arising from a transient test could have caused the observed damage. In addition, the above approach assumes (1) that the pressure pulse is transmitted through the piping (~ 15-ft long with several right angle bends) with no attenuation, and (2) that all of its energy is dissipated in distorting the pressurizer. Both of these assumptions are quite unrealistic.

The possibility of progressive damage by several lesser pressure pulses also seems remote. First, the amplitude of each pressure pulse entering the pressurizer must be sufficient to cause a stress beyond the yield point. Second, unless the ultimate stress is exceeded in some portion of the vessel wall (no cracks have been found), each pressure pulse would tend to strengthen the vessel by producing a residual compressive stress in the wall. (Timoshenko, "Strength of Materials - Part II", pp. 291-2.)

Thus, the postulate that the damage to the Spert III pressurizer was caused by pressure pulses generated during transient tests does not appear to be credible.

XIII. APPENDIX D

CALCULATIONS

6. ESTIMATE OF ERROR IN LEVEL INDICATION

(S. O. Johnson)

ESTIMATE OF ERROR IN LEVEL INDICATION

In the following analysis, estimated values of heat transfer coefficients are used to determine the parameters and no assumption on the temperature difference is required.

For purposes of the analysis, the water-filled portion of the pressurizer is assumed to be a right circular cylinder extending from the water level to the lower pressure tap. The temperature at one end (the water surface) is assumed to be 670°F. The horizontal temperature gradient is assumed to be zero, but heat is lost to the atmosphere through an effective film coefficient. In a horizontal layer, then, the heat conducted into the layer must be equal to the heat lost to the atmosphere (assumed to be at 100°F).

Thus

$$K_{eff} A_{eff} \frac{\partial^2 T(x)}{\partial x^2} = h_{eff} C T(x) \tag{1}$$

where

K_{eff} = effective conductivity in axial direction

A_{eff} = effective cross-sectional area of cylinder

h_{eff} = effective heat transfer coefficient to the atmosphere

C = circumference ($\sim$ mean) of cylinder

The boundary conditions are:

$$T(o) = 570°F$$

and

$$K_{eff} A_{eff} \leftrightarrow \frac{\partial T}{\partial x}(L) = -h_{eff} A_B T(L)$$

The second boundary condition states that the heat conducted into the bottom of the cylinder is lost to the atmosphere through an effective area A_B. For purposes of this problem it is assumed that the bottom area to be used is that of a hemisphere of the same diameter as the cylinder.

$$A_B = \frac{\pi D^2}{2} = \frac{CD}{2}$$

The solution to Equation (1) with the above boundary conditions is

$$T(x) = T(o) \left\{ \cosh R\frac{x}{L} - \left[\frac{1 + \sinh R \left(\frac{R}{LK} \cosh R + \sinh R\right)}{\cosh R \left(\frac{R}{LK} \cosh R + \sinh R\right)} \right] \sinh R\frac{x}{L} \right\} \tag{2}$$

where

$$R = \frac{h_{eff} \quad C}{K_{eff} \quad A_{eff}} \quad L \text{ and}$$

$$K = \frac{h_{eff} \quad A_B}{K_{eff} \quad A_{eff}}$$

Due to the heater fittings, approximately 14% of the lower section of the pressurizer is not insulated. It is assumed that the heater fittings are at the same temperature as the adjacent vessel wall, and that heat is transferred from them to the air through a film coefficient. According to Brown and Marco*, for the temperature range of interest

$$h_{air} = \frac{1 \pm 0.05}{3600\ (A)^{1/4}} \ \frac{\text{Btu}}{\text{ft}^2 - {}^\circ\text{F} - \text{sec}}$$

where A is the vertical height of the wall. (The maximum value to be used for A is 2 ft.) Thus, the heat transfer coefficient for the fittings is

$$h_{fittings} = 2.78\ (10^{-4})\ \frac{\text{Btu}}{\text{ft}^2 - {}^\circ\text{F} - \text{sec}}$$

For the insulated portion of the vessel

$$h_{air} = 2.36\ (10^{-4})\ \frac{\text{Btu}}{\text{ft}^2 - {}^\circ\text{F} - \text{sec}}$$

For the insulation, assuming that the thermal conductivity is 0.833 (10^{-4}) $\frac{\text{Btu - in.}}{\text{ft}^2 - {}^\circ\text{F} - \text{sec}}$, and that the insulation thickness is 3.5 in.,

$$h_{ins} = 0.238\ (10^{-4})\ \frac{\text{Btu}}{\text{ft}^2 - {}^\circ\text{F} - \text{sec}}$$

The effective heat transfer coefficient for the insulated region is

$$h_I = \frac{1}{\frac{1}{h_{air}} + \frac{1}{h_{ins}}} = 0.216\ (10^{-4})\ \frac{\text{Btu}}{\text{ft}^2 - {}^\circ\text{F} - \text{sec}}$$

(*) A. I. Brown and S. M. Marco, Introduction to Heat Transfer, p 115, New York, McGraw-Hill (1942).

The over-all effective heat transfer coefficient is

$$h_{eff} = 0.14\ h_{fittings} + 0.86\ h_I$$

$$h_{eff} = 0.58\ (10^{-4})\ \frac{Btu}{ft^2 - °F - sec}$$

To obtain $K_{eff}\ A_{eff}$ it is assumed that the conductivities of the wall and the water are 0.08 and 0.8 (10^{-3}) $\frac{Btu\text{-}in.}{ft^2 - °F - sec}$ respectively, that the cross-sectional area of the wall is 2.3 ft^2, and that the cross-sectional area of the water is 6.18 ft^2. Then

$$K_{eff}\ A_{eff} = 0.0157\ \frac{Btu - ft}{°F - sec}$$

The circumference C is taken to be 11.5 ft.

Then

$$R = 0.207\ L$$

$$K = 0.0636\ ft$$

$$\frac{R}{LK} = 3.25$$

In order to obtain a solution it is necessary to assume a value of L, the height of the water above the lower pressure tap. Having obtained the temperature distribution for an assumed value of L, the average density can be obtained. With the assumption that the density used in the level indicator was the minimum value (0.560), the actual level corresponding to the indicated level can be computed by

$$(L_{actual} - 1) = \frac{0.560}{\text{Average density}}\ (L_{indicated} - 1)$$

where the form is due to the fact that the pressure tap is at the 1-ft level. That is, an indicated level of 7.5 ft means that the indicated level is 6.5 ft above the lower pressure tap. A proper solution is obtained when the assumed value of L is equal to the resultant value of $(L_{actual} - 1)$.

The computation was carried out for L = 4 ft and 5 ft and $L_{indicated}$ = 7.5 ft. The results are tabulated below.

Assumed L	Computed (L - 1)	T (L)
4 ft	4.8 ft	447°F
5 ft	4.6 ft	389°F

Linear interpolation of these results yields an actual level of ~ 5.7 ft and a minimum temperature of ~ 410°F. The 5.7-ft level is located at the center-line of the lower heater in the top bank.

The heat loss from the pressurizer can be estimated using the film coefficients derived on the previous page. Here h_I is used above the seam weld and h_{eff} below the weld. In each portion of the vessel,

$$\text{Heat Loss} = (\text{Area})\ (\text{Temp. diff})\ (h)$$

Assuming the steam temperature to be 1000°F, the heat losses from various portions of the vessel are as follows:

Top head	$1/2\ \pi (3.6)^2$	x 900	x 0.216 (10^{-4})	= 0.40	$\frac{\text{Btu}}{\text{sec}}$
Top cylinder	5.75 x 11.5	x 900	x 0.216 (10^{-4})	= 1.29	
Middle cylinder	1.3 x 11.5	x 900	x 0.58 (10^{-4})	= 0.78	
Lower cylinder *	5.7 x 11.5	x 570	x 0.58 (10^{-4})	= 2.16	
Bottom head	$1/2\ \pi (3.6)^2$	x 310	x 0.58 (10^{-4})	= 0.37	
				5.00	$\frac{\text{Btu}}{\text{sec}}$
				or ~ 5.3	Kw

* Assuming all water to be at 670°F.

XIII. APPENDIX D

CALCULATIONS

7. PRESSURIZER HEATING RATE WITH EXPOSED HEATERS

(J. F. Koenig)

PRESSURIZER HEATING RATE WITH EXPOSED HEATERS

The Spert III pressurizer failure could have originated from superheating the steam to 1000°F while maintaining the pressure at 2500 psi, if the pressurizer level controller gave an erroneously high reading which permitted some of the heaters to be exposed to the steam thereby superheating the steam. The superheated steam would condense on the cooler walls of the pressurizer thus heating the pressurizer vessel. Figure 1 shows the time required to heat up the steam and pressurizer vessel to 1000°F if the inital conditions are 668°F and 2500 psi and if the following assumptions are made:

1. Pressure remains constant during superheating of steam.
2. Heat loss of pressurizer is neglected.
3. Heat transferred to water below heaters is negligible.
4. Only that part of the pressurizer vessel that is in contact with the steam is heated.

These assumptions will result in the fastest heat-up time. In assumption 1, the pressurizer pressure would build up while the steam is being superheated. However, steam leaks and operation of blowdown or blowoff valves might compensate for this.

If the steam is at 1000°F, it has been calculated that the heater surface would be approximately 1400°F. The manufacturer's recommended maximum operating temperature for the heaters is 1500°F. Therefore, it is possible to superheat the steam to 1000°F without burning out the heaters.

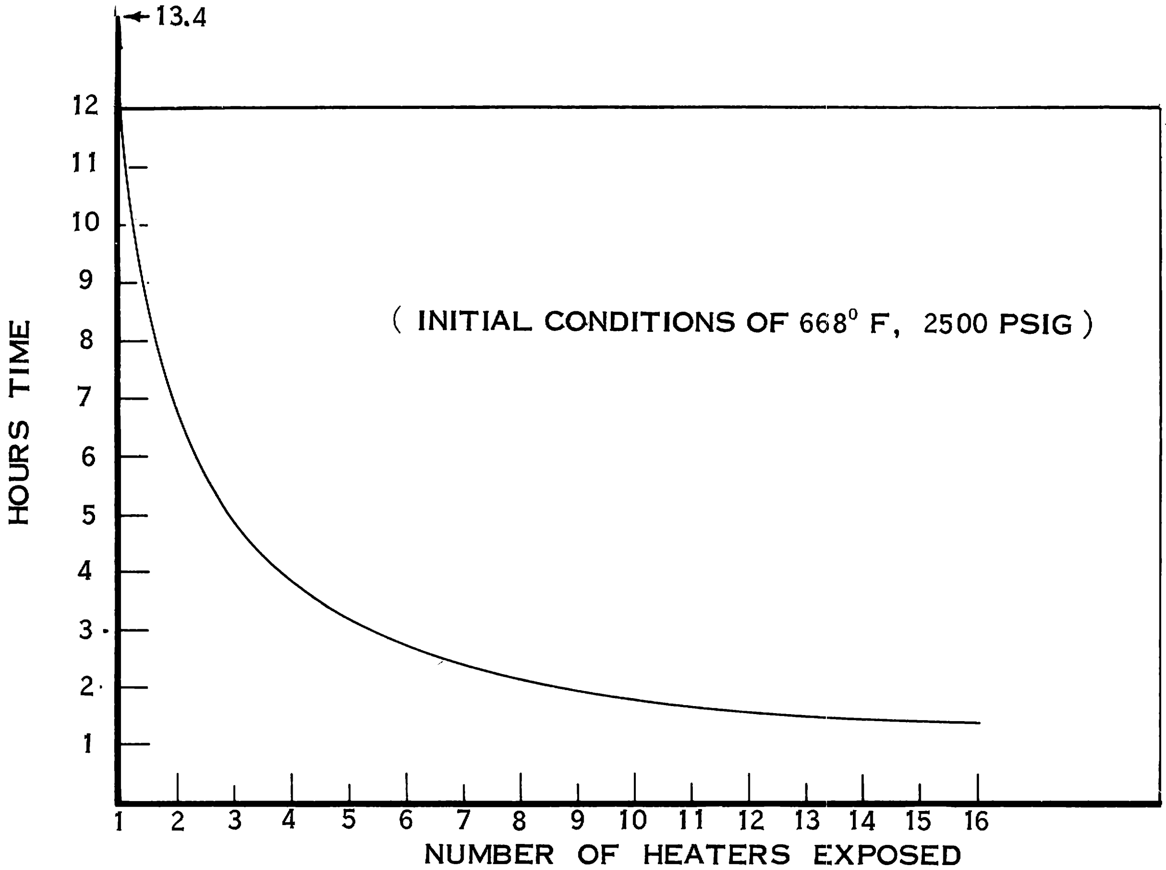


Fig. 1 Pressurizer heat-up time as a function of number of heaters exposed.

SAMPLE CALCULATIONS

In the calculation, the energy from the heaters will be transferred to the steam by radiation and natural convection. Also, the heaters will radiate directly to the walls of the pressurizer. However, the emissivitv of water vapor at high pressure is greater than 0.7 (Figures 4-15, McAdams [*]). Therefore, the steam will absorb a large percentage of the radiated energy. The steam will radiate energy to the pressurizer walls. Also, the superheated steam will condense on the cooler pressurizer walls. Therefore, eventually all the heater energy will go into heating the steam and pressurizer vessel. Therefore, an enthalpy balance will be made.

$$\text{Heater input} = \frac{192 \text{ Kw}}{16 \text{ heaters}} \times 3413 \frac{\text{Btu}}{\text{Kw-Hr}} = 4.10 \times 10^4 \text{ Btu/hr}$$

If one heater is exposed

level = 5 ft 8 in. volume steam = 34.4 ft^3

Physical properties of steam P = 2500 psia:

	$v_g(\frac{ft^3}{lb})$	$h_g(\frac{Btu}{lb})$	lbs vapor in steam dome
T = 668°F	0.1307	1091.1	263

* W. H. McAdams, Heat Transmission, 3rd Ed., New York, McGraw-Hill (1954).

	$v_g\ (\frac{ft^3}{lb})$ (*)	$h_g\ (\frac{Btu}{lb})$	lb vapor in steam dome
T = 1000°F	0.3061	1458.4	112

Average weight of vapor in steam dome = 187 lb

Heat required to superheat steam from 668°F to 1000°F:

$$h = (1458.4 - 1091.1)\ \frac{Btu}{lb} \times 187\ lb$$

$$= 367.3 \times 187 = 68{,}600\ Btu$$

Heat required to heat pressurizer vessel from 668°F to 1000°F:
(for that portion of vessel exposed to the steam)

Top head volume of metal ------------------- = $2.26\ ft^3$

Cylindrical volume (to 5-ft 8-in. level) $29.1\ \frac{12\ ft\ 7\text{-}1/2\ in. - 5\ ft\ 8\ in.}{12\ ft\ 7\text{-}1/2\ in.}$ = $16.04\ ft^3$

Total --------------------------------- = $18.3\ ft^3$

$$\text{Weight} = 18.3\ ft^3\ \frac{0.283\ lb}{in.^3} \times \frac{1728\ in.^3}{ft^3} = 8940\ lb$$

$$C_{p\ steel} = 0.16\ Btu/lb°F$$

$$h = 8940\ lb \times 0.16\ \frac{Btu}{lb°F}\ (1000 - 668)°F = 4.75 \times 10^5\ Btu$$

Time required to heat to 1000°F:

$$\phi = \frac{(4.75 + 0.686)\ 10^5\ Btu}{4.1 \times 10^4\ Btu/hr} = 13.25\ hr$$

Further calculations are tabulated in Table I.

Heaters are 4 in. apart

Volume steam increase with each heater exposed = $1.7\ ft^3$

$$\text{Weight steam (Avg)} = \text{New volume} \times \frac{187}{34.4}$$

$$h\ \text{steam} = \text{weight steam} \times 367.3$$

Volume increase of pressurizer vessel steel exposed to steam as each heater is exposed = $0.768\ ft^2$

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* Symbols are defined on page 183.

Weight increase of pressurizer vessel steel exposed to steam as each heater is exposed = 370.5 lb

Wt of pressurizer required to heat up = old wt + 370

$$h \text{ pressurizer} = (\text{New wt pressurizer})\ (0.16)\ (332)$$

$$\text{heat input} = X \times 4.1 \times 10^4 \text{ Btu/hr}$$

$$\text{heat up time } \emptyset = \frac{h \text{ steam} + h \text{ pressurizer}}{\text{heat input}} = \text{hr}$$

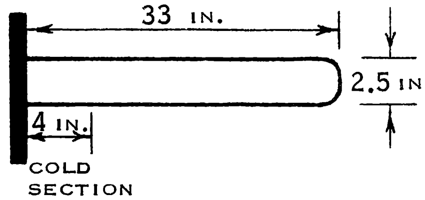
TABLE I

HEAT-UP TIME CALCULATIONS

No. of heaters exposed	Volume steam	Avg wt of H_2O in steam dome	Δh steam	wt pressurizer in contact with steam	Δh pressurizer	Δh pressurizer + Δh steam	heat input	time required to heat steam and pressurizer
1	2	3	4	5	6	7	8	9
X		2 x 5.44	2 x 367.3		5 x 53.1	4 + 6	1 x 4.1 x 10^4	
	Ft^3	Lb	Btu x 10^{-4}	Lb	Btu x 10^{-4}	Btu x 10^{-4}	$\frac{Btu}{hr}$ x 10^{-4}	Hr
1	34.4 +1.7	187	6.86	8,940 +370	47.45	54.31	4.1	13.3
2	36.1	196	7.20	9,310	49.45	56.65	8.2	6.9
3	37.8	205.5	7.54	9,680	51.4	58.94	12.3	4.78
4	39.5	215	7.90	10,050	53.4	61.3	16.4	3.74
5	41.2	224	8.22	10,420	55.3	63.52	20.5	3.09
6	42.9	233	8.56	10,790	57.2	65.76	24.6	2.68
7	44.6	242.5	8.90	11,160	59.2	68.1	28.7	2.38
8	46.3	252	9.25	11,530	61.2	70.45	32.8	2.14
9	48.0	261	9.58	11,900	63.2	72.78	36.9	1.97
10	49.7	270	9.92	12,270	65.1	75.02	41.0	1.83
11	51.4	279	10.25	12,640	67.1	77.35	45.1	1.71
12	53.1	289	10.6	13,010	69.0	79.6	49.2	1.62
13	54.8	298	10.95	13,380	71.0	81.95	53.3	1.53
14	56.5	307	11.26	13,750	73.0	84.26	57.4	1.47
15	58.2	316	11.6	14,120	75.0	86.6	61.5	1.40
16	59.9	325	11.94	14,490	76.9	88.84	65.6	1.35

Heater surface temperature radiating to steam at 1000°F is calculated as follows:

Each heater is 4 kw



$$\text{Heater surface} = 19/16\ [\ 2(33 - 1\tfrac{1}{4}) + \pi \times 1\tfrac{1}{4}\ ] = 105.4\ \text{in.}^2$$

$$\frac{q}{A} = \frac{4000\ \text{watts}}{105.4\ \text{in.}^2} \times \frac{3.413\ \ \text{Btu}}{\text{watt}\ \ \text{hr}} \times \frac{144\ \text{in.}^2}{\text{ft}^2}$$

$$= 18{,}650\ \text{Btu/hr ft}^2$$

$$= \frac{q}{A}\sigma \left[\epsilon_G T_G^4 - \alpha_{G1}\, T_1^4\right] \frac{\epsilon_1 + 1}{2}\ [*]$$

Assume $\epsilon_G = 0.8$

According to McAdams, [**] the ϵ_G approaches 0.7 at a P_wL of 20 ft atm. However, out P_w = 170 atm. Therefore, the pressurizer $P_wL \gg 20$. Also a correction C_w, (McAdams,[***]) has to be made for high pressure. As P_wL increases and P_w increases, the correction decreasingly approaches 1, However, the correction is greater than 1 and would tend to increase the ϵ_G.

Assume $\epsilon_1 = 0.8$.

Although no data is available for oxidized chromalloy, its emissivity should be approximated the same as oxidized steel $\epsilon \sim 0.8$.

$$\alpha_{G1} = \epsilon_G \left(\frac{T_G}{T_1}\right)^{0.65} C_w$$

Solving for T_1;

$$18650 \frac{\text{Btu}}{\text{Hr Ft}^2} = 0.1713 \times 10^{-8} \frac{\text{Btu}}{\text{Hr Ft}^2 R^4} \left[0.8\ (1000 + 460)^4 \right.$$

$$\left. + 0.8\ \frac{(1000 + 460)}{(T_1)}^{0.45} T_1^4\right] \times \frac{0.8 + 1}{2}$$

$$\frac{18650}{0.8 \times 0.9 \times 0.1713} = \left[\frac{1460}{10^2}\right]^4 + \frac{1460}{10^8}^{0.45} T_1\ \ 3.55$$

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* Ibid., Equation 4-57

** Ibid., Figures 4-15.

*** Ibid., Figures 4-16.

Solving

$T_1 = 1860°R$

or

$T_1 = 1400°F$

If the ϵ_G or ϵ_1 is greater than assumed, the heater surface temperature would be less. The heaters may have views of each other which would raise the surface temperature. Also, the heaters would be cooled by natural convection which would tend to reduce the surface temperature.

SYMBOL DEFINITIONS

P = Pressure (psia)

T = Temperature (°F)

v_g = Specific volume of steam (ft^3/lb)

h_g = Enthalpy of steam (Btu/lb)

h = Enthalpy (Btu/lb)

C_p = Specific heat (Btu/lb°F)

ϕ = Time (hr)

q = Rate of heat flow (Btu/hr)

A = Heat transfer area (ft^2)

σ = Stefan Boltzmann constant ($0.1713 \times 10^{-8} \frac{Btu}{Ft^2 - hr - °R^4}$)

ϵ_G = Emissivity of steam

ϵ_1 = Emissivity of surface

α_{G1} = Absorptivity for radiation of a gas from surface 1.

T_G = Temperature of gas (°R)

T_1 = Temperature of emitting surface (°R)

P_w = Partial pressure of radiating steam (atm)

L = Radiation mean beam length (ft)

C_w = Pressure correction factor for steam emissivity

Errata for IDO-16743

Fig. 60 on page 65 should be a repeat of Fig. 33, "Photograph of sample removed from vessel wall", which appears on page 44.

On Fig. 42, page 57, the labels on the curves "Yield Strength" and "Ultimate Strength" should be interchanged.

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