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STRESS ANALYSIS OF 100-K
PRIMARY PIPING

by

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STRESS ANALYSIS OF 100-K
PRIMARY PIPING

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INTRODUCTION

The program to assess the integrity of the K Reactor primary coolant piping systems with regard to the potential for brittle failure⁽¹⁾ includes as an essential part an analysis of stresses in the piping materials both during normal operation and during emergency conditions. This program of stress analysis has recently been completed and has included the physical measurement of stress (strain) during normal operation, calculation of working stresses under design and actual conditions, and analysis of the probable effects on these stresses of emergency loadings.

PURPOSE

It is the intent of this report to present the results of the analysis of the measured and calculated stresses and to define to the extent possible the conditions of stress in the primary coolant piping systems during both normal and emergency conditions. The data is presented for further evaluation in conjunction with the results of the investigations being conducted to determine fracture toughness properties, crack growth rates, and nil ductility transition temperature ranges of the K Reactor coolant piping materials.

SUMMARY

The stress analysis program consisted of in-service measurement of actual strain ranges to which the piping is subjected during all phases of normal operation coupled with analytical determination of stresses resulting from loadings other than those due to normal operating conditions.

Stresses obtained from measured strain ranges in straight runs of pipe were found to compare favorably with calculated stresses due to internal pressure. Nominal working stress ranges in the K piping were computed, from measured strains, to be approximately 6,000 psi (hoop stress) in the 36-inch piping and 5,500 psi in the 24-inch piping during operation at 462 top of riser pressure (TORP). These may be compared with calculated hoop stresses, also based on an operating pressure of 462 psig TORP, of 6,800 psi and 5,800 psi, respectively.

As expected, strain ranges measured at discontinuities such as branch connections indicated stress ranges considerably higher than stresses calculated for straight runs of pipe. For example, strain measurements at one of the 24" x 36" branch connections in the 165-K Building indicated stress ranges as high as 35,000 psi in the 36-inch pipe and 46,800 psi in the 24-inch pipe immediately adjacent to the juncture weld.

The stresses computed from strain measurements taken during the course of this study, particularly at branch connections and nozzles, have been compared with stresses obtained in photoelastic model studies⁽²⁾ conducted for the ASME Pressure Vessel Research Committee (PVRC). The measured stress ranges compare reasonably in most cases with what one might expect in ideal connections (unreinforced) of the types installed in the piping systems.

A summary of the measured and calculated stresses is shown in Table I. The measured stresses tabulated represent the stresses computed from the maximum strain values measured at a given joint or fitting. Figures 1 and 2, Appendix A, show the location of all measurement points and the values of stress computed from strain readings. A complete tabulation of all measured stresses at each location examined is listed in Tables VII through XIV in Appendix A. The stresses shown on Figures 1 and 2 represent the maximum principal stresses at the pressures shown; whereas, in Tables VII through XIV, the measured stresses have been corrected to 462 psig TORP and reduced to their circumferential (hoop) and axial components.

The nominal working stress measured on the surface of the inlet risers compares favorably with calculated values of stress due to internal pressure. Some regions of relatively high stress were found adjacent to crossheader nozzles (up to 7,100 psi) and adjacent to the vent nozzle in the riser head (approximately 7,260 psi). However, these stresses are not significantly greater than the calculated stress in the riser (6,500 psi to 6,800 psi, top to bottom).

Relatively high strain ranges were also measured in branch connections in the 24" coolant supply piping. Stress ranges indicated by these strain measurements, however, were considerably less than those in the 24" x 36" branch connections. The maximum measured stress range in the 24" branch connections was 18,000 psi.

In addition to the stresses imposed by internal hydraulic pressure in the system, other forces are acting during normal operation which contribute to the general stress level in the piping materials. Such forces are caused by forming and welding (residual stress), vibration, structural loading (at supports), and thermal expansion. All of these stresses, with the exception of residual stress, are assumed to be subject to similar concentration or intensification effect at discontinuities as are the stresses due to internal pressure.

- Thermal Stresses -- The maximum thermal stress in the coolant piping systems was determined to be 1,400 psi, based on a 35 F thermal gradient. This maximum (axial) stress occurred at the vertical 15° bend in the 36-inch tunnel piping. The maximum thermal stress in the section of piping between the pumps and the valve pit in 165-K (24-inch) was determined to be 1,200 psi at the high lift pump discharge elbow.

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TABLE I
100-K PRIMARY COOLANT PIPING
COMPARISON OF MEASURED AND COMPUTED STRESSES

LOCATION	MAXIMUM ¹ MEASURED STRESS RANGE		² CALCULATED FOR STRAIGHT PIPE RUNS		CONCENTRATION FACTOR ³ (CF)		COMPUTED STRESS RANGE ⁴ (CALCULATED X CF)	
	CIRCUM PSI	AXIAL PSI	CIRCUM	AXIAL PSI	CIRCUM	AXIAL	CIRCUM PSI	AXIAL PSI
24" Pipe Continuous Runs	5,500	1,200	5,800	2,600	1	1	5,800	2,600
36" Pipe Continuous Runs	6,000	3,200	6,800	3,100	1	1	6,000	3,200
24" x 24" Tees								
Near crotch	13,400	3,000	5,800	2,600	0.8-1.5	0.3-1.5	4,640-8,700	1,740-8,700
At side	20,700	15,100			1.4-2.2	1.4-2.8	8,100-12,800	8,100-16,200
24" x 24", 45°								
Near crotch	11,300	650	5,800	2,600	0.9-1.6	1.0-1.4	5,200-9,280	5,800-8,100
At side	14,300	10,200			2.0-2.4	1.0-2.5	11,600-13,900	5,800-14,500
24" x 12" Tees								
24"	10,000	502	5,800	2,600	0.8-1.8	1.6-2.9	4,600-10,400	9,280-16,800
12"	13,000	9,800	5,200	2,400	1.6-2.4	0.5-2.2	8,300-12,500	2,600-11,400
24" x 36" Branch (A & D Lines)								
24" At side	46,800	9,000	5,800	2,600	Photoelastic model analysis not applicable to this fitting because of its unorthodox geometry.			
36" At side	35,000	12,800	6,700	3,100				
24" x 36" Branch (B & C Lines)								
24"	12,800	6,800	5,800	2,600	1.6-2.2	2.5-3.6	9,300-12,750	14,500-20,000
36"	12,000	2,100	6,700	3,100	0.8-1.2	0.5-1.0	5,400-8,100	3,400-6,700
36" Riser								
Crossheader Nozzles	7,100	6,370	6,700	3,000	0.5-1.3	0.5-1.0	3,300-8,700	3,300-6,700
Vent Nozzle	6,100	7,200	6,250	6,250	1.2-1.5	1.2-1.5	7,800-9,800	7,800-9,800
Elliptical Head	6,100	7,700	6,250	6,250		1.6		9,600

¹Measured stresses were computed from strain measurements using $E=29.5 \times 10^6$ in/in, $\sigma = \epsilon E$.

²Calculated stresses were obtained from the following formulae:

$$\text{Hoop Stress, } \sigma_t = \frac{PD}{2t} - P_y \quad \text{Axial Stress, } \sigma_n = \frac{Pd^2}{D_o^2 - d^2}$$

Internal pressures were based in all cases on 462 psig, TORP, and were adjusted to compensate for difference in head between top of riser and point at which stress was calculated.

³Concentration factors were derived from photoelastic model studies sponsored by ASME Pressure Vessel Research Committee (PVRC) and reported in Welding Research Council Bulletin No. 113, April 1966 (Tables VII to XIV).

⁴Computed stresses were obtained by application of concentration factors to nominal circumferential stresses.
calculated for continuous pipe runs.

- . Vibrational Stresses -- Vibrational stress in the pipe wall varied from 600 psi measured at the pumps to 150 psi measured at the inlet risers. The predominant frequency of vibration was 165 to 180 cycles per second with a maximum amplitude of 18.6 μ -inches/inch. There were no low frequency, large amplitude vibrations detected in the system.
- . Structural Load Stresses -- Structural dead load stresses imposed on the piping due to bending at and between supports do not exceed 1,800 psi.
- . Residual Stress -- Residual stresses in the system resulting from forming and welding of the piping are estimated to be not greater than 4,000 psi.

Under accident, unusual event, or partial system failure conditions, the piping system could be subjected to a number of additional loads beyond those considered during normal operation. Such loads might be imposed by earthquake or by pressure surges due to line break.

- . Seismic Load Stresses -- The maximum stresses expected to result from a design basis earthquake (DBE) have been calculated to be 400 psi in the 24-inch piping, 250 psi in the 36-inch riser approach piping, and 250 psi in the risers.
- . Pressure Transient Stresses -- Pressure transients experienced during normal operation of the plants are nominal and are limited to or below normal system pressure.

For purposes of evaluation, a guillotine line break was assumed and the effects of the shock wave resulting from the subsequent check valve closure was analyzed.⁽³⁾ The maximum stress due to internal pressure in the 36-inch piping resulting from such an event was determined to be approximately 8,550 psi and in the 24-inch piping 7,320 psi, based on a maximum internal pressure buildup of 610 psig.⁽³⁾

The results of summing all normal operating stresses are given in Table IV, page 12. A summation of total stress which might be expected during a seismic occurrence of design basis magnitude is shown in Table V for points of known stress concentration. A similar summation of stress for the maximum theoretical pressure transient is given in Table VI. The methods of calculation and sample calculations are given in Appendix B.

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BACKGROUND

The piping systems under consideration in this study include all high pressure (600 psig, design pressure)⁽⁴⁾⁽⁵⁾ piping from the high lift primary coolant pumps (six at each reactor) to and including the inlet risers (four at each reactor). Six 24-inch diameter pump discharge lines transpose to four 36-inch diameter lines at the north wall of each 165-K Building. The four 36-inch lines extend through a pipe tunnel to the 105 Building and terminate in four vertical 36-inch risers, two on each side of the reactor. A schematic diagram, typical of each reactor plant, is shown in Figure 3, Appendix A. The systems were designed for 600 psig, cold water (80 F maximum) service in accordance with the requirements of then current editions of the ASME Boiler and Pressure Code and the ASA Code for Pressure Piping.

The water plant portions of the piping systems (from 190-K to the 105-K Buildings) were designed by C. T. Main, Inc., and the piping within the 105-K Buildings was designed by General Electric Company. Construction of the piping systems, as well as the rest of the reactor and water plants, was performed by Kaiser Engineers, Inc., during the period 1952-1955. Construction was specified to be in accordance with the above Codes.

The piping systems are primarily of all welded construction with some flanged connections at valves and other equipment. All field welds were stress relieved at 1,150 F in accordance with Code requirements. The 24-inch pipe was specified to be ASTM A 106, Grade B, seamless pipe. Although documentation (procurement records) is not available to confirm that the pipe is seamless, the pipe is assumed to be seamless as a result of chemical etching at random locations which revealed no longitudinal seam welds. All 36-inch pipe was specified to be ASTM A 155, Grade B, welded pipe fabricated from ASTM A 285, Grade B, steel plate. The material used is as specified except for the 36-inch mitered elbows (6 in each plant) which are of ASTM A 285, Grade C, steel. The inlet risers were fabricated off-site and are of the same material as the 36-inch pipe. A summary of materials used and their known or specified properties is shown in Table II. Wall thicknesses of pipe and fittings were confirmed by ultrasonic thickness measurements, and the results are shown in Figure 18.

On completion of all welding and stress relieving of welds, the entire systems, from pumps to and including the inlet risers, were subjected to a hydrostatic test at 900 psig. The pressure in the systems was raised to 900 psig four times, then released for repair of minor leaks at flanged connections, before the final hydrostatic test which was held (at 900 psig) for thirty minutes. The inlet risers, in addition, were hydrostatically tested at the fabricator's plant at 1,000 psig, pressure cycled five times from zero to 1,000 psig, and subjected to the hammer test while under 1,000 psig pressure as specified in the ASME Unfired Pressure Vessel Code - 1952. The risers were furnace stress relieved at 1,150 F after fabrication and prior to testing.

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TABLE II
100-K PRIMARY COOLANT PIPING
SUMMARY OF MATERIAL PROPERTIES

	SIZE	NOMINAL WALL THICKNESS	TENSILE STRENGTH	YIELD STRENGTH	MATERIAL	(1) ALLOWABLE STRESS
HIGH LIFT PUMP DISCHARGE PIPING	24"	0.968	60,000	35,000	ASTM A106, GR B, SEAMLESS. 30% C MAXIMUM	15,000
RISER APPROACH PIPING	36"	1.25	50-60,000	27,000	ASTM A285, GR B, PLATE FABRICATED TO ASTM A155 GR B PIPE	12,500
RISERS	36"	1.25	49,635 ⁽²⁾ 51,550 ⁽²⁾ 52,250 ⁽²⁾ 51,500 ⁽²⁾	29,885 ⁽²⁾ 31,400 ⁽²⁾ 35,600 ⁽²⁾ 33,000 ⁽²⁾	ASTM A285, GR B, PLATE FABRICATED TO ASTM A155 GR B PIPE ⁽²⁾ CHEMICALS: C=0.15% P=0.05% Mn=1.0% Mo=0.05%	12,500
RISER HEADS ELLIPTICAL KE-B,C, KW-A,C,D	36"	1.25	57,000 ⁽²⁾	35,000 ⁽²⁾	ASTM A285, GR B, HEAT NO. 7-59145	12,500
RISER HEADS HEMISPHERICAL KW-B KE-A,D	36"	1.375	68,000 ⁽²⁾	43,500 ⁽²⁾	ASTM A516, GR 60, REPLACEMENT HEADS	20,000
RISER REPL. MATERIAL KW-B KW-A,D	36"	1.25	65,400 ⁽²⁾	46,600 ⁽²⁾	ASTM A516, GR 60,	20,000
FLANGES	36"	400 PSI ASA RATING	70,000	36,000	ASTM A105, GR 11, FORGED STEEL	17,500
FLANGES	24"	600 PSI ASA RATING	70,000	30,000	ASTM A105, GR 1, FORGED STEEL	15,000
MITERED ELBOWS	36"	1.25	55-65,000	30,000	ASTM A285, GR C, PLATE FABRICATED TO ASTM A155 GR C PIPE	13,750
FORGED ELBOWS	24"	1.25	60,000	30,000	ASTM A234 GRADE A OR B	15,000
GATE VALVES	24"	600 PSI ASA RATING	70,000	36,000	ASTM A216 GRADE WCB	
CHECK VALVES	36"	400 PSI ASA RATING	70,000	36,000	ASTM A216, GR WCB CAST CARBON STEEL	17,500
CHECK VALVES	24"	600 PSI ASA RATING	70,000	36,000	ASTM A216, GR WCB CAST CARBON STEEL	17,500

(1) ALLOWABLE STRESS PER USAS B 31.1.0, 1967.

(2) THESE DATA OBTAINED FROM LABORATORY TESTS OF SPECIMENS REMOVED FROM PIPING. OTHER DATA BASED ON ASTM SPECIFICATIONS FOR MATERIALS USED IN PIPING SYSTEMS.

The K Reactor plants have been in more or less continuous operation since 1955 (KW startup January 4, 1955, and KE April 17, 1955), with the exception of normal outages for charge-discharge and a few outages for special maintenance. The precise number of pressure cycles, from shut-down to operating pressure, to which the systems have been subjected is not known but is estimated to be on the order of 30-45 per year or a total of less than 650 cycles per reactor in the 14 years service of the plants.

DISCUSSION

The stress analysis of the K Reactor coolant piping systems has included a review of measured, calculated, and estimated conditions of stress resulting from both normal and emergency conditions. Measured stresses were obtained through conventional methods and based on normal operating pressures, and estimated stresses were obtained through use of somewhat less conventional methods. Strain gages were affixed on continuous pipe runs to obtain nominal working stresses and at points of obvious stress concentration to obtain maximum stress intensifications. These measured principal stresses are summarized in Figures 1 and 2, Appendix A.

Experimental Measurement of Stresses

The purpose of the experimental measurement phase of the study was to determine the operating stress ranges in components for which theoretical stress evaluations were indeterminate.

The measurement of strain during operation of the piping systems was performed by attaching rosette strain gages to the polished metal surface of the pipe to be analyzed. Rosette gages are elements that contain three basic strain gages oriented at 120° spacing or at 0°, 45°, and 90° spacing (see Figure 4). The rosette gages are attached to the piping component with an epoxy bonding compound and leads are attached to readout equipment.

The principle utilized in this method of strain measurement is that the strain is proportional to the change in resistance of the gage during controlled loadings of the metal to which the gage is attached. The values of resistance change are read by means of a precision strain gage electrical bridge circuit and converted to a strain displacement of the metal under measurement analysis. A description of the equipment utilized in the program is shown below. Method of installation of gages and readout equipment are pictured in Figures 5 through 9.

- Strain Gages -- Micro Measurement, Division of the Vishay Intertechnology Company, temperature compensated for the materials to be tested.

Rectangular Rosette (see Figure 6), Type EA-06-125RA-120 gage factor 2.08, accuracy $\pm 1.5\%$, Delta Rosette (see Figure 5), Type EA-06-25-YA, gage factor 2.05, accuracy $\pm 1\%$.

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- . Cables -- Spectra Strip Corporation
Flat multi-conductor cable.
- . Indicator -- Budd Company (see Figure 8), Model P 350, manually
adjusted, null balanced meter for static strain readout (pre-
cision wheatstone bridge).
- . Amplifier -- Honeywell (see Figure 9), No. 119, six channel
developed or recording oscillograph.
- . Recorder -- (see Figure 9), Model 1508, visicorder oscillograph.
- . Connector Box -- (see Figure 8)

Strain measurements were made by first draining that portion of the system under test to obtain a "no-load" condition. The strain gages were then affixed to the pipe at the desired location with an epoxy cement. The cement was allowed to cure, electrical connections were made at the gage terminals and at the strain measuring equipment, and pressure and flow were restored to the system. Strain measurement readings were then recorded at various conditions of flow and pressure up to normal maximum operating conditions. At the completion of the measurements, the system was returned to "no-load" condition and gages checked for zero. In every case the gages zeroed out indicating elastic behavior in the piping materials.

The three readings from a given rosette are then included in a computer program entitled STRESS⁽⁶⁾ which converts these three readings to the maximum principal stress, the minimum principal stress, the shear stress, and the angle of orientation of the principal stress (see Table III). This method of determining the principal stress provides the resolved bi-axial principal stress which is usually slightly larger than the calculated value because all of the local point stress are resolved into the maximum principal stresses. The experimental strain measurement system used here has a resolution capability of 0.000001 in/in which is higher than is required by recent issues of the Codes (0.00005 in/in).

Vibrational strain measurements were resolved using the same mathematical relationships. The equipment used in obtaining vibrational strain measurements are slightly different from static strain measurements. The equipment is assembled to eliminate static values and record only the dynamic strain oscillations, each three gage rosette is recorded on a three channel recorder. This system picks up all frequencies and records the total strain values which are converted to the principal stress values and dynamic angle by the same computer program as used for static strain computations.

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TABLE III

STRAIN ANALYSIS PROGRAM

A computer code for the resolution of strain measurements into their principal vector values is described in DUN-5862. This computer code is named "STRESS" and is based on the vectorial resolution relationships founded by Mohr's theory.

The mathematical equations used for the 60° Delta, Rosette strain measuring gage is included below:

$$\sigma_{\text{max. Principal}} = E \left[\frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3(1-\mu)} + \frac{1}{1+\mu} \sqrt{\left(\epsilon_1 - \frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3} \right)^2 + \left(\frac{\epsilon_2 - \epsilon_3}{\sqrt{3}} \right)^2} \right]$$

$$\sigma_{\text{min. Principal}} = E \left[\frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3(1-\mu)} - \frac{1}{1+\mu} \sqrt{\left(\epsilon_1 - \frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3} \right)^2 + \left(\frac{\epsilon_2 - \epsilon_3}{\sqrt{3}} \right)^2} \right]$$

$$T_{\text{max. Shear}} = \frac{E}{1+\mu} \sqrt{\left(\epsilon_1 - \frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3} \right)^2 + \left(\frac{\epsilon_2 - \epsilon_3}{\sqrt{3}} \right)^2}$$

$$\text{Angle with respect to the No. 1 gage orientation} = \frac{1}{2} \text{Arctan} \left[\frac{\frac{1}{\sqrt{3}} (\epsilon_2 - \epsilon_3)}{\epsilon_1 - \frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3}} \right]$$

These relationships are coded for the 1108 Univac computer and are named in the data reports as follows:

C1, C2, C3	Gage Conversion Constants
$\epsilon_1, \epsilon_2, \epsilon_3$	Strain Measurements (micro inches)
μ	Poisson Ratio
E	Young's Elastic Modulus lbs/in ²
$\sigma_{\text{max.}}$	Principal Max. Stress lbs/in ²
$\sigma_{\text{min.}}$	Principal Min. Stress lbs/in ²
T max.	Maximum Shear Stress lbs/in ²
Angle	Reference Angle to the No. 1 Gage (Prints out in degrees)

A brittle coating was applied to some of the typical branch connections to verify that the strain gages were placed within the zones of highest stress in fittings and at various pipe and hanger locations. The results are considered qualitative only and are used only to identify extent of high stress zones. The threshold cracking strengths of the plastic films used were 800 and 1,000 micro inches per inch tensile strain. A typical application showing effect on the brittle coating of operating pressure in the pipe is shown in Figures 10 and 11.

Stress Concentration Factors at Branch Connections

The Pressure Vessel Research Committee (PVRC) has for approximately thirteen years sponsored extensive studies of stresses at openings in pressure vessels under internal pressure. The greater part of this work has been accomplished using machined, three-dimensional photoelastic models. The studies have resulted in the development of stress concentration factors for various locations around openings in pressure vessels. These concentration factors when applied to the nominal membrane stress in the vessel will provide the stress to be expected at the point near the vessel opening corresponding to the point in the idealized model at which the factor was obtained. The concentration factors developed by PVRC have been utilized in this study to determine the probable working stresses in the 24-inch and 36-inch branch connections at locations at which it was not possible to install strain gages to obtain actual strain measurements. Thus, numerical values have been obtained for the probable maximum stress at each of the branch connections in the K Reactor coolant piping.

The apparent validity of the comparison between measured stresses at the outer surface of the piping and stresses obtained through application of the concentration factors to calculated membrane stresses permits extension of the analysis to the inner surfaces of the piping by application of similarly obtained concentration factors related to the inner surfaces of the idealized models. It is at the inner surfaces of the branch connections under discussion, where localized stress concentrations will be most severe (except where yielding has occurred), that failure will most likely be initiated. Concentration factors at interior discontinuities (such as at the acute angle of re-entrant branch connections such as the 24" x 24", 45 degree laterals) can range as high as six and, when applied to the nominal hoop membrane stress in the 24-inch piping (5,500 psi) indicate stress ranges as high as 33,000 psi (yield strength 35,000 psi). It is apparent that some localized regions where measured stress ranges approach or exceed the yield point (24" x 36" wye) probably yielded during the initial hydrostatic test and the indicated stress range in those regions includes a significant residual stress in the opposite direction. A summary of the maximum stress values and the location of the maximum stress is shown in Table I, page 3, for each of the branch connections, except for the 24-inch x 36-inch branch connections in the A and D riser approach lines. These connections are not of a geometry recognized by current codes, and no stress concentration factors are available which are applicable. These connections present the highest measured stress ranges in the entire piping system.

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It should be noted that the stress concentration factors discussed above were obtained experimentally through application of internal hydraulic pressure and are based on the equivalent nominal hoop stress in straight pipe. They are therefore to be applied only to the nominal hoop stress due to internal pressure. Where stresses due to moment producing loads are encountered at branch connections, stress intensification factors from USAS B31.1.0 were utilized (see Table IV).

The measured stress ranges at the pipe fittings are not unusual for fittings of this type. Tests under the sponsorship of the ASME Pressure Vessel Research Committee on full size standard fittings have shown stresses three times the nominal hoop stress at the interior corner of a 90° branch connection. As the angle of entry becomes more acute or if the fitting is unreinforced, the concentration of stress at the acute corner can range to approximately six times the nominal hoop stress. The report on these PVRC tests has not yet been published.

Thermal Stress

The thermal stresses in the coolant supply piping systems were determined through use of the computer code MEC-21, "Piping Flexibility Analysis Program," developed through the cooperative efforts of the Mare Island Naval Shipyard Design Division and the Los Alamos Scientific Laboratory. The program was adapted for use in the Univac 1108 computer for this study. The computational methods used in the program are based on tensor analysis treatment of Castigliano's theorem of structural analysis⁽⁷⁾. The program is widely used by the U. S. Navy and the Los Alamos Laboratory and was used by General Electric Company in the design of N Reactor piping. The program provides a tabulation of stresses and forces at selected points in the piping system such as fittings, bends, and supports.

For purposes of analysis, the system was analyzed in three sections; from the high lift pumps to the valve pit in the 165-K, from the valve pit to the reactor risers, and the risers. The system was considered fixed at the pumps, fixed at the base of the inlet risers, and, since the piping in 165-K is relatively rigid (tie-down straps at all supports), fixed in 165-K.

The maximum thermal stress in the K Reactor inlet coolant piping was determined to be 1,400 psi, based on a 35 F thermal gradient. This maximum axial stress point occurred in the vertical mitered 15° bend in the 36-inch tunnel piping.

The nominal thermal stress in the section between the pumps and the valve pit in 165-K was determined to be 1,200 psi at the high lift pump discharge elbow. In the risers, at the crossheader connections, a negligible stress is induced as a result of thermal expansion of the crossheaders.

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TABLE IV
SUMMARY OF NORMAL OPERATING STRESSES

	STRESS DESCRIP.	CALC. STRESS AT 462 PSI TORP	VIB. PRESS. STRESS	PRESS. STRESS CONC. FACTOR*	COMPUTED MAX. STRESS FROM PRESSURE	MOMENT-INDUCED STRESS (T+D)		MOMENT STRESS CONC. FACTOR**	COMPUTED MOMENT STRESS (T+D) (CF)	RESIDUAL STRESS	TOTAL NORMAL OPERATING STRESS (I+II+III)
						(T) THERMAL	(D) DEAD LOAD				
1. 24" PIPE	AXIAL	2600	300	1	2900	1200	800	--	2000	4000	8,900
	CIRCUM.	5800	300	1	6100	--	--	--	--		10,100
2. 36" PIPE	A	3100	200	1	3300	1400	800	--	2200		9,500
	C	6800	200	1	7000	--	--	--	--		11,000
3. 36" RISER	A	3000	100	1	3100	NEG.	NEG.	--	--		7,100
	C	6700	100	1	6800	NEG.	NEG.	--	--		10,800
4. 36" RISER AT CROSSHEADER	A	3000	150	1	3150	NEG.	NEG.	--	--		7,150
	C	6700	150	1.3	8850	NEG.	NEG.	2.0	--		12,850
5. 36" RISER ELLIPT. HEAD	-	--	--	--	--	--	--	--	--		--
	-	6250	100	1	6350	--	--	--	--		10,350
6. 36" RISER HEAD AT NOZZLE	-	--	--	--	--	--	--	--	--		--
	-	6250	150	1.5	9550	--	--	--	--		13,550
7. 36" RISER HEMI. HEAD	-	--	--	--	--	--	--	--	--		--
	-	3150	100	1	3250	--	--	--	--		7,250
8. 36" RISER HEMI. HEAD AT NOZZLE	-	--	--	--	--	--	--	--	--		--
	-	3150	150	1.4	4550	--	--	--	--		8,550
9. 36" ELBOW RISER BASE	A	3100	100	--	3200	1200	--	1.0	1200		8,400
	C	6800	100	--	6900	--	--	1.0	--		10,900
10. 36" ELBOW AT TUNNEL	A	3100	200	--	3300	1200	800	2.2	4400	...	11,700
	C	6600	200	--	6800	--	--	1.0	--	...	10,800
11. 36X36X24 AT TUNNEL	A	3100	200	--	3300	1200	800	2.1	4200		11,500
	C	6600	200	4.5****	31,700	--	--	1.0	--		35,700
12. 24" ELL AT TUNNEL	A	2600	200	--	2800	1200	800	2.2	4400	...	11,200
	C	5800	200	--	6000	--	--	1.0	--	...	10,000
13. 36X36X24 BRANCH FLUSH CONN.	A	3100	200	--	3300	NEG.	800	2.0	1600		8,900
	C	6800	200	--	7000	NEG.	--	1.0	--		11,000
14. 24X24X24 45° LATERAL	A	2600	300	--	2900	1200	800	2.5	5000		11,900
	C	5800	300	5.9	37,700	--	--	1.0	--		41,700
15. 24X24X24 TEE	A	2600	300	--	2900	1200	800	2.1	4200		11,100
	C	5800	300	4.1	24,100	--	--	1.0	--		27,800
16. 24X24X12 TEE	A	2600	300	--	2900	1200	800	2.1	4200		11,100
	C	5800	300	2.0	11,900	--	--	1.0	--		15,900
17. 24" ELL AT PUMP	A	2600	300	--	2900	--	800	2.2	1760	...	8,660
	C	5800	300	--	6100	--	--	1.0	--	...	10,100

NOTE: STRESSES ARE IN POUNDS PER SQUARE INCH.

* STRESS CONCENTRATION FACTORS BASED ON PHOTOELASTIC MODELS. (SEE TABLES VII THROUGH XIV.

** STRESS CONCENTRATION FACTORS FROM USAS B 31.1.0 CODE FOR POWER PIPING, 1967.

*** RESIDUAL STRESSES IN FORGED FITTINGS ARE ACTUALLY EXPECTED TO BE NEGLIGIBLE.

**** ESTIMATED -- NO MODEL DATA AVAILABLE.

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The thermal stress in the crossheader-to-riser assembly was also analyzed by means of the MEC 21. The program considered thermal expansion in all piping components and effects of moments at all fittings due to thermal expansion and computed the resulting moment at the crossheader to riser juncture. The crossheader was assumed to be empty and weightless to obtain purely thermal effects. A maximum thermal range of 35 F has been considered.

Vibrational Stress

Vibrational stresses were determined through use of strain gages, and the same computational methods were used as in determination of the hydraulic pressure stresses as presented in Table III, page 9. The coupling arrangement of the strain gages and equipment used in resolution and readout were modified, however, to eliminate the static strain values and provide only that increment of strain due to dynamic oscillations. Incremental strain values were resolved into principal vibrational stress components. Vibrational stress in the pipe wall varied from 600 psi, measured at the pumps, to 150 psi, measured at the inlet risers. The predominant vibrational frequency was 165 to 180 cycles per second with a maximum amplitude of 18.6 μ -inches/inch. The amplitudes of vibration steadily diminished as distance from the pumps increased until at the risers strain was measured at 4.4 μ -inches/inch. There were no low frequency large amplitude vibrations detected in the system. The locations where these measurements were made are presented in Figure 12.

Structural Load Stress

Structural dead load stresses imposed on the piping by the pipe support system result in added stress due to bending. The maximum stress due to bending does not exceed 800 psi. This was determined by an analytical evaluation of support loadings as illustrated in Appendix A, Sample Calculations.

Stress in the riser due to structural load is limited to that stress resulting from the weight of a short section of crossheader (full of water) which exerts a vertical moment at the crossheader-to-riser juncture. This moment is considered to result from the weight of the crossheader between the last support and the riser.

Another structural load stress is the compressive stress in the lower section of the riser due to the supported dead weight of the riser section. This stress has been computed to be a maximum of 203 psi. Since it is a compressive stress and acts to reduce the predominant axial stresses, it has not been considered in the stress summation as an added degree of conservatism.

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Residual Stress

The presence of residual stresses in the piping system was also considered. Residual stresses normally result from incompatibilities of strain within a material rather than from external loading. In piping, the two primary sources of residual stress are 1) rolling of the plate into a cylindrical shape, and 2) welding. All pipe used in the K coolant systems was stress relieved after forming and welding at 1,150 F for one hour per inch of thickness to minimize the residual stresses. Field welds were similarly stress relieved. The cyclic hydrostatic test, 900 psig for the total system and 1,000 or 1,200 psig for pre-fabricated items such as the risers and mitered elbows, further served to relax these residual stresses.

Applying creep data, residual stresses at weldments in the K piping do not exceed 4,000 psi. Although data are not available for the exact materials used in these piping systems, data from typical low carbon steels shown in Figure 15 reveals relaxation of stress to approximately 4,000 psi at stress relief temperatures.⁽⁸⁾

Values of residual stress in the risers remote from the welds were obtained experimentally when material was removed for purposes of fracture toughness testing. A section of the riser was removed, ends machined, and slit longitudinally. Applying the Crampton Deflection Technique⁽¹⁵⁾, the internal stress was determined to be approximately 300 psi. Figure 14 presents the detailed results of this test. The residual stress thus measured is not, however, representative of the maximum residual stress in the piece (i.e., at the welds and heat affected zones) but, rather, represents the residual stress in the pipe section at the slit.

Investigation into the effects of preheat and thermal and mechanical stress relief in welded structures on residual stresses has been reported by Nordell and Hall.⁽¹⁴⁾ It is shown that thermal stress relief after welding results in a significant decrease in residual stress while preheat or mechanical stress relief were far less beneficial. Residual stress (longitudinal) in a welded specimen was reduced by thermal stress relief at 1,150 F to approximately 1/6 of the stress measured in the as-welded condition at the centerline of the weld. Approximately four inches from the centerline of the weld, residual stress was zero. Transverse residual stress following thermal stress relief averaged less than 2,000 psi (see Figure 16).

At the intersection of longitudinal and girth welds in the 36-inch piping under study, this same analysis indicates that circumferential residual stress resulting from the girth weld would not be reduced by thermal stress relief below the transverse stress in the longitudinal weld; i. e., approximately 2,000 psi. This relationship is shown in Figure 17.

The residual stress used in this study was 4,000 psi which is considered conservative. A lower value may be more realistic, although published data does not support the use of the lower value. Larger residual stresses exist at

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pipe junctions, but these stresses are of a localized nature. Fracture toughness testing of the 36" pipe material has led to a preliminary relationship between the nominal stress required to propagate an existing crack and the existing crack length.⁽¹⁶⁾ For a crack 4" long, the nominal stress would have to exceed 12,000 psi to propagate the crack. If the crack were to grow to 6-inches long, the stress required to propagate it would be 9,000 psi. Current information indicates that the high local stresses reduce to nominal values (7,000 psi) in distances shorter than the critical lengths mentioned above.

Emergency Condition Stress

Under accident, unusual event, or partial system failure conditions, the piping system could be subjected to a number of additional loads beyond those considered during normal operation. The results of an evaluation of these additional stresses resulting from these conditions are summarized in Tables V and VI. It should be noted that the particular stresses which would be present in a particular event would depend on the nature of the event. However, for evaluation purposes, the maximum total stress listed in the tables represents the combination of all of the stress factors considered. The method utilized in the development of these stresses is illustrated in Appendix B.

In addition to the normal operating stresses discussed above, stresses which might be imposed on the piping systems during accident or emergency conditions must be considered. These factors are discussed in the following sections.

Earthquake

Basic criteria currently applied to the reactor plants regarding earthquake protection are as follows.

- a) The operational basis earthquake (OBE) is defined as the most severe earthquake expected during the life of the plant. The currently accepted intensity of such an earthquake is one which would result in a peak ground acceleration of 0.12g.
- b) The design basis earthquake (DBE) is defined as the most destructive earthquake that would ever be expected to occur at the site. The currently accepted intensity of such an earthquake is one which would result in a peak ground acceleration of 0.25g.

The above criteria have been recommended by Newmark (1968)⁽⁹⁾ and USC&GS (1967).⁽¹⁰⁾

In determining the maximum stress which might be sustained in the piping systems, a damping factor of 2 percent was assumed.⁽¹¹⁾ The natural frequencies of various portions of the piping system were calculated using Euler's method for clamped, continuous beams (see Figures 19 and 20, Appendix B).

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TABLE V
SEISMIC STRESS SUMMARY

	<u>STRESS DESCRIP.</u>	<u>CALC. SEISMIC STRESS</u>	<u>MOMENT CONC. FACTOR</u>	<u>FACTORED SEISMIC STRESS</u>	<u>TOTAL NORMAL OPERATING STRESS (TABLE IV)</u>	<u>TOTAL STRESS DURING DBA QUAKE</u>
1. 24" PIPE	AXIAL	400	-	400	8,900	9,300
	CIRCUM.	-	-	-	10,100	10,100
2. 36" PIPE	A	250	-	250	9,500	9,750
	C	-	-	-	11,000	11,000
3. 36" RISER	A	-	-	-	7,100	7,100
	C	NEG.	-	-	10,800	10,800
4. 36" RISER A1 CROSSHEADER	A	-	-	-	7,150	7,150
	C	NEG.	2.0	-	12,900	12,900
5. 36" RISER ELLIPT. HEAD	-	-	-	-	-	-
	-	-	-	-	10,350	10,350
6. 36" RISER AT HEAD NOZZLE	-	-	-	-	-	-
	-	-	-	-	13,600	13,600
7. 36" RISER HEMI. HEAD	-	-	-	-	-	-
	-	-	-	-	7,250	7,250
8. 36" RISER HEMI. HEAD AT NOZZLE	-	-	-	-	-	-
	-	-	-	-	8,550	8,550
9. 36" ELBOW AT RISER BASE	A	-	1.0	-	8,400	8,400
	C	-	1.0	-	10,900	10,900
10. 36" ELBOW AT TUNNEL	A	250	2.2	550	11,700	12,250
	C	-	1.0	-	10,800	10,800
11. 36X36X24 AT TUNNEL (45°)	A	400	2.1	840	11,500	12,340
	C	-	1.0	-	35,700	35,700
12. 24" ELL AT TUNNEL	A	400	2.2	880	11,200	12,080
	C	-	1.0	-	10,000	10,000
13. 36X36X24 BR. FLUSH CONN.	A	250	1.0	250	8,900	9,150
	C	250	1.0	250	11,000	11,250
14. 24X24X24 45° LATERAL	A	400	2.5	1000	11,900	12,900
	C	-	1.0	-	41,700	41,700
15. 24X24X24 TEE, 190 BLDG.	A	400	2.1	840	11,100	11,940
	C	-	1.0	-	27,800	27,800
16. 24X24X12 TEE, 190 BLDG.	A	400	2.1	840	11,100	11,940
	C	-	1.0	-	15,900	15,900
17. 24" ELL AT PUMP	A	400	2.2	880	8,860	9,540
	C	-	1.0	-	10,100	10,100

(SEE FOOTNOTES ON TABLE IV)

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TABLE VI
PRESSURE TRANSIENT STRESS SUMMARY

	STRESS DESCRIP.	VIB. PRESS. STRESS. FROM TABLE IV	PRESS. TRANS- IENT STRESS	PRESS. STRESS CONC. FACTOR	COMPUTED MAX. STRESS FROM PRESSURE	COMPUTED MOMENT & RESIDUAL STRESSES TABLE IV	TOTAL STRESS DURING PRESSURE TRANSIENT
1. 24" PIPE	AXIAL	300	3,330	1.0	3,630	6,000	9,630
	CIRCUM.	300	7,320	1.0	7,620	4,000	11,620
2. 36" PIPE	A	200	3,950	1.0	4,150	6,200	10,350
	C	200	8,550	1.0	8,750	4,000	12,750
3. 36" RISER	A	100	3,810	1.0	3,910	4,000	7,910
	C	100	8,250	1.0	8,350	4,000	12,350
4. 36" RISER AT CROSSHEADER	A	150	3,810	1.0	3,960	4,000	7,960
	C	150	8,250	1.3	10,850	4,000	14,850
5. 36" RISER ELLIPT. HEAD	-	-	-	-	-	-	-
	-	100	7,700	1.0	7,800	4,000	11,800
6. 36" RISER AT HEAD NOZZLE	-	-	-	-	-	-	-
	-	150	7,700	1.5	11,700	4,000	15,700
7. 36" RISER HEMI. HEAD	-	-	-	-	-	-	-
	-	100	3,870	1.0	3,970	4,000	7,970
8. 36" RISER HEMI. HEAD AT NOZZLE	-	-	-	-	-	-	-
	-	150	3,870	1.4	5,570	4,000	9,570
9. 36" ELBOW AT RISER BASE	A	100	3,950	-	4,050	5,200	9,250
	C	100	8,550	-	8,650	4,000	12,650
10. 36" ELBOW AT TUNNEL	A	200	3,950	-	4,150	8,400	12,550
	C	200	8,550	-	8,750	4,000	12,750
11. 36X36X24 AT TUNNEL (45°)	A	200	3,950	-	4,150	8,200	12,350
	C	200	8,550	4.5****	38,600	4,000	42,600
12. 24" ELL AT TUNNEL	A	200	3,330	-	3,530	8,400	11,930
	C	200	7,320	-	7,520	4,000	11,520
13. 36X36X24 BR. FLUSH CONN.	A	200	3,950	-	4,150	5,600	9,750
	C	200	8,550	-	8,750	4,000	12,750
14. 24X24X24 45° LATERAL	A	300	3,330	-	3,630	9,000	12,630
	C	300	7,320	5.9	43,500	4,000	47,500
15. 24X24X24 TEE, 190 BLDG.	A	300	3,330	-	3,630	8,200	11,830
	C	300	7,320	4.1	30,300	4,000	34,300
16. 24X24X12 TEE, 190 BLDG.	A	300	3,330	-	3,630	8,200	11,830
	C	300	7,320	2.0	14,940	4,000	18,940
17. 24" ELL AT PUMP	A	300	3,330	-	3,630	5,760	9,400
	C	300	7,320	-	7,620	4,000	11,620

(SEE FOOTNOTES ON TABLE IV)

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These natural frequencies (varying from 3.4 to 51 cycles per second, Period = 0.3 seconds to 0.02 seconds) were applied, together with the damping factor, to the response acceleration spectra based on Housner's recommendations (see Figure 13). The data curves in Figure 13 are translated to the configuration shown from the basic Housner data.

From the above data, a factor of 0.6g was obtained and used for the static analysis of various portions of the piping systems to determine the maximum stress which might result from a seismic event of this magnitude. The maximum stress was calculated to be 400 psi in the 24-inch piping system, 250 psi in the 36-inch piping, and 250 psi in the risers. At points of stress concentration (fittings or other discontinuities), the stress would be higher depending on the concentration factors as shown in Table IV.

Pressure Transient

Pressure transients experienced during routine operation of the plant are negligible, and, more importantly, are limited to or below the normal system pressure. There are no quick-acting, mechanically actuated valves present in the system. Flow transients through the check valves are very slow in relation to the check valve closing time, so that no check valve slam can occur.

The pressure transient of significant concern in evaluating the piping system is the transient following a hypothetical major pipe break accident. In this case relatively rapid flow transients occur, including a flow reversal through a 24-inch check valve. The analysis of the pressure wave which occurs during this event indicates that this is the maximum emergency stress which would occur in the piping system.⁽³⁾

The analytic methods utilized to determine the magnitude of the pressure pulse which might result from check valve closure following a pipe break accident are presented in HW-74095, Supp. 5.⁽³⁾ Momentum balance calculations were utilized to establish the pressure transient in the system. A dynamic analysis of check valve closure as a function of time and flow rate was used to establish the time (310 milliseconds) at which the valve is closed.

The maximum internal pressure in the 36-inch pipe during the transient is $250 + 360 = 610$ psig, resulting in a maximum stress of 7,320 psi in the 24-inch piping and 8,550 psi in the 36-inch piping. The pipe stresses associated with this value are given in Table VI.

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C. W. Harrison, Sr. Engineer
Process Design

:tba

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STRESS ANALYSIS OF 100-K
PRIMARY PIPING

APPENDIX A

UNCLASSIFIED

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LEGEND: 571

105-K

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PARTIAL PLAN

VIEW C-C

JAFS, FOR DISSEMINATION WATER DESIGN CATEGORY		DATE		U. S. ATOMIC ENERGY COMMISSION RICHLAND OPERATIONS OFFICE	
BY JAFS		TITLE		DOUGLAS UNITED NUCLEAR, INC.	
APPROVED C. W. HARRISON		DATE 11/10/62		PROJECT STRESS ANALYSIS STUDY	
CHECKED J. M. MEYER		DATE 11/10/62		DRAWING NO. 190-K TO 105-K	
CLASSIFICATION UNCLASSIFIED		PROJECT NO. R 400		DRAWING NO. H-1-72220	

[illegible]

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IN P.S.I. AT 405 TOP OF RISER PRESSURE
IN P.S.I. AT 393 TOP OF RISER PRESSURE
IN P.S.I. AT 411 TOP OF RISER PRESSURE

REV. NO.	DATE	DESCRIPTION
1	10/1/68	ISSUED FOR CONSTRUCTION
2	10/1/68	REVISION

BY DATE	FOR	BY DATE	FOR	BY DATE	FOR
10/1/68	10/1/68	10/1/68	10/1/68	10/1/68	10/1/68
DESCRIPTION					
2 B-E WAS 26758. NSW 47247					
REVISIONS					
CHECK PAY ISSUE NO					
DRAWING STATUS					

APPROVAL FOR CONSTRUCTION DATE 10/1/68		U. S. ATOMIC ENERGY COMMISSION RICHLAND OPERATIONS OFFICE	
DOUGLAS UNITED NUCLEAR, INC.		PROCESS PIPING STRESS ANALYSIS STUDY	
SCALE NONE		190-KE TO 105-KE	
H-1-72194		H-1-72194

FIGURE 3
100-K PRIMARY COOLANT SUPPLY PIPING

DELTA ROSETTE

RECTANGULAR ROSETTE

FIGURE 4. STRAIN GAGE ORIENTATION (SHOWING
GAGE MEASURING AXIS)

STRAIN GAGE
60 Degree Delta Rosette
Micro Measurements Co.
Type EA-06-250YA-120

STRAIN GAGE
45 Degree (rectangular) Rosette
(with solder tabs)
Micro Measurements Co.
Type EA-06-125-RA-120
Gage Factor 2.08 1.5% Nom.

TYPICAL STRAIN GAGE INSTALLATION

EQUIPMENT USED FOR STATIC STRAIN READING
 INDICATOR, CONNECTOR BOX,
 CABLE TO GAGES

EQUIPMENT USED TO RECORD DYNAMIC STRAIN
CONNECTOR BOX, INDICATOR, AMPLIFIER,
RECORDER.

BRITTLE COATING

36" X 24" Branch Connection at 36" 90° Elbow

The photograph reveals the extent of the high stressed area. The stress coat used had a crack sensitivity threshold of 800 micro inches (24,000 psi) and indicates these areas of piping to be stressed above that level. See Figure A-11 for close-up photo of outlined area.

BRITTLE COATING
36" X 24" Branch Connection at 36" 90° Elbow

FIGURE 12
LOCATION OF STRAIN GAGES
FOR
VIBRATION ANALYSIS

190 - K

FIGURE 13. SEISMIC RESPONSE ACCELERATION SPECTRA

FIGURE 14

RESIDUAL STRESS DETERMINATION

MEASUREMENT ACCURACY
0.0005"

Crampton Deflection Technique
for Residual Stress Determination

$$S = E't \left(\frac{1}{D_o} - \frac{1}{D_1} \right)$$

$$= \frac{29.5 \times 10^6}{1-(0.3)^2} \times 1.25 \left(\frac{1}{36.00} - \frac{1}{36.01} \right)$$

$$= 313 \text{ psi}$$

Effect of Tempering Time and Temperature on the Remaining Stress in Carbon Steel Boiler Plate Prestressed to Yield (25,000psi) Before Tempering⁽⁸⁾

FIGURE 15

--Distribution of residual stress in a 1 in. thick thermally stress-relieved specimen. A (left)—distribution of average longitudinal residual stress across plate width; B (right)—distribution of residual stress through thickness of weld

REFERENCE: TWO STAGE FRACTURING IN MILD STEEL PLATES,
W. J. NORDELL AND W. J. HALL,
WELDING RESEARCH SUPPLEMENT - MARCH 1965

FIGURE 16

PROBABLE RESIDUAL STRESS DISTRIBUTION IN THERMALLY STRESS-RELIEVED
GIRTH AND LONGITUDINAL WELDS IN 36-INCH PIPING.

FIGURE 17

PROBABLE RESIDUAL STRESS DISTRIBUTION AT INTERSECTION
OF LONGITUDINAL AND GIRTH WELDS IN 36-INCH PIPING
(THERMALLY STRESS RELIEVED)

190-KE

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STRESS RUN NO. 1392-9
TEST NO. 3
RECTANGULAR STRAIN GAGES

WRC
BULLETIN #113
FIG. # P-4D & E-6

CALCULATED HOOP STRESS = 6700 PSI

GAGE NO.	MEASURED STRESS (PSI)		CONCENTRATION FACTOR RANGE		COMPUTED STRESS RANGE (PSI)	
	CIRCUM.	AXIAL	CIRCUM.	AXIAL	CIRCUM.	AXIAL
21	5480	2095	0.8 - 1.8	0.6 - 1.5	5360 - 12080	4020 - 10050
22	7105	6370	0.5 - 1.3	0.5 - 1.0	3340 - 8710	3340 - 6700

1. CONCENTRATION FACTORS ARE DERIVED FROM PHOTOELASTIC MODEL STUDIES SPONSORED BY THE ASME PRESSURE VESSEL RESEARCH COMMITTEE (PVRC) AND REPORTED IN WELDING RESEARCH COUNCIL BULLETIN NO. 113, APRIL 1966.
2. COMPUTED STRESSES OBTAINED BY MULTIPLYING CALCULATED STRESS X CONCENTRATION FACTOR.

TABLE VII. BRANCH CONNECTION STRESS ANALYSIS - 36" x 5" x 90°

STRESS RUN NO. 1392-8
TEST NO. 4
DELTA STRAIN GAGES

WRC
BULLETIN #113
FIG. # C-8A

CALCULATED HOOP STRESS = 5800 PSI

GAGE NO.	MEASURED STRESS (PSI)		CONCENTRATION FACTOR RANGE		COMPUTED STRESS RANGE (PSI)	
	CIRCUM.	AXIAL	CIRCUM.	AXIAL	CIRCUM.	AXIAL
11	13000	5560	1.2 - 2.0	1.0 - 1.3	6955 - 11600	5800 - 7540
12	20225	15560	1.4 - 2.2	1.4 - 2.8	8120 - 12760	8120 - 16210
13	11240	67	0.8 - 1.5	0.3 - 0.6	4640 - 8700	1740 - 3480
14	10495	33	0.8 - 1.3	0.2 - 0.6	4640 - 7540	1160 - 3480

1. CONCENTRATION FACTORS ARE DERIVED FROM PHOTOELASTIC MODEL STUDIES SPONSORED BY THE ASME PRESSURE VESSEL RESEARCH COMMITTEE (PVRC) AND REPORTED IN WELDING RESEARCH COUNCIL BULLETIN NO. 113, APRIL 1966.
2. COMPUTED STRESSES OBTAINED BY MULTIPLYING CALCULATED STRESS X CONCENTRATION FACTOR.

TABLE VIII. BRANCH CONNECTION STRESS ANALYSIS - 24" x 24" x 90°

STRESS RUN NO. 1392-9
TEST NO.
RECTANGULAR STRAIN GAGES

GAGE NO.	MEASURED STRESS (PSI)		CONCENTRATION FACTOR RANGE		COMPUTED STRESS RANGE (PSI)	
	CIRCUM.	AXIAL	CIRCUM.	AXIAL	CIRCUM.	AXIAL
7	10715	2740	PHOTOELASTIC MODEL ANALYSIS NOT APPLICABLE TO THIS FITTING BECAUSE OF ITS UNORTHODOX GEOMETRY.			
8	14300	-1209				
9	46800	6195				
10	5050	7145				
11	11440	-6110				
35	34950	5020				
36	18690	12850				
37	27650	9750				
38	33090	9060				
39	28000	-2478				
40	-3750	-7820				
41	-13650	1888				

TABLE IX. BRANCH CONNECTION STRESS ANALYSIS
36" x 24" x 45°

STRESS RUN NO. 1392-9
TEST NO. 3
RECTANGULAR STRAIN GAGES

WRC
BULLETIN #113
FIG. # C-7C & UW-1

CALCULATED HOOP STRESS (36") = 6700 PSI
CALCULATED HOOP STRESS (24") = 5800 PSI

GAGE NO.	MEASURED STRESS (PSI)		CONCENTRATION FACTOR RANGE		COMPUTED STRESS RANGE (PSI)	
	CIRCUM.	AXIAL	CIRCUM.	AXIAL	CIRCUM.	AXIAL
1	11950	-30	0.8 - 1.2	0.4 - 0.6	5350 - 8050	2680 - 4020
2	12750	2509	1.6 - 2.2	0.6 - 1.2	9280 - 12750	3480 - 6960
4	8950	6845	2.5 - 3.2	2.5 - 3.6	14500 - 18600	14500 - 20900
5	12180	2919	1.8 - 2.2	1.0 - 2.0	10430 - 12750	5800 - 11600
6	8380	2095	1.0 - 1.6	0.5 - 1.0	6700 - 10700	3350 - 6700

1. CONCENTRATION FACTORS ARE DERIVED FROM PHOTOELASTIC MODEL STUDIES SPONSORED BY THE ASME PRESSURE VESSEL RESEARCH COMMITTEE (PVRC) AND REPORTED IN WELDING RESEARCH COUNCIL BULLETIN NO. 113, APRIL 1966.
2. COMPUTED STRESSES OBTAINED BY MULTIPLYING CALCULATED STRESS X CONCENTRATION FACTOR.

TABLE X. BRANCH CONNECTION STRESS ANALYSIS - 36" x 24" x 65°

STRESS RUN NO. 1392-8 WRC
TEST NO. 4 BULLETIN #113
DELTA STRAIN GAGES FIG. # 8C & UW-1

CALCULATED HOOP STRESS = 5800 PSI

GAGE NO.	MEASURED STRESS (PSI)		CONCENTRATION FACTOR RANGE		COMPUTED STRESS RANGE (PSI)	
	CIRCUM.	AXIAL	CIRCUM.	AXIAL	CIRCUM.	AXIAL
17	7380	1828	0.8 - 1.6	0.8 - 1.0	4640 - 9280	4640 - 5800
18	9708	2300	0.1 - 1.8	0.6 - 1.2	580 - 10430	3480 - 7060
19	13877	1828	2.0 - 2.6	0.4 - 0.8	11600 - 15200	2320 - 4640
20	14258	10200	2.0 - 2.4	1.0 - 2.5	11600 - 13900	5800 - 14500
21	11330	648	0.9 - 1.6	1.0 - 1.4	5220 - 9280	5800 - 8110

1. CONCENTRATION FACTORS ARE DERIVED FROM PHOTOELASTIC MODEL STUDIES SPONSORED BY THE ASME PRESSURE VESSEL RESEARCH COMMITTEE (PVRC) AND REPORTED IN WELDING RESEARCH COUNCIL BULLETIN NO. 113, APRIL 1966.
2. COMPUTED STRESSES OBTAINED BY MULTIPLYING CALCULATED STRESS X CONCENTRATION FACTOR.

TABLE XI. BRANCH CONNECTION STRESS ANALYSIS - 24" x 24" x 45°

STRESS RUN NO. 1392-8 TEST NO. 4 DELTA STRAIN GAGES	WRC BULLETIN #113 FIG. # 8A
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CALCULATED HOOP STRESS = 5800 PSI

GAGE NO.	MEASURED STRESS (PSI)		CONCENTRATION FACTOR RANGE		COMPUTED STRESS RANGE (PSI)	
	CIRCUM.	AXIAL	CIRCUM.	AXIAL	CIRCUM.	AXIAL
7	9694	5960	1.2 - 2.0	1.0 - 1.3	6955 - 11600	5800 - 7540
8	20690	15110	1.4 - 2.2	1.4 - 2.8	8120 - 12760	8120 - 16210
9	13410	2978	0.8 - 1.5	0.3 - 0.6	4640 - 8700	1740 - 3480
10	12870	1843	0.8 - 1.3	0.2 - 0.6	4640 - 7540	1160 - 3480

1. CONCENTRATION FACTORS ARE DERIVED FROM PHOTOELASTIC MODEL STUDIES SPONSORED BY THE ASME PRESSURE VESSEL RESEARCH COMMITTEE (PVRC) AND REPORTED IN WELDING RESEARCH COUNCIL BULLETIN NO. 113, APRIL 1966.
2. COMPUTED STRESSES OBTAINED BY MULTIPLYING CALCULATED STRESS X CONCENTRATION FACTOR.

TABLE XII. BRANCH CONNECTION STRESS ANALYSIS - 24" x 24" x 90°

STRESS RUN NO. 1392-8
TEST NO. 4
DELTA STRAIN GAGES

WRC
BULLETIN #113
FIG. H 4E & E4A

CALCULATED HOOP STRESS (24") = 5800 PSI
CALCULATED HOOP STRESS (12") = 5200 PSI

GAGE NO.	MEASURED STRESS (PSI)		CONCENTRATION FACTOR RANGE		COMPUTED STRESS RANGE (PSI)	
	CIRCUM.	AXIAL	CIRCUM.	AXIAL	CIRCUM.	AXIAL
3	12962	9860	1.6 - 2.4	0.5 - 2.2	8320 - 12480	2600 - 11430
4	9377	502	1.4 - 2.4	1.6 - 2.9	8120 - 13910	9280 - 16800
5	9920	-88	0.8 - 1.8	0.2 - 1.0	4640 - 10420	1160 - 5800
6	6083	354	0.8 - 1.8	0.4 - 1.2	4160 - 9360	2080 - 6250

1. CONCENTRATION FACTORS ARE DERIVED FROM PHOTOELASTIC MODEL STUDIES SPONSORED BY THE ASME PRESSURE VESSEL RESEARCH COMMITTEE (PVRC) AND REPORTED IN WELDING RESEARCH COUNCIL BULLETIN NO. 113, APRIL 1966.
2. COMPUTED STRESSES OBTAINED BY MULTIPLYING CALCULATED STRESS X CONCENTRATION FACTOR.

TABLE XIII. BRANCH CONNECTION STRESS ANALYSIS - 24" x 12" x 90°

STRESS RUN NO. 1392-9
TEST NO. 3
RECTANGULAR STRAIN GAGES

WRC
BULLETIN #113
FIG. # 19D & 19E

CALCULATED HOOP STRESS = 6500 PSI

GAGE NUMBER	MEASURED STRESS (PSI)		CONCENTRATION FACTOR RANGE		COMPUTED STRESS RANGE (PSI)	
	CIRCUM.	AXIAL	CIRCUM.	AXIAL	CIRCUM.	AXIAL
25	4395	2445	0.87	0.4 - 0.5	5650	2600 - 3250
26	-590	-1560	-0.1 - 0.0	0.0 - 0.4	-650 - 0	0 - 2600
27	-2445	-1265	-0.4 - (-0.3)	-0.3 - (-0.2)	-2600 - (-1950)	-1950 - (-1300)
28	561	2300	0.4 - 0.7	0.4 - 0.7	2600 - 4550	2600 - 4550
29	3895	6140	0.4 - 0.9	0.4 - 0.9	2600 - 5850	2600 - 5850
30	6075	7160	1.2 - 1.5	1.2 - 1.5	7800 - 9750	7800 - 9750

1. CONCENTRATION FACTORS ARE DERIVED FROM PHOTOELASTIC MODEL STUDIES SPONSORED BY THE ASME PRESSURE VESSEL RESEARCH COMMITTEE (PVRC) AND REPORTED IN WELDING RESEARCH COUNCIL BULLETIN NO. 113, APRIL 1966.
2. COMPUTED STRESSES OBTAINED BY MULTIPLYING CALCULATED STRESS X CONCENTRATION FACTOR.

TABLE XIV. BRANCH CONNECTION STRESS ANALYSIS - 36" x 4" HEAD

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STRESS ANALYSIS OF 100-K
PRIMARY PIPING
APPENDIX E

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APPENDIX B
SAMPLE CALCULATIONS
K REACTOR PRIMARY PIPING ANALYSIS
STRESS ANALYSIS

To illustrate the method of analysis and summation of stresses shown in Tables IV, V, and VI, the conditions of stress in a typical crossheader-to-riser connection (see sketch) are derived below.

MANUFACTURER TAYLOR FORGE CO.

MATERIAL - ASTM A-105 1952

MINIMUM PHYSICAL VALUES OF MATERIAL	
TENSILE STRENGTH	70,000 psi
YIELD STRENGTH	36,000 psi
ELONGATION (2" MIN %)	22%

NOTE: VENT NOZZLE AT TOP OF RISER SIMILAR
EXCEPT IS 4" NPS

Although not detailed in this tabulation, similar analyses were made upon other portions of the process piping system and the results are reported in Tables IV, V, and VI. Where appropriate, the considerations given to other sections of the piping system are noted.

I. INTERNAL PRESSURE STRESS

Nominal Hoop Stress in Riser

$$\begin{aligned}
 S &= \frac{PD}{2t} - 0.4 P \\
 &= \frac{479 \times 36}{2 \times 1.25} - (0.4)(479) \\
 &= \underline{6700 \text{ psi}}
 \end{aligned}$$

Nominal Axial Stress in Riser

$$\begin{aligned}
 S &= \frac{Pd^2}{(D^2 - d^2)} \\
 &= \frac{479 (33.5)^2}{(36)^2 - (33.5)^2} \\
 &= \underline{3000 \text{ psi}}
 \end{aligned}$$

P = pressure at lowest crossheader
based on a 462 TORP is 479 psig
D = 36-inch outside diameter pipe
t = actual wall thickness
d = inside diameter of 36-inch pipe

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Membrane Stress Due to Pressure Transient from Pipe Break

Hoop Stress in Riser

$$\begin{aligned} S &= \frac{PD}{2t} - 0.4 P \\ &= \frac{590 \times 36}{2 \times 1.25} - (0.4)(590) \\ &= \underline{8550} \text{ psi} \end{aligned}$$

Axial Stress in Riser

$$\begin{aligned} S &= \frac{Pd^2}{(D^2 - d^2)} \\ &= \frac{590(33.5)^2}{(36)^2 - (33.5)^2} \\ &= \underline{3810} \text{ psi} \end{aligned}$$

P = transient pressure
= 610 at 24 x 36 wye
= 590 bottom of riser
= 570 TORP

II. VIBRATIONAL STRESS

The predominant range of frequencies of measured stress generating vibrations lies between 160 cps to 185 cps. The maximum strain measured vectorially within this range of frequencies was measured near the pump discharge wye fitting and was 18.6 μ -inches/inch. The amplitudes of vibration steadily diminished as distance from the pumps increased until at the risers strain was measured at 4.4 μ -inches/inch. The strains were converted to stress by means of the computer program STRESS⁽⁶⁾ and ranged from 130 psi in the risers to 550 psi near the pumps. The vibrational forces act in both directions (circumferentially and axially) and were so combined in the summation stresses. In the crossheader connection being discussed here, 150 psi was added.

III. THERMAL STRESS

The thermal stress in the crossheader-to-riser assembly was analyzed with the aid of the MEC 21 Piping Flexibility Analysis Program⁽⁷⁾ developed

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III.

through the cooperative efforts of the Mare Island Naval Shipyard Design Division and the Los Alamos Scientific Laboratory. The program was adapted for use in the Univac 1108 computer for this study. The program considered thermal expansion and computed the resulting moment at the crossheader to riser juncture. The crossheader was assumed to be empty and weightless to obtain purely thermal effects. A maximum thermal range of 35 F has been considered.

The moment diagram shown below was obtained from the program and shows the resulting moment at the riser to be 715 lb-ft.

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III.

The resulting horizontal moment at the riser is assumed transposed to the annular weld at the base of the heavily reinforced nozzle (see sketch) and is calculated as follows:

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$$\begin{aligned}
 S &= \frac{M}{Z} \\
 &= \frac{715 \times 12}{93} \\
 &= 92 \text{ psi}
 \end{aligned}$$

M = moment = 715 lb-ft
 Z = Section modulus at weld throat
 (Tube-Turn Engineering Data
 Book No. 211)
 D = o.d. of weld at throat
 d = i.d. of weld at throat

$$\begin{aligned}
 &= 0.098 \frac{D^4 - d^4}{D} \\
 &= 0.098 \frac{14.174^4 - 12.76^4}{14.174} \\
 &= 93 \text{ in}^3
 \end{aligned}$$

This stress (S = 92 psi) represents the principal stress in the weld and is assumed to be transmitted to the riser through the weld. It can be resolved into its circumferential (tangential) component by the

III.

relationship $\sigma_{\phi} = \sigma_p \cos^2 \phi$, (12) in which σ_p is the principal stress, ϕ is the angle of resolution, and σ_{ϕ} is the desired component of stress.

$$\begin{aligned}\sigma_{\phi} &= \sigma_p \cos^2 \phi \\ &= 92 \times \cos^2 45^\circ \\ &= \underline{46} \text{ psi}\end{aligned}$$

This represents the circumferential stress in the riser resulting from maximum thermal effect.

Thermal stresses in the 24- and 36-inch pipe runs were also computed utilizing the MEC-21 program. The maximum stress was computed to be 1200 psi at the first 90 degree elbow downstream from a high-lift pump and 1400 psi in the 36-inch piping at the 15 degree mitered vertical elbows in the pipe tunnel.

IV. STRUCTURAL LOAD STRESS

Stress in the riser due to structural load is limited to that stress resulting from the weight of a short section of crossheader (full of water) which exerts a vertical moment at the crossheader-to-riser juncture. This moment is considered to result from the weight of the crossheader between the last support and the riser as indicated below:

IV.

The maximum moment (at the riser) is calculated from the relationship

$$M_{\max} = \frac{wL^2}{8} \quad (13) \quad \text{considering the section of crossheader as a uniformly}$$

loaded beam fixed at one end (at the riser) and supported at the other.

$$\begin{aligned} M &= \frac{wL^2}{8} \\ &= \frac{2.3 \times 108^2}{8} \\ &= \underline{3400} \text{ lb-in} \end{aligned}$$

w = Unit weight of 5-inch pipe full of water

$$= 28 \text{ lb/ft} = 2.3 \text{ lb/in.}$$

L = length of beam (pipe) = 108 inches

This moment, acting at the crossheader nozzle weld at the riser and in a plane parallel to the riser axis, may be resolved into the principal stress in the weld as follows:

$$\begin{aligned} S &= \frac{M}{Z} \\ &= \frac{3400}{70} \\ &= \underline{48} \text{ psi} \end{aligned}$$

M = Moment = 3400 lb-in.

Z = Section modulus of weld at throat

$$\begin{aligned} &= 0.098 \frac{D^4}{d} - d^4 \quad (\text{Tube-Turn Engineering Data Book No. 211}) \\ &= .098 \frac{13.82^4 - 12.76^4}{13.82} \\ &= \underline{70} \text{ in}^3 \end{aligned}$$

D = O.D. of weld at throat

d = I.D. of weld at throat

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IV.

This stress (48 psi) represents the principal stress in the weld and is assumed to be transmitted to the riser through the weld. It can be resolved into its axial (with respect to the riser) component by the relationship

$$\sigma_{\phi} = \sigma_p \cos^2 \phi$$

in which σ_p is the principal stress, ϕ is the angle of resolution, and σ_{ϕ} is the desired component of stress.

$$\begin{aligned}\sigma_{\phi} &= \sigma_p \cos^2 \phi \\ &= 48 \cos^2 45^\circ \\ &= \underline{24} \text{ psi}\end{aligned}$$

The stress in the riser due to structural loading of the crossheader is thus seen to be very small and has been neglected in the summation of stresses.

Another structural load stress is the compressive stress in the lower section of the riser due to the supported dead weight of the riser section. This stress has been computed to be a maximum of 203 psi. Since it is a compressive stress and acts to reduce the predominant axial stresses, it has not been considered in the stress summation as an added degree of conservatism.

An analysis similar to the above was used to determine torsional stress in the crossheader fitting due to structural dead load. Again the stress was found to be negligible.

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The crossheader was assumed to be hinged at the support for purposes of conservatism, and the resulting moment at the crossheader fitting to riser weld was determined as follows:

$$\begin{aligned}\sum M &= \frac{wl_1}{2} \times \frac{l_1}{2} + \frac{wl_2}{2} \times l_1; \\ &= \frac{2.3 \times 43^2}{4} + \frac{2.3 \times 48 \times 43}{2} \\ &= 3400 \text{ lb-in}\end{aligned}$$

$$\begin{aligned}w &= 2.3 \text{ lb/in.} \\ l_1 &= 43 \text{ in.} \\ l_2 &= 48 \text{ in.}\end{aligned}$$

The torsional stress in a vertical plane through the circumferential nozzle weld was calculated as follows:

$$S_{\text{torsional}} = \frac{16 T d_o}{\pi (d_o^4 - d_i^4)}$$

d_i = inside diameter of nozzle
 d_o = outside diameter of nozzle @ weld
 T = Torsional Moment

$$\begin{aligned}S_t &= \frac{16 \times 3400 \times 9}{\pi (9^4 - 5^4)} \\ &= 26 \text{ psi}\end{aligned}$$

Ref: Kent's Mechanical Engineers Handbook, Eleventh Edition, Design and Shop Practice, p.7-42

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IV.

Structural load stresses in other portions of the piping system are not negligible, however. For example, in the 24-inch pipe runs, with a typical 25-foot span, the stress at the supports due to bending was calculated as follows:

$$S = \frac{M}{Z}$$

$$= \frac{33.6 \times 300^2}{12 \times 388}$$

$$= \underline{650} \text{ psi}$$

M = maximum moment in pipe

$$= \frac{wL^2}{12} \quad w = 33.6 \text{ lb/in.}, \text{ 24-inch pipe full of water}$$

$$L = 25 \times 12 = 300 \text{ inches}$$

$$Z = 388 \text{ in}^3$$

If an intensification factor of 1.2 is applied to account for concentration of stress at the supports, the resulting stress is 800 psi, which was used in the summation of stresses in Table IV.

V. SEISMIC STRESS

In the analysis of stress due to horizontal seismic force, an approach similar to that used in the thermal stress analysis was used, except that the weight of the pipe filled with water was included. It was assumed that earthquake resulting in a ground acceleration of 0.25 g would result in relative movement between the reactor shield and the risers. The crossheaders are anchored at the center of the front face shield and the risers are anchored to the building structure. The anchor on the shield is incapable of resisting an axial force on the crossheader of greater than 500 pounds. Thus, 500 pounds is the maximum force that can be transmitted through the crossheader to the riser nozzle to which the crossheader is attached. For purposes of calculation (and conservatism) it was further assumed that the crossheader was broken, and the entire 500 pounds is assumed to be transmitted axially to the crossheader.

The axial force of 500 pounds was inserted into the MEC-21 program and the moment diagram developed as shown in the sketch:

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V.

The resulting moment at the riser is seen to be 915 lb-ft. This moment, acting in a horizontal plane, will be transmitted to the effective weld area where it is resisted by the couple as shown below:

The moment, 915 lb-ft, may be resolved into the principal stress in the weld as follows:

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V.

$$\begin{aligned} S &= \frac{M}{Z}, \\ &= \frac{915 \times 12}{93} \\ &= \underline{118} \text{ psi} \end{aligned}$$

$$\begin{aligned} M &= \text{moment} = 915 \text{ lb-ft.} \\ Z &= \text{Section Modulus at weld throat} \\ &= 0.098 \times \frac{D^4 - d^4}{D}, \quad D = \text{o.d. of weld} \\ &= .098 \frac{14.17^4 - 12.76^4}{14.17} \quad d = \text{i.d. of weld} \\ &= \underline{93} \text{ in}^3 \end{aligned}$$

Resolving S into its circumferential component:

$$\sigma_{\phi} = \sigma_p \cos^2 \phi,$$

$$\begin{aligned} \sigma_{\phi} &= 118 \times \cos^2 45^\circ \\ &= \underline{60} \text{ psi} \end{aligned}$$

This represents the circumferential stress in the riser due to horizontal seismic force.

Stress in the riser due to vertical seismic forces was calculated on the basis that the crossheader to riser approach section can be considered as a uniformly loaded beam fixed at one end (at the riser) and supported at the other (at the nearest support) as was done in the calculation of structural load stress.

Stress in the riser due to moment resulting from vertical load is:

$$\begin{aligned} S &= \frac{M}{Z}, \\ &= \frac{2040}{70} \\ &= \underline{29.2} \text{ psi} \end{aligned}$$

$$\begin{aligned} M &= \text{Moment at riser} \\ &= \frac{wL^2}{8} \\ &= \frac{28 \times 108^2}{12 \times 8} \\ &= 3400 \text{ lb-in} \end{aligned}$$

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Assuming a .6g vertical acceleration
(.6xwL) the additive moment due to
earthquake = .6 x 3400
= 2040 lb-in

Z = Section modulus of weld at throat

$$\begin{aligned} &= .098 \frac{D^4 - d^4}{D} \\ &= .098 \frac{13.82^4 - 12.76^4}{13.82} \\ &= \underline{70} \text{ in}^3 \end{aligned}$$

The principal stress in the riser, 29.2 psi resolved into its axial (with respect to the riser) component as follows:

$$\begin{aligned} \sigma_o &= \sigma_p \cos^2 \phi \\ &= 29.2 \cos^2 45^\circ = \underline{14.6} \text{ psi} \end{aligned}$$

This stress is considered negligible in the final summation.

The effect of seismic loading on the stresses in the riser barrels has been determined to be negligible. Each riser is anchored to the building structure (to the floor at the base elbow and to the building wall at four equispaced locations in the height of the riser. Static loading values of the riser based on a 0.6g response acceleration factor are not significant.

The stresses determined above are then combined in the following manner:

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SUMMATION OF STRESSES
RISER AT CROSSHEADER NOZZLE

	<u>CIRCUMFER.</u>	<u>AXIAL</u>
NOMINAL CIRCUMFERENTIAL STRESS	6700 PSI	
NOMINAL AXIAL STRESS		3000 PSI
VIBRATIONAL STRESS (MEASURED)	150 PSI	150 PSI
STRUCTURAL LOAD STRESS	---	NEGL.
THERMAL STRESS	---	NEGL.
CONCENTRATION FACTOR	*1.3/2.0**	1.0
FACTORED NORMAL STRESS		
(6700 X 1.3) + 150 (MEASURED)	<u>8850 PSI</u>	
(3000 X 1.0) + 150 (MEASURED)		<u>3150 PSI</u>
RESIDUAL STRESS	4000 PSI	4000 PSI
SEISMIC STRESS (VERTICAL & HORIZONTAL)	NEGL.	NEGL.
TOTAL COMBINED STRESS (SEISMIC)	<u>12,850 PSI</u>	<u>7150 PSI</u>
PRESSURE TRANSIENT (590 PSI)***	8550 PSI	3810 PSI
FACTORED PRESSURE TRANSIENT		
(8550 X 1.3) + 150 (MEASURED)	11,365 PSI	
(3810 X 1.0) + 150 (MEASURED)		3960 PSI
RESIDUAL STRESS	4000 PSI	4000 PSI
TOTAL COMBINED STRESS (PRESSURE TRANSIENT)	<u>15,365 PSI</u>	<u>7960 PSI</u>

* FACTOR DERIVED FROM PHOTOELASTIC MODEL AND APPLIED TO STRESSES FROM INTERNAL PRESSURE.

** FACTOR FROM USA'S B31.1.0 POWER PIPING CODE (1967) AND APPLIED TO STRESSES DUE TO MOMENTS.

*** TOTAL INTERNAL PRESSURE IN PIPE DURING PRESSURE TRANSIENT (THIS IS NOT ADDITIVE TO NOMINAL PRESSURE).

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FIGURE 19
NATURAL FREQUENCY
K PIPING SYSTEMS
(24-INCH)

Natural frequency of typical span of 24" pipe in the 165-190 plant by Euler's method for clamped continuous beams.

$$\omega_n = (\beta I)^2 \sqrt{\frac{EI}{m l^4}}$$

$$n = 1 \text{ st mode} = 1$$

$$I_{24} = 4657 \text{ in}^4$$

$$\beta I = 4.73$$

$$W = 33.8 \text{ lb/in}$$

$$(\beta I)^2 = 22.373$$

$$g = 386.4 \text{ in/sec}^2$$

$$E = 30 \times 10^6 \text{ lbs/in}^2$$

$$m = 33.8/386.4 = 0.0875$$

$$l = 300 \text{ in}$$

$$\omega_1 = 22.373 \sqrt{\frac{30 \times 10^6 \times 4657}{0.0875 \times (300)^4}}$$

$$= 22.373 \sqrt{194}$$

$$= 22.373 \times 14.1$$

$$= 314 \text{ Radians/sec}$$

$$= 314/2\pi = 51 \text{ cycles/sec}$$

$$\text{Period} = 1/51 = 0.0196 \text{ sec}$$

$$\sim 0.02 \text{ sec.}$$

FIGURE 20
NATURAL FREQUENCY
K PIPING SYSTEMS
(24-INCH)

$$\omega_1 = \frac{\pi^2}{l} \sqrt{\frac{gEI}{Wl^2}}$$

$$l = 300 \text{ in}$$

$$g = 386.4 \text{ in/sec}^2$$

$$E = 30 \times 10^6 \text{ lb/in}^2$$

$$I = 204.1 \text{ in}^4$$

$$W = 20,000 \text{ lbs} = (wl)$$

$$\omega_1 = \frac{\pi^2}{l} \sqrt{\frac{gEI}{20,000 \text{ } l}}$$

$$= \frac{9.9}{300} \sqrt{\frac{386.4 \times 30 \times 10^6 \times 204}{20,000 \times 300}}$$

$$= 21 \text{ Rad/sec}$$

$$N = 3.36 \text{ cycles/sec}$$

$$\text{Period} = 0.298 \text{ sec}$$

$$\sim \underline{0.3} \text{ sec}$$