824/21

Program: High-Temperature Piping Design Technology

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I. PROJECT OBJECTIVES

The overall objectives of the High-Temperature Piping Design Technology Program are to

- 1) Provide design methodology for elevated temperature piping systems, based on experiences with FFTF, LMEC, and CRBRP designs and on operating high temperature piping systems
- 2) Assess the validity of using current high temperature Code Case 1592 design rules on piping systems
- 3) Investigate critical piping system design problems from the design criteria viewpoint
- 4) Increase the confidence level in high temperature design rules for piping by demonstrating the conservativeness of these rules.

This project is responsible for the following major tasks, each of which is supportive of one or more of the listed overall objectives.

A long-range program plan (PIPLAN) is being developed and updated to assure piping integrity for current and future LMFBR plants. This plan will identify:

- 1) The design logic used in identifying PIPLAN program areas and needs
- 2) The specific tasks required to achieve a high level of piping structural reliability

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- 3) Tasks contributing to the support of a probabilistic assessment of piping integrity
- 4) A schedule for completion and a definition of work under way for each task.

This project also is assessing and cataloguing relevant operating experience and failures in piping of SRE, HNPF, Fermi, EBR-II, SCTL, LCTL, other operating sodium systems, and foreign reactors; this will provide:

- 1) A basis for selecting one or more elevated temperature piping loops that were designed to less stringent criteria than contained in Code Case 1592, yet have been operating successfully without failures
- 2) A supporting data base for piping integrity assessments
- 3) Substantiation for judgmental assumptions, with regard to causes and modes of piping failures.

Sodium piping construction records are being surveyed for defects detected at installation. These will be assessed and catalogued in a manner that can assist the characterization of initial defects that may be present in LMFBR primary piping at reactor startup.

Another major task of this project is the Code Case 1592 evaluation of selected sodium piping systems that have been operating successfully, yet were designed to less stringent criteria than contained in Code Case 1592. The results of both elastic and inelastic analyses will be used to demonstrate, based on actual operating experience, that the design rules contained in Code Case 1592 are conservative.

The impact on design margins of variations between actual and assumed material, geometry, and loading characterizations is being evaluated. This activity consists of data collection, testing, and analysis. The results will provide:

- 1) An identification of those design parameters whose variations have the greatest impact on design margins

- 2) Guidance for preparing design specifications, selecting fabrication processes, defining critical tests, and establishing dimensional tolerances

- 3) The groundwork for future reliability evaluations of the piping integrity obtainable using the current design codes.

This project is also conducting an experimental evaluation of the creep-fatigue damage produced by high cycle, small amplitude oscillations in the presence of sustained stresses in the creep temperature range. These data will assist in resolving the current questions on the influence of mean stress on the high cycle fatigue life of Code materials and will support extension of the Code Case 1592 fatigue curves beyond 10^6 cycles.

Another major task of this project is developing simplified analysis methods for elevated temperature piping and, eventually, incorporating these methods into the applicable design codes for LMFBR piping. Primary emphasis will be placed on developing appropriate elevated temperature stress indices for piping products currently having low temperature stress indices. Other activities will include developing stress indices for additional piping products and joints which are commonly employed in sodium piping systems, and developing a simplified method to evaluate and control the effects of local overstrain (elastic followup).

This project also is responsible for preparing interpretive reports presenting a methodology for the design of elevated temperature sodium piping systems. These reports will present practical guidelines for the design and required analysis of LMFBR high temperature piping systems. Emphasis will be given to providing practical data in terms of analytical modeling methodologies and Code evaluation techniques.

The final major task of this project is the thermal transient and mechanical testing of piping elbows and bare pipe clamps. The elbow tests, as presently scoped, will:

- 1) Verify the current analytical assumption that radial thermal gradients do not influence the load-deflection characteristics of pipe elbows nor the mechanical load-carrying capacity.

- 2) Verify the current design methodology for elevated temperature piping by running combined thermal transient and cyclic load histories in an accelerated environment.

The bare pipe clamp tests will provide a benchmark against which analytical modeling techniques can be developed for the clamp-pipe interaction.

II. MAJOR ACCOMPLISHMENTS DURING REPORT PERIOD

SUBTASK A - PIPLAN DEVELOPMENT

Typing of the general revision was completed and copies were distributed to the PIPLAN co-chairmen and ERDA representative for review and comment. Following resolution of comments, PIPLAN will be submitted for approval.

SUBTASK BI - COLLECTION OF PIPING FAILURE DATA

A summary report (N146TI35005), "Sodium Piping Survey, Pipe Hanger and Support Experience" is being issued. The report includes an update of sodium piping failure experience in the U.S. and abroad, a survey of construction defect data for sodium piping systems, and a survey of pipe support experience in light water reactor service.

The plant piping treated in this report includes runs of conventional piping which are wetted by sodium (or NaK) in the primary, secondary, and purification service of test loops and power plants. No piping smaller than 1 in. nominal pipe size has been included.

Also excluded are lines used primarily for venting, relief, overflow, draining, or for cover gas services. Most separable components have been excluded, such as the intermediate heat exchangers, steam generators, sodium coolers, and sodium filling systems. However, heater piping has been included, whether flame or electric, provided it is conventional pipe without fins. All piping runs are welded construction so no failures are contributed by gaskets, bolted flanges, swagelock fittings, mechanical seals, or bellows. The definition of pipe failure, as used in the report, implies an actual or imminent leakage of sodium while the pipe is in use or in operational standby service.

The total U.S. experience was found to be 413,000 foot-years as of July 1977. Approximately 57,500 ft of pipe has been wetted with sodium for periods ranging from 5,000 to 115,000 hours or 7.2 years. The U.S. experience is primarily with Type 304 stainless steel (402,600 foot years) with the remaining 2.5% being Type 316 stainless steel and 2-1/4 Cr - 1 Mo. Nearly all of the experience has been accumulated at pressures of less than 200 psi. The distribution of pipe size in U.S. sodium experience is skewed heavily toward the smaller pipe sizes. Of this experience, 16% is related to nominal pipe size, from 3-1/2 to 30 in., and less than 10% of the operating experience has been accumulated at temperatures which exceed 600°F. Pipe wall thickness has ranged from 0.065 to 0.500 in., or from 2% to 7% of the outside diameter.

A significant number of the 52 U.S. sodium piping installations have been dismantled or otherwise deactivated. Currently, operating sodium piping greater than 1-in. pipe size totals less than 34,000 linear feet in this country. Other facilities are becoming operational, however, and the forecasted 1978 daily average of 36,000 feet of pipe would produce a growth of about 9% in the accumulated foot-years of experience by July 1978.

Total foreign experience is estimated to be 438,000 foot-years. There are approximately 118 foreign sodium installations with piping of 1 inch or larger. At least one third of those facilities contain no Type 304 stainless steel piping. The use of Type 316 stainless steel is very common, and somewhat higher design temperatures are employed. Documentation of piping material is lacking for many loops.

About 85,000 feet of sodium piping is currently installed and operating in foreign countries. That is 2-1/2 times greater than current domestic in-service inventory, and the annual growth factor for the foreign cumulative experience data base is about 20%.

The distribution of foreign pipe sizes is considered more representative of a large plant since nine sodium-cooled fast reactors represent 67% of those currently installed. However, the cumulative operating experience to July 1977 consists primarily of small pipe sizes. It is estimated that 35% of the accumulated foreign operating data is for pipe sizes greater than 3-1/2 inches, with 11% being applicable to Type 304 stainless steel.

Domestic pipe failures of 1-inch pipe size and larger were described in Table 1 of the April-June 1977 Quarterly Report. Human error in installation and operation predominated, with an overall failure rate of the order of one per 34,000 foot-years for pipe sizes 1 inch and larger. Only one failure has occurred in U.S. piping of 3-1/2 inches and larger, giving an indicated failure rate of the order of one per 68,000 foot-years in the larger size range.

While fewer details are available on foreign piping failures, Table 1 lists the known failures in foreign piping. The indicated failure rate for 3-1/2 inch pipe size and larger is of the order of one per 50,000 foot-years (excluding cases where the pipe size is unknown). As with domestic pipe failures, principal causes appear to be fabrication construction defects and maloperation rather than design related failures.

SUBTASK BII – SURVEY OF CONSTRUCTION DATA FOR DEFECT RECORDS

Even though semiautomatic and automatic welding procedures are employed for circumferential pipe welds, the initial rejection rate can be expected to average 10 to 15% for butt welds in 6 in. or larger pipe sizes when radiographic inspection is employed with another nondestructive test procedure. It is noted that there have been instances of sodium piping operation over several years with rejectable defects.

A number of rigid pipe support and snubber restraint (predominantly hydraulic type) failures have been observed in light-water reactor plant operation. These support failures generally appear to have a minor effect upon the integrity of the piping pressure boundary. The visibility and redundancy of pipe supports are considered to be beneficial contributing factors.

SUBTASK BIII – ANALYSIS OF SYSTEMS WITH PRIOR SUCCESSFUL OPERATING HISTORY

The inelastic analysis of the SNAP-8ER outlet line has been completed through the initial loading, unloading, and reloading to the maximum temperature of 1300°F. The same maximum strain value was obtained after reloading to establish a stable cycle with a strain range of 0.39%. During the first quarter hour hold period after reloading to 1300°F, the maximum stress relaxed to approximately 30% of its initial value. The inelastic evaluation was

TABLE 1
SODIUM PIPING FAILURE IN FOREIGN INSTALLATIONS
(Sheet 1 of 2)

Facility	Date of Event	Line Size	Cause and/or Remarks
BR-5 (USSR)	1959	Unknown	Drain pipe heater short-circuited. Burned hole in pipe.
BR-5 (USSR)	June 1966	25/40 mm	Drainage piping transition - crack in expanded section between weld caused by pipe movement restriction.
DFR (UK)	Aug 1966	1 in.	External thermal siphon circuit pipe failure - hair line cracks, NaK leakage - small fires. Caused by wet insulation, stress corrosion due to salt water.
5 MW Test Facility (Bensberg, Germany)	1967	Unknown	Sodium partially frozen on heater startup. Tube in oil fired sodium heater burst. Sodium spill-fire.
5 MW Test Facility (Bensberg, Germany)	Prior to 1971	Unknown	Leak on outer surface of sodium inlet to KNK tube-in-tube steam generator prototype. No fire. Continued operation for 600 hours. Severe corrosion and nitriding near leak.
5 MW Test Facility (Bensberg, Germany)	Prior to 1971	Unknown	Small pipe damaged - Electrical trace heater short circuited and overheated pipe - 3 heaters in series. 1st heater failed, T/C located at 3rd heater - pin-hole leaks - no fire.
REML (UK)	Prior to 1971	300 mm	Fillet weld fusion zone penetrated thin wall of thermocouple "pocket". Thermocouple "pocket" through pipe wall extending into flow stream had broken off from its junction with the pipe wall - Small sodium leak - no fire. Design standard revised.

TABLE 1
SODIUM PIPING FAILURE IN FOREIGN INSTALLATIONS
(Sheet 2 of 2)

Facility	Date of Event	Line Size	Cause and/or Remarks
Phenix (France)	Approx. 1973	Unknown	Three mixing tee leaks detected. Small fires in all cases.
DFR (UK)	May 1967	4 in.	Pipe stub to vessel nozzle - 1-1/4 in. long crack in weld on the 4 in. pipe near a thermocouple block. No fire.
DFR (UK)	July 1967	1 in.	Siphon dump circuit - surface pitting and corrosion found in 2 places - airborne seaspray and alkyd paint spots on pipe - no leak or fire reported.
DFR (UK)	Sept. 1967	2 in.	Primary circuit dump line - 1/4 in. diameter hole - pitting corrosion mechanism, high temperatures, air, and residual sodium.
DFR (UK)	Aug. 1970	Unknown	Heat exchanger feed pipe-leak at weld - small fire.
DFR (UK)	Mar. 1974	16 in.	Leak (1 mm hole) in secondary loop between IHX and secondary sodium pump. Accidental application of welders torch - found heat affected zone leading from nearby weld. Moderate leakage - small fire when seal jacket removed.
Rapsodie (France)	1963	Unknown	Air exchanger tubes - weld porosity external corrosion of faulty tube and adjacent tubes.
Rapsodie (France)	Sept. 1966	Unknown	Secondary loop, fill and drain. Double walled pipe - while preheating the inner wall burst, flooding all IHX internal structures with sodium.

performed based upon the assumption that the 0.39% strain range would persist during load-unload cycles subsequent to the creep-relaxation occurring during hold periods at operating temperatures. Details of the analysis are presented in Report N146SR250002, which is to be issued shortly.

SUBTASK C – EVALUATION OF SENSITIVITY OF PIPING DESIGN MARGINS TO VARIABILITY IN MATERIALS, LOADING, TEMPERATURE AND GEOMETRY

Elastic analyses were made for the effects of uniform wall thickness variation, of axisymmetric weld mismatch, and of ovalization. An interim report summarizing the results of these analyses with respect to expected variations in material properties is being prepared.

SUBTASK D – HIGH CYCLE FATIGUE BEHAVIOR AT ELEVATED TEMPERATURES

The report on this subtask summarizes high cycle fatigue testing performed during FY 77. Some background on experimental procedure is presented, but detailed discussion of experimental procedure, evaluation of data trends, and design recommendations will be deferred to topical reports.

Presently, fatigue design curves for nuclear power plant components are presented in Section III of the ASME Boiler and Pressure Vessel Code for metal temperatures up to 427°C (800°F) and in ASME Code Case 1592 for metal temperatures of 427°C (800°F) and above. However, these curves cover only the range of 10 to 10^6 cycles. Extended curves to 10^8 cycles recently have been recommended for use with low-strength austenitic alloys and Inconel Alloy 718 in the low-temperature regime ($\leq 427^\circ\text{C}$)⁽¹⁾. The objective of the present program is to develop experimental data on the long-life (10^5 to 10^8 cycles to failure) fatigue resistance of two nuclear piping steels – 2-1/4 Cr - 1 Mo steel and Type 316 stainless steel. Temperatures of interest ranged from 21° to 538°C (70° to 1000°F). Baseline fatigue data were for constant amplitude, fully reversed strain cycling conditions. Additional data, to evaluate the influence of tensile mean stress, were developed.

Specimens were taken from well-characterized heats of 2-1/4 Cr - 1 Mo steel (Heat 3P5601)⁽²⁾ and Type 316 stainless steel (Heat 8092297, Plate

13 C)⁽³⁾. Before sectioning the sample plates* into blanks, the 2-1/4 Cr - 1 Mo steel plate was isothermally annealed using the heat treatment schedule that has been used in past studies of this material⁽²⁾. Specimens cut from Type 316 stainless-steel plate were given the solution anneal recommended for use in laboratory studies of this alloy⁽⁴⁾. All specimens were machined and polished to the cylindrical test-section configuration shown in Figure 1.

Figure 1. Specimen Configuration

All experiments were conducted using an electrohydraulic system operated in either closed-loop strain or load control. Specimens were loaded axially, and the load was measured using a standard strain gauged cell in series with the specimen. Axial strain was measured directly over a 12.7 mm (0.50-in.) gauge length using a special extensometer.⁽⁵⁾

Heating of the specimens was accomplished by high-frequency (450 kHz) induction, and temperature was controlled by using a standard proportional-type power controller. The geometry of the heating coil was designed to minimize temperature gradients in the test section to 3°C (5°F) or less. A specimen instrumented with six Chromel-Alumel thermocouples was used to verify the final coil design. The locations of these thermocouples are illustrated in Figure 2. Once the appropriate calibrations were established,

*Both samples were furnished by Dr. C. R. Brinkman of Union Carbide Corporation, Nuclear Division.

Figure 2. Locations at Which Thermocouples Were Attached to Specimen Used for Calibration of Heating Unit

thermocouples were attached to the actual test specimens only at locations A, B, and F. The one at location B was then used to provide a feedback signal to the temperature controller. The other two were used for periodic checking of the specimen temperature during each test.

Laboratory experiments normally are conducted under either deformation control or load control. Under actual service situations, cyclic straining may be caused by either type of loading. For example, simple thermal loadings tend to be deformation-controlled; whereas, pressure loadings are generally load-controlled. In structures with complex three-dimensional geometries and/or with combined types of loadings, the cyclic straining is governed by a combination of both control modes. Because of the exploratory nature of the present work and for purposes of simplicity, the experimental approach has been limited to controlling just one rather than a combination of these two types of loadings.

For the carbon steels, low-alloy steels, and austenitic steels used in nuclear reactor structural components, even long-life (10^6 to 10^8 cycles to

failure) fatigue damage involves inelastic cyclic deformation. Thus, the simplifications based on nominally elastic deformation cannot be used in conducting fatigue experiments on these steels. For alloys that cyclically harden, the initial cyclic stress-strain history shows significant inelasticity [see Figures 3(a) and (b)], even though the stable response may approach nominally elastic conditions.* In contrast, a cyclically softening material may stabilize to much different conditions under both strain- and load-controlled cycling. [See Figures 3(c) and (d)].

Such differences in stress-strain response necessitate the measurement of both strain and stress until the response has cyclically stabilized in long-life fatigue experiments. Once stable conditions have been reached, it is normally more economical and convenient to measure only load and to conduct the experiment under load control. Provided that the initial cyclic stress-strain behavior has a negligible influence on fatigue life, it is acceptable to conduct the entire fatigue experiment under load control while monitoring the strain behavior to make sure that stable conditions are truly achieved and that excessive mean strains are not developed. For alloys that cyclically harden to a stable condition, the experiment can be started under strain-controlled conditions, until stable stress-strain response is obtained, and then switched to the load-control mode for the remainder of the experiment. To the extent possible, this latter approach was used in the present study.

All experiments were initiated in strain control, and strain was programmed to follow a triangular waveform at a constant rate of about $4 \times 10^{-3} \text{ sec}^{-1}$. When the stress range hardened to a stable level, the control mode was switched to load. The specimen was then cycled under sinusoidal loading at constant frequency. The frequency was increased to as high a level as possible for stable operation of the closed-loop test system. After changing control modes, the cyclic frequency was periodically slowed down to measure the strain level to make sure that large inelastic strains did not develop. In some cases, specimens were cycled to failure completely under strain control so that the influence of changing control mode could be assessed. To assess possible frequency effects, a few specimens were load cycled completely

*The stable response can, of course, remain inelastic.

INITIALLY
INELASTIC

STABLY
ELASTIC

(a) STRAIN CONTROL AND
CYCLIC HARDENING

INITIALLY
INELASTIC

STABLY
ELASTIC

(b) LOAD CONTROL AND
CYCLIC HARDENING

INITIALLY
INELASTIC

STABLY
ELASTIC

(c) STRAIN CONTROL AND
CYCLIC SOFTENING

INITIALLY
INELASTIC

STABLY
ELASTIC

(d) LOAD CONTROL AND
CYCLIC SOFTENING

Figure 3. Idealized Representation of the Relation of Test Control
Mode and Cyclic Hardening Behavior to Cyclic
Stress (σ) - Strain (ϵ) Response

to failure at a low frequency. In tests conducted at 21°C (70°F), care was taken to limit the frequency increase so that the specimen temperature increased by no more than 5°C (9°F). Thus, the entire experiment was essentially under strain-controlled conditions, even though load control was employed to allow faster cycling rates and more economical testing where possible.

Discussion of Results

Results of all fatigue experiments that have been conducted during this program are summarized in three tables.

- Table 2 — fully reversed strain cycling of 2-1/4 Cr - 1 Mo steel
- Table 3 — strain cycling with tensile mean stress of 2-1/4 Cr - 1 Mo steel
- Table 4 — fully reversed strain cycling of Type 316 stainless steel.

The control mode history of each specimen is listed in these tables. Values of the controlled total strain range, $\Delta\epsilon_t$, stable stress range, $\Delta\sigma$, and total time to fracture are listed for each experiment also. For fully reversed cycling (Tables 2 and 4), stable values of inelastic and elastic strain range, $\Delta\epsilon_{in}$ and $\Delta\epsilon_e$, are given. For the tensile mean stress experiments on 2-1/4 Cr - 1 Mo steel (Table 3), the values of $\Delta\epsilon_{in}$ were insignificant and $\Delta\epsilon_e \simeq \Delta\epsilon_t$; however, it was necessary to report values of maximum total strain, ϵ_{tmax} , and maximum stable stress, σ_{max} , for these experiments. The tensile mean stress levels were induced by straining to indicated maximum levels with subsequent strain cycling between ϵ_{tmax} and $\epsilon_{tmax} - \Delta\epsilon_t$. Since the maximum stress decreased or relaxed cyclically, the initial values of σ_{max} are also listed in Table 3. Three values of fatigue life are reported for the short-life (near 10^5 cycles or less) experiments--

N_0 — Number of kilocycles where tension load starts to drop more than compression load

N_5 — Number of kilocycles where tension load drops 5 percent more than compression load

N_f — Number of kilocycles to complete fracture.

For the long-life experiments, only values of N_f were determined.

TABLE 2
FATIGUE OF ISOTHERMALLY ANNEALED 2-1/4 Cr - 1 Mo PLATE
(HEAT 3P5601) UNDER FULLY REVERSED CYCLING IN AIR

Specimen Number	Control Mode	Total Axial Strain Range, percent	Stable Axial Inelastic Strain Range, percent		Stable Stress Range		Fatigue Life, kilocycles (kc)			Time to Fracture, hour
			Inelastic	Elastic	MPa	ksi	N ₀	N ₅	N _f	
At 316 C (600 F)										
FL14	Strain at $4 \times 10^{-3} \text{ s}^{-1}$	0.40	0.09	0.31	606	87.9	23.0	26.2	28.4	15.8
FL6	Strain at $4 \times 10^{-3} \text{ s}^{-1}$	0.31	0.02	0.29	549	79.6	225	--	236.8	105
FL15	Strain at $6 \times 10^{-3} \text{ s}^{-1}$ to 73.9 kc, load at 6 Hz thereafter	0.30	0.02	0.28	557	80.7	--	--	170.6	24.5
FL18	Strain at $6 \times 10^{-3} \text{ s}^{-1}$ to 85.7 kc, load at 1 Hz to 1267.6 kc, load at 5.8 Hz thereafter	0.29	--	0.25	553	80.2	--	--	1,441.6	50.2
FL16	Strain at $5 \times 10^{-3} \text{ s}^{-1}$ to 32.2 kc, load at 15 Hz thereafter	0.27	--	0.27	502	72.8	--	--	17,437.8	322
At 427 C (800 F)										
FL19	Strain $4 \times 10^{-3} \text{ s}^{-1}$ to cycle 80, strain at 7×10^{-3} thereafter	0.33	0.01	0.32	557	80.8	137	137	138.0	38.3
FL7	Strain at $4 \times 10^{-3} \text{ s}^{-1}$ to 60 kc, load at 1.2 Hz to 1,742.3 kc, load at 5 Hz thereafter	0.31	0.01	0.30	577	83.7	--	--	>2,242.6 ^(a)	>528 ^(a)
FL9	Strain at $4 \times 10^{-3} \text{ s}^{-1}$ to 63.6 kc, load at 0.6 Hz to 65.0 kc, load at 1 Hz to 334.5 kc, load at 3 Hz to 357 kc, load at 10 Hz thereafter	0.29	--	0.29	564	81.7	--	--	829.6	41.2
FL23	Strain at $2 \times 10^{-3} \text{ s}^{-1}$ to cycle 485, strain at $4 \times 10^{-3} \text{ s}^{-1}$ to 168.6 kc, load at 1 Hz to 169.5 kc, load at 5 Hz to 175.7 kc, load at 6.9 Hz thereafter	0.29	--	0.29	540	78.3	--	--	1,011.5	102
FL10	Strain at $4 \times 10^{-3} \text{ s}^{-1}$ to 2.95 kc, load at 10 Hz to 666.9 kc, load at 21 Hz to 2,526 kc, load at 35 Hz thereafter	0.29	--	0.29	510	74.0	--	--	31,919.4	277
At 538 C (1000 F)										
FT1	Strain at $4 \times 10^{-3} \text{ s}^{-1}$	0.30	0.08	0.22	385	55.8	105.8	107.3	108.5	45.2
FL20	Strain at $4 \times 10^{-3} \text{ s}^{-1}$	0.29	0.08	0.21	394	57.1	75.6	75.8	76.49	30.8
FT3	Strain at $4 \times 10^{-3} \text{ s}^{-1}$ to 47.5 kc, load at 0.74 Hz	0.26	0.03	0.23	376	54.6	--	--	1,279.7	41.8
FT4	Strain at $4 \times 10^{-3} \text{ s}^{-1}$ to 4 kc, load at 19 Hz thereafter	0.24	0.02	0.22	367	53.3	--	--	2,225.2	33.8
FT5	Strain at $4 \times 10^{-3} \text{ s}^{-1}$ to 1 kc, load at 24.6 Hz thereafter	0.21	--	0.21	351	50.9	--	--	5,993.4	68.0
FT6	Strain at $4 \times 10^{-3} \text{ s}^{-1}$ to 1.63 kc, load at 27 Hz thereafter	0.19	--	0.19	297	43.1	--	--	10,736.4	111

(a) Specimen damaged by equipment malfunction.

TABLE 3

FATIGUE OF ISOTHERMALLY ANNEALED 2-1/4 Cr - 1 Mo PLATE
(HEAT 3P5601) STRAIN CYCLED IN AIR
WITH POSITIVE MEAN STRESS

Specimen Number	Control Mode	Strain, percent		Stable Stress				Initial Maximum Stress		Number of kilocycles (kc) to Failure, N _f	Time to Fracture, hour
		Total Range	Maximum	Range		Maximum		Stress			
				MPa	ksi	MPa	ksi	MPa	ksi		
<u>At 316 C (600 F)</u>											
FL3	Strain at 2 x 10 ⁻³ to cycle 213, strain at 4 x 10 ⁻³ to 189 kc, load at 1 Hz to 320 kc, load at 5.1 Hz thereafter	0.25	4.10	452	65.5	452	65.5	452	65.5	>10,756(a)	>685(a)
FL1	Strain at 2 x 10 ⁻³ to cycle 345, load at 0.2 Hz to cycle 442, load at 21.6 Hz thereafter	0.23	2.16	438	63.6	375	54.4	385	55.9	>33,910(a)	>437(a)
<u>At 427 C (800 F)</u>											
FL21	Strain at 2 x 10 ⁻³ s ⁻¹ to 1.82 kc, load at 0.3 Hz to 2.04 kc, load at 15 Hz thereafter	0.32	2.15	571	82.8	330	47.5	364	52.8	848.1	17.6
FL13	Strain at 1.6 x 10 ⁻³ s ⁻¹ to cycle 895, strain at 2.6 x 10 ⁻³ s ⁻¹ to 6.62 kc, load at 11 Hz to 3,537 kc, load at 22 Hz thereafter	0.28	2.31	484	70.2	332	48.1	367	53.2	24,132	358.2
FL8	Strain at 2 x 10 ⁻³ s ⁻¹ to cycle 606, load at 1 Hz to 8.48 kc, load at 35 Hz to 5,444 kc, load at 32 Hz thereafter	0.22	2.21	367	53.3	367	53.3	367	53.8	60,541	527
<u>At 538 C (1000 F)</u>											
FT7	Strain at 4 x 10 ⁻³ s ⁻¹ to 6.85 kc, load at 26 Hz thereafter	0.23	2.60	344	50.0	206	30.0	295	42.8	230.80	4.7
FT8	Strain at 2 x 10 ⁻³ to 60.3 kc, load at 1 Hz to 62.8 kc, load at 32 Hz thereafter	0.18	2.47	273	39.6	149	21.6	285	41.3	3,750	63

(a) Specimen did not fail and test was stopped.

TABLE 4

FATIGUE DATA FOR SOLUTION ANNEALED TYPE 316 STAINLESS STEEL
PLATE (HEAT 8092297) TESTED IN AIR UNDER FULLY
REVERSED STRAIN CYCLING

Specimen Number	Control Mode	Total Axial Strain Range, %	Stable Axial ^(a) Strain Range, %		Stable ^(a) Strain Range		Fatigue Life, kilocycles (kc)			Time to Failure, hour
			Inelastic	Elastic	MPa	ksi	N ₀	N ₅	N _f	
<u>At 21 C (70 F)</u>										
A1	Strain at 4 x 10 ⁻³ s ⁻¹	1.01	0.70	0.31	629	91.2	8.952	9.372	9.543	13.1
A2	Strain at 4 x 10 ⁻³ s ⁻¹ to 45.17 kc, strain at 8 x 10 ⁻³ s ⁻¹ thereafter	0.45	0.23	0.22	444	64.4	--	146.0	147.9	60.7
A4	Strain at 4 x 10 ⁻³ s ⁻¹ to 17 kc, load at 2.7 Hz thereafter	0.31	0.10	0.21	417	60.5	--	--	1,340	143
A3	Strain at 4 x 10 ⁻³ s ⁻¹ to 50 kc, load at 8.4 Hz to 70 kc, strain at 4 x 10 ⁻³ s ⁻¹ to 70.5 kc, load at 8.4 Hz thereafter	0.29	0.09	0.20	418	60.6	--	--	1,190	66
A5	Strain at 4 x 10 ⁻³ s ⁻¹ to 26 kc, load at 5.9 Hz to 36,000 kc, load at 15 Hz thereafter	0.24	0.04	0.20	399	57.8	--	--	>66,000 ^(b)	>2,279 ^(b)
<u>At 427 C (800 F)</u>										
A7	Strain at 4 x 10 ⁻³ s ⁻¹	0.70	0.41	0.29	467	67.8	16.2	16.4	17.46	18.2
A9	Strain at 4 x 10 ⁻³ s ⁻¹	0.55	0.27	0.28	445	64.6	29.1	30.7	32.02	24.4
A10	Strain at 4 x 10 ⁻³ s ⁻¹	0.48	0.22	0.26	441	63.9	49.5	54.4	55.15	36.8
A8	Strain at 4 x 10 ⁻³ s ⁻¹	0.40	0.06	0.33	549	79.7	--	--	668	370
<u>At 538 C (1000 F)</u>										
A11	Strain at 4 x 10 ⁻³ s ⁻¹	0.49	0.17	0.32	517	75.0	--	--	22.15	15.2
A14	Strain at 4 x 10 ⁻³ s ⁻¹	0.44	0.12	0.32	538	78.0	26.2	28.5	29.65	18.1
A19	Strain at 4 x 10 ⁻³ s ⁻¹ to 150 kc, strain at 8 x 10 ⁻³ s ⁻¹ thereafter	0.41	0.02	0.39	622	90.2	243	--	245.2	128
A12	Strain at 4 x 10 ⁻³ s ⁻¹	0.40	0.08	0.32	511	74.2	--	--	>50.3 ^(c)	>27.9 ^(c)
A13	Strain at 4 x 10 ⁻³ s ⁻¹	0.40	0.02	0.38	573	83.1	--	--	>373.4 ^(c)	>207 ^(c)

(a) Stable values were not necessarily at $N_f/2$ but were at saturated cyclic stress levels. For 427 and 538 C, these were generally at much greater than $N_f/2$.

(b) Specimen did not fail and test was discontinued.

(c) Equipment malfunction damaged specimen before failure.

Fatigue of 2-1/4 Cr - 1 Mo Steel: Fatigue data for the 2-1/4 Cr - 1 Mo steel are shown, with stress amplitude as the independent variable, in Figure 4. Comparative data from other studies^(2,6), which used hourglass-shaped specimens exclusively, are also included in these figures.

Except for one half-filled triangle symbol, the filled and half-filled symbols in Figure 4 represent data from the present study. In terms of stress amplitude, the present data fell at the upper end of the scatter band of those from other work, at lower temperatures (Figure 4a), and at the lower end of a similar scatter band at 538°C (1000°F) (Figure 4b).

The experiments on effect of mean stress were exploratory and limited. Their results give indications of trends, but more data are needed before definite conclusions can be reached. The effect of mean stress can be seen by examining the plots of Figure 4 in detail. If mean stress has a detrimental effect, the stress amplitude will be lower than that for fully reversed cycling at comparable fatigue lives. At 316°C (600°F), as indicated by the filled circles in Figure 4a, there was no effect of mean stress and neither specimen failed. As shown by the half-filled diamond symbols in the same figure, only the specimen at zero minimum stress (FL8) showed a reduction in life due to mean stress at 427°C (800°F). Finally, at 538°C (1000°F), the half-filled diamond symbols in Figure 4b show that mean stress had a detrimental effect on fatigue life. The latter is true because at this high temperature a great deal of cyclic relaxation of the maximum stress occurred (compare initial and stable values in Table 3). Thus, the early cycles of loading were actually more damaging than predicted by the stable-stress levels. This effect should be accounted for in a proper cumulative creep/fatigue damage analysis.

Fatigue of Type 316 Stainless Steel: Fatigue results for the Type 316 stainless steel are presented in Figure 5. Only fully-reversed cycling experiments have been conducted on this material to date. Data from past studies^(7,8) of another heat of this alloy are included in this figure for comparison purposes. Also, the average trend fatigue curve that has been developed by Jaske and O'Donnell⁽⁷⁾ for temperatures up to 427°C (800°F) is shown in the figure. Initial experiments were conducted in the region of 10^4 to 10^5 cycles to failure to provide transition from existing past data. Subsequent experiments were

Figure 4. Fatigue Life of 2-1/4 Cr - 1 Mo Steel as a Function of Stable Stress Amplitude

Figure 5. Fatigue Life of Type 316 Stainless (Heat 8093397)
as a Function of Strain Range
(Fully-Reversed Cycling)

then conducted in the long-life ($>10^5$ cycles) regime. The present results agree well with those of past work and with the Jaske-O'Donnell curve. At lives below 10^5 cycles to failure, the fatigue resistance decreased as temperature increased. However, at longer lives ($>10^5$ cycles), the high-temperature fatigue resistance was slightly superior to that at 21°C (70°F), indicating a crossover in the fatigue curves with increasing temperature. Of course, this trend needs to be substantiated through additional long-life tests in the future.

One complicating aspect of the experimental work on Type 316 stainless steel was the unexpected cyclic hardening behavior at 427° and 538°C (800° and 1000°F). Figure 6 shows the stress response as a function of cycles, and Figure 7 shows a cyclic stress-strain plot of the same data. At the highest strain range at 21° and 427°C (70° and 800°F), the classical type of response was observed. Namely, the material rapidly hardened to a stable level where

a. AT 21°C (70°F)

b. AT 427°C (800°F)

c. AT 538°C (1000°F)

Figure 6. Cyclic Stress Response of Type 316 Stainless Steel (Heat 8092297) Under Strain Cycling at $4 \times 10^{-3} \text{ Sec}^{-1}$ in Air

Figure 7. Cyclic Stress-Strain Response of Type 316 Stainless Steel (Heat 8092297) Under Strain Cycling at $4 \times 10^{-3} \text{ Sec}^{-1}$ in Air

it remained for most of the life. However, such was not the case for the lower strain ranges. At 21°C (70°F), the low strain range data showed initial cyclic softening to a stable level, but at 427° and 538°C (800° and 1000°F), almost continuous cyclic hardening behavior was observed throughout the cyclic life. As a result, specimens at the lowest strain ranges actually hardened to higher levels than those at slightly higher strain ranges.

The cyclic hardening behavior at 427° and 538°C (800° and 1000°F) made it necessary to conduct these experiments entirely under strain control rather than switching to load control when the response stabilized as was done at 21°C (70°F). Because of the limited frequency response of the extensometer, such strain cycling required relatively low cyclic frequencies (about 1 Hz or less). This results in long test times and costly experiments. In the interest of performing future tests more rapidly and economically, three aspects should be investigated on an exploratory basis: (1) load controlled testing before stable response is developed, (2) strain-controlled experiments on another heat of material, and (3) experiments on aged material. A few experiments should be performed under load control to see if the cyclic hardening response has a significant effect on fatigue life. Some comparable long-life data on another heat (65808) would help indicate whether this cyclic hardening behavior is generic to the alloy or an anomaly of heat-to-heat variation. Aging the material by long-term exposure at temperature prior to testing may cause the cyclic response to stabilize more rapidly, and hence, enable more rapid testing. Once results of these exploratory tests have been analyzed and interpreted, plans for additional long-life fatigue tests can be formulated.

Summary

Long-life fatigue curves (up to 10^8 cycles) have been developed for annealed 2-1/4 Cr - 1 Mo steel. The effect of mean stress noted for specimens tested at higher temperatures will be evaluated, with respect to code analysis procedures, in a future topical report.

A limited amount of long-life fatigue data has been developed for solution-annealed Type 316 stainless steel at 21°, 427°, and 538°C (70°, 800°, and 1000°F). At less than 10^5 cycles to failure, fatigue resistance decreased as increased temperature. A tendency toward an opposite trend was seen at more

than 10^5 cycles to failure. For low strain ranges of less than 0.5% at 427° and 538°C (800° and 1000°F), continuous cyclic hardening was noted throughout the fatigue life, which required relatively slow cycling in strain control. In order to increase cyclic frequency and shorten test times, three types of experiments need to be conducted on an exploratory basis: (1) load controlled tests with monitoring of strain, (2) tests of specimens from another heat of Type 316 stainless steel, and (3) tests of aged material. Results of these experiments will then be used to plan further work. Mean-stress experiments also need to be carried out for this alloy.

SUBTASK E – SIMPLIFIED HIGH TEMPERATURE PIPING ANALYSES

The correlation of PIRAX2 predictions with ORNL beam test data was evaluated; it was found that when the material properties obtained directly from the test article material were used in the analysis, the correlation for deflection controlled testing was quite satisfactory, but that the correlation between predicted and measured strains for load controlled testing was very poor. A satisfactory correlation was also noted between PIRAX2 predictions and ORNL elbow test data. In all cases, the effect of variations in material characterization was considerable. Additional details can be obtained from the topical report AI-DOE-13216, "Evaluations of the Piping System Inelastic Analysis Computer Program, PIRAX2," by E. C. Rodabaugh.

In addition, currently available time-to-failure data on branch connections were reviewed, and the following evaluation is reported.

A branch connection in piping, or the analogous nozzle in a pressure vessel, poses a very formidable problem in stress analysis, even in the elastic regime. The problem is compounded in the plastic-creep regime because there often is a weld between nozzle and vessel (the weld material has different plastic characteristics than the base material, and irregularities of weld contours may significantly influence the useful life of the nozzle).

In general, the loadings on a branch connection of nozzle are very complex, consisting of pressure, moments, and thermal gradients. Progress in establishing design methods for elastic regime design was made by considering one load at a time, starting with the simplest loading: static internal pressure.

This report starts the same way: by looking at available data on branch connections or nozzles that are subjected to a static pressure but at temperatures that involve time-dependent characteristics of the material.

Test Data

Test data from three references are used in this report:

Reference 1: W. Sys⁽⁹⁾ test on a nozzle in a cylindrical vessel; base material was 2-1/4 Cr - 1 Mo, test temperature was 575°C (1067°F).

Reference 2: T. E. Taylor⁽¹⁰⁾ tests on nozzles in spherical vessels; base material was carbon steel, test temperature was 450°C (842°F).

Reference 3: Goodall and Ainsworth⁽¹¹⁾ tests on nozzles in cylindrical vessels; base material was a special aluminum alloy, test temperature was 150°C (302°F).

The references contain additional data and/or evaluation concepts that are not discussed in this report. Of interest here is the time-to-failure, location of the failure, and description of any part-thru cracking.

Attempts are underway to obtain additional data of this type. When and if obtained, the report will be expanded to include that data and will be issued as a phase report.

Reference 1 Tests: The test specimen, shown in Figure 8, consists of a cylindrical-shell nozzle welded to a cylindrical-shell vessel. The base material is stated to be 2-1/4 Cr - 1 Mo; nothing is said concerning the welding or weld material. However, a photograph of the specimen after tests indicates that cracking occurred in the weld (see Figure 8), with apparently no cracks in the base material.

The pressurizing fluid was argon gas. The major portion of this work concerns the test apparatus, and implies that additional tests are being conducted.

DIMENSIONS OF REFERENCE (1) TEST MODEL (END CLOSURES NOT SHOWN)
 PRESSURE: 153 kg/cm² TEMPERATURE: 575°C
 FIRST CRACKS OBSERVED AT 2,200 HOURS
 FAILURE (LEAK) AT 2760 HOURS
 BOTH FIRST CRACK AND FAILURE OCCURRED IN THE WELD, TRANSVERSE
 PLANE, AS INDICATED IN ABOVE SKETCH

Figure 8. Reference 1 Test Specimens and Failures

Reference 2 Tests: The nozzles used in these tests are described in Figure 9. Three spherical vessels were tested, each with three nozzles.

Leakage occurred in only one nozzle in each vessel: nozzle CD in vessels 1 and 3, and nozzle EF in vessel 2. In vessel No. 1, a loss of temperature control occurred at 1250 hours; hence, there are only two quantitative time-to-failures.

The pressurizing fluid was lead. The authors investigated the influence of lead on the steel, and state that:

The randomness and equal severity of cracks starting from both surfaces seem to preclude lead having an effect on the behaviour. Certainly evidence from the literature and observations at B.W.R.A. all indicate that lead is inert with respect to mildly alloyed iron at 45°C.

(The 45°C is probably a typo error, should be 450°C.)

Reference 3 Tests: The nozzles used in the tests are described in Figure 10. The test specimens were machined from a solid block of aluminum alloy. The particular aluminum alloy (not described in detail) was selected because it creeps at relatively low temperature and is stable for several thousand hours at the testing temperature.

The pressurizing fluid was not identified, nor was the location or description of the failures given. Presumably the failures occurred in the vicinity of the nozzle-to-vessel intersection region.

Evaluation

Comparison With Unperforated Vessel: A pertinent comparison is between the time-to-failure obtained in the nozzle tests and the expected time-to-failure of the unperforated vessel. This, as will become apparent, leads to an evaluation of the C_1 index of 2.0 for fabricated branch connections.

Data are available which relate the stress-to-rupture in an uniaxial test to the material, temperature, and time. For the present purpose, such data are given in convenient form in API RP530⁽¹²⁾; this includes the average stress-to-rupture as a function of the Larson-Miller parameter for a large number of materials including two of interest herein, i.e., medium carbon

MATERIAL: ALUMINUM KILLED MILD STEEL, WITH COMPOSITION (PERCENT) OF:

C	Mn	Si	S	P	Cu	Al
0.16	1.46	0.14	0.033	0.017	0.11	0.023

PRESSURE*: 1920 psi, TEMPERATURE: 450°C (842°F)

NOZZLE	t, in.	WELDING ELECTRODES
AB	0.4	CLASS 6 LOW HYDROGEN
CD	0.4	CLASS 3 RUTILE
EF	0.3	CLASS 6 LOW HYDROGEN

FAILURE DATA:

VESSEL	NOZZLE	HOURS	FAILURE LOCATION
1	CD	>1250**	CRACKS IN WELDS – ALONG WELDS
2	EF	1780	CRACKS IN WELDS, TRANSVERSE TO WELDS
3	CD	730	CRACKS IN WELDS, TRANSVERSE TO WELDS

*CONTINUOUS EXCEPT IN VESSEL 3, REDUCED TO ZERO 1 HOUR IN EVERY 8 HOURS

**TEMPERATURE CONTROL LOST AT 1250 HOURS. FAILURE OCCURRED IN 1257 HOURS AT SOME TEMPERATURE GREATER THAN 500°C

Figure 9. Reference 2 Test Specimens and Failures

DIMENSIONS ARE GIVEN AS RATIOS. LENGTHS L_n AND L_r OF TEST ASSEMBLIES BETWEEN CLOSURES (NOT SHOWN) ARE NOT GIVEN IN ANY FORM BUT, FROM A PHOTOGRAPH IN REFERENCE (3), IT APPEARS THAT L_n IS ABOUT 1.5 R, L_r IS ABOUT 4R.

IDEN.	r/R	R/T	t/T	PR/T (N/mm ²)	TEMPERATURE (°C)	RUPTURE TIME (h)
B	0.5	12.4	0.52	111	150	3400
C	0.25	12.6	0.30	174	150	500
D	0.38	23.4	0.80	137	150	400
E	0.25	45.0	0.00	93	150	1050

MATERIAL IS A SPECIAL ALUMINUM ALLOY WITH UNIAXIAL STRESS-TO-RUPTURE OF 200 N/mm² AT 10,000 HOURS AT 150°C. SPECIMENS WERE MACHINED FROM SOLID FORGINGS. RADIUS r_i IS NOT GIVEN. RUPTURE LOCATION OR EXTENT IS NOT GIVEN.

Figure 10. Reference 3 Test Specimens and Failures

steel for the Reference 2 tests and 2-1/4 Cr - 1 Mo for the Reference 1 tests. The Reference 3 test gives the information that the stress-to-rupture for the aluminum alloy used in the nozzle tests is 200 N/mm^2 at 10^4 hours plus a graph (Figure 6 in the Reference), which together were used to develop the equation:

$$S_r = 462 \theta^{-0.0908}$$

where S_r = stress-to-rupture in N/mm^2 at time θ in hours.

Assuming that the membrane stress calculated by the equations

$$S_n = PR/T \text{ for cylindrical shells}$$

$$S_n = PR/2T \text{ for spherical shells}$$

is equivalent to the stress in uniaxial tests (where P = internal pressure, R = mean radius, and T = wall thickness), we can calculate the time-to-rupture for unperforated shells. Table 5 summarizes these evaluations in the column headed "Hours-to-Failure, Nominal". It can be seen that these expected times are all longer than the test time-to-failures. This indicates that the nozzles weakened the test assemblies as compared to unperforated vessels.

Inclusion of the C_1 -Index: Table NB-3682.7-1 of the Code⁽¹³⁾ gives indices for pressure loading of "Branch Connections per NB-3643" of $B_1 = 1.0$, $C_1 = 2.0$, and $K_1 = 1.7$. The C_1 -index represents the primary-plus-secondary stress. The column of Table 5 headed "With $C_1 = 2$ " shows the expected hours-to-failure using $C_1 S_n$ as equivalent to the uniaxial stress-to-rupture data. Comparison of this column with the "Test" column shows that now the predictions are conservative (less hours than test) except for the Reference 3 nozzle E.

Meeting NB-3643 Requirements: It should be noted that the C_1 -index of 2.0 is applicable if and only if the reinforcement requirements of NB-3643 are met. Table 5 shows the percentage of reinforcement for the nozzles used in tests. If this percent is less than 100, the requirements of NB-3643 are not met if T is taken as the required minimum thickness of the vessel.

TABLE 5
SUMMARY OF EVALUATIONS OF TIME-TO-FAILURE
DATA ON BRANCH CONNECTIONS

Reference	Identification	$S_n^{(1)}$ (psi)	Hours-to-Failure			Percent Reinforced ⁽⁴⁾	$\frac{T'}{T}$ ⁽⁵⁾
			Test	Nominal ⁽²⁾	With $C_1 = 2$ ⁽³⁾		
1	—	12000	2760	5089	75	100	1.00
2	2-EF	17400	1780	13100	~5	73	0.94
	3-CD	17400	730	13100	~5	52	0.83
3	B	16100	3400	6.52×10^6	3150	1	0.54
	C	25200	500	48300	23	4	0.56
	D	19800	400	6.62×10^5	320	13	0.59
	E	13500	1050	4.59×10^7	22200	0	0.50

- (1) $S_n = PR/T$ for cylindrical shells, $PR/2T$ for spherical shells, where P is the internal pressure, R = mean radius, T = wall thickness.
- (2) Expected hours-to-failure for unperforated shells, based on the equivalence of S_n to the stress in uniaxial tests.
- (3) Expected hours-to-failure for nozzles, using $2S_n$ for equivalence with uniaxial stress-to-rupture data.
- (4) The percentage of cut-out area replaced by excess thickness in the nozzle. The cut-out area is equal to $d_i T$, where d_i = nozzle inside diameter, T = vessel wall thickness. The reinforcement zone boundary was that in the ASME Boiler and Pressure Vessel Code, Section III, Div. 1, Fig. NB-3339.4-1.
- (5) T' is the vessel thickness such that the excess area in the vessel wall and excess thickness in the nozzle wall is equal to the cut out area $d_i T'$. Reinforcement zone boundary parallel to the vessel wall was taken as d_i on each side of the nozzle centerline.

It can be seen in Table 5 that the model which is unconservative using $C_1 = 2$ (Reference 3 Tests, Identification E) is essentially an unreinforced opening; indeed, it is just a hole in the vessel. In order to bring these tests into Code perspective, there is some required minimum thickness, T' , for which the opening is fully reinforced. The ratio of T'/T is shown in the last column of Table 5. To evaluate the test specimens as Code-reinforced nozzles, $S'_n = C_1 S_n / (T'/T)$ must be used. When this is done, $C_1 = 2$ is conservative for all tests. For example, for the Reference 3 test, Identification E, the predicted hours-to-failure is 11 hours, as compared to the test hours-to-failure which is 1050.

Other Evaluation Aspects: The Reference 1 and 2 tests, where the nozzles were welded to the vessels, gave failures in the welds. From this aspect, use of the base material stress-rupture data is mostly "nominal" evaluation.

The base material properties probably were significant in controlling the amount of strain imposed on the welds but obviously a critical aspect of the results was the capacity of the weld material to absorb strain before cracking.

The Reference 2 test vessels were spherical-vessels, fabricated from twelve "pentagonal segments". The segments were butt welded together with class 6 low hydrogen weld metal. Note in Table 5 that the expected life of a spherical vessel made of the base material is 13100 hours. None of the welds in the spherical vessel failed during the tests. However, quoting from the Reference 3 test:

Cracking was observed in the membrane butt welds of all vessels. This was first observed in vessel No. 2 where a large number of cracks transverse and extending the full width of the weld were present. On re-inspection of vessel No. 1 five very fine cracks were observed. Because of the presence of these cracks testing of vessels Nos. 2 and 3 was continued after leakage and repair and strengthening of the nozzles.

The cracks in vessel No. 2 increase in number and opening width as further testing progressed. The largest had opening widths of the order of 0.25 in. and depths of the order of 0.5 in. at the termination of testing, although leakage through them did not occur. Cracks in vessel No. 3 were first observed at the termination of testing. These were only six in number, longitudinal and less than 1/2 in. long."

It is pertinent to note that the Reference 2 test data provides the additional information that a number of nozzles survived the test without failure; i.e., nozzles AB in all three vessels, nozzle CD in vessel 2, and nozzle EF in vessel 3. Cracks had started in nozzles CD and EF in all three vessels. No mention is made of any cracks in nozzle AB. Apparently this nozzle, due to a combination of heavier nozzle wall and low hydrogen weld rods (more ductile weld material?), survived the tests without cracks.

Evaluation in Relation to Code Case 1592⁽¹³⁾: Code Case 1592 covers only one of the three base materials used in the test assemblies discussed herein: the 2-1/4 Cr - 1 Mo material used in the Reference 1 tests. Case 1592 states that,

for "Design Conditions", the primary membrane stress must not exceed S_o . The value of S_o is, at most, 67% of the average stress-to-rupture in 100,000 hours and, for 2-1/4 Cr - 1 Mo at 1067°F, S_o is equal to 5,300 psi. The membrane stress developed by the test pressure was $S_n = PR/T = 12,000$ psi; the test life was 2,760 hours.

A more informative comparison can be made from Case 1592 values of S_t ; these are allowable stresses as a function of time. For 2-1/4 Cr - 1 Mo at 1067°F, S_t is equal to about 7,300 psi at 3,000 hours (about the test time to leakage). The value of $C_1 S_n / (T'/T)$ was 24,000 psi. Accordingly, the Case 1592 safety factor on stress for time-to-failure of around 3,000 hours is about 3.3.

The Reference 2, vessel 3, test involved a fatigue aspect, in that the pressure was reduced to zero 1 hour in every 8 hours. The time at pressure was about 640 hours, with about 90 cycles from zero pressure to full pressure and return to zero pressure. It is of interest to examine this test using an equation analogous to Equation 5 of ASME Code Case 1592:

$$\sum \frac{n}{N} + \sum \frac{\theta}{\theta'} = 1.00 \quad \dots (1)$$

where

n = cycles of loading (90 in this test)

N = cycles of loads to produce fatigue failure

θ = time at loading (640 in this test)

θ' = time for stress-to-rupture at load.

It also is of interest to see if Equation 1 predicts the relative time-to-failure of vessels 2 and 3. For vessel 2, $n = 0$, hence θ' must equal 1,780 hours*. Equation 1, for vessel 3, is then:

*It is recognized that nozzle EF failed in vessel 3, and nozzle CD failed in vessel 3. However, the description of part-through cracks suggests that nozzle CD in vessel 2 was about to fail, hence for this crude evaluation, it is assumed that the only difference between vessels 2 and 3 was that vessel 3 was pressure-cycled, vessel 2 was not.

$$\frac{90}{N} + \frac{640}{1780} = 1.00 \quad \dots (2)$$

While Code Case 1592 does not cover carbon steel, an approximation can be obtained for N from the ASME Code Criteria Document, which gives a failure (not design) fatigue S-N curve. This curve is applicable to carbon steel at temperatures up to 700°F. It is estimated that at the test temperature of 842°F the S-value for a given N is decreased by a factor of 2.0. The value of N is then a known function of the calculated stress range, S. The obtained calculated values of 90/N are shown in Table 6. In order to satisfy Equation 2, 90/N must be equal to $1 - 640/1780 = 0.64$. It is apparent in Table 6 that the procedure leads to far lower values of 90/N.

TABLE 6
EVALUATION OF FATIGUE CYCLES FOR THE
REFERENCE 2 TEST VESSEL 3

Stress Equation	Stress Range (psi)	N(1)	$\frac{90}{N}$
$S_n = PR/2T$	17400	$>10^6$	<0.00009
$C_1 S_n$	34800	10^6	0.00009
$C_1 S_n / (T'/T)$	41930	2×10^5	0.00045
$C_1 K_1 S_n / (T'/T)$	71300	3×10^4	0.003

(1) N is the cycles-to-failure obtained from Figure 9 of Reference 14, assuming that the stress, for a given N, at 842°F is one-half of that given directly by the figure. The figure gives N as a function of stress amplitude, the above tabulation shows stress range which is two times the stress amplitude.

This isolated comparison indicates that the $C_1 K_1$ product of $2 \times 1.7 = 3.4$ is not adequate to explain the relative time-to-failure of vessels 2 and 3. Actually, of course, scatter in test data of this type of testing is such that no conclusion can be drawn from a single pair of tests. It also is pertinent to note that the C_1 and K_1 indices are meant to be applied to nozzles with full penetration welds with the surfaces of the welds ground to a radius; such welds would

be expected to have significantly better fatigue life than the welds used in the test assemblies (see Figure 9).

Summary

As indicated in Table 5, the presence of nozzles or openings in pressure vessels, particularly if unreinforced, lead to a significant reduction in time-to-failure as compared to the expected (nominal) time-to-failure for unperforated vessels. The use of a C_1 -index of 2.0, along with adjustment for the amount of reinforcing in the test vessels, leads to highly conservative predictions of the time-to-failure.

Comparison of the Reference 1 test with Code Case 1592 (only test with base material covered by the Case) indicates that if $C_1(PR/T)/(T'/T)$ is limited to S_t of the Case (~ 7300 psi for 3,000 hours at 1067°F), the Case 1592 factor of safety on stress is about 3.3. An evaluation of relative time-to-failures of the Reference 2 test vessels 2 (static pressure) and 3 (cyclic pressure), suggests that the C_1 and K_1 indices are inadequate for the types of welds used in the test specimens.

Failures in the Reference 1 and 2 tests were in or at welds, not in the base material. This emphasizes the importance of weld properties at elevated temperatures; unfortunately, very little is known about these properties. The Reference 3 test specimens did not contain welds.

SUBTASK F – DESIGN METHODS AND CRITERIA FOR LMFBR PIPING

Work on this subtask has been deferred, and remaining funds for the current fiscal year have been transferred to Subtask G.

SUBTASK G – PIPE COMPONENT TESTING

A report (N146TI250002), "Stress Analysis of a Conventional Type Pipe Clamp has been issued during the report period. The report covers the elastic analysis of the pipe-to-pipe clamp interaction under clamp bolt-up, pipe pressurization, differential thermal expansion, and tensile and compressive support loading. Iterative solutions were required, since the contact forces can only be compressive and the solutions are nonlinear. Preliminary review indicates the contact pressure to be highly localized and future refinement of the model may be necessary.

Contact pressure was found to be localized in 30 degrees of arc or less. The contact load and clamp bending moment distribution due to clamp bolt-up, pipe pressurization, and temperature loading are similar and show approximate quadrant symmetry. Figure 11 shows the contact pressure and clamp bending moment distributions for a 1000 psi internal pressure.

For the tensile support load, the clamp bending moment is skewed towards the support load application point and the contact at the clamp heel nearest the load application point disappears, (see Figure 12).

Under compressive support load, the majority of the load is reacted in direct bearing at the clamp heel nearest the load application point, and the bending moment in the clamp is substantially less, (see Figure 13).

Maximum stresses in the pipe and clamp for each load condition are listed in Table 7. It should be noted that the internal pressure of 1000 psi was selected to give significant stress levels rather than being taken equal to the pipe clamp test pressure of 150 psi. The bolt-up load of 1000 pounds is also arbitrary since the clamp test bolt-up load has not been determined.

When considering the tabulated stresses, it is important to note the following computer modeling effects: the clamp was modeled with beam elements, the pipe with plate elements, and the contact pressure as discrete loads at respective node points. Under these conditions, the contact pressure reduces to a line load. In addition, the stress values for the plate elements reflect an average stress level in the element rather than the peak stress at the corner of the plate element where the contact load is applied.

The line load simulation of the contact pressure results in calculated pipe stresses higher than actual, while the plate element stresses would underestimate the local stress. Smaller sized plate elements were used near the clamp interface to mitigate the effects of discrete element modeling.

III. IMPACT ON LMFBR PROGRAMS

This project specifically responds to currently identified tasks in the Piping Integrity Plan. It relates to substantiation and improvement of piping integrity for CRBR, PLBR, and CRB programs. Subtasks BI and BII are in response to CRBR DRS 02.22, Rev. 1.

Pipe Clamp Moment Loading for the Piping Subjected
to an Internal Pressure of 1000 psi*

Figure 11. Pipe Clamp Loading for the Piping Subjected to
an Internal Pressure of 1000 psi*

Pipe Clamp Moment Loading for the Clamp Subjected to a Total Axial Load of 4000 lb

Figure 12. Pipe Clamp Loading for the Clamp Subjected to a Total Axial Load of 4000 lb

Pipe Clamp Moment Loading for the Clamp Subjected
to a Total Axial Load of 4000 lb

Figure 13. Pipe Clamp Loading for the Clamp Subjected
to a Total Axial Load of 4000 lb

TABLE 7
MAXIMUM STRESS INTENSITY FOR THE CLAMP AND PIPE
AND THE CORRESPONDING ELEMENT

Loading Condition	Clamp Maximum Stress Intensity (psi)	Beam Element Number	Pipe Maximum Stress Intensity	Plate Element Number
Axial Tension, 4000 lb	38,945	8	19,916	74
Axial Compression, 4000 lb	18,956	27	23,858	89
Internal Pressure, 1000 psi	3,071	8	14,621	74
Steady State Thermal Expansion	23,307	8	12,348	74
Bolt Preload, 1000 lb	36,840	8	20,257	74

IV. SCHEDULE

The various subtasks, milestones, and milestone completion dates are shown in the schedule below.

▽ - Milestone

△ - Completion Date

Subtask	GFY 1976				1976 T	GFY 1977			
	1	2	3	4	3 mos	1	2	3	4
A. PIPLAN Development			▽1						▽2
BI. Collection of Piping Data for Defect Records			△1	▽1		▽2			▽3
BII. Survey of Construction Data for Defect Records				△1		△2			△3
BIII. Analysis of Systems with Successful Prior Operating History								▽1	△1
C. Evaluation of Sensitivity of Piping Design Margins to Variability in Material and Geometry			▽1		▽2	▽3			▽4
D. High Cycle Fatigue Behavior at Elevated Temperature (BNL)			△1			△2			△3
E. Simplified High Temperature Pipe Analysis (BNL)						▽1		▽2	
F. Design Methods and Criteria for LMFBR Piping - (deferred)						△1			△2
G. Pipe Component Testing									▽1

Milestones

A.1 Complete initial draft PIPLAN. A.2 Continue support of Piping Integrity Plan (PIPLAN) development. BI.1 Identification data acquisition plan for assessment of failure data and operating experience. BI.2 Complete interim report on failure data in piping systems. BI.3 Submit final report on failure data and failure rates in piping systems. BII.1 Complete survey of construction records and prepare final report on defect in piping systems. BIII.1 Selection of sample pipeline and operating history for 1592 evaluation.	BIII.2 Completion of elastic evaluation of sample pipeline. BIII.3 Complete inelastic analysis of SNAP-8-ER Loop. BIII.4 Select additional piping systems for CC 1592 evaluation and report progress. C.1 Complete sensitivity planning. C.2 Submit interim report/sensitivity study. D.1 Interim report/high cycle fatigue. D.2 Interim report/high cycle fatigue. E.1 Interim report and recommendations for methods of analysis. G.1 Complete AIP with LMEC for tests at TTF.

V. NEXT REPORTING PERIOD ACTIVITIES

SUBTASK A — Following resolution of comments, the PIPLAN revision will be submitted for approval.

SUBTASK BI AND BII — No effort has been funded in FY 78.

SUBTASK BIII — No effort has been funded in FY 78.

SUBTASK C — An interim report will be issued on preliminary sensitivity analysis results.

SUBTASK D — A topical report on high cycle - mean stress fatigue testing of 2-1/4 Cr - 1 Mo will be prepared.

SUBTASK E — Work will be started on simplified equations for deformation control and creep-fatigue evaluation.

SUBTASK F — No effort has been funded in FY 78.

SUBTASK G — Initial testing of bare pipe clamps will be started.

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