CENC - 1110

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ANALYTICAL REPORT
FOR
INDIAN POINT REACTOR VESSEL
UNIT NO. 2

COMBUSTION ENGINEERING, INC.
CHATTANOOGA, TENN.

Cockrell
Lowry

REPORT NUMBER 1110
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"ANALYTICAL REPORT"

COMBUSTION ENGINEERING, INC.
NUCLEAR COMPONENTS ENGINEERING DEPARTMENT
C.E. CONTRACT NO. 17765

ANALYTICAL REPORT

FOR

INDIAN POINT REACTOR VESSEL UNIT NO. 2

C. R. COCKRELL AND J. C. LOWRY

ACKNOWLEDGEMENT

Acknowledgement is hereby made of the important contributions rendered by the following individuals:

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1.000 ABSTRACT

The structural integrity of the 173 in. I.D. Indian Point Reactor Vessel - Unit No. 2 designed and fabricated under contract to the Atomic Power Division of the Westinghouse Electric Corporation is established by the results of the detailed structural and thermal analysis contained in this report.

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2.000 INTRODUCTION

2.010 Subject

The Indian Point Reactor Vessel - Unit No. 2 is a 173 in. I.D. pressurized water reactor. The vessel is of cylindrical shape terminating in a hemispherical head at the bottom and a bolted flange at the top. Four inlet and four outlet nozzles are located in the cylindrical wall section. The vessel is supported by four weld built-up pads located on the underside of two inlet nozzles and two outlet nozzles. The closure head is of the hemispherical type. The closure seal is of the O-ring type.

2.020 Purpose

This report contains the detailed structural and thermal analysis required to substantiate the adequacy of the design of the 173 in. I.D. Indian Point Reactor Vessel.

2.030 Scope

The detailed analytical work necessary to justify the reactor vessel and its associated parts included in the contract are contained in this report. All equations used are shown and intermediate answers and final answers are usually presented in tabular form.

3.000 DESIGN CRITERIA

The design shall be in accordance with the ASME Boiler and Pressure Vessel Code Section III, Nuclear Vessels and Special Case Rulings in effect on the date of purchase order.

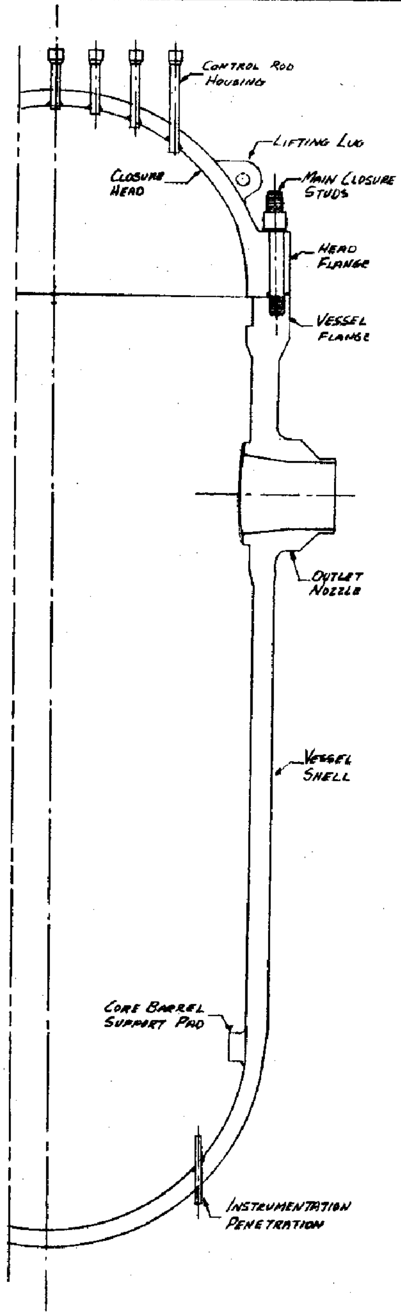
The design parameters used were

Design Pressure............................ 2500 psia
Normal Operating Pressure .... 2235 psig
Design Temperature .... 650°F
Normal Operating Inlet Water Temp........... 557°F
Normal Operating Outlet Water Temp..........611.7°F
Design Life .... 40 years

	Transient Condition	Occurrences	Ref. Fig. in Ref. 9
1.	Plant heatup at 100°F/Hr.	200	4.4.1
2.	Plant cooldown at 100°F/Hr.	200	4.4.1
3.	Plant loading at 5% of full power per min.	14,500	4.4.2
4.	Plant unloading at 5% of full power per min.	14,500	4.4.2
5.	Step load increase 10% of full power - not to exceed full power	2,000	4.4.3
6.	Step load decrease of 10% from 50% power	2,000	4.4.3
7.	Step load decrease of 50% of full power	200	4.4.4
8.	Reactor trip	400	4.4.5
9.	Hydro test 3125 psia	5	4.4.6
10.	Hydro test 2500 psia	5	4.4.7
11.	Steady state fluctuations	10^6	None
12.	Loss of flow, one pump	80	4.4.8
13.	Loss of load	80	4.4.9
14.	Steam break	5	See PAR 27

Material Allowables	S_m @ 70°F	S_m @ 550°F
SA-240 Typ. 316	20.0 KSI	17.6 KSI
SA-302B	26.7	26.7
SA-336	26.7	26.7
ASTM-A540-B24	43.3	36.8
Inconel	23.3	23.3

4.000 GEOMETRY AND GENERAL CONFIGURATION



This sketch shows the general configuration and relative locations of component parts. The nomenclature shown is used consistently throughout this report. For detailed drawings of component parts, see Appendix C.

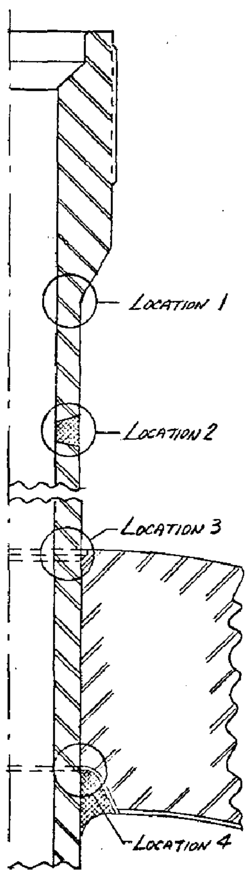
Reference Drawings

Title	Drawing No.
1. Control Rod Mechanism Housing Details	E-232-051
2. Control Rod Penetration Details	E-232-052
3. Closure Head Forming & Welding	E-232-046
4. Closure Head Machining	E-232-047
5. Stud, Nut & Washer Detail	E-232-049
6. Pressure Vessel Forming & Welding	E-232-042
7. Pressure Vessel Final Machining	E-232-044
8. Nozzle Details	E-232-045
9. Miscellaneous Attachments	E-232-050
10. Miscellaneous Details	E-232-055
11. Bottom Head Forming & Welding	E-232-043
12. Instrumentation Penet. Assembly & Details - Bottom Head	E-232-056

5.000 SUMMARY OF RESULTS

Results of the detailed structural analysis presented in Appendix A are summarized on Pages 7 through 17 for locations of major interest.

5.010 CONTROL ROD HOUSINGS



Location - 1

Stresses Due to Operating Pressure of 2.25 KSI

SURFACE	STRESS			STRESS INTENSITY		
	σ_x	σ_θ	σ_r	$\sigma_x - \sigma_\theta$	$\sigma_x - \sigma_r$	$\sigma_\theta - \sigma_r$
INSIDE	0.27	4.20	-2.25	-3.93	2.52	6.45
OUTSIDE	3.76	5.25	0	-1.49	3.76	5.25

The maximum stress intensity for operating pressure is $\sigma_\theta - \sigma_r = 6.45$ KSI on the inside surface.

The overall usage factor for fatigue was U = 0.

Location - 2

Stresses Due to Operating Pressure of 2.5 KSI and Design Temperature of 650°F

SURFACE	STRESS			STRESS INTENSITY		
FOR SA-182 TYPE 304SS FLANGE						
	σ_x	σ_θ	σ_r	$\sigma_x - \sigma_\theta$	$\sigma_x - \sigma_r$	$\sigma_\theta - \sigma_r$
INSIDE	3.21	-10.47	-2.5	13.68	5.71	-7.97
OUTSIDE	1.27	-11.07	0	12.34	1.27	-11.07
FOR SB-167 INCONEL TUBE						
INSIDE	3.21	23.14	-2.5	-19.93	5.71	25.64
OUTSIDE	1.27	22.54	0	-21.27	1.27	22.54

The maximum stress intensity for design pressure of 2.5 KSI and the design temperature of 650°F is $\sigma_\theta - \sigma_r = 25.6$ KSI and was located on the inside surface of the inconel tube material.

The overall usage factor for fatigue was U = 0.

Location - 3

Stresses Due to Operating Pressure of 2.25 KSI and Maximum and Minimum Interference Fit with Closure Head

SURFACE	INTERFERENCE = 0.003 PRESSURE = 0						INTERFERENCE = 0.003 PRESSURE = 2.25 KSI					
	STRESS			STRESS INTENSITY			STRESS			STRESS INTENSITY		
	σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$	σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$
INSIDE	27.35	-19.35	0	46.70	27.35	-19.35	14.87	-5.03	-2.25	19.90	17.12	-2.78
OUTSIDE	-27.35	-35.75	0	8.40	-27.35	-35.75	-12.09	-13.49	0	1.40	-12.09	-13.49
	INTERFERENCE = 0 PRESSURE = 0						INTERFERENCE = 0 PRESSURE = 2.25 KSI					
INSIDE	0	0	0	0	0	0	-12.48	14.32	-2.25	-26.80	-10.23	16.57
OUTSIDE	0	0	0	0	0	0	15.26	22.26	0	-7.00	15.26	22.26

The maximum stress intensity is $\sigma_x - \sigma_\theta$ = 46.7 KSI and occurs for the condition of maximum interference and zero internal pressure. This stress intensity is on the inside surface.

The maximum range of stress is 55.3 KSI and occurs on the inside surface.

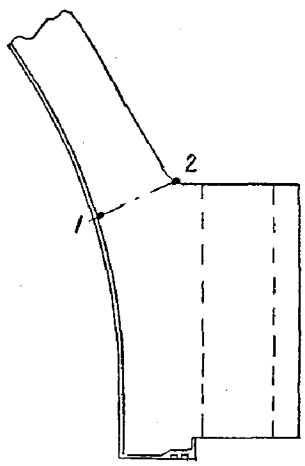
The overall usage factor for fatigue was U = 0.0003.

Location - 4

The maximum stress intensity for the J-weld is $\sigma_x - \sigma_\theta$ = 42.2 KSI and occurs on the inside surface. The maximum range of stress intensity is 42.8 at the same location.

The overall usage factor for fatigue was U = 0.06 and occurred on the outside surface.

5.020 CLOSURE HEAD FLANGE AND SHELL



Primary Stress Intensities

Bolt Up Condition

LOCATION	STRESS			STRESS INTENSITY		
	σ_x	σ_θ	σ_r	$\sigma_x - \sigma_\theta$	$\sigma_x - \sigma_r$	$\sigma_\theta - \sigma_r$
1	-21.7	-2.5	0	-19.2	-21.7	-2.5
2	23.4	13.9	0	9.5	23.4	13.9
	BOLT-UP PLUS: DESIGN PRESS. = 2.5 KSI					
1	-12.2	6.1	-2.5	-18.3	-9.6	8.6
2	35.9	24.0	0	11.8	35.9	24.0

Range of Stress Intensity

The highest range of stress intensity for the head flange to closure head juncture was located on the inside surface (Location - 1). The value of this range of stress intensity is 50.4 KSI and compares favorably with the allowable of 80 KSI.

Fatigue Evaluation

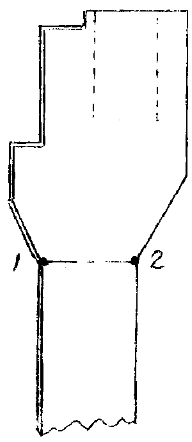
The following overall usage factors were calculated at the two above locations:

Location - 1 U = 0.004

Location - 2 U = 0.015

The maximum allowable usage factor is 1.0.

5.030 VESSEL FLANGE AND SHELL



Primary Stress Intensity

Bolt Up Condition

LOCATION	STRESS			STRESS INTENSITY		
	σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$
1	-20.8	-4.7	0	-16.1	-20.8	-4.7
2	20.8	7.8	0	13.0	20.8	7.8
	BOLT-UP PLUS DESIGN PRESS. = 2.5 KSI					
1	-10.2	14.1	-2.5	-24.3	-7.7	16.6
2	28.8	25.8	0	3.0	28.8	25.8

Range of Stress Intensity

The highest range of stress intensity for the vessel flange to vessel shell juncture was located on the inside surface (Location - 1). The value of this range of stress intensity is 45.4 KSI and compares favorably with the allowable of 80 KSI.

Fatigue Evaluation

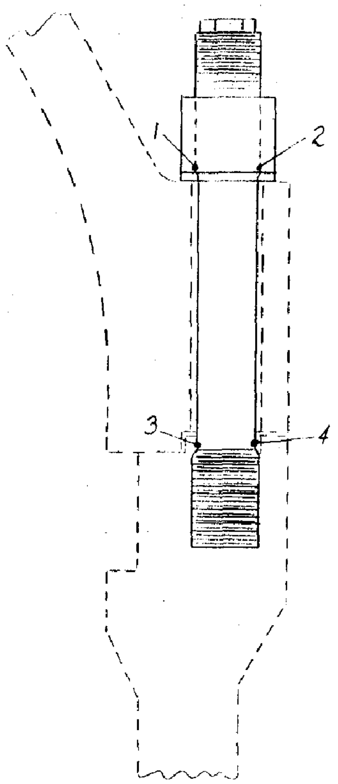
The following overall usage factors were calculated at the two above locations:

Location - 1 U = 0.005

Location - 2 U = 0.00002

The maximum allowable usage factor is 1.0.

5.040 MAIN CLOSURE STUDS



Average Bolt Stress:

The average bolt stress resulting from the design pressure flow off load plus O-ring seating load was 34.4 KSI. The allowable stress is 34.8 KSI.

Maximum Average Bolt Service Stress

The maximum average bolt service stress for the bolt-up condition was 36.8 KSI compared to the allowable of 86.6 KSI. For the bolt-up plus operating pressure condition, this stress was 39.6 KSI compared to the allowable of 73.5 KSI.

Maximum Bolt Service Stress

The maximum bolt service stress was 95.9 KSI and occurred at Location - 3. This stress occurred during the heat-up cycle of the 2500 psi hydrostatic test. This stress compares favorably with the allowable of 116.4 KSI.

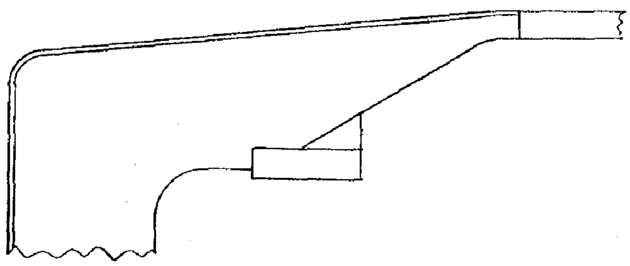
Fatigue Evaluation

The maximum overall usage factor for the closure studs was U = 0.313 and occurred at the point where it enters the vessel flange (Location - 3). This usage factor compares favorably with the allowable of 1.0.

Maximum Bearing Stress

The maximum bearing stress between the closure stud washer and closure head flange was 39.8 KSI and compares with the allowable of 40 KSI for the flange material. This value occurred during the heatup cycle of the 2500 psi hydrostatic test.

5.050 INLET NOZZLE AND VESSEL SUPPORT



Primary Membrane Stress Intensity

The maximum average primary membrane stress intensity for the inlet nozzle was at the juncture of the nozzles to the vessel wall on the longitudinal axis. The value of this stress intensity was 21.1 KSI and compares favorably with the allowable of 26.7 KSI.

The same location gave the highest value of average primary plus local primary stress. The value of this stress intensity was 32.3 KSI and compares favorably with the allowable of 1.5 S_m = 40 KSI.

Range of Stress Intensity

The highest range of stress intensity occurred at the juncture of the nozzle to vessel wall on the outside surface in the longitudinal direction. The value of this range of stress intensity was 45.5 KSI and compares favorably with the allowable of 80 KSI.

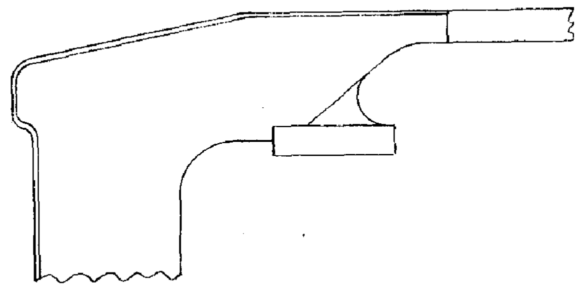
Bearing Stress on Support Pad

The bearing stress on the underside of the support pad for dead weight and thermal pipe reactions only was 3.0 KSI. This stress was not to exceed 5.0 KSI for this condition.

Fatigue Evaluation

The maximum overall usage factor for the inlet nozzle was U = 0.042 and occurred at the nozzle-vessel wall juncture on the outside surface in the circumferential direction. This value compares favorably with the allowable of 1.0.

5.060 OUTLET NOZZLE AND VESSEL SUPPORT



Primary Membrane Stress Intensity

The maximum average primary membrane stress intensity for the outlet nozzle was at the juncture of the nozzle to the vessel wall on the longitudinal axis. The value of this stress intensity was 21.1 KSI and compares favorably with the allowable of 26.7 KSI.

The same location gave the highest value of average primary plus local primary stress. The value of this stress intensity was 32.3 KSI and compares favorably with the allowable of 1.5 S_m = 40 KSI.

Range of Stress Intensity

The highest range of stress intensity occurred at the juncture of the nozzle to vessel wall on the outside surface in the longitudinal direction. The value of this range of stress intensity was 45.5 KSI and compares favorably with the allowable of 80 KSI.

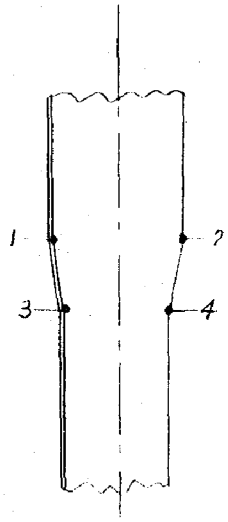
Bearing Stress on Support Pad

The bearing stress on the underside of the support pad for dead weight and thermal pipe reactions only was 3.6 KSI. This stress was not to exceed 5.0 KSI for this condition.

Fatigue Evaluation

The maximum overall usage factor for the outlet nozzle was U = 0.022 and occurred at the nozzle-vessel juncture on the inside surface in the longitudinal direction. This value compares favorably with the allowable of U = 1.0.

5.070 VESSEL WALL TRANSITION



Primary Membrane Stress Intensity

The maximum average primary membrane stress intensity for the vessel wall transition occurs in the thin portion of the vessel wall. The value of this stress intensity was 26.3 KSI and compares favorably with the allowable stress intensity of 26.7 KSI.

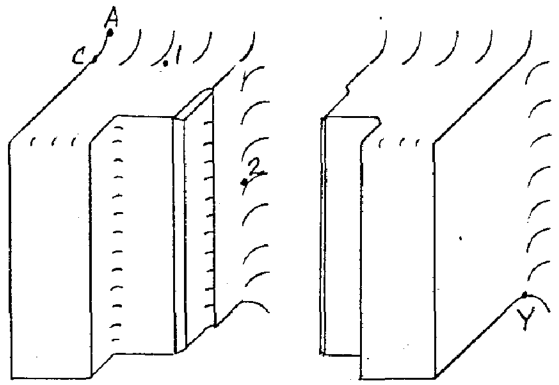
Range of Stress Intensity

The highest range of stress intensity occurred at Location - 1 as shown above. The value of this range of stress intensity was 37.9 KSI and compares favorably with the allowable of 80 KSI.

Fatigue Evaluation

The maximum overall usage factor for the vessel wall transition was U = 0.002 and occurredat Location - 2 as shown above. This value compares favorably with the allowable of U = 1.0.

5.080 CORE BARREL SUPPORT PADS



Stresses Due to Insertion of Core

The maximum stress intensity during insertion of the core occurred at Location - 1. The value of this stress intensity was 10.7 KSI and compares favorably with the allowable of 35 KSI. The maximum shear stress occurs at Location -/2 and is 10.4 KSI which compares favorably with the allowable of 18.6 KSI.

Stresses Due to Steady Loads

The most critical stress intensity for the steady 125 KIP side load and steady 125 KIP vertical load (due to thermal growth) occurred at the vessel wall (Location - Y as shown above). The value of this stress intensity was 31.1 KSI and compares favorably with the allowable of 35 KSI.

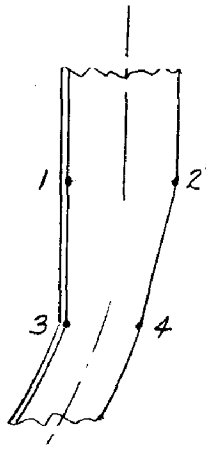
Range of Stress Intensity

The highest range of stress intensity occured at Location - C as shown above. The value of this range of stress intensity was 40.8 KSI and compares favorably with the allowable of 69.9 KSI.

Fatigue Evaluation

The fatigue evaluation disclosed that the highest overall usage factor for the pads was 0.02 and occurred at the upper corner of the pad at the pad-to-vessel juncture Location - A. This value compares favorably with the allowable of U = 1.0.

5.090 BOTTOM HEAD TO SHELL JUNCTURE



Primary Membrane Stress Intensity

The maximum average primary membrane stress intensity for the bottom head to shell juncture occurs in the cylindrical shell portion of the juncture. The value of this stress intensity was 26.3 KSI and compares favorably with the allowable of 26.7 KSI.

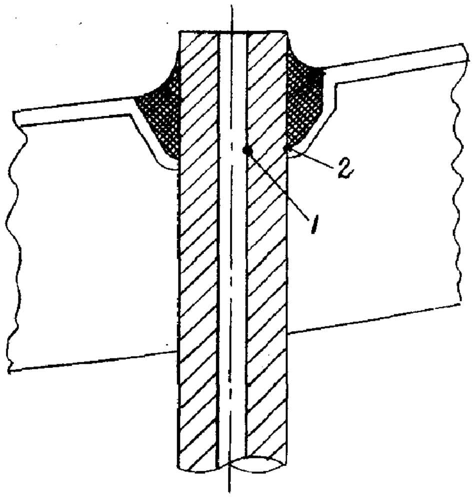
Range of Stress Intensity

The highest range of stress intensity occurred at Location - 3 as shown above. The value of this range of stress intensity was 34.1 KSI and compares favorably to the allowable range of stress intensity of 80 KSI.

Fatigue Evaluation

The fatigue evaluation disclosed that the highest overall usage factor for the bottom head to shell juncture was 0.003 and occurred at Location - 3 as shown above. This value compares favorably with the allowable of U = 1.0.

5.100 BOTTOM HEAD INSTRUMENTATION PENETRATIONS



Primary Membrane Stress Intensity

The maximum average primary membrane stress intensity occurs in the bottom head when taking into consideration the ligament efficiency. The value of the stress intensity is 26.5 KSI and compares favorably with the allowable of 26.7 KSI.

Range of Stress Intensity

The highest range of stress intensity occurred on the inside surface of the tube - Location - 1 as shown above. The value of this range of stress intensity is 53.9 KSI and compares favorably with the allowable of 69.9 KSI.

Fatigue Evaluation

The fatigue evaluation disclosed that the highest overall usage factor for the bottom head instrumentation was 0.14 and occurred on the outside surface of the tube - Location - 2 as shown above. The value compares favorably with the allowable of U = 1.0.

6.000 DISCUSSION OF RESULTS & METHOD OF ANALYSIS

6.010 Control Rod Housings

A. Discussion of Results

Location - 1

For the juncture of the CRDM flange to tube, the maximum stress intensity for design pressure is 7.2 KSI and occurs on the inside surface. The allowable for the 304 stainless steel at the design temperature is 1.5 S_m = 23 KSI.

Location - 2

For the 304 stainless steel at the bi-metallic weld, the maximum primary plus secondary stress intensity was 13.7 KSI on the inside surface. This stress intensity occurred for the design temperature and pressure. For the operating conditions, the maximum range of stress intensity was 11.1 KSI and compares favorably with the allowable of 3 S_m = 45.9 KSI.

For the inconel portion of the tube, the maximum stress intensity for the design conditions occurred on the inside surface. The value of this stress intensity was 25.6 KSI. For the operating transients, the maximum range of stress intensity was 21.8 KSI and compares favorably with the allowable of 3 S_m = 69.9 KSI.

Location - 3

At the point where the CRDM housing enters the closure head, stresses are induced in the tube at zero pressure due to the interference fit. The stress intensity at the maximum interference and zero pressure is 46.7 KSI. For the operating transients, the maximum range of stress intensity was 55.3 KSI and compares favorably with the allowable of 3 S_m = 69.9 KSI.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.0003 for the inside surface. This compares favorably with the allowable of 1.0.

Location - 4

At the location where the CRDM housing is attached to the closure head by the J-weld, the maximum range of stress intensity is 42.8 KSI and compares favorably with the allowable of 3 S_m = 69.9 KSI. This range of stress intensity occurs on the inside surface.

From the standpoint of fatigue, the most critical location will be on the outside portion of the tube where a stress concentration factor of four was used. The cumulative usage factor at this location was 0.06 and compares favorably with the allowable of 1.0.

B. Method of Analysis

Location - 1

An interaction analysis was performed at cut one assuming the CRDM housing flange to be a ring and the tube a long cylinder.

Location - 2

An interaction analysis was performed at cut two taking into consideration that elements 2 and 3 are long cylinders having different values of Young's Modulus of Elasticity and coefficients of thermal expansion.

Location - 3

An interaction analysis was performed at cut three by taking the housing as a cylinder and setting its deflection equal to the deflection of the radius of the head penetration and conservatively assuming its rotation equal to the local flexibility as if it were solidly attached. It was assumed that the forces exerted on the head by the tube have negligible effect on the head.

Location - 4

An interaction analysis was performed by dividing the actual structure into the following analytical model: the closure head was treated as a perforated spherical shell with modified elastic constants and the CRDM housing as a long cylinder. The effects of the redundants on the closure head were assumed to be local only. It was assumed that

for any condition where there is interference between the tube and head, no bending at the weld can exist. Using mechanical and thermal stresses from this analysis, a fatigue evaluation was made for the J-weld.

6.020 CLOSURE HEAD FLANGE AND SHELL

A. Discussion of Results

The maximum primary stress intensity at the closure head flange to shell juncture was 35.8 KSI for the bolt-up plus design pressure condition. This stress occurs on the outside surface of the juncture of the head to flange and compares favorably with the allowable of 1.5 S_m = 40 KSI.

The highest range of stress intensity for this juncture was 50.4 KSI on the inside surface and compares favorably with the allowable of 3 S_m = 80 KSI.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.015 and occurred for the outside surface. This value compares favorably with the allowable of 1.0.

B. Method of Analysis

The closure head, closure head flange, vessel flange, vessel shell, and closure studs were all evaluated in the same analysis. The actual structure was divided into the following elements: the closure head dome was treated as a long sphere, the closure head flange was treated as a ring, the vessel flange and studs were combined as one element with the flange treated as a ring and the studs as cantilever beams fixed to the flange, and the vessel shell was treated as a long cylinder.

Using the above described analytical model, an interaction analysis was performed to determine the stresses due to the mechanical and thermal loadings for the heatup and cooldown cycle. For the remaining transients, the conservative skin stress method was used for determining thermal stresses. These stresses were evaluated in light of the strength and fatigue requirements of the ASME Boiler and Pressure Vessel Code, Section III.

6.030 VESSEL FLANGE AND SHELL

A. Discussion of Results

The maximum primary stress intensity at the vessel flange to vessel shell juncture was 28.8 KSI for the boltup plus design pressure condition. This stress occurred on the outside surface of the juncture flange to vessel shell and compares favorably to the allowable of 1.5 S_m = 40 KSI.

The highest range of stress intensity for this juncture was 45.4 KSI on the inside surface and compares favorably with the allowable of 3 S_m = 80 KSI.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.005 and occurred for the inside surface. This value compares favorably with the allowable of 1.0.

B. Method of Analysis

See Section 6.020-B, Method of Analysis, Closure Head Flange and Shell.

6.040 MAIN CLOSURE STUDS

A. Discussion of Results

The maximum average bolt service stress for the cold boltup condition was 36.8 KSI and compares favorably with the allowable of 2 S_m = 86.6 KSI. For the boltup plus operating pressure, the average bolt service stress is 39.6 KSI and compares favorably with the allowable of 2 S_m = 73.5 KSI at temperature.

The maximum bolt service stress was 95.9 KSI and occurred on the inside surface of the stud where it enters the vessel flange. This stress occurred during the heatup cycle of the 2500 PSI hydrostatic test and compares favorably with the allowable of 3 S_m = 116.4 KSI.

A fatigue evaluation was performed on the studs using the method outlined in Para. N-416.2 of the ASME Boiler and Pressure Vessel Code, Section III. The maximum cumulative usage factor for the studs was 0.313 on the inside surface of the stud where it enters the vessel flange. The allowable usage factor is 1.0.

The maximum bearing stress between the closure stud washers and closure head flange was 39.8 KSI. This stress occurred during the heatup cycle of the 2500 PSI hydrostatic test and compares favorably with the allowable of 1.5 S_m for the flange material.

B. Method of Analysis

See Section 6.020-B, Method of Analysis, Closure Head Flange and Shell.

6.050 INLET NOZZLE AND VESSEL SUPPORTS

A. Discussion of Results

The maximum average primary membrane stress intensity for the inlet nozzle occurred at the juncture of the nozzle to the vessel wall on the longitudinal axis. The value of this stress intensity was 21.1 KSI and compares favorably with the allowable of 26.7 KSI.

The same location gave the highest value of average primary plus local primary stress. The value of this stress intensity was 32.3 KSI and compares favorably with the allowable of 1.5 S_m = 40 KSI.

The highest range of stress intensity for the operating transients occurred at the juncture of the nozzle to vessel wall on the outside surface in the longitudinal direction. The value of this range of stress intensity was 45.5 KSI and compares favorably with the allowable of 80 KSI.

The bearing stress on the underside of the support pad for dead weight and the thermal pipe reactions only was 3.0 KSI. This stress was to be limited to 5.0 KSI under this condition.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.042 and occurred at the nozzle to vessel wall juncture on the outside surface in the circumferential direction. This value compares favorably with the allowable of 1.0. The cumulative usage factor through the nozzle wall and weld built-up support pad was found to be 0.007 on the outside surface of the pad.

B. Method of Analysis

For the analysis of the nozzle and nozzle to shell juncture, the loads considered were internal pressure, operating transients, thermally induced and seismic pipe reactions, static

weight of vessel, earthquake loading, and expansion and contraction.

The stresses resulting from all external loads were determined in the nozzle by the use of the standard formula for direct stress plus bending stress in a beam. At the juncture of the nozzle to vessel wall, these stresses were determined by the methods presented in references 19, 20, and 21.

The pressure stresses were determined in the nozzle by performing an interaction analysis. The actual structure was divided into the following elements: the thin portion of the nozzle was treated as a cylinder, the tapered portion was treated as a tapered cylinder, the reinforcement portion was treated as a cylinder, and the vessel was treated by idealizing it as a spherical segment of the same thickness as the vessel and with a mid-radius 1.5 times the actual radius of the vessel.

The thermal stresses for the operating transients were determined by performing an interaction with the above analytical model.

For the fatigue evaluation, pressure stresses were determined by the stress index method set forth in Article I-6 of the ASME Boiler and Pressure Vessel Code, Section III. Peak stresses resulting from the external loads and the thermal transients were determined by concentrating the stresses as determined by the above described methods. Combining these stresses enabled the fatigue evaluation to be performed.

6.060 OUTLET NOZZLE AND VESSEL SUPPORT

A. Discussion of Results

The maximum average primary membrane stress intensity for the outlet nozzle occurred at the juncture of the nozzle to the vessel wall on the longitudinal axis. The value of this stress intensity was 21.1 KSI and compares favorably with the allowable of 26.7 KSI.

The same location gave the highest value of average primary plus local primary stress. The value of this stress intensity was 32.3 KSI and compares favorably with the allowable of 1.5 S_m = 40 KSI.

The highest range of stress intensity for the operating transients occurred at the juncture of the nozzle to vessel wall on the outside surface in the longitudinal direction. The value of this range of stress intensity was 54.1 KSI and compares favorably with the allowable of 80 KSI.

The bearing stress on the underside of the support pad for the dead weight and the thermal pipe reactions only was 3.6 KSI. This stress was to be limited to 5.0 KSI under this condition.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.022 and occurred at the nozzle to vessel wall juncture on the inside surface in the longitudinal direction. This value compares favorably with the allowable of 1.0. The cumulative usage factor through the nozzle wall and the weld built-up support pad was found to be 0.011 on the outside surface of the pad.

B. Method of Analysis

See Section 6.050, Method of Analysis, Inlet Nozzle and Vessel Supports for the method of analysis.

6.070 VESSEL WALL TRANSITION

A. Discussion of Results

The maximum average primary stress intensity for the vessel wall transition occurs in the thin portion of the vessel wall. The value of this stress intensity is 26.3 KSI and compares favorably with the S_m value of 26.7 KSI.

The highest range of stress intensity for the operating transients occurred on inside surface at the large end of the taper. The value of this range of stress intensity was 37.9 KSI and compares favorably with the allowable of $3\ S_m = 80$ KSI.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.002 and occurred on the outside surface at the large end of the taper. This value is well below the allowable of 1.0.

B. Method of Analysis

Stresses due to internal pressure were determined by means of a standard interaction analysis. For the purpose of this analysis, the actual structure was divided into

the following elements: the thick portion of the vessel was treated as a cylinder, the tapered portion was treated as a short tapered cylinder, and the thin portion of the vessel wall was treated as a long cylinder.

The thermal stresses were determined by the skin stress method where it is assumed that the inside surface of the vessel is at the same temperature as the reactor coolant and the mean temperature of the shell remains at the steady state temperature. This method is considered conservative.

The fatigue evaluation was made on a cumulative basis where superposition of all transients is taken into consideration.

6.080 CORE BARREL SUPPORT PADS

A. Discussion of Results

The most critical stress intensity occurred for the steady 125 KIP side load and steady 125 KIP vertical load. This stress intensity of 31.1 KSI occurred at the lower outside corner of the juncture of the pad to the vessel wall and compares favorably with the allowable of 35 KSI.

The highest range of stress intensity occurred at the upper outside corner of the juncture of the pad to vessel wall. The value of this range of stress intensity was 40.8 KSI and compares favorably with the 3 S_m allowable of 69.9 KSI.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.02 and occurred at the upper outside corner of the juncture of the pad to vessel wall. This value is well below the allowable of 1.0.

B. Method of Analysis

Thermal, mechanical, and pressure stresses were calculated at various locations on the pad and at the vessel wall for the loads specified in reference *10*.

Mechanical stresses were calculated by the flexure formula for bending stress in a beam, pressure stresses were taken from the analysis of the vessel to bottom head juncture, and thermal stresses were determined by the conservative

method of skin stresses. These stresses were resolved into stress intensities and compared with the allowables set forth in reference-/. Stresses due to the cyclic loads were multiplied by a stress concentration factor where applicable and used in a fatigue evaluation.

6.090 BOTTOM HEAD TO SHELL JUNCTURE

A. Discussion of Results

The maximum average primary stress intensity for the bottom head to shell juncture occurs in the cylindrical shell portion of the juncture. The value of this stress intensity is 26.3 KSI and compares favorably with the S_m value of 26.7 KSI.

The highest range of stress intensity for the operating transients occurred on the inside surface at the start of the hemispherical shell. The value of this range of stress intensity was 34.1 KSI and compares favorably with the allowable of 3 S_m = 80 KSI.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.003 and occurred on the inside surface at the start of the hemispherical shell. This value is well below the allowable of 1.0.

B. Method of Analysis

Stresses due to internal pressure were determined by means of a standard interaction analysis. For the purpose of this analysis, the actual structure was divided into the following elements: the cylindrical shell was treated as a long cylinder, the tapered portion of the cylindrical shell was treated as a short tapered cylinder, and the hemispherical section was treated as a long spherical shell.

The thermal stresses were determined by the skin stress method where it is assumed that the inside surface of the vessel is at the same temperature as the reactor coolant and the mean temperature of the shell remains at the steady state temperature. This method is considered conservative.

The fatigue evaluation was made on a cumulative basis where superposition of all transients is taken into consideration.

6.100 BOTTOM HEAD INSTRUMENTATION PENETRATIONS

A. Discussion of Results

The maximum average primary membrane stress intensity for the bottom head is 26.5 KSI when taking into consideration of the ligament efficiency. This value compares favorably with the S_m value of 26.7 KSI.

At the location where the instrumentation penetration is attached to the bottom head by the J-weld, the maximum range of stress intensity is 53.9 KSI. This range of stress intensity occurs on the inside surface and compares favorably with the allowable of 69.9 KSI.

The fatigue evaluation revealed that the highest cumulative usage factor was 0.14 and occurred on the outside surface of the tube. This value compares favorably with the allowable of 1.0.

B. Method of Analysis

An interaction analysis was performed by dividing the actual structure into the following analytical model: the bottom head was treated as a perforated spherical shell with modified elastic constants and the instrumentation tube as a long cylinder. The effects of the redundants on the bottom head were assumed to be local only. It was assumed that for any condition where there is interference between the tube and head, no bending at the weld can exist. Using mechanical and thermal stresses from this analysis, a fatigue evaluation was made for the J-weld.

7.000 REFERENCES

1. ASME Boiler and Pressure Vessel Code, Section III for Nuclear Vessels
2. Section III Code Case 1332-3
3. Section III Code Case 1335-2
4. Section III Code Case 1336
5. Section III Code Case 1366
6. Tentative Structural Design Basis for Reactor Pressure Vessels and Directly Associated Components (Pressurized, Water Cooled Systems) PB151987, U.S. Department of Commerce.
7. Screw-Thread Standard for Federal Service, 1956
8. Code for Pressure Piping, ASA-B 31.1-1953
9. Westinghouse Equipment Spec. 676497 dated 3-23-67
10. Westinghouse PAR's
11. S. Timoshenko and J. N. "Theory of Elasticity", McGraw-Hill, 1934
12. S. Timoshenko and S. Woinowsky Krieger, "Theory of Plates and Shells", McGraw-Hill, 1940
13. R. B. Heywood, "Design by Photoelasticity", Chapman and Hill Ltd., 1952
14. R. E. Peterson, "Stress Concentration Factor", John Wiley and Sons, Inc., New York 1953
15. E. F. Bruhn, "Analysis and Design of Airplane Structures", 1949
16. P. G. Lawson, W. J. Cox, "Mechanics of Materials", John Wiley & Sons, Inc., New York, 1947
17. B. F. Langer, "Applications of Stress Concentration Factors", WAPD-BT-18

18. W. J. O'Donnell, "The Effects of Local Flexibility on Stresses in a Structure", WAPD(CE)-170

19. P. O. Bijlaard, "Stresses from Radial Loads in Cylindrical Pressure Vessels", The Welding Journal, Research Supplement, Dec., 1954

20. P. O. Bijlaard, "Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessels", The Welding Journal, Dec., 1954

21. Welding Research Council Bulletin No. 107, Local Stresses in Spherical and Cylindrical Shells due to External Loadings, K. R. Wichman, A. G. Hooper, and J. L. Mershon, August 1965.

22. Sampson, R. C., "Photoelastic Investigation of Stress Distribution in a Perforated Hemispherical Head of Reduced Thickness" Westinghouse Research Report 100FF996-R4.

APPENDIX A

DETAILED STRUCTURAL ANALYSIS

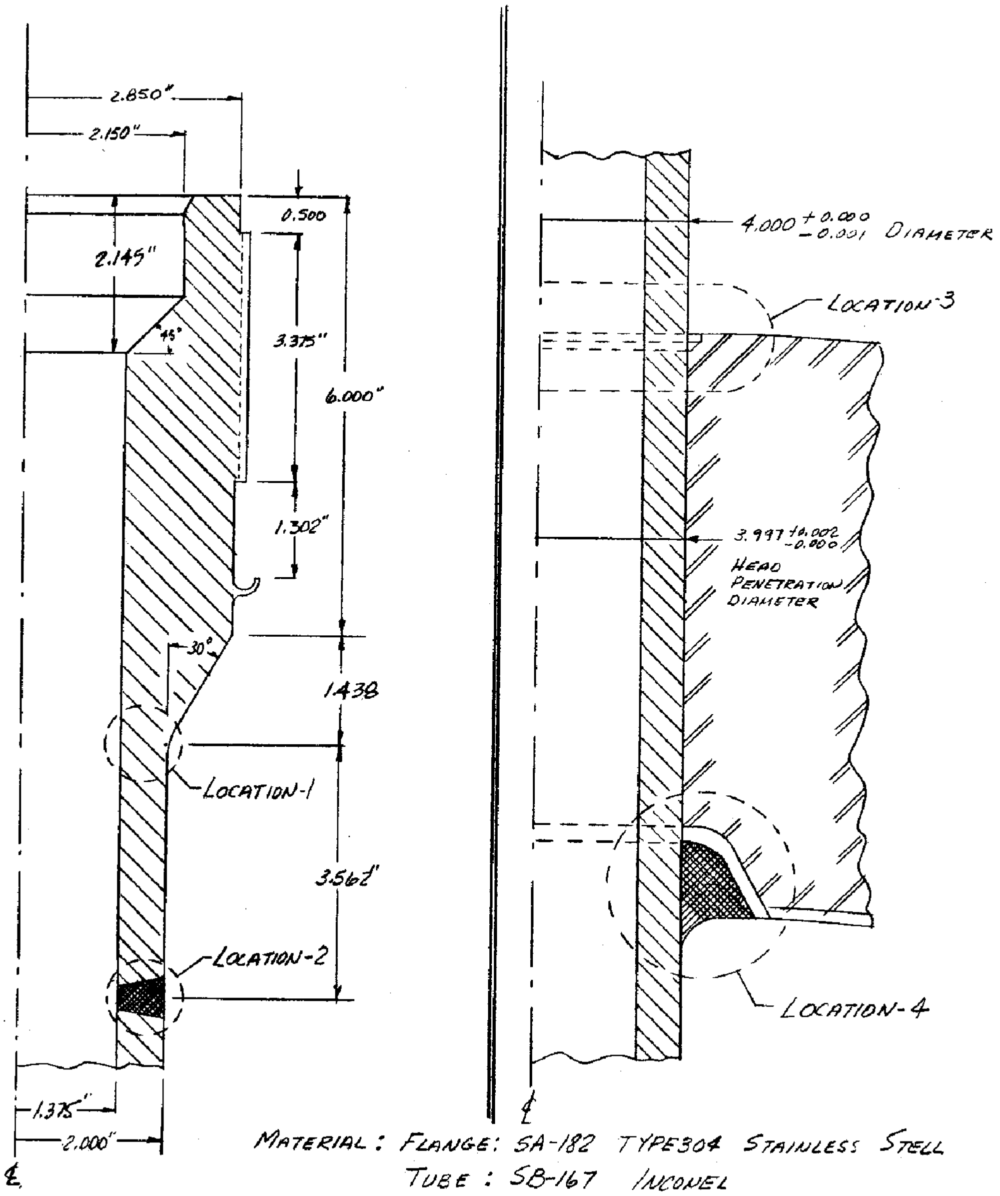
APPENDIX A

DETAILED STRUCTURAL ANALYSIS

TABLE OF CONTENTS

5- Detailed Analysis:

a. System Geometry:



Material: Flange: SA-182 Type 304 Stainless Stell
Tube: SB-167 Inconel

5- Detailed Analysis:

b- System Loads:

The Control Rod Housing Assembly Shown On Sheet 5 Will Be Investigated For The Following Loads In This Analysis:

1- Design Pressure Of 2.5 ksi At Design Temperature Of 650°F.
2- Operating Pressure Of 2.25 ksi At Operating Temperature.
3- The Thermal And Pressure Transients As Given In Ref. 9 For The Control Rod Housing To Closure Head Weld.

c- System Allowables:

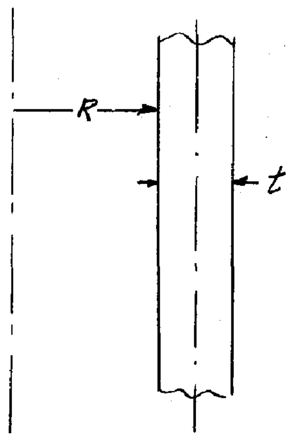
The Following Allowable Stresses Are Based On The A.S.M.E. Nuclear Code Section III, Reference 1 And Are Relevant For This Analysis.

1- The Average Primary Stress Intensity Across A Solid Section Shall Not Exceed S_m At Design Temp. (650°F) And Design Pressure 2.5 ksi.
2- The Local Primary Stress Plus The Average Primary Stress Shall Not Exceed 1.5 S_m At 650°F.
3- The Range Of Primary Plus Secondary Stress Intensity Resulting From Mechanical And Thermal Loads Shall Not Exceed 3 S_m At Actual Metal Temperature And Operating Pressure.
4- Show That Each Point Meets The Requirements For Peak Stress Intensity Given In N-414.5 Of The ASME Code Section III. The Procedure Will Be As Outlined In N-415.2 Of Section III.

5- DETAILED ANALYSIS:

d- DESIGN SIZING:

CONSIDER THE CONTROL ROD HOUSING WALL:



DESIGN PRESSURE = 2.5 KSI

DESIGN TEMPERATURE = 650°F

MATERIALS: INCONEL, S_m = 23.3 KSI (FROM REF. 4)

SA 182-F304, S_m = 15.3 KSI (FROM REF. 1)

$R = 1.375''$

$t_{ACT} = 0.625''$

$P = 2.5\,KSI$

FROM N-431 OF SECTION III, ASME NUCLEAR CODE:

$$t_{REQ'D} = \frac{PR}{S_m - 0.5P}$$

$$= \frac{2.5(1.375)}{23.3 - 0.5(2.5)} = \underline{0.156''} \text{ (FOR INCONEL TUBE)}$$

AND

$$= \frac{2.5(1.375)}{15.3 - 0.5(2.5)} = \underline{0.245''} \text{ (FOR SA-182-F304 STAINLESS STEEL)}$$

THE 0.625" ACTUAL THICKNESS IS ADEQUATE FOR BOTH MATERIALS; HENCE, CRITERION 5-C-1 IS SATISFIED.

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ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER S-100-P | A-5
SHEET 8 OF 27
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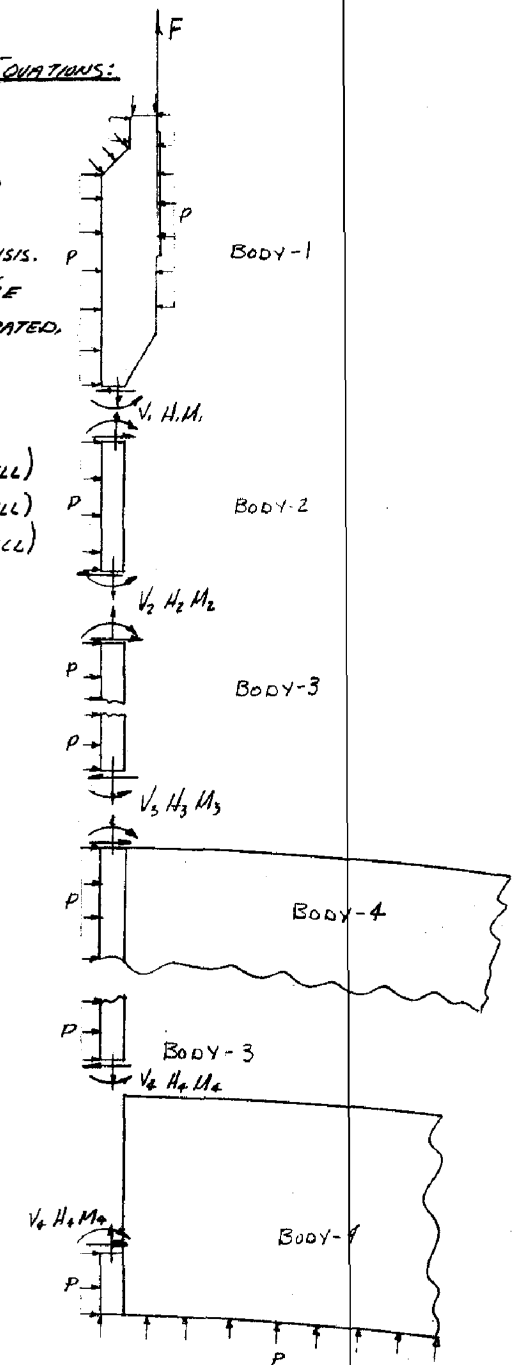
DESCRIPTION CONTROL ROD HOUSING STRESS ANALYSIS AND FATIGUE EVALUATION
CHECK DATE 1-24-67 BY CAUDLE

5- DETAILED ANALYSIS:

C. DEVELOPMENT OF CONTINUITY EQUATIONS:

1- ANALYTICAL MODEL:

THE ACTUAL STRUCTURE IS DIVIDED INTO THE FOLLOWING ANALYTICAL MODEL TO FACILITATE THE ANALYSIS. THE ASSUMED DIRECTIONS OF THE REDUNDANT FORCES ARE ILLUSTRATED.



BODY-1 (RING)
BODY-2 (LONG CYLINDERICAL SHELL)
BODY-3 (LONG CYLINDERICAL SHELL)
BODY-4 (LONG SPHERICAL SHELL)

$H_1 \rightarrow H_4$, $M_1 \rightarrow M_4$ ARE THE UNKNOWN INTERACTION FORCES

P IS THE APPLIED PRESSURE

F IS THE BLOW OFF LOAD IMPOSED ON THE CONTROL ROD DRIVE MECHANISM ATTACHMENT

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CHARGE NO.

DESCRIPTION Control Rod Housing Stress Analysis And Fatigue Evaluation

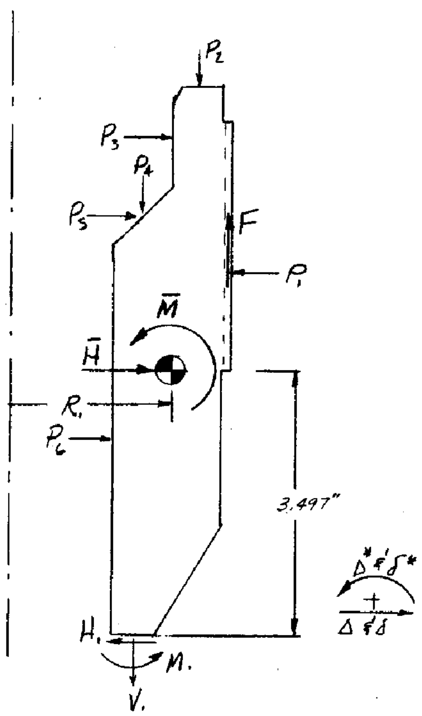
NUMBER S-100-P | A-6
SHEET 9 OF 27
DATE 1-24-67 BY Cockrell
CHECK DATE 1-24-67 BY Caudle

5- Detailed Analysis:

e Development Of Continuity Equations:

2. Movements Due To Redundant And Applied Loads:

Body-1



Force	Magnitude	Radius	Moment Arm
P_1	5.177P	2.880	1.353
F	1.414P	2.933	0.793
P_2	0.730P	2.515	0.375
P_3	1.370P	2.150	3.256
P_4	0.775P	1.763	0.377
P_5	0.775P	1.763	2.184
P_6	5.293P	1.375	0.851
V_1	0.560P	1.688	0.452
H_1	Unknown	1.688	3.497
M_1	Unknown	1.688	~

$A_1 = 9.011 in^2$ $I_1 = 32.069 in^4$ $R_1 = 2.140 in$

$\frac{R_1^2}{I_1} = 0.14058$ $\frac{R_1^2}{A_1} = 0.50822$

$$\bar{H} = -\frac{1.688}{2.140}H_1 - \frac{2.880}{2.140}(5.177P) + \frac{2.150}{2.140}(1.370P) + \frac{1.763}{2.140}(0.775P) + \frac{1.375}{2.140}(5.293P)$$

$$= \underline{-0.78878 H_1 - 1.55142 P}$$

$$\bar{M} = \frac{1.688}{2.140}\left[-3.497H_1 + M_1 + (0.452)(0.560P)\right] + \frac{2.880}{2.140}(1.353)(5.177P) + \frac{2.933}{2.140}(0.793)(1.414P)$$

$$- \frac{2.515}{2.140}(0.375)(0.730P) - \frac{2.150}{2.140}(3.256)(1.370P) + \frac{1.763}{2.140}\left[0.377(0.775P) - 2.184(0.775P)\right]$$

$$+ \frac{1.375}{2.140}(0.851)(5.293P) = \underline{-2.75838 H_1 + 0.78878 M_1 + 8.10020 P}$$

5. DETAILED ANALYSIS:

e. DEVELOPMENT OF CONTINUITY EQUATIONS

2. MOVEMENTS DUE TO REDUNDANT AND APPLIED LOADS:

BODY-1 :

DISPLACEMENTS DUE TO REDUNDANT FORCES:

$$E\Delta_{11} = \frac{R_1^2}{A_1}\bar{H} + h_j \frac{R_1^2}{I_1}\bar{M} = -1.7569 H_1 + 0.3878 M_1$$

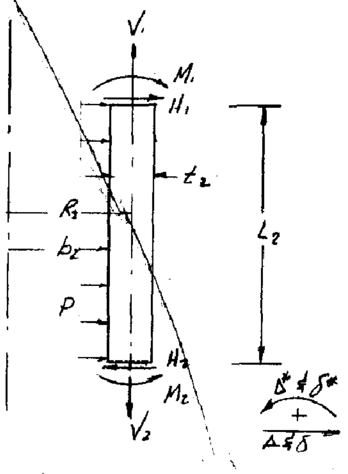
$$E\Delta_{11}^{*} = \frac{R^2}{I}\bar{M} = -0.3878 H_1 + 0.1109 M_1$$

DISPLACEMENTS DUE TO APPLIED FORCES:

$$E\delta_{11} = \frac{R_1^2}{A_1}\bar{H} + h_j \frac{R_1^2}{I_1}\bar{M} = 3.1937P$$

$$E\delta_{11}^{*} = \frac{R_1^2}{I_1}\bar{M} = 1.1387P$$

BODY-2:



$R_2 = 1.688''$ $\quad L_2 = 3.562''$

$b_2 = 1.375''$

$t_2 = 0.625''$

$$\beta^4 = \frac{3(1-\nu^2)}{R^2 t^2} = 2.45278 \qquad \beta^2 = 1.56613$$

$$\beta = 1.25145$$

$$D = \frac{E t^3}{12(1-\nu^2)} = 0.02236 E$$

$V_2 = 0.560P$

NOTE THAT FOR BODY-2 TO BE A LONG CYLINDER, βL MUST BE GREATER THAN 3. THE ACTUAL $\beta L = 4.46$; HENCE, BODY-2 IS A LONG CYLINDER.

5. DETAILED ANALYSIS:

e. DEVELOPMENT OF CONTINUITY EQUATIONS:

2. MOVEMENTS DUE TO REDUNDANT AND APPLIED FORCES:

BODY-2:

DISPLACEMENTS DUE TO REDUNDANT FORCES:

$$E\Delta_{21} = \frac{E}{2\beta^2 D}\left[\frac{1}{\beta}H_1 + M_1\right] = 11.4090 H_1 + 14.2776 M_1$$

$$E\Delta^*_{21} = \frac{-E}{2\beta^2 D}\left[H_1 + 2\beta M_1\right] = -14.2776 H_1 - 35.7398 M_1$$

$$E\Delta_{22} = \frac{-E}{2\beta^3 D}\left[\frac{1}{\beta}H_2 - M_2\right] = -11.4090 H_2 + 14.2776 M_2$$

$$E\Delta^*_{22} = \frac{-E}{2\beta^2 D}\left[H_2 - 2\beta M_2\right] = -14.2776 H_2 + 35.7398 M_2$$

DISPLACEMENTS DUE TO APPLIED LOADS:

$$E\delta_{21} = E\delta_{22} = \frac{b_2^2}{t_2}\left(\frac{R_2}{b_2} - \frac{\nu}{2}\right)P = 3.2599P$$

$$E\delta^*_{21} = E\delta^*_{22} = 0$$

DISPLACEMENTS DUE TO THERMAL EFFECTS:

$$E\delta_{22} = R_2 E\alpha_m (T_m - 70) = 241.823$$

FOR $E\alpha = 0.247$ KSI/°F, $T_m = 650°F$

$$E\delta^*_{22} = 0$$

DUE TO THE DIFFERENCE IN COEFFICIENTS OF THERMAL EXPANSION FOR THE 304 STAINLESS STEEL AND THE INCONEL MATERIALS, A MISMATCH IN RADIAL EXPANSION OCCURS AT CUT-2 ON A RISE IN TEMPERATURE. THIS MISMATCH WILL CAUSE THERMAL STRESSES. AN INTERACTION ANALYSIS OF THE JUNCTURE WILL ENABLE STRESSES TO BE CALCULATED FOR BOTH PRESSURE AND TEMPERATURE. NOTE ALSO THAT THE DIFFERENCE IN YOUNG'S MODULUS IS TAKEN INTO ACCOUNT.

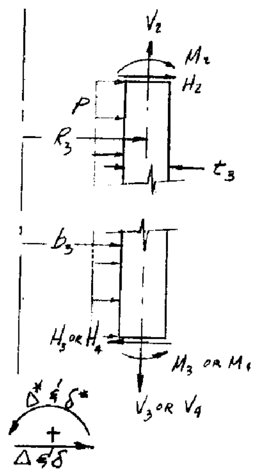
DESCRIPTION CONTROL ROD HOUSING STRESS ANALYSIS AND FATIGUE EVALUATION

5- DETAILED ANALYSIS:

e- DEVELOPMENT OF CONTINUITY EQUATIONS:

2- MOVEMENTS DUE TO REDUNDANT AND APPLIED FORCES:

BODY-3:



$R_3 = 1.688''$

$b_3 = 1.375''$

$t_3 = 0.625$

$\beta^4 = \frac{3(1-\nu^2)}{R^2 t^2} = 2.45278 \qquad \beta^2 = 1.56613$

$\beta = 1.25145$

$D = \frac{Et^3}{12(1-\nu^2)} = 0.02236\,E$

$V_2 = V_3 = 0.560\,P$

$\frac{E_{304SS}}{E_{INCONEL}} = 0.88013 \quad @\ 650°F$

DISPLACEMENTS DUE TO REDUNDANT FORCES

$E\Delta_{32} = \frac{E_{inc}}{2\beta^3 D}\left[\frac{1}{\beta}H_2 + M_2\right]\frac{E_{304}}{E_{INCONEL}} = 10.0414\,H_2 + 12.5661\,M_2$

$E\Delta^*_{32} = \frac{-E_{inc}}{2\beta^2 D}\left[H_2 + 2\beta M_2\right]\frac{E_{304}}{E_{INCONEL}} = -12.5661\,H_2 - 31.4557\,M_2$

} IN TERMS OF YOUNG'S MODULUS OF 304 SS.

$E\Delta_{33} = \frac{-E}{2\beta^2 D}\left[\frac{1}{\beta}H_3 - M_3\right] = -11.4090\,H_3 + 14.2776\,M_3$

$E\Delta^*_{33} = \frac{-E}{2\beta^2 D}\left[H_3 - 2\beta M_3\right] = -14.2776\,H_3 + 35.7398\,M_3$

$E\Delta_{34} = \frac{-E}{2\beta^2 D}\left[\frac{1}{\beta}H_4 - M_4\right] = -11.4090\,H_4 + 14.2776\,M_4$

$E\Delta^*_{34} = \frac{-E}{2\beta^2 D}\left[H_4 - 2\beta M_4\right] = -14.2776\,H_4 + 35.7398\,M_4$

5- DETAILED ANALYSIS:

C. DEVELOPMENT OF CONTINUITY EQUATIONS:

2- MOVEMENTS DUE TO REDUNDANT AND APPLIED FORCES

BODY-3:

DISPLACEMENTS DUE TO APPLIED FORCES:

$$E\delta_{32} = \frac{b_3^2}{t_3}\left(\frac{R_3}{b_3} - \frac{\nu}{2}\right)\frac{E_{304SS}}{E_{INCONEL}} = \frac{(1.375)^2}{0.625}\left(\frac{1.688}{1.375} - \frac{0.3}{2}\right)(0.88013) = 2.8691P \quad *$$

$$E\delta_{32}^* = 0$$

$$E\delta_{33} = E\Delta_{INT} + \frac{b_3^2}{t_3}\left(\frac{R_3}{b_3} - \frac{\nu}{2}\right) = E\Delta_{INT} + 3.2599P$$

(SEE NOTE BELOW) $(\Delta_{INT})_{MAX} = 0.0015''$

$$E\delta_{33}^* = 0$$

$$E\delta_{34} = \frac{b_3^2}{t_3}\left(\frac{R_3}{b_3} - \frac{\nu}{2}\right) = 3.2599P$$

$$E\delta_{34}^* = 0$$

* IN TERMS OF YOUNGS MODULUS OF 304SS.

NOTE THAT STRESSES ARE DEVELOPED WHERE THE CONTROL ROD HOUSING ENTERS THE CLOSURE HEAD DUE TO THE INTERFERENCE FIT (DESIGNATED BY $E\Delta_{INT}$ ABOVE) AND THE EXPANSION OF THE PIPE DUE TO PRESSURE. THE NET DEFLECTION OF THE HOUSING IS ASSUMED TO BE EQUAL TO THE DEFLECTION OF THE HOLE IN THE HEAD TAKING INTO ACCOUNT OF THE ORIGINAL INTERFERENCE FIT. ROTATION OF THE HOUSING IS ASSUMED TO BE ZERO.
THEORETICALLY THE HEAD EXPANDS MORE UNDER PRESSURE THAN THE HOUSING DOES (SEE $E\Delta_{43}$ DUE TO PRESSURE AS GIVEN ON SHEET-15). WHEN THE POINT IS REACHED WHERE THERE IS NO LONGER CONTACT BETWEEN THE HEAD AND THE HOUSING, THE INTERACTION EQUATIONS ARE NO LONGER VALID. WHEN THIS OCCURS, THE STRESSES IN THE HOUSING BECOMES EQUAL TO THE PRESSURE STRESSES ALONE.
NOTE ALSO THAT THE INTERFERENCE FIT IS RELIEVED DUE TO A RISE IN TEMPERATURE; THIS EFFECT WILL BE ACCOUNTED FOR IN THE FATIGUE EVALUATION.

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NUMBER S-100-P | A-11
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DESCRIPTION CONTROL ROD HOUSING STRESS ANALYSIS AND FATIGUE EVALUATION
CHECK DATE 1-24-67 BY [illegible]

5- DETAILED ANALYSIS:

C- DEVELOPMENT OF CONTINUITY EQUATIONS:

2- MOVEMENTS DUE TO REDUNDANT AND APPLIED FORCES:

BODY-3

DEFLECTIONS DUE TO THERMAL EFFECTS:

$E\delta_{32} = R_3 E\alpha_m (T_m - 70) \frac{E_{304SS}}{E_{INCONEL}} = 189.570$ (IN TERMS OF YOUNG'S MODULUS OF 304SS)

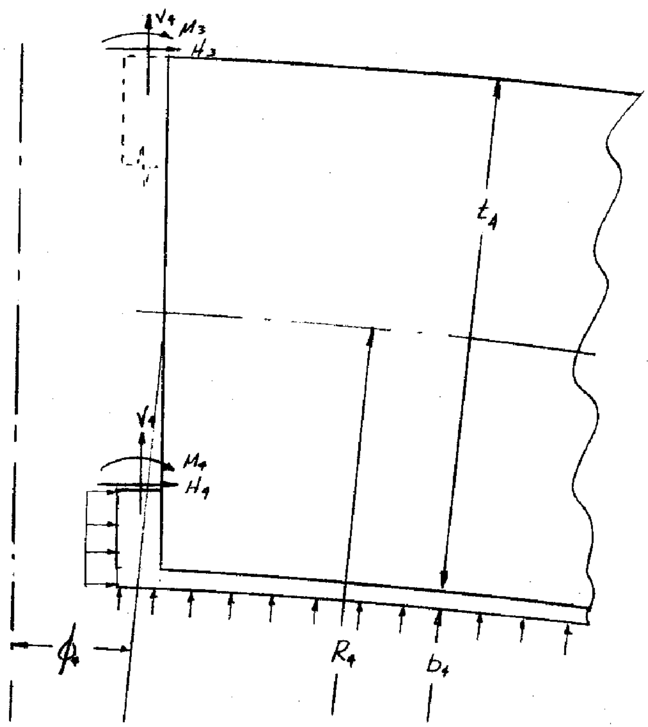
$E\alpha_m = 0.220$
$T_m = 650°F$

$E\delta^*_{32} = E\delta^*_{33} = E\delta^*_{34} = 0$

$E\delta_{33} = R_3 E\alpha_m (T_m - 70) = 1.688 (E\alpha_m)_3 (T_{m_3} - 70)$

$E\delta_{34} = R_3 (E\alpha_m)_3 (T_{m_3} - 70) = 1.688 (E\alpha_m)_3 (T_{m_3} - 70)$

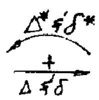
BODY-4:



$R_4 = 89.500''$
$b_4 = 85.781''$
$t_4 = 7.000''$
$\phi_4 = 1°16.8'$

$\frac{E^*}{E} = 0.72$
$\nu^* = 0.285$
} FOR L.E = 0.666

$\frac{E_{INCONEL}}{E_{304SS}} = 1.13793$



5. Detailed Analysis:

E. Development Of Continuity Equations:

2. Movements Due To Redundant And Applied Forces:

Body - 4:

Displacements Due To Redundant Forces:

$$E\Delta_{43} = 0$$

$$E\Delta^*_{43} = -\frac{6.36(1-\nu^2)}{t_3^2}M_3 = \underline{-14.8161 M_3}$$

Note That The Deflection Is Conservatively Ignored, And That The Rotation Is Taken As Equal To The Local Flex. Term For Moment Only.

$$E\Delta_{44} = 1.408(1-\nu^2)\left[\left(\frac{R_3}{t_3}+\frac{1}{2}\right)\ln\left(\frac{R_3}{t_3}+\frac{1}{2}\right)-\left(\frac{R_3}{t_3}-\frac{1}{2}\right)\ln\left(\frac{R_3}{t_3}-\frac{1}{2}\right)+\frac{1}{2}\right]H_4$$

$$+\frac{2.38(1-\nu-2\nu^2)}{t_3}M_4 = \underline{3.1873 H_4 + 1.9802 M_4}$$

$$E\Delta^*_{44} = -\frac{2.38(1-\nu-2\nu^2)}{t_3}H_4 - \frac{6.36(1-\nu^2)}{t_3^2}M_4 = \underline{-1.9802 H_4 - 14.8161 M_4}$$

From Ref. 18

Displacements Due To Pressure:

$$E\delta_{43} = E\delta_{44} = \frac{(1-\nu^*)b_4^2}{2t_4}P\sin\theta\left(\frac{E}{E^*}\right)\left(\frac{E_{INCONEL}}{E_{302B}}\right) = \frac{(1-0.285)(85.781)^2}{2(7)}(0.02234)\left(\frac{1}{0.72}\right)(1.13793)P$$

$$= \underline{13.2686 P}$$

$$E\delta^*_{43} = E\delta^*_{44} = 0$$

Displacements Due To Thermal Effects:

$$E\delta_{43} = R_4\sin\theta\, E\alpha_m(T_m-70)\frac{E_{INCONEL}}{E_{302B}} = 2.276(E\alpha_m)_4(T_{m4}-70)$$

$$E\delta^*_{43} = E\delta^*_{44} = 0$$

$$E\delta_{44} = R_4\sin\theta(E\alpha_m)_4(T_{m4}-70)\frac{E_{INCONEL}}{E_{302B}} = 2.276(E\alpha_m)_4(T_{m4}-70)$$

5- DETAILED ANALYSIS:

e- DEVELOPMENT OF CONTINUITY EQUATIONS:

3- CONTINUITY MATRIX AND LOADING VECTORS:

A PRESSURE SOLUTION WILL BE REQUIRED FOR EACH JUNCTURE, A THERMAL SOLUTION WILL BE REQUIRED AT JUNCTURES 2, 3, & 4, AND AN INTERFERENCE FIT SOLUTION AT JUNCTURE-3 ONLY. THE MATRIX AND COLUMN VECTORS WILL BE ARRANGED AS SHOWN BELOW.

$$\left.\begin{aligned}
E\Delta_{11} - E\Delta_{21} &= E\delta_{21} - E\delta_{11} \\
E\Delta^*_{11} - E\Delta^*_{21} &= E\delta^*_{21} - E\delta_{11} \\
E\Delta_{22} - E\Delta_{32} &= E\delta_{32} - E\delta_{22} \\
E\Delta^*_{22} - E\Delta^*_{32} &= E\delta^*_{32} - E\delta^*_{22} \\
E\Delta_{33} - E\Delta_{43} &= E\delta_{43} - E\delta_{33} \\
E\Delta^*_{33} - E\Delta^*_{43} &= E\delta^*_{43} - E\delta^*_{33} \\
E\Delta_{34} - E\Delta_{44} &= E\delta_{44} - E\delta_{34} \\
E\Delta^*_{34} - E\Delta^*_{44} &= E\delta^*_{44} - E\delta^*_{34}
\end{aligned}\right\}$$

SUBSTITUTING THE DEFLECTIONS AND ROTATIONS INTO THESE COMPATIBILITY EQUATIONS AND WRITING IN MATRIX FORM YIELDS THE MATRIX AND COLUMN VECTORS AS SHOWN BELOW,

$$\begin{bmatrix}
-13.1659 & -13.8898 & 0 & 0 & 0 & 0 & 0 & 0 \\
13.8898 & 35.8507 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & -21.4504 & 1.7115 & 0 & 0 & 0 & 0 \\
0 & 0 & -1.7115 & 67.1955 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & -11.4090 & 14.2776 & 0 & 0 \\
0 & 0 & 0 & 0 & -14.2776 & 50.5559 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & -14.5963 & 12.2974 \\
0 & 0 & 0 & 0 & 0 & 0 & -12.2974 & 50.5559
\end{bmatrix}
\begin{bmatrix} H_1 \\ M_1 \\ H_2 \\ M_2 \\ H_3 \\ M_3 \\ H_4 \\ M_4 \end{bmatrix} =$$

$$\begin{bmatrix} 0.0662P \\ -1.1387P \\ -0.3908P \\ 0 \\ W_P \\ 0 \\ W_P \\ 0 \end{bmatrix} + \begin{bmatrix} \\ \\ -52.253 \\ 0 \\ W_T \\ 0 \\ W_T \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ -E\Delta_{INT} \\ 0 \\ 0 \\ 0 \end{bmatrix}$$

WHERE:

$$W_P = 10.0087\,P$$

$$W_T = 2.276\,E\alpha_{m4}(T_{m4}-70) - 1.688\,E\alpha_{m3}(T_{m3}-70)$$

$$E\Delta_{INT} = 31(-0.0015)\times 10^3 = 46.5 \text{ KIP/in}$$

5. DETAILED ANALYSIS:

e. DEVELOPMENT OF CONTINUITY EQUATIONS:

4- REDUNDANT LOAD VALUES:

INVERTING THE ABOVE MATRIX AND MULTIPLYING BY THE COLUMN VECTORS YIELDS THE FOLLOWING REDUNDANT SHEARS AND MOMENTS.

REDUNDANT FORCE	DESIGN PRESSURE $P = 2.5$ KSI	TEMPERATURE WHERE APPLICABLE	INTERFERENCE FIT WHERE APPLICABLE
H_1	$0.04816P = 0.12040$	—	—
M_1	$-0.05042P = -0.12605$	—	—
H_2	$0.01825P = 0.04563$	2.44095	—
M_2	$0.00046P = 0.00115$	0.06217	—
H_3	$-0.13558W_p = -3.39245$	$-0.13558W_T$	$0.13558E\Delta_{INT} = 6.30447$
M_3	$-0.03829W_p = -0.95808$	$-0.03829W_T$	$0.03829E\Delta_{INT} = 1.78049$
H_4	$-0.08617W_p = -2.15612$	$-0.08617W_T$	—
M_4	$-0.02096W_p = -0.52446$	$-0.02096W_T$	—

NOTE THAT THE ABOVE VALUES FOR THE REDUNDANTS H_3 & M_3 ARE VALID AS LONG AS THERE IS INTERFERENCE BETWEEN THE HOUSING AND HEAD. THE VALUES OF H_3 & M_3 BECOME ZERO IN THE ABSENCE OF INTERFERENCE.

f. STRESSES:

STRESSES WILL BE CALCULATED AT THE TEN LOCATIONS AS SHOWN ON SHEET-18. STRESSES AT LOCATION-3 & 4 WILL BE CALCULATED DUE TO A TEMP. OF 650°F. THIS WILL REPRESENT THE MAXIMUM POSSIBLE EFFECT OF THE DIFFERENCE IN COEFF. OF EXPANSION AND YOUNG'S MODULUS. STRESSES AT LOCATION-7 & 8 WILL BE CALCULATED DUE TO THE MAXIMUM INTERFERENCE FIT. STRESSES WILL BE CALCULATED AT ALL LOCATIONS DUE TO THE DESIGN PRESSURE OF 2.5 KSI.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.______

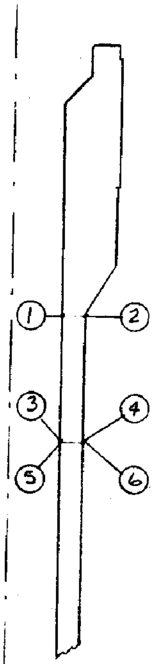
NUMBER 5-100-P | A-15
SHEET 18 OF 27
DATE 1-24-67 BY COCKRELL
CHECK DATE 1-24-67 BY CAUDLE

DESCRIPTION CONTROL ROD HOUSING STRESS ANALYSIS AND FATIGUE EVALUATION

5- DETAILED ANALYSIS:

f- STRESSES:

STRESSES WILL BE CALCULATED AT THE LOCATIONS AS SHOWN BELOW.



POINTS 1 & 2:

$$\sigma_x = \pm\frac{6M_1}{t_2^2} + \frac{b_2^2 P}{2R_2 t_2} = \pm 15.3602 M_1 + 0.896 P$$

$$\sigma_\theta = \pm\frac{\nu 6M_1}{t_2^2} + \frac{E\Delta_{21}}{R} + \frac{b_2 P}{t_2} = \pm 4.6081 M_1 + 0.59241 E\Delta_{21} + 2.2P$$

POINTS 3 & 4:

$$\sigma_x = \pm\frac{6M_2}{t_2^2} + \frac{b_2^2 P}{2R_2 t_2} = \pm 15.3602 M_2 + 0.896 P$$

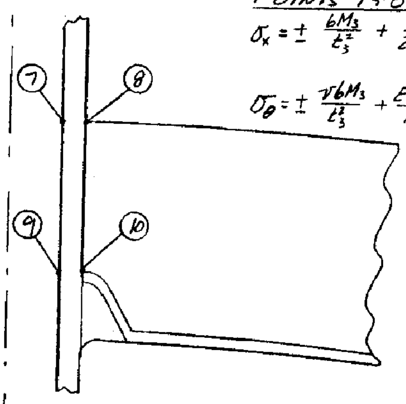
$$\sigma_\theta = \pm\frac{\nu 6M_2}{t_2^2} + \frac{E\Delta_{22}}{R_2} + \frac{b_2 P}{t_2} = \pm 4.6081 M_2 + 0.59241 E\Delta_{22} + 2.2P$$

POINTS 5 & 6:

$$\sigma_x = \pm\frac{6M_2}{t_3^2} + \frac{b_3^2 P}{2R_3 t_3} = \pm 15.3602 M_2 + 0.896 P$$

$$\sigma_\theta = \pm\frac{\nu 6M_2}{t_3^2} + \frac{E\Delta_{32}}{R_3} + \frac{b_3 P}{t_3} = \pm 4.6081 M_2 + 0.59241 E\Delta_{32} + 2.2P$$

POINTS 7 & 8:



$$\sigma_x = \pm\frac{6M_3}{t_3^2} + \frac{b_3^2 P}{2R_3 t_3} = \pm 15.3602 M_3 + 0.896 P$$

$$\sigma_\theta = \pm\frac{\nu 6M_3}{t_3^2} + \frac{E\Delta_{33}}{R_3} + \frac{b_3 P}{t_3} = \pm 4.6081 M_3 + 0.59241 E\Delta_{33} + 2.2P$$

POINTS 9 & 10

$$\sigma_x = \pm\frac{6M_4}{t_3^2} + \frac{b_3^2 P}{2R_3 t_3} = \pm 15.3602 M_4 + 0.896 P$$

$$\sigma_\theta = \pm\frac{\nu 6M_4}{t_3^2} + \frac{E\Delta_{34}}{R_3} + \frac{b_3 P}{t_3} = \pm 4.6081 M_4 + 0.59241 E\Delta_{34} + 2.2P$$

NOTE THAT POINTS 3 & 4 ARE ON THE SA-182 TYPE 304 STAINLESS STEEL AND THAT POINTS 5 & 6 ARE ON THE SB-167 INCONEL MATERIAL.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER 5-100-P | A-16
SHEET 19 OF 27
CHARGE NO. ____
DATE 1-24-67 BY COCKRELL
DESCRIPTION CONTROL ROD HOUSING STRESS ANALYSIS AND FATIGUE EVALUATION
CHECK DATE 1-24-67 BY CAUDLE

5- DETAILED ANALYSIS:

f. STRESSES:

LOCATION	$\frac{6M}{t^2}$	$\frac{PR}{2t}$	σ_x	$\frac{\nu 6M}{t^2}$	$E\Delta$	$\frac{E\Delta}{R}$	$\frac{PR}{t}$	σ_θ	σ_r	STRESS INTENSITY		
										$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$
1	-1.94	2.24	0.30	-0.58	-0.426	-0.25	5.50	4.67	-2.50	-4.37	2.80	7.17
2	1.94		4.18	0.58	-0.426	-0.25		5.83	0	-1.65	4.18	5.83
3	0.02		2.26	0.01	-0.504	-0.30		5.21	-2.50	-2.95	4.76	7.71
4	-0.02		2.22	-0.01	-0.504	-0.30		5.19	0	-2.97	2.22	5.19
5	0.02		2.26	0.01	0.537	0.32		5.83	-2.50	-3.57	4.76	8.33
6	-0.02		2.22	-0.01	0.537	0.32		5.81	0	-3.59	2.22	5.81
7	-14.72		-12.48	-4.41	25.023	14.82		15.91	-2.50	-28.37	-9.98	18.41
8	14.72		16.96	4.41	25.023	14.82		24.73	0	-7.77	16.96	24.73
9	-8.06		-5.82	-2.42	17.111	10.14		13.22	-2.50	-19.04	-3.32	15.72
10	8.06	↓	10.30	2.42	17.111	10.14	↓	18.06	0	-7.76	10.30	18.06
THERMAL STRESSES DUE TO DESIGN TEMP. OF T = 650°F												
3	0.95	0	0.95	0.29	-26.961	-15.97	0	-15.68	0	16.63	0.95	-15.68
4	-0.95		-0.95	-0.29	-26.961	-15.97		-16.26		15.31	-0.95	-16.26
5	0.95		0.95	0.29	28.736	17.02		17.31		-16.36	0.95	17.31
6	-0.95	↓	-0.95	-0.29	28.736	17.02	↓	16.73	↓	-17.68	-0.95	16.73
INTERFERENCE FIT STRESSES (Δ_{INT} = 0.0015 AT JUNCTURE - 3 ONLY)												
7	27.35	0	27.35	8.20	-46.500	-27.55	0	-19.35	0	46.70	27.35	-19.35
8	-27.35	0	-27.35	-8.20	-46.500	-27.55	0	-35.75	0	8.40	-27.35	-35.75
COMBINED STRESS (P = 2.5 KSI; T = 650°F; Δ_{INT} = 0.0015"												
1	-1.94	2.24	0.30	-0.58	-0.426	-0.25	5.50	4.67	-2.50	-4.37	2.80	7.17
2	1.94		4.18	0.58	-0.426	-0.25		5.83	0	-1.65	4.18	5.83
3	0.97		3.21	0.30	-27.465	-16.27		-10.47	-2.50	13.68	5.71	-7.97
4	-0.97		1.27	-0.30	-27.465	-16.27		-11.07	0	12.34	1.27	-11.07
5	0.97		3.21	0.30	29.273	17.34		23.14	-2.50	-19.93	5.71	25.64
6	-0.97		1.27	-0.30	29.273	17.34		22.54	0	-21.27	1.27	22.54
7	12.63		14.87	3.79	-21.477	-12.73		-3.44	-2.50	18.31	17.37	-0.94
8	-12.63		-10.39	-3.79	-21.477	-12.73		-11.02	0	0.63	-10.39	-11.02
9	-8.06		5.82	-2.42	17.111	10.14		13.22	-2.50	-19.04	-3.32	15.72
10	8.06	↓	10.30	2.42	17.111	10.14	↓	18.06	0	-7.76	10.30	18.06

5- DETAILED ANALYSIS:

f. STRESSES:

$E\Delta_{33} = -E\Delta_{INT} + (W_P + W_T) \leq 0$ MUST

$E\Delta_{34} = 0.68386(W_P + W_T) \geq 0$ FOR

TRANSIENT		INTERNAL PRESSURE KSI	W_P	T_4 °F	$(E\alpha_m)_4$ KSI/°F	T_3 °F	$(E\alpha_m)_3$ KSI/°F	W_T	W_P+W_T	$E\Delta$ FORCED $E\Delta_{33}$	$E\Delta$ FORCED $E\Delta_{34}$
a	0	0.315	3.153	70	0.177	99	0.185	-9.056	-5.903	-52.403	-5.903
	4.47hrs	2.250	22.520	518	0.186	547	0.217	14.931	37.451	-9.049	25.611
STEADYSTATE		2.250	22.520	547	0.186	547	0.217	27.208	49.728	0	34.007
b	0	2.250	22.520	547	0.186	518	0.217	37.831	60.351	0	41.272
	4.47hrs	0.315	3.153	99	0.178	70	0.180	11.749	14.902	-31.598	10.191
c	20min	2.250	22.520	547	0.186	554.8	0.218	23.536	46.056	-0.444	31.496
d	20min	2.250	22.520	554.8	0.186	547	0.217	30.510	53.030	0	36.265
e	100sec	2.140	21.419	583.7	0.186	572.8	0.218	32.560	53.979	0	36.914
	225sec	2.275	22.770	583.7	0.186	582	0.218	29.064	51.834	0	35.447
f	40sec	2.320	23.220	583.7	0.186	593	0.219	24.132	47.352	0	32.382
	100sec	2.260	22.620	583.7	0.186	597	0.219	22.654	45.274	-1.226	30.961
	260sec	2.140	21.419	583.7	0.186	585	0.218	27.960	49.379	0	33.768
g	2min	2.370	23.721	555	0.186	567	0.218	22.434	46.155	-0.345	31.564
	3.2min	2.350	23.520	555	0.186	570	0.218	21.330	44.850	-1.650	30.671
	10.4min	2.150	21.519	555	0.186	555	0.218	26.850	48.369	0	33.078
h	10sec	2.220	22.219	554.8	0.186	564	0.218	23.453	45.672	-0.828	31.233
	65sec	1.910	19.117	554.8	0.186	546	0.217	30.876	49.993	0	34.188
i	220min	3.125	31.277	70	0.177	70	0.180	0	31.277	-15.223	21.389
j	HEATUP 0hrs	0.315	3.153	70	0.177	86	0.183	-4.942	-1.789	-48.289	-1.789
	STEADY STATE	2.500	25.022	400	0.185	400	0.212	20.859	45.881	-0.619	31.376
	COOLDOWN 0.7hrs	1.250	12.511	400	0.185	384	0.211	27.116	39.627	-6.873	27.099
k	~	2.350	23.520	554.8	0.186	548.8	0.217	29.851	53.371	0	36.498
	~	2.150	21.519	554.8	0.186	560.8	0.218	24.631	46.150	-0.350	31.560
l	12sec	2.250	22.520	554.8	0.186	521.8	0.217	39.743	62.263	0	42.579
m	10sec	2.760	27.624	554.8	0.186	585	0.218	15.726	43.350	-3.150	29.645
	28sec	2.120	21.218	554.8	0.186	596	0.219	10.788	32.006	-14.494	21.888
	160sec	1.440	14.413	554.8	0.186	550	0.218	28.605	43.018	-3.482	29.418
n	33sec	0.300	3.003	547.0	0.186	430	0.213	72.497	75.500	0	51.631
	54sec	0.700	7.006	547.0	0.186	350	0.209	103.151	110.157	0	75.332

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER 5-100-P | A-18
SHEET 21 OF 27
CHARGE NO. ________
DATE 1-24-67 BY Cockrell
DESCRIPTION Control Rod Housing Stress Analysis And Fatigue Evaluation
CHECK DATE 1-24-67 BY Grubie

5- Detailed Analysis:
 f- Stresses:

Transient		$\frac{6M}{t^2}$	$\frac{PR}{2t}$	$\frac{\nu 6M}{t^2}$	$\frac{E\Delta}{R}$	$\frac{PR}{t}$	Point-7 σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$	Point-8 σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$
P=0 T=70°F		27.35	0	9.20	-27.55	0	27.35	-19.35	0	46.70	27.35	-19.35	-27.35	-35.75	0	8.40	-27.35	-35.75
a	0	30.92	0.28	9.25	-31.04	0.69	31.10	-21.10	-0.32	52.20	31.42	-20.78	-30.54	-39.60		9.06	-30.54	-31.66
	4.47hrs	5.32	2.02	1.60	-5.36	4.95	7.34	1.19	-2.25	6.15	9.59	3.44	-3.30	-2.01		-1.29	-3.30	-7.01
Steady state		0	2.02	0	0	4.95	2.02	4.95	-2.25	-2.93	4.27	7.20	2.02	4.95		-2.93	2.02	4.95
b	0	0	2.02	0	0	4.95	2.02	4.95	-2.25	-2.93	4.27	7.20	2.02	4.95		-2.93	2.02	4.95
	4.47hrs	18.58	0.28	5.58	-18.72	0.69	18.36	-12.45	-0.32	31.31	19.18	-12.13	-18.30	-23.61		5.31	-18.30	-23.61
c	20min	0.26	2.02	0.08	-0.26	4.95	2.28	4.77	-2.25	-2.49	4.53	7.02	1.76	4.61		-2.85	1.76	4.61
d	20min	0	2.02	0	0	4.95	2.02	4.95	-2.25	-2.93	4.27	7.20	2.02	4.95		-2.93	2.02	4.95
e	100sec	0	1.92	0	0	4.71	1.92	4.71	-2.14	-2.79	4.06	6.85	1.92	4.71		-2.79	1.92	4.71
	225sec	0	2.04	0	0	5.01	2.04	5.01	-2.28	-2.97	4.32	7.29	2.04	5.01		-2.79	2.04	5.01
f	40sec	0	2.08	0	0	5.10	2.08	5.10	-2.32	-3.02	4.40	7.42	2.08	5.10		-3.02	2.08	5.10
	100sec	0.72	2.02	0.22	-0.73	4.97	2.74	4.46	-2.26	-1.72	5.00	6.72	1.30	4.02		-2.72	1.30	4.02
	260sec	0	1.92	0	0	4.71	1.92	4.71	-2.14	-2.79	4.06	6.85	1.92	4.71		-2.79	1.92	4.71
g	2min	0.20	2.12	0.06	-0.20	5.21	2.32	5.07	-2.37	-2.75	4.69	7.44	1.92	4.95		-3.03	1.92	4.95
	12min	0.17	2.11	0.29	-0.98	5.17	5.09	4.48	-2.35	-1.40	5.43	6.73	1.14	3.90		-2.76	1.14	3.90
	34min	0	1.93	0	0	4.73	1.93	4.73	-2.15	-2.80	4.08	.98	1.93	4.73		-2.80	1.93	4.73

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER 5-100-P | A-19
SHEET 22 OF 27
CHARGE NO.
DATE 1-24-67 BY COCKRELL
DESCRIPTION CONTROL ROD HOUSING STRESS ANALYSIS AND FATIGUE EVALUATION
CHECK DATE 1-24-67 BY CAUDLE

5- DETAILED ANALYSIS:

f- STRESSES:

TRANSIENT		$\frac{6M}{t^2}$	$\frac{PR}{2t}$	$\frac{\nu 6M}{t^2}$	$\frac{E\Delta}{R}$	$\frac{PR}{t}$	POINT - 7						POINT - 8					
							σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$	σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$
h	10sec	0	1.99	0	0	4.88	1.99	4.88	-2.22	-2.89	4.21	7.10	1.99	4.88	0	-2.89	1.99	4.88
	65sec	0	1.71	0	0	4.20	1.71	4.20	-1.91	-2.49	3.62	6.11	1.71	4.20		-2.49	1.71	4.20
i	220min	8.95	2.80	2.69	-9.02	6.88	11.75	2.55	-3.13	11.20	14.88	3.68	-6.15	-4.83		-1.32	-6.15	-4.83
j	HEATUP 0hrs	29.40	0.28	8.52	-28.61	0.69	28.68	-19.40	-0.32	48.08	29.00	-19.08	-28.12	-36.44		8.32	-28.12	-36.44
	STEADY STATE	0.36	2.24	0.11	-0.37	5.50	2.60	5.24	-2.50	-2.64	5.10	7.74	1.88	5.02		-3.14	1.88	5.02
	COOLDOWN 0hrs	4.04	1.12	1.21	-4.07	2.75	5.16	-0.11	-1.25	5.27	6.41	1.14	-2.92	-2.53		-0.39	-2.92	-2.53
k	~	0	2.11	0	0	5.17	2.11	5.17	-2.35	-3.06	4.46	7.52	2.11	5.17		-3.06	2.11	5.17
	~	0.21	1.93	0.06	-0.21	4.73	2.14	4.58	-2.15	-2.44	4.29	6.73	1.72	4.46		-2.74	1.72	4.46
l	12sec	0	2.02	0	0	4.95	2.02	4.95	-2.25	-2.93	4.27	7.20	2.02	4.95		-2.93	2.02	4.95
m	0sec	1.85	2.47	0.56	-1.87	6.07	4.32	4.76	-2.76	-0.44	7.08	7.52	0.62	3.64		-3.02	0.62	3.64
	28sec	8.52	1.90	2.56	-8.59	4.66	10.42	-1.37	-2.12	11.79	12.54	0.75	-6.62	-6.49		-0.13	-6.62	-6.49
	160sec	2.05	1.29	0.61	-2.06	3.17	3.34	1.72	-1.44	1.62	4.78	3.16	-0.76	0.50		-1.26	-0.76	0.50
n	33sec	0	0.27	0	0	0.66	0.27	0.66	-0.30	-0.39	0.57	0.96	0.27	0.66		-0.39	0.27	0.66
	54sec	0	0.63	0	0	1.54	0.63	1.54	-0.70	-0.91	1.33	2.24	0.63	1.54		-0.91	0.63	1.54

$S.I._{MAX\ RANGE}$ = 55.3 KSI @ POINT 7 CRITERION 5-C-3

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO.

DESCRIPTION *Control Rod Housing Stress Analysis And Fatigue Evaluation*

NUMBER *S-100-P* | A-20

DATE *1-24-67* BY *Cockrell*
CHECK DATE *1-24-67* BY *Caudle*

5. Detailed Analysis:

f. Stresses

		$\frac{6M}{t^2}$	$\frac{PR}{2t}$	$\frac{\nu 6M}{t^2}$	$\frac{E\Delta}{R}$	$\frac{PR}{t}$	Point-9 σ_x	Point-9 σ_θ	Point-9 σ_r	Point-9 $\sigma_x-\sigma_\theta$	Point-9 $\sigma_x-\sigma_r$	Point-9 $\sigma_\theta-\sigma_r$	Point-10 σ_x	Point-10 σ_θ	Point-10 σ_r	Point-10 $\sigma_x-\sigma_\theta$	Point-10 $\sigma_x-\sigma_r$	Point-10 $\sigma_\theta-\sigma_r$	Point-10 $4(\sigma_\theta-\sigma_r)$
a	0	0	0.28	0	-3.50	0.69	0.28	-2.31	-0.32	3.09	0.60	-2.49	0.28	-2.81	0	3.09	0.28	-2.81	-11.24
	4.47hrs	-12.06	2.02	-3.62	15.17	4.95	-10.04	16.50	-2.25	-26.54	-7.79	18.75	14.08	23.74		-9.66	14.08	23.74	94.96
Steady State		-16.01	2.02	-4.80	20.15	4.95	-13.99	20.30	-2.25	-34.29	-11.74	22.55	18.03	29.90		-11.87	18.03	29.90	119.60
b	0	-19.43	2.02	-5.83	24.45	4.95	-17.41	23.57	-2.25	-40.98	-15.16	25.82	21.45	35.23		-13.78	21.45	35.23	140.92
	4.47hrs	-4.80	0.28	-1.44	6.04	0.69	-4.52	5.29	-0.32	-9.81	-4.20	5.61	5.08	8.17		-3.09	5.08	8.17	32.68
c	20min	-14.83	2.02	-4.45	18.66	4.95	-12.81	19.16	-2.25	-31.97	-10.56	21.41	16.85	28.06		-11.21	16.85	28.06	112.24
d	20min	-17.07	2.02	-5.12	21.48	4.95	-15.05	21.31	-2.25	-36.36	-12.80	23.56	19.09	31.55		-12.46	19.09	31.55	126.20
e	105sec	-17.38	1.92	-5.21	21.87	4.71	-15.46	21.37	-2.14	-36.83	-13.32	23.51	19.30	31.79		-12.49	19.30	31.79	127.16
	225sec	-16.69	2.04	-5.01	21.00	5.01	-14.65	21.00	-2.28	-35.65	-12.37	23.28	18.73	31.02		-12.29	18.73	31.02	124.08
f	40sec	-15.24	2.08	-4.57	19.18	5.10	-13.16	19.71	-2.32	-32.87	-10.84	22.03	17.32	28.85		-11.53	17.32	28.85	115.40
	100sec	-14.58	2.02	-4.37	18.34	4.97	-12.56	18.94	-2.26	-31.50	-10.30	21.20	16.60	27.68		-11.08	16.60	27.68	110.72
	240sec	-15.90	1.92	-4.77	20.00	4.71	-13.98	19.94	-2.14	-33.92	-11.84	22.08	17.82	29.48		-11.66	17.82	29.48	117.92
g	2min	-14.86	2.12	-4.46	18.70	5.21	-12.74	19.45	-2.37	-32.19	-10.37	21.82	16.98	28.37		-11.39	16.98	28.37	113.48
	3.2min	-14.44	2.11	-4.33	18.17	5.17	-12.33	19.01	-2.35	-31.34	-9.98	21.36	16.55	27.67		-11.12	16.55	27.67	110.68
	10.4min	-15.57	1.93	-4.67	19.60	4.73	-13.64	19.66	-2.15	-33.30	-11.49	21.81	17.50	29.00	↓	-11.50	17.50	29.00	116.00

COMBUSTION ENGINEERING, INC.

ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO. ______

DESCRIPTION: Control Rod Housing Stress Analysis And Fatigue Evaluation

NUMBER: 5-100-P | A-21

DATE: 1-24-67 BY: Cockrell

CHECK DATE: 1-24-67 BY: Caudle

5- Detailed Analysis:

f. Stresses:

Transient		$\frac{6M}{t^2}$	$\frac{PR}{2t}$	$\frac{\nu 6M}{t^2}$	$\frac{E\Delta}{R}$	$\frac{PR}{t}$	Point-9						Point-10						
							σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$	σ_x	σ_θ	σ_r	$\sigma_x-\sigma_\theta$	$\sigma_x-\sigma_r$	$\sigma_\theta-\sigma_r$	$4(\sigma_\theta-\sigma_r)$
h	10sec	-14.70	1.99	-4.41	18.50	4.88	-12.71	18.97	-2.22	-31.68	-10.49	21.19	16.69	27.79	0	-11.10	16.69	27.79	111.16
	65sec	-16.10	1.71	-4.83	20.25	4.20	-14.39	19.62	-1.91	-34.01	-12.48	21.53	17.81	29.28		-11.47	17.81	29.28	117.12
i	220min	-10.07	2.80	-3.02	12.67	6.88	-7.27	16.53	-3.13	-23.80	-4.14	19.66	12.87	22.57		-9.70	12.87	22.57	90.28
j	Heatup 0hrs	0	0.28	0	-1.06	0.69	0.28	-0.37	-0.32	0.65	0.60	-0.05	0.28	-0.37		0.65	0.28	-0.37	-1.48
	Steady State	-14.77	2.24	-4.43	18.59	5.50	-12.53	19.66	-2.50	-32.19	-10.03	22.16	17.01	28.52		-11.51	17.01	28.52	114.08
	Cooldown 0hrs	-12.76	1.12	-3.83	16.05	2.75	-11.64	14.97	-1.25	-26.61	-10.39	16.22	13.88	22.63		-8.75	13.88	22.63	90.52
k	~	-17.18	2.11	-5.15	21.62	5.17	-15.07	21.64	-2.35	-36.71	-12.72	23.99	19.29	31.94		-12.65	19.29	31.94	127.76
	~	-14.86	1.93	-4.46	18.70	4.73	-12.93	18.97	-2.15	-31.90	-10.78	21.12	16.79	27.89		-11.10	16.79	27.89	111.56
l	12sec	-20.05	2.02	-6.01	25.22	4.95	-18.03	24.16	-2.25	-42.19	-15.78	26.41	22.07	36.18		-14.11	22.07	36.18	144.72
m	10sec	-13.96	2.47	-4.19	17.56	6.07	-11.49	19.44	-2.76	-30.93	-8.73	22.20	16.43	27.82		-11.39	16.43	27.82	111.28
	29sec	-10.30	1.90	-3.09	12.97	4.66	-8.40	14.54	-2.12	-22.94	-6.28	16.66	12.20	20.72		-8.52	12.20	20.72	82.88
	160sec	-13.85	1.29	-4.16	17.43	3.17	-12.56	16.44	-1.44	-29.00	-11.12	17.88	15.14	24.76		-9.62	15.14	24.76	99.04
n	33sec	-24.31	0.27	-7.29	30.59	0.66	-24.04	23.96	-0.30	-48.00	-23.74	24.26	24.58	38.54		-13.96	24.58	38.54	154.16
	54sec	-35.47	0.63	-10.64	44.63	1.54	-34.84	35.53	-0.70	-70.37	-34.14	36.23	36.10	56.81		-20.71	36.10	56.81	227.24

$S.I._{\substack{MAX\\RANGE}}$ = 42.94 KSI @ Point-9
= 34.8 KSI @ Point-10
} Criterion 5-C-3

5- Detailed Analysis:

f- Stresses:

Criterion 5-C-1:

$S.I._{MAX} = \frac{PR}{t} + \frac{P}{2} = \underline{6.75 KSI} < S_m = $ 15.3 KSI For SA-182 Type 304 SS / 23.3 KSI For SB-167 Inconel @ 650°F

Criterion 5-C-2:

$S.I._{MAX} = \frac{PR}{t} + \frac{P}{2} + \frac{E\Delta}{R} = \underline{16.9 KSI} < 1.5 S_m = 34.95 KSI$ @ 650°F [For Points 9 & 10]

Criterion 5-C-3:

$S.I._{MAX} = \sigma_x - \sigma_\theta = \underline{55.3 KSI} < 3 S_m = 69.9 KSI$ @ 550°F [For Point-7]

See Sheets 21 & 22

g- Fatigue Evaluation:

Consider Juncture-1

$S.I._{MAX} = (\sigma_\theta - \sigma_r) = 8.96 KSI$ @ Point-1 [For T=100°F & P=3.125 KSI Transient-i (Plant Hydro)]

$S.I._{MIN} = 0$ For T = 70°F & P = 0

$S_{ALT} = \frac{S.I._{MAX} - S.I._{MIN}}{2} = \underline{4.5 KSI}$ From Fig. N-415(B) Of Section III

$N = \infty$ ∴ $U_{OVERALL} = \underline{0}$

Consider Juncture-2

$S.I._{MAX} = (\sigma_\theta - \sigma_r) = 22.72 KSI$ @ Point 5 [For T = 550°F & P = 2.76 KSI Transient-m (Loss Of Load)]

$S.I._{MIN} = 0$ For T = 70°F & P = 0

$S_{ALT} = \frac{S.I._{MAX} - S.I._{MIN}}{2} = \underline{11.36 KSI}$ From Fig. N-415(B) Of Section III

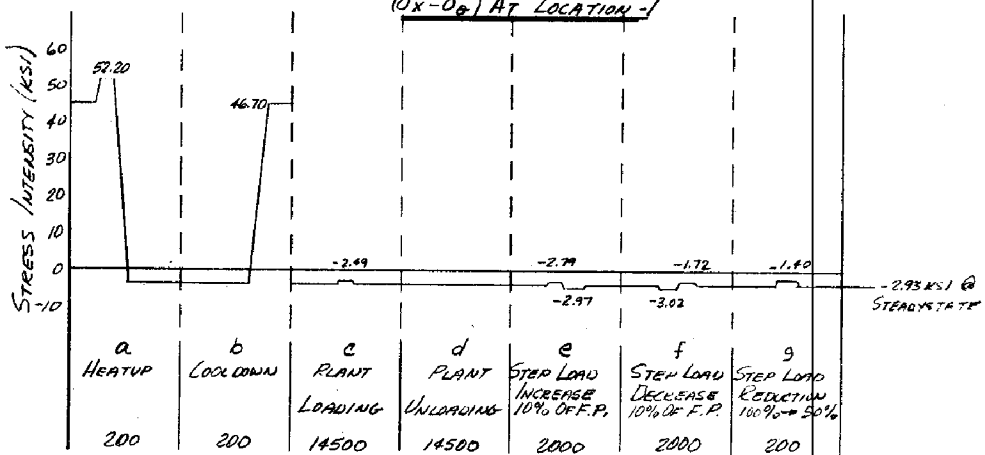
$N = \infty$ ∴ $U_{OVERALL} = \underline{0}$

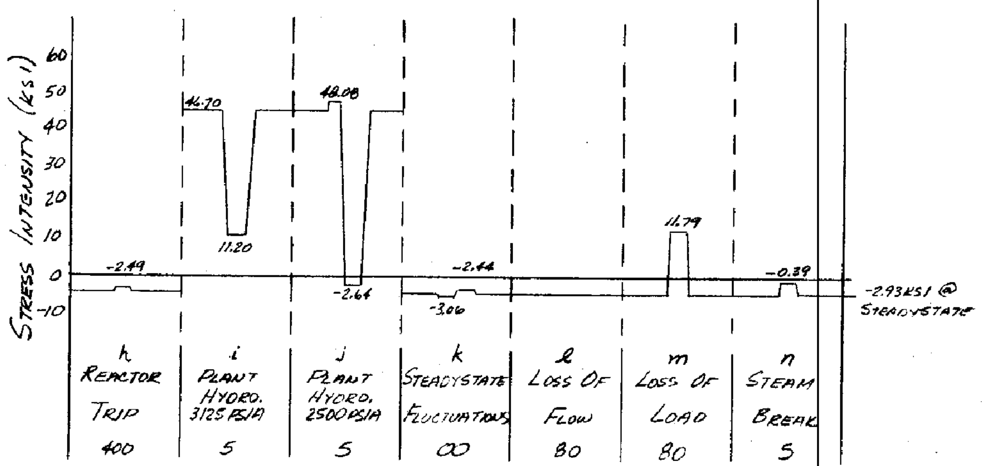
COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER S-100-P | A-23
SHEET 26 OF 27
CHARGE NO.
DATE 1-24-67 BY Cockrell
DESCRIPTION Control Rod Housing Stress Intensity And Fatigue Evaluation
CHECK DATE 1-24-67 BY Caudle

5- Detailed Analysis:
g- Fatigue Evaluation:

$(\sigma_x - \sigma_\theta)$ At Location -7





S_{max}	S_{min}	Number of Occurrences	S_{alt}	N^*	U
52.20	-3.06	200	2763	650,000	0.00030
48.08	-2.64	5	25.36	∞	0

*From Fig. N-415(B) Of Ref. 1

$U_{overall}$ = 0.00030

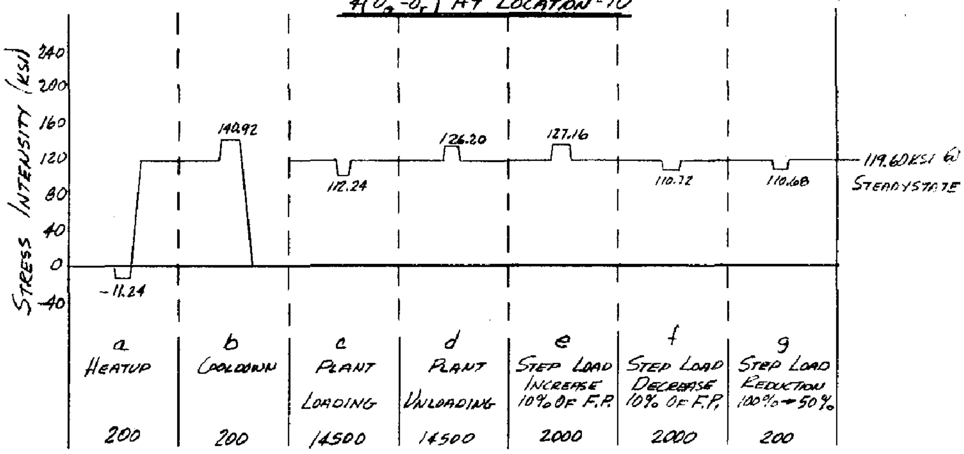
COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

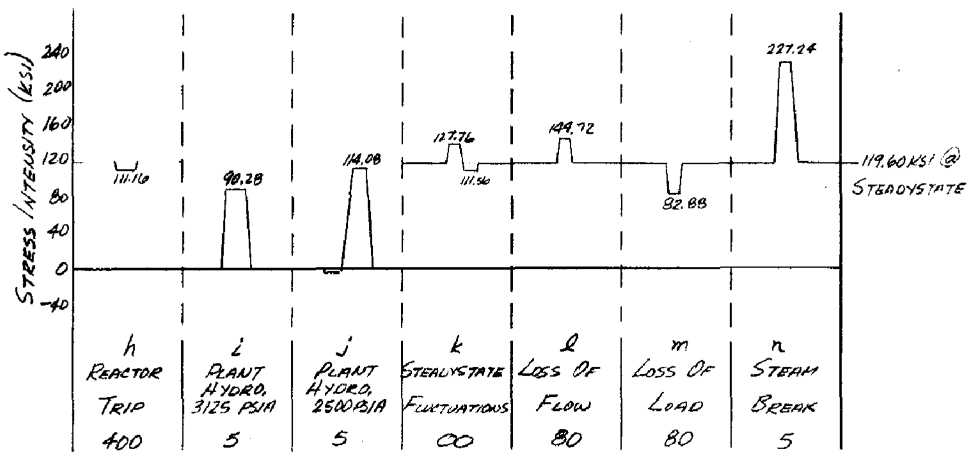
NUMBER 5-100-P | A-24
SHEET 27 OF 27
CHARGE NO.
DATE 1-24-67 BY COCKRELL
DESCRIPTION CONTROL ROD HOUSING STRESS INTENSITY AND FATIGUE EVALUATION
CHECK DATE 1-24-67 BY CAUDLE

5- DETAILED ANALYSIS:

g. FATIGUE EVALUATION:

$4(\sigma_a - \sigma_r)$ AT LOCATION-10





S_{MAX}	S_{MIN}	NUMBER OF OCCURRENCES	S_{ALT}	N*	U
227.24	-11.24	5	119.24	750	0.00666
144.72	-11.24	80	77.98	3600	0.02222
140.92	-11.24	115	76.08	3700	0.03108
114.08	-1.48	5	57.78	11000	0.00045
90.28	0	5	45.14	36000	0.00014
127.80	82.88	80	22.46	∞	

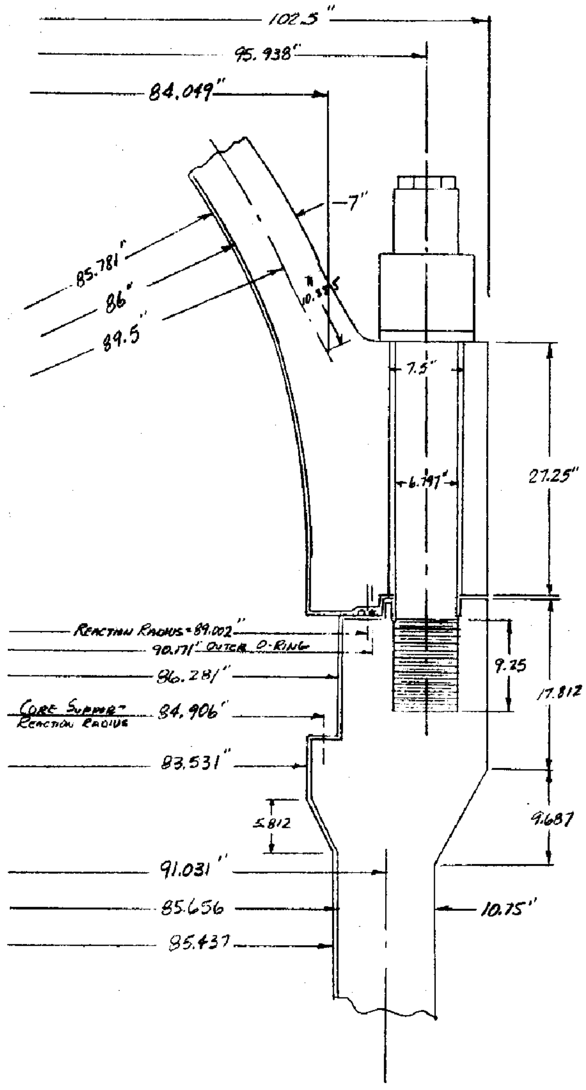
*FROM FIG. N-415(B) OF REF. 1

$U_{OVERALL} = 0.06055$

5- Detailed Analysis:

a- System Geometry:

A Cross Section Of The Closure Head, Closure Head Flange, Vessel Flange, Vessel, And Closure Studs Is Shown Below. Certain Dimensions Of Interest Are Listed.



All Materials Are SA-302-B Or Have Same Allowables Except Nuts, Studs And Washers Which Are SA-320 Gr. L43

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER S-150-P | A-26
SHEET 6 OF 29
CHARGE NO.
DATE 10-22-65 BY COCKRELL
DESCRIPTION STRUCTURAL ANALYSIS OF THE CLOSURE HEAD AND VESSEL ASSEMBLY
CHECK DATE 10-27-65 BY ALEXANDER

5- DETAILED ANALYSIS:

b. SYSTEM LOADS:

THE CLOSURE HEAD AND VESSEL ASSEMBLY WILL BE INVESTIGATED FOR THE FOLLOWING LOADS IN THIS ANALYSIS:

1- DESIGN PRESSURE OF 2.5 KSI AT DESIGN TEMP. OF 650°F.
2. INITIAL BOLT-UP 10% GREATER THAN THE DESIGN PRESSURE BLOW OFF LOAD BASED ON THE ℄ OF THE OUTER O-RING.
3. THE CORE AND CORE SUPPORT WEIGHT OF 675 KIPS PLUS THE CORE HOLDDOWN SPRING FORCE OF 660 KIPS.
THE ABOVE LOADS ARE BASED ON WESTINGHOUSE EQUIPMENT SPEC. 676208 REF. 9.

c. SYSTEM ALLOWABLES:

THE FOLLOWING ALLOWABLE STRESSES ARE BASED ON THE A.S.M.E. NUCLEAR CODE SECTION III, REFERENCE-1 AND ARE RELEVANT FOR THIS ANALYSIS.

1- THE AVERAGE PRIMARY STRESS INTENSITY ACROSS A SOLID SECTION SHALL NOT EXCEED S_m AT DESIGN TEMP. AND PRESSURE.
2- THE LOCAL PRIMARY STRESS PLUS THE AVERAGE PRIMARY STRESS SHALL NOT EXCEED $1.5 S_m$ AT DESIGN TEMP. AND PRESSURE.
3- THE PRIMARY BENDING STRESS ALONE OR COMBINED WITH THE ABOVE SHALL NOT EXCEED $1.5 S_m$ AT DESIGN TEMP. AND PRESSURE.
4- THE AVERAGE BOLT STRESS RESULTING FROM THE DESIGN PRESS. BLOW-OFF LOAD PLUS O-RING SEATING LOAD SHALL NOT EXCEED THE S_m VALUE AT THE 650°F DESIGN TEMP.
5- THE MAXIMUM AVERAGE BOLT SERVICE STRESS SHALL NOT EXCEED $2 S_m$ AT OPERATING TEMP. AND PRESSURE.
6- THE MAXIMUM BOLT SERVICE STRESS SHALL NOT EXCEED $3 S_m$ AT OPERATING TEMP. AND PRESSURE.
7- THE AVERAGE PRIMARY SHEAR STRESS ACROSS A SECTION IN PURE SHEAR (SUCH AS SCREW THREADS) SHALL NOT EXCEED $0.6 S_m$ AT DESIGN TEMP.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO.

DESCRIPTION: STRUCTURAL ANALYSIS OF THE CLOSURE HEAD AND VESSEL ASSEMBLY

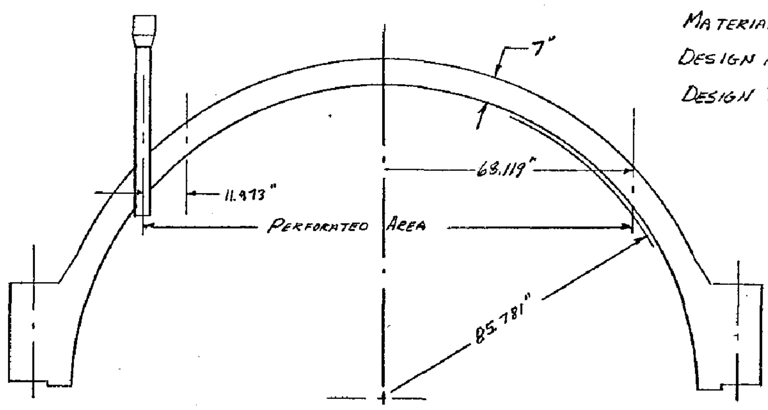
NUMBER: S-150-P | A-27
SHEET 7 OF 29
DATE 10-22-65 BY COCKRELL
CHECK DATE 10-22-65 BY ALEXANDER

5- DETAILED ANALYSIS:

C- SYSTEM ALLOWABLES:

B- THE AVERAGE BEARING STRESS SHALL NOT EXCEED S_y AT ACTUAL METAL TEMPERATURE.

d- DESIGN SIZING:



MATERIAL: SA-302B
DESIGN PRESS: 2.5 KSI
DESIGN TEMP: 650°F

CONSIDER THE REQUIRED THICKNESS IN THE UN-PERFORATED AREA: REFERENCE PARAGRAPH N-431, SECTION III NUCLEAR CODE:

$$t_{REQ'D\ (UNPERFORATED)} = \frac{PR}{2S_m - P} = \frac{2.5(85.781)}{2(26.7)-2.5} = 4.213 \text{ in}$$

USE 7" MIN TO ALLOW FOR PENETRATIONS

WHERE: t = SHELL THICKNESS (EXCLUDING CLAD)
P = DESIGN PRESSURE
R = INSIDE RADIUS OF HEAD (TO CLAD SURFACE)
S_m = 26.7 KSI (ALLOWABLE STRESS AT 650°F)

NOTE:
SEE APPENDIX-C FOR JUSTIFICATION OF PENETRATION SPACING.

5- DETAILED ANALYSIS:

d- DESIGN SIZING:

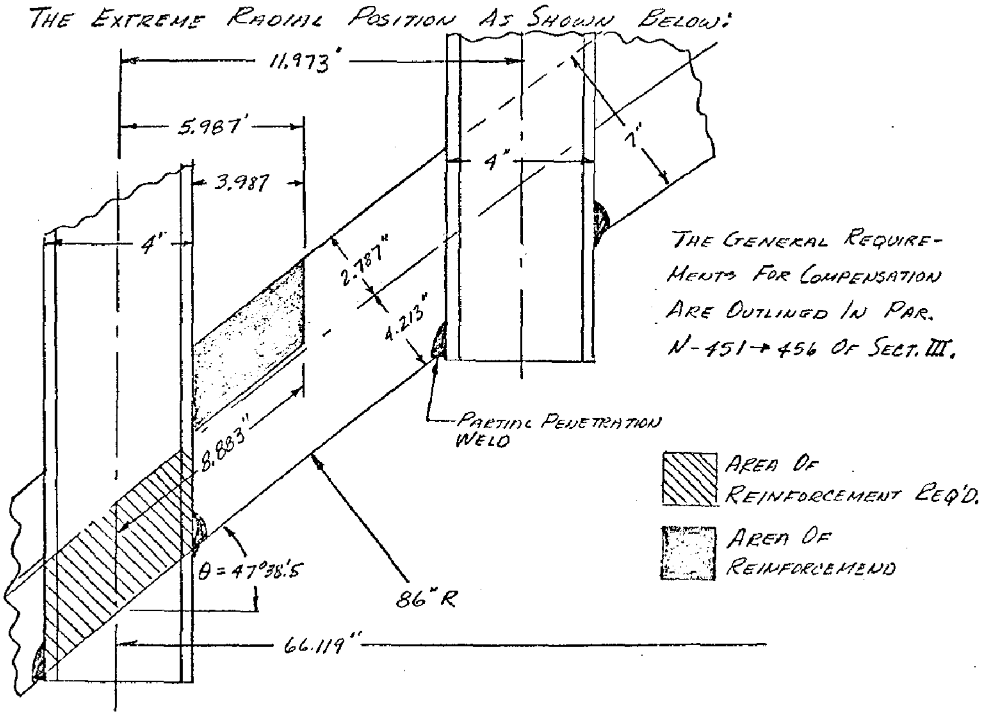
CONSIDER THE REQUIRED THICKNESS IN THE PERFORATED AREA:

SINCE THE HOLES ARE IN A SQUARE PATTERN THE REQUIRED THICKNESS MAY BE CONSERVATIVELY ESTIMATED BY DIVIDING THE REQUIRED SOLID THICKNESS BY THE LIGAMENT EFFICIENCY. THIS INSURES SATISFACTION OF THE PRIMARY STRESS REQUIREMENT IN THE PERFORATED AREA.

$$\eta = \text{LIGAMENT EFFICIENCY} = \frac{11.973 - 4}{11.973} \doteq 0.666$$

$$t_{\substack{REQ'D \\ PERFORATED}} = \frac{t_{SOLID}}{\eta} = \frac{4.213}{0.666} \doteq 6.326'' < 7.000'' \text{ (CRITERION 5-c-1)}$$

ANOTHER APPROACH WHICH MAY BE USED TO INSURE ADEQUATE HEAD THICKNESS IS TO ASSUME THE CONTROL ROD HOUSING AS A CONNECTION WHICH REQUIRES REINFORCEMENT. THE CONTROLLING LOCATION IS AT THE EXTREME RADIAL POSITION AS SHOWN BELOW:



THE GENERAL REQUIREMENTS FOR COMPENSATION ARE OUTLINED IN PAR. N-451 → 456 OF SECT. III.

5. DETAILED ANALYSIS:

d. DESIGN SIZING:

THE AREA OF REINFORCEMENT REQUIRED IS:

$$A_{REQ'D} = \frac{4 \times 4.213}{\cos\theta} = 25.00\ in^2 \qquad \theta = \sin^{-1}\frac{66.119}{89.5} = 47°38.5'$$

CONSIDER LIMITS OF REINFORCEMENT:

THE LIMIT OF REINFORCEMENT IS MEASURED ALONG THE MIDSURFACE OF THE NOMINAL WALL THICKNESS AND IS THE LARGER OF: (SEE N-454 & 455).

a - THE DIAMETER OF THE FINISHED OPENING 4 in., (OR)
b - THE FINISHED RADIUS OF THE OPENING (2") PLUS THE THICKNESS OF THE CLOSURE HEAD (7"). NOTE THAT THE CONTROL ROD HOUSING WALL THICKNESS CANNOT BE COUNTED BECAUSE OF THE PARTIAL PENETRATION WELD. THIS TOTAL IS 9.0 in.

THE HORIZONTAL PROJECTION OF b IS:

$$HORIZ.\ PROJ. = 9.0 \cos\theta = 6.066''$$

$$SINCE\ 6.066 > \frac{11.973}{2} \quad USE \frac{11.973}{2} = 5.987''$$ (LIMIT OF REINFORCEMENT)

THE AREA FURNISHED FOR REINFORCEMENT IS:

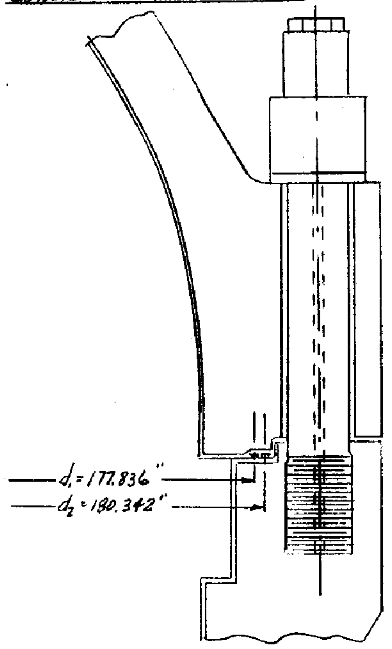
$$A_{FURNISHED} = \frac{2(3.987)(2.787)}{\cos\theta} = 32.97\ in^2 \qquad \cos\theta = 0.67398$$

SINCE THE AREA FURNISHED FOR COMPENSATION EXCEEDS THE AREA REQUIRED, THE REINFORCEMENT REQUIREMENT IS SATISFIED. AT PENETRATIONS CLOSER TO THE VERTICAL AXIS OF THE HEAD, THERE IS A LARGER MARGIN BETWEEN THE REINFORCEMENT FURNISHED AND THAT WHICH IS REQUIRED. NOTE THAT THE VENT NOZZLE DOES NOT REQUIRE REINFORCEMENT SINCE IT MEETS THE REQUIREMENTS OF N-452(a) OF SECT III.

5- DETAILED ANALYSIS:

d- DESIGN SIZING:

CONSIDER CLOSURE STUDS:



P = DESIGN PRESSURE: 2.5 KSI

DESIGN TEMP.: 650 °F

d_s = STUD SIZE: 6.797" SHANK DIA.

d_H = HOLE THRU CENTER OF STUD: 1.125"

N = NUMBER OF STUDS: 54

STUD MATERIAL: SA-320 GR. L43 CLASS 3 WITH $S_m = 34.8$ KSI AT 650°F.

NOTE: REFER TO ARTICLE I-12 OF SECTION III, NUCLEAR CODE FOR DESIGN SIZING OF STUDS.

THE DESIGN BLOW OFF LOAD IS:

$$\frac{\pi}{4} d_o^2 P = \frac{\pi (180.342)^2 (2.5)}{4} = 63,859 \text{ KIPS}$$

IT WILL BE CONSERVATIVELY ASSUMED THAT 1.5 KIPS PER INCH OF O-RING IS REQUIRED FOR SEALING.

THE O-RING SEATING LOAD $= 1.5\pi(d_1 + d_2) = 1.5\pi(177.836 + 180.342) = 1688$ KIPS

THE TOTAL LOAD IS $= F = 63859 + 1688 = 65547$ KIPS

THE TOTAL BOLT AREA $= \frac{\pi}{4}\left[d_s^2 - d_H^2\right] N = \frac{\pi}{4}\left[6.797^2 - 1.125^2\right] 54 = 1905.7 \text{ in}^2$

THE STUD DESIGN STRESS IS:

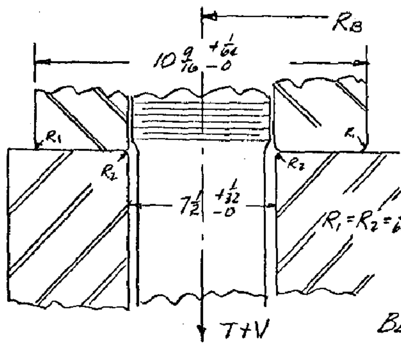
$\sigma_x = \frac{F}{A} = \frac{65547}{1905.7} = 34.4$ KSI

THIS STRESS IS BLOW THE ALLOWABLE OF $S_m = 34.8$ KSI, CRITERION 5-C-4

5- DETAILED ANALYSIS:

d- DESIGN SIZING:

CONSIDER BEARING STRESS BETWEEN STUD WASHER AND HEAD FLANGE:



LOAD PER STUD $= \frac{2\pi R_B (T+V)}{54} = \frac{2\pi (45.938)(116.532+9.767)}{54}$

$= \underline{1410 \text{ KIPS}}$

MINIMUM O.D. OF WASHER

$10.5625 - 0.0313 = 10.5312$

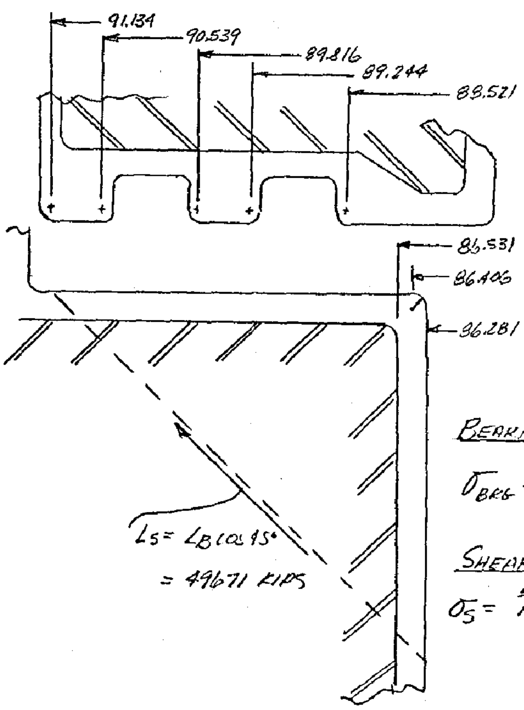
MAXIMUM DIA OF STUD HOLE

$7.500 + 0.0625 = 7.5625$

BEARING AREA/STUD $= \frac{\pi}{4}(10.5312^2 - 7.5625^2) = 42.19 \text{ IN}^2$

$\sigma_{BRG} = \frac{F}{A} = \frac{1410}{42.19} = \underline{33.5 \text{ KSI}} < S_y = 42.6 \text{ KS @ } 550°F$ FOR HEAD FLANGE
SEE N-417.1(a) OF REF. 1

CONSIDER HEAD AND VESSEL FLANGE MATING SURFACE:



THE CONTROLLING CONDITION WILL BE FOR BOLT-UP. THE BOLT-UP LOAD IS $L_B = 1.1(63859.2) = 70245.1$ KIPS

BEARING AREA $= \pi(91.134^2 - 90.539^2 + 89.816^2 - 89.244^2 + 88.521^2 - 86.531^2)$

$= 1755.7 \text{ IN}^2$

SHEAR AREA $= \frac{\pi}{\cos 45°}(91.134^2 - 86.281^2)$

$= 3825.3 \text{ IN}^2$

BEARING STRESS:

$\sigma_{BRG} = \frac{L_B}{A_B} = \underline{40.0 \text{ KSI}} < S_y = 50 \text{ KSI @ } 100°F$
SEE N-417.1(a) OF REF. 1.

SHEAR STRESS

$\sigma_S = \frac{L_S}{A_S} = \underline{13.0 \text{ KSI}} < 0.6 S_m = 16.0 \text{ KSI}$ FOR FLANGE MATERIAL AND 16.1 KSI FOR CLAD MATL.
SEE N-417.1(b) OF REF. 1.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

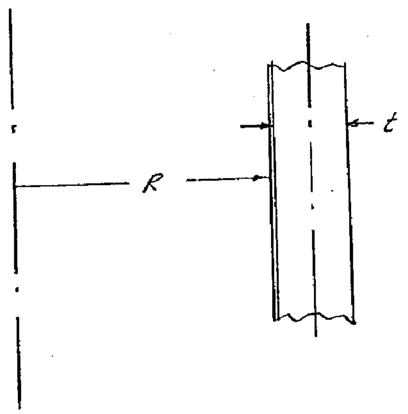
CHARGE NO.

DESCRIPTION Structural Analysis Of The Closure Head And Vessel Assembly

NUMBER 5-150-P | A-32
SHEET 12 OF 29
DATE 10-22-65 BY Cockrell
CHECK DATE 10-22-65 BY Alexander

5- Detailed Analysis:
d- Design Sizing:

Consider The Vessel Wall:



Material: SA-302 B
Design Press.: 2.5 ksi
Design Temp: 650°F
$S_m = 26.7$ ksi @ 650°F
$R = 86.5''$
$t_{ACT} = 8.625$ in

From N-431 Of Section III, ASME Nuclear Code:

$$t_{REQ'D} = \frac{PR}{S_m - 0.5P} = \frac{2.5(86.5)}{26.7 - 0.5(2.5)} = 8.497'' < 8.625 \text{ in.}$$

Actual Thickness Away From Any Openings, Therefore Satisfaction Of Criterion 5-C-1

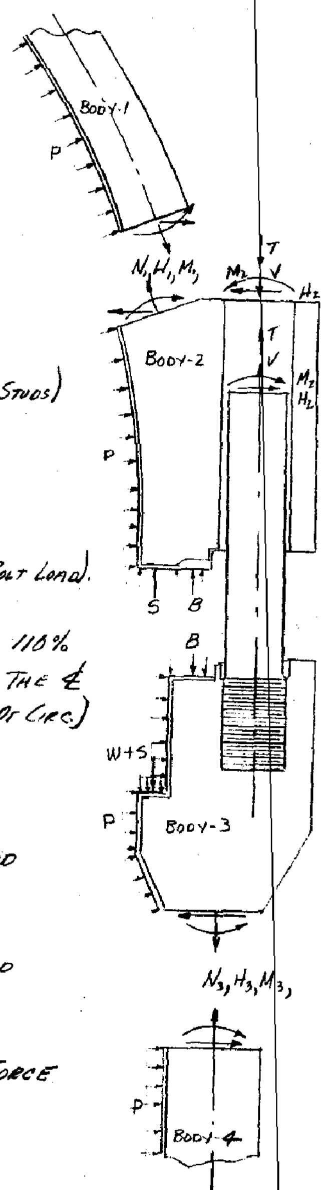
COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.
DESCRIPTION Structural Analysis Of The Closure Head And Vessel Assembly
NUMBER S-150-P | A-33
SHEET 13 OF 29
DATE 10-22-65 BY Cockrell
CHECK DATE 10-22-65 BY Alexander

5. Detailed Analysis:

e- Development Of Continuity Equations:

1- Analytical Model:

The Actual Structure Is Divided Into The Following Analytical Model To Facilitate The Analysis. The Assumed Directions Of The Redundants Are Illustrated.



Body-1 (Long Spherical Shell)
Body-2 (Ring)
Body-3 (Ring With A Beam Appentage - Studs)
Body-4 (Long Cylindrical Shell)

$H_1 \rightarrow H_3$, $M_1 \rightarrow M_3$, And V Are The Unknown Interaction Forces. (V Is The Change In Bolt Load).

T Is The Initial Bolt Load Based Upon 110% Of The Blow-Off Load Calculated To The ℄ Of The Outer O-Ring. (In-Kip Per In. Of Circ.)

P Is The Applied Pressure.

B Is The Bearing Ledge Reaction And Is A Function Of T, V, And P

W Is The Weight Of The Core And Core Support

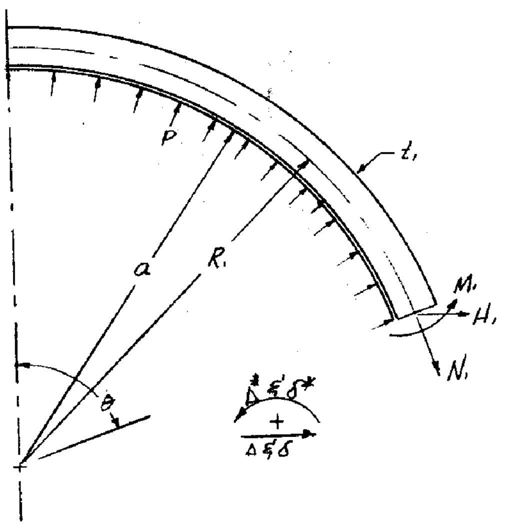
S Is The Core Holddown Spring Force

5. Detailed Analysis:

e. Development Of Continuity Equations:

2. Movements Due To Redundant And Applied Forces:

Body-1:



$R_1 = 89.500''$

$t_1 = 7.000''$

$a = 85.781''$

$\theta = 69°54'$

$N_1 = \frac{a^2 P}{2R_1} = 41.108P$

$\beta^4 = 3(1-\nu^2)\left(\frac{R_1}{t_1}\right)^2$

$\beta = 4.59624$

$K_1 = 1 - \left(\frac{1-2\nu}{2\beta}\right)\cot\theta = 0.98408$

$K_2 = 1 - \left(\frac{1+2\nu}{2\beta}\right)\cot\theta = 0.93631$

Displacements Due To Redundant Forces:

$$E\Delta_{11} = \frac{2\beta^2 \sin\theta}{t_1}\left[\frac{R\sin\theta}{2\beta}\left(\frac{1}{K_1} + K_2\right)H_1 + \frac{1}{K_1}M_1\right]$$

$$= 101.1878\, H_1 + 5.7599\, M_1$$

$$E\Delta^*_{11} = \frac{2\beta^2 \sin\theta}{t_1}\left[\frac{1}{K_1}H_1 + \frac{2\beta}{R_1 \sin\theta}\,\frac{1}{K_1}M_1\right]$$

$$= 5.7599\, H_1 + 0.6300\, M_1$$

Displacements Due To Applied Forces:

$$E\delta_{11} = \frac{(1-\nu)a^2 P \sin\theta}{2t} = 345.5091\, P$$

$$E\delta^*_{11} = 0$$

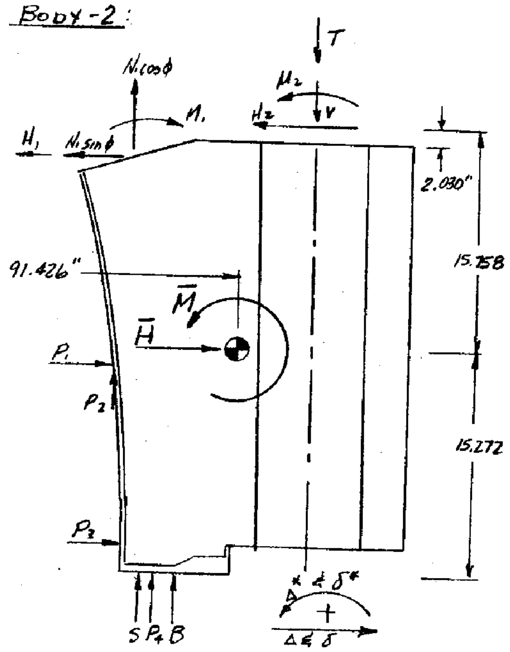
DESCRIPTION STRUCTURAL ANALYSIS OF THE CLOSURE HEAD AND VESSEL ASSEMBLY

5- DETAILED ANALYSIS:

e. DEVELOPMENT OF CONTINUITY EQUATIONS:

2- MOVEMENTS DUE TO REDUNDANT AND APPLIED LOADS:

BODY-2:



LOAD	MAGNITUDE	RADIUS	MOMENT ARM
$N_1 \sin\phi$	14.1276P	84.049	12.486
$N_2 \cos\phi$	38.6052P	84.049	7.377
H_1	UNKNOWN	84.049	12.486
M_1	UNKNOWN	84.049	—
T	116.532 KIP/IN	95.938	4.512
V	UNKNOWN	95.938	4.512
H_2	UNKNOWN	95.938	15.758
M_2	UNKNOWN	95.938	—
P_1	22.969P	81.889	0.412
P_2	3.322P	81.889	9.537
P_3	3.376P	83.500	13.584
P_4	6.671P	86.836	4.590
S	1.242 KIP/IN	84.578	6.848
B	1.0779(T+V) − 45.677P	89.002	2.424

$$I = 27499\ in^4 \qquad \frac{R^2}{I} = 0.30396$$

$$A = 441.967\ in^2 \qquad \frac{R^2}{A} = 18.9125$$

$$\bar{H} = -H_1 \frac{84.049}{91.426} - H_2 \frac{95.938}{91.426} - 14.1276P \frac{84.049}{91.426} + 22.969P \frac{81.889}{91.426} + 3.376P \frac{83.5}{91.426}$$

$$= \underline{-0.9193 H_1 - 1.0494 H_2 + 10.6687P}$$

$$\bar{M} = \left[(14.1276P)(12.486) - (38.6052P)(7.377) + 12.486 H_1 - M_1\right]\frac{84.049}{91.426} + \left[-4.512(T+V) + 15.758 H_2 + M_2\right]\frac{95.938}{91.426} + \left[(22.969P)(0.412) - (3.322P)(9.537)\right]\frac{81.889}{91.426} + (3.376P)(13.584)\frac{83.5}{91.426} - (6.671P)(4.590)\frac{86.836}{91.426} - 6.848S\frac{84.578}{91.426} - 2.424B\frac{89.002}{91.426}$$

$$= \underline{11.4785 H_1 - 0.9193 M_1 + 16.5357 H_2 + 1.0494 M_2 - 7.2782(T+V) - 6.33515 + 1.0401P}$$

5. Detailed Analysis:
C. Development Of Continuity Equations:
2. Movements Due To Redundant And Applied Loads:

Body-2:

Displacements Due To Redundant Forces:

$E\Delta_{21} = \frac{R^2}{A}\bar{H} - h_j \frac{R^2}{I}\bar{M} = -60.9499H_1 + 3.4890M_1 - 82.6039H_2 - 3.9830M_2 + 27.6228V$

$E\Delta^*_{21} = \frac{R^2}{I}\bar{M} = 3.4890H_1 - 0.2794M_1 + 5.0262H_2 + 0.3190M_2 - 2.2123V$

$E\Delta_{22} = \frac{R^2}{A}\bar{H} - h_j \frac{R^2}{I}\bar{M} = -72.3659H_1 + 4.4028M_1 - 99.0496H_2 - 5.0268M_2 + 34.8614V$

$E\Delta^*_{22} = \frac{R^2}{I}\bar{M} = 3.4890H_1 - 0.2794M_1 + 5.0262H_2 + 0.3190M_2 - 2.2123V$

$EV_{22} = 6.936\frac{R^2}{I}\bar{M} = 24.1997H_1 - 1.9379M_1 + 34.8617H_2 + 2.2126M_2 - 15.3445V$

Displacements Due To Applied Forces:

$E\delta_{21} = \frac{R^2}{A}\bar{H} - h_j \frac{R^2}{I}\bar{M} = 27.6228T + 24.0433S + 197.8250P$

$E\delta^*_{21} = \frac{R^2}{I}\bar{M} = -2.2123T - 1.9256S + 0.3161P$

$E\delta_{22} = \frac{R^2}{A}\bar{H} - h_j \frac{R^2}{I}\bar{M} = 34.8614T + 30.3439S + 196.7907P$

$E\delta^*_{22} = \frac{R^2}{I}\bar{M} = -2.2123T - 1.9256S + 0.3161P$

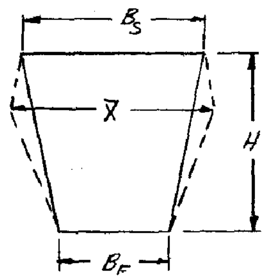
$Ev_{22} = 6.936\frac{R^2}{I}\bar{M} = 2.1925P$

5- DETAILED ANALYSIS:

e- DEVELOPMENT OF CONTINUITY EQUATIONS:

2- MOVEMENTS DUE TO REDUNDANT AND APPLIED FORCES:

CONSIDER SQUASAGE OF HEAD FLANGE:



B_S = WIDTH OF BEARING SURFACE UNDER STUDS = 10.5625"

B_F = WIDTH OF BEARING SURFACE BETWEEN FLANGES = 5.25"

H = DEPTH OF FLANGE = 28.5625"

THE EFFECTIVE LOAD PATH IS ASSUMED TO FOLLOW A 4 TO 1 SLOPE RELATIVE TO VERTICAL DISTANCE; SUCH THAT,

$$\bar{X} = \left(\frac{10.5625+5.25}{2}\right) + \left(\frac{28.5625}{2}\right)\left(\frac{1}{4}\right)(2) = 15.047"$$

$$\text{AVG. WEIGHTED THICKNESS} = \frac{1}{2}\left[\frac{10.563+15.047}{2} + \frac{5.25+15.047}{2}\right] = 11.476"$$

$$\text{AVG. BEARING RADIUS} = \frac{1}{2}(95.938 + 89.002) = 92.470"$$

$$\text{LOAD} = 2\pi\left[(89.002)(-1.0779V + 45.6777P) - 6.671P(86.836)\right] = -602.7804V + 21903.9P$$

$$\text{SQUASAGE} = \frac{H(\text{LOAD})}{\text{BEARING AREA}} = \frac{28.5625(-602.78V + 21903.9P)}{2\pi(92.470)(11.476)} = -2.5822V + 93.8307P$$

ADDING THE SQUASAGE TO THE VERTICAL DISPLACEMENT YIELDS:

$$EV_{22} = 24.1997H_1 - 1.9379M_1 + 34.8617H_2 + 2.2126M_2 - 17.9267V$$

$$EV_{22} = +96.0232P$$

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO. ________

DESCRIPTION: STRUCTURAL ANALYSIS OF THE CLOSURE HEAD AND VESSEL ASSEMBLY

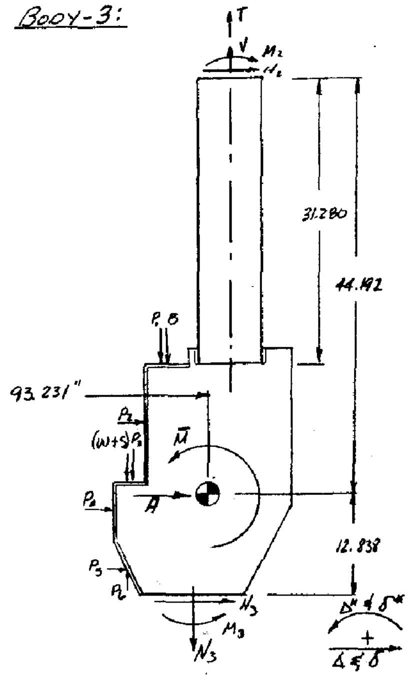
DATE 10-22-65 BY COCKRELL
CHECK DATE 10-22-65 BY ALEXANDER

5. DETAILED ANALYSIS:

e. DEVELOPMENT OF CONTINUITY EQUATIONS

2. MOVEMENTS DUE TO REDUNDANT AND APPLIED FORCES:

BODY-3:



LOAD	MAGNITUDE	RADIUS	MOMENT ARM
T	116.532 KIPS/IN	95.938	2.707
V	UNKNOWN	95.938	2.707
H_2	UNKNOWN	95.938	44.192
M_2	UNKNOWN	95.938	—
B	1.0779(T+V) −45.6777P	89.002	4.229
P_1	3.890P	88.226	5.005
P_2	13.000P	86.281	6.662
P_3	2.781P	84.891	8.340
W	1.270 KIP/IN	84.578	8.653
S	1.242 KIP/IN	84.578	8.653
P_4	7.187P	83.500	3.432
P_5	5.8125P	84.469	9.932
P_6	1.9370P	84.469	8.762
N_3	40.0934P	91.031	2.200
H_3	UNKNOWN	91.031	12.838
M_3	UNKNOWN	91.031	—

$I = 20945 \text{ in}^4 \quad A = 378.957 \text{ in}^2 \quad \frac{R^2}{I} = 0.41409 \quad \frac{R^2}{A} = 22.9367$

$$\bar{H} = \frac{95.938}{93.231} H_2 + \frac{91.031}{93.231} H_3 + (13.000P)\frac{86.281}{93.231} + (7.187P)\frac{83.500}{93.231} + (5.8125P)\frac{84.469}{93.231}$$

$$= \underline{1.0290 H_2 + 0.9764 H_3 + 23.7340 P}$$

$$\bar{M} = \left[2.707(T+V) - 44.192 H_2 - M_2\right]\frac{95.938}{93.231} + 4.229 B \frac{89.002}{93.231} + 3.890P(5.005)\frac{88.226}{93.231}$$
$$- 13.000P(6.662)\frac{86.281}{93.231} + 2.781P(8.340)\frac{84.891}{93.231} + (W+S)(8.653)\frac{84.578}{93.231} + 7.187P(3.432)\frac{83.500}{93.231}$$
$$+ \left[5.8125P(9.932) - 1.937P(8.762)\right]\frac{84.469}{93.231} + \left[12.838 H_3 + M_3 + 2.200 N_3\right]\frac{91.031}{93.231}$$

$$= \underline{-45.4751 H_2 - 1.0290 M_2 + 12.5351 H_3 + 0.9765 M_3 + 7.1372(T+V) + 7.8499(W+S) - 79.8729P}$$

5. Detailed Analysis:

c. Development Of Continuity Equations:

2. Movements Due To Redundant And Applied:

Body-4:

Displacements Due To Redundant Forces:

$$E\Delta_{32} = \frac{R^2}{A}\bar{H} - h_j\frac{R^2}{I}\bar{M} = 857.5801H_2 + 18.8700M_2 - 207.4870H_3 - 17.9066M_3 - 130.8923V$$

$$E\Delta^*_{32} = \frac{R^2}{I}\bar{M} = -18.8717H_2 - 0.4270M_2 + 5.2019H_3 + 0.4052M_3 + 2.9619V$$

$$E\Delta_{33} = \frac{R^2}{A}\bar{H} - h_j\frac{R^2}{I}\bar{M} = -218.6730H_2 - 5.4818M_2 + 89.1174H_3 + 5.2020M_3 + 38.0249V$$

$$E\Delta^*_{33} = \frac{R^2}{I}\bar{M} = -18.8717H_2 - 0.4270M_2 + 5.2019H_3 + 0.4052M_3 + 2.9619V$$

$$EV_{32} = 6.936\frac{R^2}{I}\bar{M} = -130.8941H_2 - 2.9617M_2 + 36.0804H_3 + 2.8105M_3 + 20.5437V$$

Displacements Due To Applied Forces:

$$E\delta_{32} = \frac{R^2}{A}\bar{H} - h_j\frac{R^2}{I}\bar{M} = -130.8923T - 143.9612(W+S) + 2009.1897P$$

$$E\delta^*_{32} = \frac{R^2}{I}\bar{M} = 2.9619T + 3.2576(W+S) - 33.1465P$$

$$E\delta_{33} = \frac{R^2}{A}\bar{H} - h_j\frac{R^2}{I}\bar{M} = 38.0249T + 41.8215(W+S) + 118.8449P$$

$$E\delta^*_{33} = \frac{R^2}{I}\bar{M} = 2.9619T + 3.2576(W+S) - 33.1465P$$

$$Ev_{32} = 6.936\frac{R^2}{I}\bar{M} = -229.9041P$$

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO. ______

NUMBER S-150-P | A-40
SHEET 20 OF 29
DATE 10-22-65 BY COCKRELL
CHECK DATE 10-22-65 BY ALEXANDER

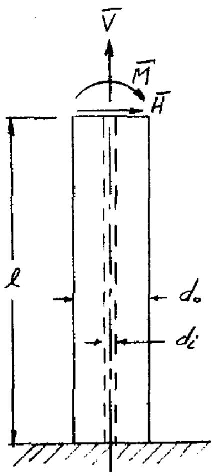
DESCRIPTION STRUCTURAL ANALYSIS OF THE CLOSURE HEAD AND VESSEL ASSEMBLY

5- DETAILED ANALYSIS:

e- DEVELOPMENT OF CONTINUITY EQUATIONS:

2- MOVEMENTS DUE TO REDUNDANT AND APPLIED FORCES:

NOTE THAT THE MOVEMENTS OF BODY-3 AT JOINT 2 (TOP OF STUDS) RESULTING FROM THE REDUNDANTS (H_2, M_2, & V) DO NOT AS YET INCLUDE THE BOLT DEFLECTION AND ROTATION. THERE ARE 54 STUDS ON A 95.938 in. RADIUS, OR A STUD EVERY 11.1629 in. SINCE H_2, M_2, & V ON BODY-2 AND 3 ARE SET UP IN TERMS OF KIPS PER INCH AND IN-KIP PER INCH OF CIRCUMFERENCE, THE ACTUAL LOADS ON THE STUDS WILL BE $11.1629 H_2$, $11.1629 M_2$, AND $11.1629 V$.



$\bar{H} = 11.1629\, H_2$ $\qquad I = \frac{\pi}{64}(d_o^4 - d_i^4) = 104.692\ in^4$

$\bar{M} = 11.1629\, M_2$ $\qquad A = \frac{\pi}{4}(d_o^2 - d_i^2) = 35.291\ in^4$

$\bar{V} = 11.1629\, V$ $\qquad \ell = 31.280\ in$

THE STUD MOVEMENTS ARE:

$$E\Delta = \left(\frac{\bar{H}\ell^3}{3I} + \frac{\bar{M}\ell^2}{2I}\right)\frac{E_{312}}{E_{310}} = 1061.4292\, H_2 + 50.8997\, M_2$$

$$E\Delta^* = \left(-\frac{\bar{H}\ell^2}{2I} - \frac{\bar{M}\ell}{I}\right)\frac{E_{312}}{E_{310}} = -50.8997\, H_2 - 3.2545\, M_2$$

$$EV = \left(\frac{\bar{V}\ell}{A}\right)\frac{E_{302}}{E_{320}} = 9.6545\, V$$

COMBINING THE STUD MOVEMENTS WITH THE FLANGE MOVEMENTS (SHEET 19) YIELDS THE FOLLOWING EQUATIONS FOR THE TOTAL MOVEMENTS.

$$E\Delta_{32} = 1919.0093\, H_2 + 69.7697\, M_2 - 207.4870\, H_3 - 17.9066\, M_3 - 130.8923\, V$$

$$E\Delta^*_{32} = -69.7697\, H_2 - 3.6815\, M_2 + 5.2019\, H_3 + 0.4052\, M_3 + 2.9619\, V$$

$$EV_{32} = -130.8941\, H_2 - 2.9617\, M_2 + 36.0804\, H_3 + 2.8105\, M_3 + 30.1982\, V$$

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT. CHATTANOOGA. TENN.
CHARGE NO.
DESCRIPTION Structural Analysis Of The Closure Head And Vessel Assembly

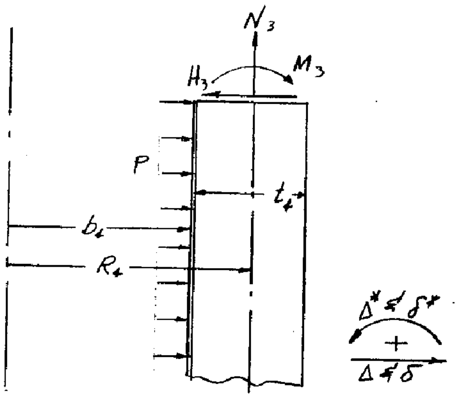
NUMBER 5-150-P | A-41
SHEET 21 OF 29
DATE 10-22-65 BY Cockrell
CHECK DATE 10-22-65 BY Alexander

5- Detailed Analysis:

e- Development Of Continuity Equations:

2. Movements Due To Redundant And Applied Loads:

Body-4:



$R_4 = 91.031''$
$b_4 = 85.437''$
$t_4 = 10.75''$

$\beta^4 = \frac{3(1-\nu^2)}{R^2 t^2}$

$\beta = 0.04108$

$D = \frac{Et^3}{12(1-\nu^2)} = 113.76345E$

Displacements Due To Redundant Forces:

$$E\Delta_{43} = \frac{E}{2\beta^3 D}\left[-\frac{1}{\beta}H_3 + M_3\right]$$

$$= -63.2918\,H_3 + 2.6006\,M_3$$

$$E\Delta^*_{43} = -\frac{E}{2\beta^2 D}\left[-H_3 + 2\beta M_3\right]$$

$$= 2.6006\,H_3 - 0.2137\,M_3$$

Displacements Due To Applied Forces:

$$E\delta_{43} = \frac{b_4^2}{t_4}\left(\frac{R_4}{b_4} - \frac{\nu}{2}\right)P = \frac{(85.437)^2}{10.75}\left(\frac{91.031}{85.437} - 0.15\right)P$$

$$= 621.6238\,P$$

$$E\delta^*_{43} = 0$$

5. DETAILED ANALYSIS:

e. DEVELOPMENT OF CONTINUITY EQUATIONS:

3. CONTINUITY MATRIX AND LOADING VECTORS:

TWO DIFFERENT SOLUTIONS WILL BE REQUIRED FOR THE LOADINGS CONSIDERED IN THIS ANALYSIS; THEY ARE,

SOLUTION 1: BOLT-UP ONLY PLUS CORE SUPPORT LOADING PLUS CORE HOLDDOWN SPRING FORCE. THESE LOADINGS WILL BE SOLVED BY REQUIRING CONTINUITY OF RADIAL DISPLACEMENTS AND ROTATIONS (VERTICAL CONTINUITY IS NOT A CONSIDERATION FOR THE BOLT-UP CONDITION SINCE T, W, AND S ARE KNOWN LOADS). THE SOLUTION FOR THIS CONDITION WILL RESULT IN A 6×6 MATRIX WITH A COLUMN VECTOR FOR T, W, AND S.

SOLUTION 2: FOR INTERNAL PRESSURE ONLY. THIS SOLUTION WILL RESULT IN THE SAME NUMERICAL MATRIX AS FOR THE FIRST 6 COLUMNS AND ROWS AS SOLUTION-1 BUT ONE COLUMN AND ONE ROW WILL BE ADDED TO REFLECT THE VERTICAL CONTINUITY REQUIREMENT AT CUT-2. THE COLUMN VECTORS ASSOCIATED WITH THIS 7×7 MATRIX WILL BE IN TERMS OF PRESSURE (P).

THE TWO MATRICES WILL BE ARRANGED AS FOLLOWS:

$$E\Delta_{11} - E\Delta_{21} = E\delta_{21} - E\delta_{11}$$
$$E\Delta^*_{11} - E\Delta^*_{21} = E\delta^*_{21} - E\delta^*_{11}$$
$$E\Delta_{22} - E\Delta_{32} = E\delta_{32} - E\delta_{22}$$
$$E\Delta^*_{22} - E\Delta_{32} = E\delta^*_{32} - E\delta^*_{22}$$
$$E\Delta_{33} - E\Delta_{43} = E\delta_{43} - E\delta_{33}$$
$$E\Delta^*_{33} - E\Delta_{43} = E\delta^*_{43} - E\delta^*_{33}$$
$$EV_{22} - EV_{32} = Ev_{32} - Ev_{22}$$

(First six equations:) 6×6 MATRIX FOR SOLUTION-1 WITH TERMS FOR T, W, & S

(All seven equations:) 7×7 MATRIX FOR SOLUTION-2 WITH TERMS FOR PRESSURE

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO.________

DESCRIPTION STRUCTURAL ANALYSIS OF THE CLOSURE HEAD AND VESSEL ASSEMBLY

NUMBER 5-150-P | A-43
SHEET 23 OF 29
DATE 10-22-65 BY COCKRELL
CHECK DATE 10-22-65 BY ALEXANDER

5- DETAILED ANALYSIS:

e- DEVELOPMENT OF CONTINUITY EQUATIONS:

3- CONTINUITY MATRIX AND LOADING VECTORS:

SUBSTITUTING THE DEFLECTIONS AND ROTATIONS INTO THE ABOVE COMPATIBILITY EQUATIONS AND WRITING IN MATRIX FORM YIELDS:

$$
\begin{bmatrix}
162.1377 & 2.2709 & 82.6039 & 3.9830 & 0 & 0 & -27.6228 \\
2.2709 & 0.9094 & -5.0262 & -0.3190 & 0 & 0 & 2.2123 \\
-72.3659 & 4.4028 & -2018.0589 & -74.7959 & 207.4870 & 17.9066 & 165.7537 \\
3.4890 & -0.2744 & 74.7959 & 4.0005 & -5.2019 & -0.4052 & -5.1742 \\
0 & 0 & -218.6730 & -5.4818 & 152.4692 & 2.6013 & 38.0249 \\
0 & 0 & -18.8717 & -0.4270 & 2.6013 & 0.6189 & 2.9619 \\
24.1997 & -1.9379 & 165.7558 & 5.1743 & -36.0804 & -2.8105 & -48.1249
\end{bmatrix}
\begin{Bmatrix} H_1 \\ M_1 \\ H_2 \\ M_2 \\ H_3 \\ M_3 \\ V \end{Bmatrix}
=
\begin{bmatrix} 27.6228 \\ -2.2123 \\ -165.7537 \\ 5.1742 \\ -38.0249 \\ -2.9619 \end{bmatrix} T +
\begin{bmatrix} 0 \\ 0 \\ -143.9612 \\ 3.2576 \\ -41.8215 \\ -3.2576 \end{bmatrix} W +
\begin{bmatrix} 24.0433 \\ -1.9256 \\ -174.3051 \\ 5.1832 \\ -41.8215 \\ -3.2576 \end{bmatrix} S +
\begin{bmatrix} -147.6841 \\ 0.3161 \\ 1812.3990 \\ -33.4426 \\ 493.7789 \\ 33.1465 \\ -325.9273 \end{bmatrix} P
$$

5. DETAILED ANALYSIS:

e. DEVELOPMENT OF CONTINUITY EQUATIONS:

4. REDUNDANT LOAD VALUES:

THE ABOVE MATRICES YIELD THE FOLLOWING VALUES FOR THE REDUNDANT SHEARS AND MOMENTS.

REDUNDANT	BOLT UP ONLY $T = 116.532$ KIPS/IN	CORE SUPPORT WEIGHT ONLY $W = 1.270$ KIP/IN	CORE HOLDDOWN SPRING ONLY $S = 1.242$ KIP/IN	DESIGN PRESSURE ONLY $P = 2.5$ KSI
H_1	0.19919T = 23.2120	-0.00885W = -0.01124	0.17153S = 0.21304	-0.04139P = -0.1035
M_1	-2.82716T = -329.4546	0.06068W = 0.07706	-2.44812S = -3.04057	-8.38228P = -20.9557
H_2	0.02907T = 3.3876	0.04329W = 0.05498	0.03434S = 0.04265	-1.09028P = -2.7257
M_2	-0.16448T = -19.1672	-0.57218W = -0.72667	-0.26258S = -0.32612	19.09019P = 47.7255
H_3	-0.15637T = -18.2221	-0.17102W = -0.21720	-0.17178S = -0.21335	1.22229P = 3.0557
M_3	-3.35564T = -391.0394	-3.61932W = -4.59654	-3.67571S = -4.56523	9.64912P = 24.1228
V	~	~	~	3.90667P = 9.7667

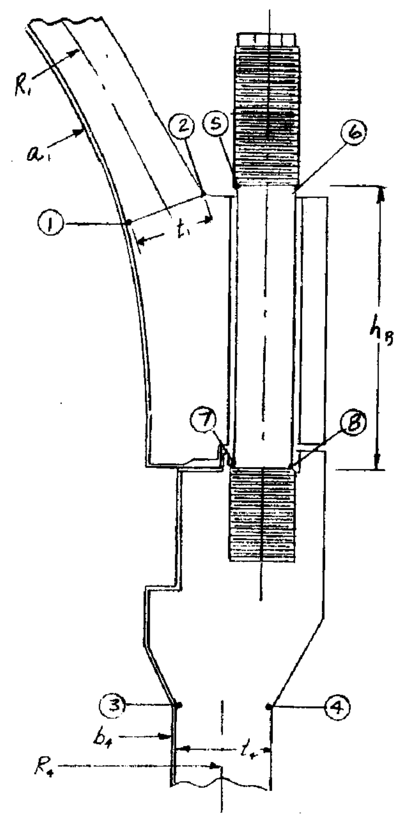
COMBINING THE VALUES OF THE REDUNDANT FORCES FOR THE BOLT-UP CONDITION PLUS CORE SUPPORT WEIGHT PLUS THE CORE HOLDDOWN SPRING FORCE, WE GET;

REDUNDANT	BOLT-UP PLUS CORE SUPPORT WEIGHT PLUS CORE HOLDDOWN SPRING
H_1	23.4238
M_1	-332.4181
H_2	3.4852
M_2	-20.2200
H_3	-18.6527
M_3	-400.2012
V	~

5. DETAILED ANALYSIS:

f. STRESSES:

STRESSES WILL BE CALCULATED AT THE LOCATIONS AS SHOWN BELOW.



POINTS 1 & 2:

$t_1 = 9.405$ IN

$$\sigma_x = \pm\frac{6M_1}{t_1^2} + \frac{H_1\cos\theta}{t_1} + \frac{a_1^2 P}{2R t_1}$$
$$= \pm 0.06783 M_1 + 0.03654 H_1 + 4.3709 P$$

$$\sigma_\theta = \pm\frac{\nu 6M_1}{t_1^2} + \frac{\nu H_1\cos\theta}{t_1} + \frac{E\Delta_{11}}{R_1\sin\theta} \pm \frac{E\Delta_{11}^{*}\,t_1\cos\theta}{2R_1\sin\theta} + \frac{a_1^2 P}{2R_1 t_1}$$
$$= \pm 0.02036 M_1 + 0.01096 H_1 + 0.01189 E\Delta_{11} \pm 0.01922 E\Delta_{11}^{*} + 4.30709 P$$

POINTS 3 & 4:

$$\sigma_x = \pm\frac{6M_3}{t_4^2} + \frac{b_4^2 P}{2R_4 t_4}$$
$$= \pm 0.05191 M_3 + 3.72961 P$$

$$\sigma_\theta = \pm\frac{\nu 6M_3}{t_4^2} + \frac{E\Delta_{43}}{R_4} + \frac{b_4 P}{t_4}$$
$$= \pm 0.01557 M_3 + 0.01098 E\Delta_{43} + 7.94762 P$$

POINTS 5 & 6:

$$\sigma_x = \frac{\bar{T}+\bar{V}}{A} \pm \frac{\bar{M}c}{I} = 0.31631(T+V) \pm 0.36236 M_2$$

POINTS 7 & 8:

$$\sigma_x = \frac{\bar{T}+\bar{V}}{A} \pm \frac{(\bar{M}+\bar{H}\ell_B)c}{I} = 0.31631(T+V) \pm 0.36236 M_2 \pm 11.33485 H_2$$

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.

NUMBER 5-150-P | A-46
SHEET 26 OF 29
DATE 10-22-65 BY COCKRELL
CHECK DATE 10-22-65 BY ALEXANDER

DESCRIPTION: STRUCTURAL ANALYSIS OF THE CLOSURE HEAD AND VESSEL ASSEMBLY

5- DETAILED ANALYSIS:

f- STRESSES:

BOLT-UP STRESSES ($T = 116.532$ KIP/IN ; $W = 1.270$ KIP/IN ; $S = 1.242$ KIP/IN)

LOCATION	$\frac{6M}{t^2}$	$\frac{H}{t}$	$\frac{PR}{2t}$	σ_x	$\frac{\nu 6M}{t^2}$	$\frac{\nu H}{t}$	EW	$\frac{EW}{R}$	EW*	$\frac{tEW^*}{2R}$	$\frac{PR}{2t}$ / $\frac{PR}{t}$	σ_θ	σ_r	Stress Intensity $\sigma_x-\sigma_\theta$	Stress Intensity $\sigma_x-\sigma_r$	Stress Intensity $\sigma_\theta-\sigma_r$
1	-22.55	0.86	—	-21.69	-6.77	0.26	455.5	5.42	-74.5	-1.43	—	-2.52	0	-19.17	-21.69	-2.52
2	22.55	0.86	—	23.41	6.77	0.26	455.5	5.42	-74.5	1.43	—	13.88	0	9.53	23.41	13.88
3	-20.77	—	—	-20.77	-6.23	—	139.8	1.54	—	—	—	-4.69	0	-16.08	-20.77	-4.69
4	20.77	—	—	20.77	6.23	—	139.8	1.54	—	—	—	7.77	0	13.00	20.77	7.77
PRESSURE STRESSES ($P = 2.5$ KSI)																
1	-1.42	0	10.93	9.51	-0.43	0	-131.2	-1.56	-13.8	-0.27	10.93	8.67	-2.5	0.84	12.01	11.17
2	1.42	0	10.93	12.35	0.43	0	-131.2	-1.56	-13.8	0.27	10.93	10.07	0	2.28	12.35	10.07
3	1.25	—	9.32	10.57	0.38	—	-130.7	-1.43	—	—	19.87	18.82	-2.5	-8.25	13.07	21.32
4	-1.25	—	9.32	8.07	-0.38	—	-130.7	-1.43	—	—	19.87	18.06	0	-9.99	8.07	18.06
BOLT-UP PLUS PRESSURE STRESSES																
1	-23.97	0.86	10.93	-12.18	-7.20	0.26	324.3	3.86	-88.3	-1.70	10.93	6.15	-2.5	-18.33	-9.68	8.65
2	23.97	0.86	10.93	35.76	7.20	0.26	324.3	3.86	-88.3	1.70	10.93	23.95	0	11.81	35.76	23.95
3	-19.52	—	9.32	-10.[illegible]	-5.85	—	9.1	0.11	—	—	19.87	14.13	-2.5	-24.33	-7.70	16.63
4	19.52	—	9.32	28.84	5.85	—	9.1	0.11	—	—	19.87	25.83	0	3.01	28.84	25.83

5- DETAILED ANALYSIS:

f. STRESSES:

FOR BOLT-UP:

CRITERION 5-C-3:

$S.I._{MAX} = \sigma_x - \sigma_r = \underline{23.4\ KSI} < 1.5S_m = 40\ KSI$ @ 70°F

FOR LOCATION 2 SEE SHEET-25

BOLT-UP PLUS PRESSURE:

CRITERION 5-C-1:

$S.I._{MAX} = \frac{PR}{t} - \left(\frac{-P}{2}\right) = 19.87 + 1.25 = \underline{21.1\ KSI} < S_m = 26.7\ KSI$ @ 650°F

FOR LOCATIONS 3 & 4

CRITERION 5-C-2:

$S.I._{MAX} = \frac{PR}{t} + \frac{E\Delta}{R} + \frac{P}{2} = \underline{21.2\ KSI} < 1.5S_m = 40\ KSI$ @ 650°F

FOR LOCATIONS 3 & 4

CRITERION 5-C-3:

$S.I._{MAX} = (\sigma_x - \sigma_r) = \underline{35.8\ KSI} < 1.5S_m = 40\ KSI$ @ 650°F

FOR LOCATION 2

5. DETAILED ANALYSIS:

f. STRESSES:

CONSIDER STRESS IN STUDS:

LOCATION	BOLT-UP STRESSES $T = 116.532$ KIP/IN $W = 1.270$ KIP/IN $S = 1.242$ KIP/IN			PRESSURE STRESSES $P = 2.25$ KSI OPERATING PRESSURE			COMBINED BOLT-UP PLUS PRESSURE STRESSES		
	DIRECT STRESS	BENDING STRESS	DIRECT + BENDING STRESS	DIRECT STRESS	BENDING STRESS	DIRECT + BENDING STRESS	DIRECT STRESS	BENDING STRESS	DIRECT + BENDING STRESS
5	36.83	-7.32	29.51	2.78	15.56	18.34	39.61	8.24	47.85
6	↓	7.32	44.15	↓	-15.56	-12.78	↓	-8.24	31.37
7	↓	32.18	69.01	↓	-12.24	-9.46	↓	19.94	59.55
8	↓	-32.18	4.65	↓	12.24	15.02	↓	-19.94	19.67

FOR BOLT-UP:

CRITERION 5-C-5:

$\sigma_x = \underline{36.83}$ KSI $< 2S_m = 86.6$ KSI @ 70°F

CRITERION 5-C-6:

$\sigma_x = 36.83 + 32.18 = \underline{69.01 \text{ KSI}} < 3S_m = 129.9$ KSI @ 70°F

FOR BOLT-UP PLUS OPERATING PRESSURE:

CRITERION 5-C-5:

$\sigma_x = \underline{39.61 \text{ KSI}} < 2S_m = 73.5$ KSI @ 550°F

CRITERION 5-C-6:

$\sigma_x = 39.61 + 19.94 = \underline{59.55 \text{ KSI}} < 3S_m = 110.3$ KSI @ 550°F

5. DETAILED ANALYSIS:
f. STRESSES:

CONSIDER THE SHEARING STRESS IN THE STUD, NUT, AND VESSEL FLANGE THREADS:

THE STUD OR EXTERNAL THREAD IS 7"-8N-2A. THE MATERIAL IS SA-320-L43 WITH AN ALLOWABLE SHEAR STRESS OF $0.6 S_m = 20.88$ KSI (CRITERION 5-C-7)

THE NUT OR FLANGE INTERNAL THREAD IS 7"-8N-2B. THE ALLOWABLE STRESS FOR THE FLANGE MATERIAL IS $0.6 S_m = 16.02$ KSI

THE EQUATIONS BELOW FROM REFERENCE 7 GIVE THE SHEAR AREAS:

$$A_e = \pi N L_e K_{N_{max}}\left[\frac{1}{2N} + 0.57735(E_{S_{min}} - K_{N_{max}})\right]$$

$$A_i = \pi N L_e D_{S_{min}}\left[\frac{1}{2N} + 0.57735(D_{S_{min}} - E_{N_{max}})\right]$$

WHERE: N IS THE NUMBER OF THREADS PER INCH = 8
L_e IS THE LENGTH OF ENGAGEMENT = 9.25" FOR FLANGE; 7.5 FOR NUT

$K_{N_{max}}$ = THE MAXIMUM MINOR DIAMETER OF THE INTERNAL THREADS = 6.8897"
$E_{S_{min}}$ = THE MINIUM PITCH DIAMETER OF THE EXTERNAL THREADS = 6.9050"

$D_{S_{min}}$ = THE MINIUM MAJOR DIAMETER OF THE EXTERNAL THREADS = 6.9818"
$E_{N_{max}}$ = THE MAXIMUM PITCH DIAMETER OF THE INTERNAL THREADS = 6.9326"

$A_e = 12.351 L_e$ = 92.6 in² (TOP OF STUD); 114.2 in² (BOTTOM OF STUD)

$A_i = 15.950 L_e$ = 119.6 in² (FOR NUT); 147.5 in² (FOR FLANGE)

THE SHEAR STRESS FOR DESIGN PRESSURE PLUS BOLT-UP IS:

THE TOP OF STUD:

$$S_s = \frac{\bar{T}+\bar{V}}{A_e} = \frac{2\pi R_B(T+V)}{54 A_e} = \frac{1410}{92.6}$$
$= \underline{15.2 \text{ KSI}} < 20.88$ KSI

THE BOTTOM OF STUD:

$$S_s = \frac{\bar{T}+\bar{V}}{A_e} = \frac{2\pi R_B(T+V)}{54 A_e} = \frac{1410}{114.2}$$
$= \underline{12.3 \text{ KSI}} < 20.88$ KSI

FOR NUT:

$$S_s = \frac{\bar{T}+\bar{V}}{A_e} = \frac{2\pi R_B(T+V)}{54 A_e} = \frac{1410}{119.6}$$
$= \underline{11.8 \text{ KSI}} < 20.88$ KSI

FOR VESSEL FLANGE:

$$S_s = \frac{\bar{T}+\bar{V}}{A_e} = \frac{2\pi R_B(T+V)}{54 A_c} = \frac{1410}{147.5}$$
$= \underline{9.6 \text{ KSI}} < 16.02$ KSI

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER S-150-P APP A | A-50
SHEET 1 OF 1
CHARGE NO.
DATE 1-10-66 BY COCKRELL
DESCRIPTION ALLOWABLE DESIGN STRESS VALUE S_m FOR STAINLESS STEEL CLAD DEPOSIT
CHECK DATE BY

APPENDIX A:

THE FOLLOWING VALUES OF S_m FOR THE STAINLESS STEEL CLAD DEPOSIT WERE DETERMINED BY C.E. WELDING SPEC. MA-38G(1) AND SAA-38(C). THE RESULTS OF THE TEST WERE REPORTED IN APPENDIX-A OF REPORT NUMBER S-150-P ON CONTRACT NO. 15564 (ZORITA REACTOR VESSEL).

FOR MA-38G(1):

S_m = 27.5 ksi @ 70°F
S_m = 16.4 ksi @ 650°F

FOR SAA-38(C):

S_m = 26.8 ksi @ 70°F
S_m = 18.7 ksi @ 650°F ← USE THESE VALUES

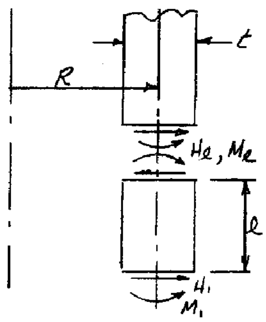
PARAGRAPH N-417-1 OF THE ASME NUCLEAR CODE SECTION III STATES THAT THE ALLOWABLE BEARING STRESS SHALL BE 1.5 S_m AT THE TEMPERATURE OF INTEREST. THE BEARING STRESS AT ROOM TEMPERATURE (70°F) UNDER BOLT-UP CONDITIONS IS THE GREATEST SINCE THE BEARING FORCE RELIEVES WHEN PRESSURE IS APPLIED. THE ALLOWABLE BEARING STRESS AT 70°F BECOMES,

$1.5 S_m = 1.5(26.8) = \underline{40.2 \text{ ksi}}$

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.
DESCRIPTION: APPENDIX TO S-150-P
NUMBER: S-150-P APP. B | A-51
SHEET 1 OF 3
DATE 4-26-68 BY COCKRELL
CHECK DATE 4-26-68 BY HEILKER

APPENDIX B:

IN THIS APPENDIX, THE SHEAR AND MOMENT AT CUT-1 (H_1 AND M_1 AS SHOWN ON SHEET 13 OF S-150-P) WILL BE ATTENUATED TO THE TOP END OF THE HUB. THE HUB IS APPROXIMATELY 11" LONG. THE HUB WILL BE TREATED AS A CYLINDRICAL SHELL OF THE SAME MID RADIUS AS THE CLOSURE HEAD. THIS IS SUFFICIENTLY ACCURATE SINCE THERE IS VERY LITTLE CURVATURE IN 11". THE ASSUMED CYLINDER WILL HAVE A THICKNESS WHICH IS THE AVERAGE THICKNESS OF THE HUB.



$R = 90.101''$
$t = 8.202''$
$\ell = 11''$
$\beta = \sqrt[4]{\frac{3(1-\nu^2)}{R^2 t^2}} = 0.04726$

FOR $\beta\ell = 0.52$ FROM REF. 3

$\phi = 0.8110$
$\psi = 0.2218$
$\theta = 0.5164$
$\xi = 0.2946$

$H_1 = 0.19919\,T - 0.00885\,W + 0.17153\,S - 0.04139\,P$
$M_1 = -2.92716\,T + 0.06068\,W - 2.44812\,S - 8.38228\,P$

FROM REF. 3

$$H_\ell = H_1\psi - 2\beta M_1\xi = 0.12306\,T - 0.00366\,W + 0.10635\,S + 0.22469\,P$$

$$M_\ell = \frac{1}{\beta}H_1\xi + M_1\phi = -1.05167\,T - 0.00593\,W - 0.91662\,S - 7.05593\,P$$

FOR $T = 116.532$ KIP/IN, $W = 1.270$ KIP/IN, $S = 1.242$ KIP/IN & $P = 2.5$ KSI

$H_\ell = 14.46787$ KIP/IN
$M_\ell = -123.69918$ IN KIP/IN
} FOR BOLT-UP ONLY

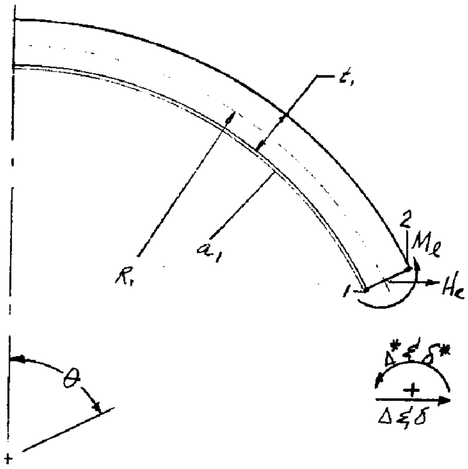
$H_\ell = 15.02959$ KIP/IN
$M_\ell = -141.3390$ IN-KIP/IN
} FOR BOLT-UP PLUS DESIGN PRESSURE

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER S-150-P App. B | A52
SHEET 2 OF 3
CHARGE NO. ____
DATE 4-26-68 BY COCKRELL
DESCRIPTION APPENDIX TO S-150-P
CHECK DATE 4-26-68 BY HEILKER

APPENDIX B:

TO DETERMINE DEFLECTIONS AND STRESSES AT THE TOP OF THE HUB, H_ℓ AND M_ℓ WILL BE PLACED ON A SPHERICAL SECTION AS SHOWN BELOW. THE ANGULAR LOCATION OF THE TOP OF THE HUB IS APPROXIMATELY $63\frac{1}{2}°$ OFF THE VERTICAL CENTER LINE.



$R_1 = 89.5''$
$t_1 = 7''$
$a_1 = 85.781$
$\theta = 63\frac{1}{2}°$

$\beta^4 = 3(1-\nu^2)\left(\frac{R}{t_1}\right)^2$
$\beta = 4.59624$
$K_1 = 1-\left(\frac{1-2\nu}{2\beta}\right)\cot\theta$
$= .97831$
$K_2 = 1-\left(\frac{1+2\nu}{2\beta}\right)\cot\theta$
$= 0.91322$

DISPLACEMENTS:

$$E\Delta_\ell = \frac{2\beta^2 \sin\theta}{t_1}\left[\frac{R\sin\theta}{2\beta}\left(\frac{1}{K_1}+K_2\right)H_\ell + \frac{1}{K_1}M_\ell\right] = \underline{91.0907\,H_\ell + 5.5214\,M_\ell}$$

$$E\Delta_\ell^* = \frac{2\beta^2 \sin\theta}{t_1}\left[\frac{1}{K_1}H_\ell + \frac{2\beta}{R\sin\theta}\frac{1}{K_1}M_\ell\right] = \underline{5.5214\,H_\ell + 0.63367\,M_\ell}$$

STRESSES

$$\sigma_x = \pm\frac{6M_\ell}{t_1^2} + \frac{H_\ell\cos\theta}{t_1} + \frac{a_1^2 P}{2Rt_1} = \underline{\pm 0.12244\,M_\ell + 0.06374\,H_\ell + 5.87260P}$$

$$\sigma_\theta = \pm\frac{\nu 6M_\ell}{t_1^2} + \frac{\nu H_\ell\cos\theta}{t_1} + \frac{E\Delta_\ell}{R\sin\theta} \pm \frac{E\Delta^* t\cos\theta}{2R\sin\theta} + \frac{a_1 P}{2R_1 t_1}$$

$$= \underline{\pm 0.03673\,M_\ell + 0.01912\,H_\ell + 0.01248\,E\Delta_\ell \pm 0.01950\,E\Delta_\ell^* + 5.87260P}$$

$\sigma_r = -P, 0$

APPENDIX B:

THE FOLLOWING TABLE GIVES THE STRESSES AND STRESS INTENSITIES FOR THE INSIDE AND OUTSIDE SURFACES OF THE CLOSURE HEAD AT THE TOP OF THE HUB. STRESSES WERE CALCULATED FOR THE BOLT-UP CONDITION AND FOR BOLT-UP PLUS DESIGN PRESSURE.

BOLT-UP ONLY

LOCATION	$\frac{6M}{t^2}$	$\frac{H}{t}$	$\frac{PR}{2t}$	σ_x	$\frac{\nu 6M}{t^2}$	$\frac{\nu H}{t}$	$\frac{E\Delta}{R}$	$\frac{tE\delta'}{2R}$	$\frac{PR}{2t}$	σ_θ	σ_r	STRESS INTENSITY $\sigma_x - \sigma_\theta$	$\sigma_x - \sigma_r$	$\sigma_\theta - \sigma_r$
IN	-15.4	0.9	0	-14.5	-4.6	0.3	7.9	.03	0	3.7	0	-18.2	-14.5	3.7
OUT	+15.4	0.9	0	16.3	4.6	0.3	7.9	-.03	0	12.8	0	3.5	16.3	12.8

BOLT-UP PLUS DESIGN PRESSURE

LOCATION	$\frac{6M}{t^2}$	$\frac{H}{t}$	$\frac{PR}{2t}$	σ_x	$\frac{\nu 6M}{t^2}$	$\frac{\nu H}{t}$	$\frac{E\Delta}{R}$	$\frac{tE\delta'}{2R}$	$\frac{PR}{2t}$	σ_θ	σ_r	$\sigma_x - \sigma_\theta$	$\sigma_x - \sigma_r$	$\sigma_\theta - \sigma_r$
IN	-17.6	1.0	14.7	-1.9	-5.3	0.3	7.3	-.13	14.7	16.9	-2.5	-18.8	0.6	19.4
OUT	17.6	1.0	14.7	33.3	+5.3	0.3	7.3	.13	14.7	27.7	0	5.6	33.3	27.7

FROM THE ABOVE TABLE WE SEE THAT THE HIGHEST STRESS INTENSITY FOR THE BOLT-UP CONDITION IS 18.2 KSI ON THE INSIDE SURFACE. FOR THE BOLT-UP PLUS DESIGN PRESSURE CONDITION THE HIGHEST STRESS INTENSITY WAS 33.3 KSI FOR THE OUTSIDE SURFACE. NOTE THAT BOTH OF THESE STRESS INTENSITIES ARE LESS THAN THE STRESS INTENSITIE FOR THE SAME CONDITIONS AT THE BOTTOM END OF THE HUB (SEE SHEET 26 OF S-150-P).

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.

DESCRIPTION Appendix to S-150-P

NUMBER S-150-P App. C | A-54
SHEET 1 OF 2
DATE 4-25-68 BY Cockrell
CHECK DATE ____ BY ____

CLOSURE HEAD PENETRATION SPACING:

The general requirements for spacing of openings in spherical shells (such as a closure head) are given in Paragraph N-451 of the ASME Code Section III. Additional requirements are given in Article I-6 of Section III as referenced in N-451(e).

The spacing of the closure head penetrations meet the requirements of Paragraph N-451 with the exception of I-613(b) of Article I-6. Paragraph I-613(b) states that "The arc-distance measured between the center lines of adjacent nozzles along the inside surface of the shell is not less than three times the sum of their inside radii for openings in a head....".

This requirement would restrict the ligament efficiency to a minimum of 66.7%. The actual ligament efficiency of the closure head is 66.6%. Since the actual ligament efficiency deviates slightly from the required minimum ligament efficiency, Paragraph I-620 of Article I-6 must be met. Paragraph I-622 of Article I-6 states that "In accordance with I-1012, re-evaluation is not required for configurations for which there are available detailed experimental results that are consistent with the requirements of Article I-10."

The purpose here is to use the results of Westinghouse Research Report 100FF996-R4 to show that the closure head penetration spacing is satisfactory for the closure head. The ligament efficienty of the penetration spacing in this experiment was 38% which should more than suffice to show that the closure head's ligament efficiency of 66.6% is adequate.

The results from the above report show that the membrane portion of the stress due to internal pressure can be accurately calculated by dividing the primary membrane stress in the unperforated region by the ligament efficiency. See the discussion and results given on pages 23 and 24 of the above report. It is shown there

CLOSURE HEAD PENETRATION SPACING:

that the calculated value of the average primary membrane stress deviates from the experimental value by only 2%. The higher value being the calculated value; hence, the method of dividing the required thickness in the unperforated region by the ligament efficiency is valid and reasonable. This method was used to obtain the thickness of the closure head.

The stress indices to be applied to the computed membrane stress intensities for a fatigue analysis as determined from the results of the experiment are 1.05 and 1.44 for the inside and outside surfaces, respectively. These values are well below the required index of 2 for both the inside and outside surfaces from Article I-6 of Section III. It is, therefore, concluded that the slight deviation from the spacing requirement in I-613(b) is insignificant both from the standpoint of establishing the required head thickness and for fatigue considerations.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.

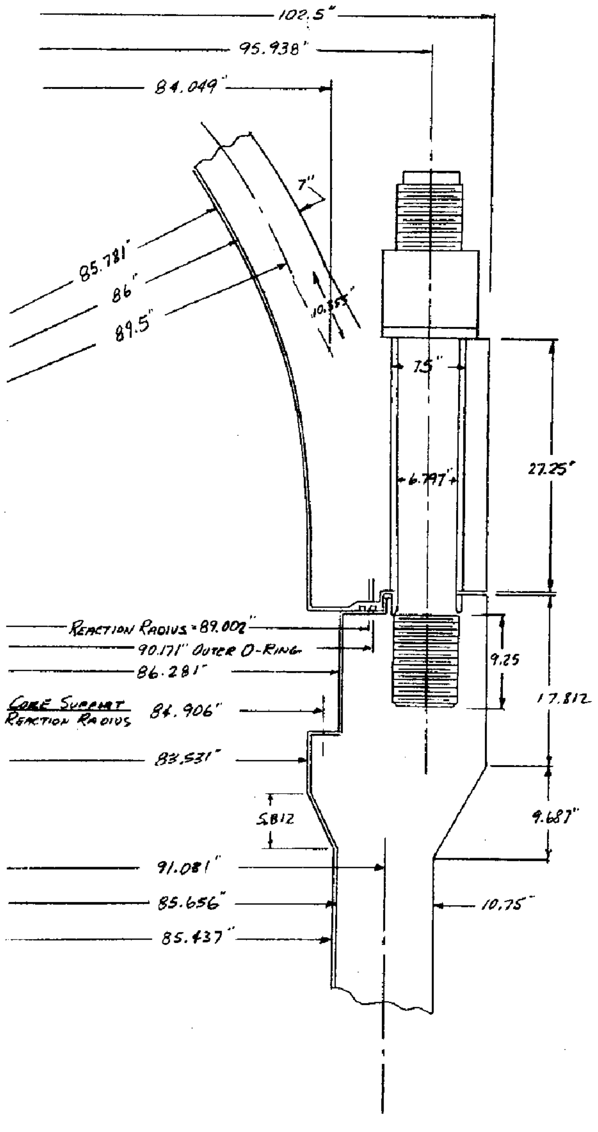
NUMBER 5-151-P | A 56
SHEET 5 OF 51
DATE 5-12-66 BY COCKRELL
CHECK DATE 5-12-66 BY ALEXANDER

DESCRIPTION Fatigue Evaluation Of Head Flange, Vessel Flange And Closure Studs

5. Detailed Analysis:

a. System Geometry:

A Cross Section Of The Closure Head, Closure Head Flange, Vessel Flange, Vessel, And Closure Studs Is Shown Below. Certain Dimensions Of Interest Are Listed.



All Materials Are SA-302B Or Have Same Allowables Except Nuts, Studs And Washers Which Are SA-320 Gr. 143.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.

NUMBER 5-151-P | A-57
SHEET 6 OF 51
DATE 5-12-66 BY COCKRELL
CHECK DATE 5-12-66 BY ALEXANDER

DESCRIPTION FATIGUE ANALYSIS OF HEAD FLANGE VESSEL FLANGE AND CLOSURE STUDS

5- DETAILED ANALYSIS:

b - SYSTEM LOADS:

THE CLOSURE HEAD AND VESSEL ASSEMBLY SHOWN ON THE PREVIOUS SHEET WILL BE ANALYZED FOR THE FOLLOWING TRANSIENT CONDITIONS.

TRANSIENT CONDITION	NUMBER OF OCCURRENCES
a. PLANT HEATUP AT 100°F PER HOUR	200
b. PLANT COOLDOWN AT 100°F PER HOUR	200
c. PLANT LOADING AT 5% OF FULL POWER PER MIN.	14500
d. PLANT UNLOADING AT 5% OF FULL POWER PER MIN.	14500
e. STEP LOAD INCREASE OF 10% OF FULL POWER BUT NOT TO EXCEED FULL POWER	2000
f. STEP LOAD DECREASE OF 10% OF FULL POWER FROM 100% POWER	2000
g - STEP LOAD REDUCTION FROM 100% TO 50% FULL POWER	200
h - REACTOR TRIP FROM FULL POWER	400
i - PLANT HYDROSTATIC TEST OF 3125 PSIA AT ROOM TEMP.	5
j - PLANT HYDROSTATIC TEST OF 2500 PSIA UP TO 400°F	5
k - STEADY STATE FLUCTUATIONS OF ±6°F & ±100°F PER MIN.	∞
l - LOSS OF FLOW, ONE PUMP	30
m - LOSS OF LOAD	30
n - STEAM BREAK	5

c - SYSTEM ALLOWABLES:

1 - THE AVERAGE BEARING STRESS UNDER THE MAXIMUM ATTAINABLE LOAD SHALL NOT BE MORE THAN $1.5 S_m$ AT ACTUAL METAL TEMPERATURE. SEE N-417-1 OF THE ASME CODE SECTION III.

2 - SHOW THAT THE RANGE OF STRESS INTENSITY AT EACH POINT DUE TO THE COMBINATION OF MECHANICAL LOADING PLUS THERMAL EFFECTS (NEGLECTING STRESS CONCENTRATIONS) IS LESS THAN $3 S_m$. SEE N-414.4 OF THE ASME CODE SECTION III.

3 - SHOW THAT EACH POINT MEETS THE REQUIREMENTS FOR PEAK STRESS INTENSITY GIVEN IN N-414.5 OF THE ASME CODE. THE PROCEDURE WILL BE THAT DESCRIBED IN N-415.2 AND N-416.2 OF THE ASME CODE SECTION III.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.

NUMBER 5-151-P | A 58
SHEET 7 OF 51
DATE 5-12-66 BY COCKRELL
CHECK DATE 5-12-66 BY ALEXANDER

DESCRIPTION FATIGUE EVALUATION OF HEAD FLANGE VESSEL FLANGE AND CLOSURE STUDS

5. DETAILED ANALYSIS
d- DEVELOPMENT OF CONTINUITY MATRIX
1- ANALYTICAL MODEL

THE ACTUAL STRUCTURE IS DIVIDED INTO THE FOLLOWING ANALYTICAL MODEL TO FACILITATE THE ANALYSIS. THE ASSUMED DIRECTIONS OF THE REDUNDANTS ARE ILLUSTRATED.

BODY-1 (LONG SPHERICAL SHELL)
BODY-2 (RING)
BODY-3 (RING WITH A BEAM APPENTAGE-STUDS)
BODY-4 (LONG CYLINDRICAL SHELL)

$H_1 \rightarrow H_3$, $M_1 - M_3$, AND V ARE THE UNKNOWN INTERACTION FORCES. (V IS THE CHANGE IN BOLT LOAD).

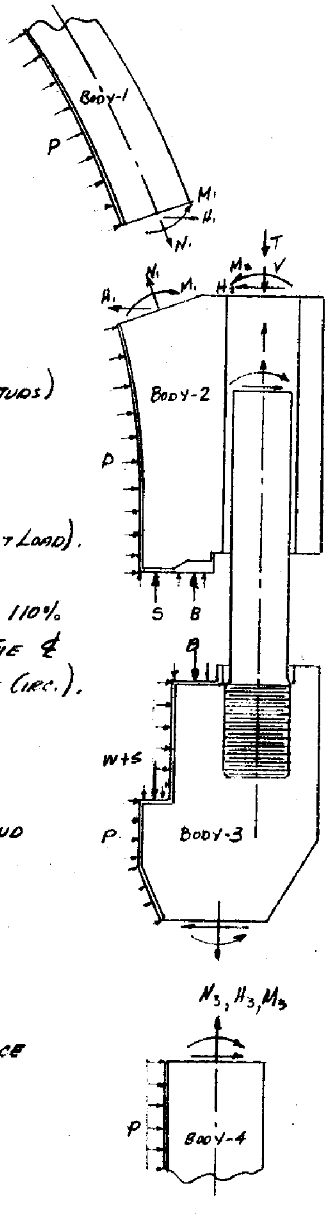
T IS THE INITIAL BOLT LOAD BASED UPON 110% OF THE BLOW-OFF LOAD CALCULATED TO THE ℄ OF THE OUTER O-RING. (IN-KIP PE IN. OF CIRC.).

P IS THE APPLIED PRESSURE.

B IS THE BEARING LEDGE REACTION AND IS A FUNCTION OF T, V, AND P.

W IS THE WEIGHT OF THE CORE AND CORE SUPPORT.

S IS THE CORE HOLDDOWN SPRING FORCE



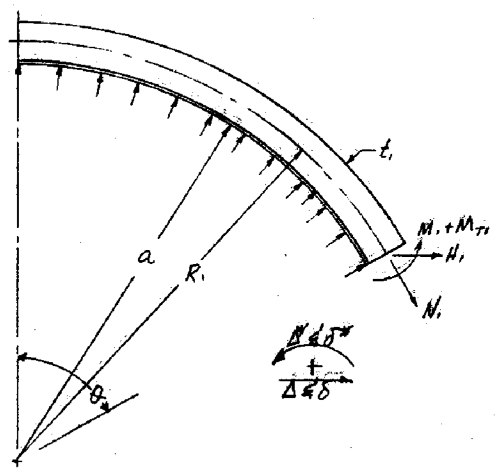
COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.

NUMBER S-151-B | A 59
SHEET 8 OF 51
DATE 5-12-66 BY Cockrell
CHECK DATE 5-12-66 BY Alexander

DESCRIPTION Fatigue Evaluation of Head Flange Vessel Flange and Closure Studs

5. Detailed Analysis:
d. Development of Continuity Equations:
2. Movements Due To Loads:

Body-1



$R_1 = 89.5''$

$t_1 = 7''$

$a = 85.781''$

$\theta = 69°54'$

$N_1 = \frac{a^2 P}{2R_1} = 41.108P$

$\beta^2 = 3(1-\nu^2)\left(\frac{R_1}{t_1}\right)^2$

$\beta = 4.59624$

$K_1 = 1-\left(\frac{1-2\nu}{2\beta}\right)\cot\theta = 0.98408$

$K_2 = 1-\left(\frac{1+2\nu}{2\beta}\right)\cot\theta = 0.93631$

Displacements Due To Redundant Forces:

$$E\Delta_H = \frac{2\beta^2\sin\theta}{t_1}\left[\frac{R\sin\theta}{2\beta}\left(\frac{1}{K_1}+K_2\right)H_1 + \frac{1}{K_1}M_1\right]$$

$$= 101.1878\,H_1 + 5.7599\,M_1$$

$$E\Delta_H^* = \frac{2\beta^2\sin\theta}{t_1}\left[\frac{1}{K_1}H_1 + \frac{2\beta}{R_1\sin\theta}\,\frac{1}{K_1}M_1\right]$$

$$= 5.7599\,H_1 + 0.6300\,M_1$$

Displacements Due To Applied Forces

$$E\delta_H = \frac{(1-\nu)a^2 P\sin\theta}{2t} = 345.5091\,P$$

$$E\delta_H^* = 0$$

5. DETAILED ANALYSIS:

d. DEVELOPMENT OF CONTINUITY MATRIX:

2. MOVEMENTS DUE TO LOADS:

DISPLACEMENTS DUE TO THERMAL EFFECTS:

$$E\delta_{11T} = E\alpha R_1 \sin\theta (T_{M1} - 70) + \frac{2\beta^2 \sin\theta}{t_1}\left(\frac{1}{K_1}\right) M_{T1}$$

$$= 84.049 E\alpha (T_{M1} - 70) + 5.7599 M_{T1}$$

$$E\delta_{11T}^{*} = E\alpha R_1 \sin\theta \left(\frac{\Delta T}{\Delta X}\right)_1 + \frac{4\beta^3}{t_1 R_1}\left(\frac{1}{K_1}\right) M_{T1}$$

$$= 84.049 E\alpha \left(\frac{\Delta T}{\Delta X}\right)_1 + 0.6300 M_{T1}$$

WITH THE ABOVE EQUATIONS FOR $E\delta_{11T}$ AND $E\delta_{11T}^{*}$ AND THE VALUES FOR T_{M1}, $(E\alpha)_{MEAN}$, AND $\left(\frac{\Delta T}{\Delta X}\right)_1$ LISTED IN THE FOLLOWING TABLE, WE GET THE FOLLOWING VALUES FOR THE DISPLACEMENT AND ROTATION OF BODY-1 AT CUT-1 DUE TO THERMAL EFFECTS.

TRANSIENT		T_{M1}	$(E\alpha)_{MEAN}$	$\left(\frac{\Delta T}{\Delta X}\right)_1$	M_{T1}	$E\delta_{11T}$	$E\delta_{11T}^{*}$
HEATUP	4.00 HRS	412	185	-3.90	332,271	7231630	148690
	4.25	434	186	-4.10	334,688	7618220	146750
	4.35	443	186	-4.20	335,628	7764330	145790
	4.47	456	186	-4.30	336,991	7975410	145080
	5.00	482	186	-4.20	259,934	7938030	98100
STEADY STATE		546	186	-0.100	3,217	7459890	470
COOLDOWN	4.00 HRS	234	182	3.90	-289,429	841610	-122680
	4.25	212	181	4.00	-292,207	477150	-123240
	4.35	203	180	4.10	-293,446	321910	-122840
	4.47	190	180	4.20	-295,197	115150	-122430
	5.00	164	180	4.10	-225,118	125450	-79790

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT. CHATTANOOGA, TENN.

CHARGE NO. ____

DESCRIPTION: FATIGUE EVALUATION OF HEAD FLANGE VESSEL FLANGE AND CLOSURE STUDS

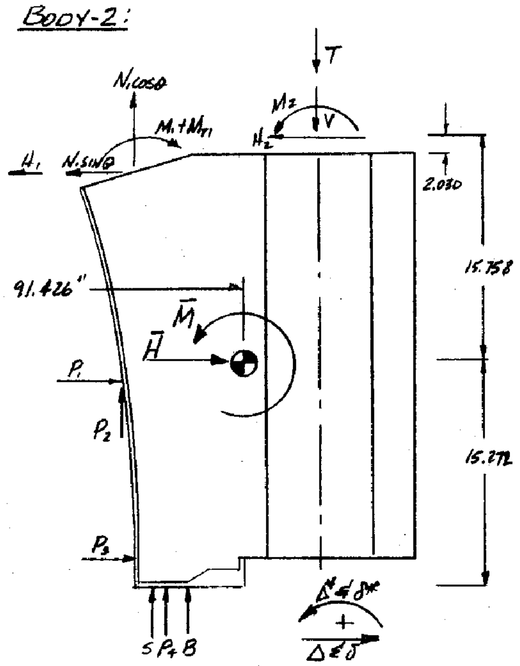
NUMBER: 5-151-P | A-61
SHEET: 10 OF 51
DATE: 5-12-66 BY: COCKRELL
CHECK DATE: 5-12-66 BY: ALEXANDER

5- DETAILED ANALYSIS:

d- DEVELOPMENT OF CONTINUITY EQUATIONS:

2- MOVEMENTS DUE TO LOADS:

BODY-2:



$A_2 = 441.967 \text{ IN}^2$

$I_2 = 27499 \text{ IN}^4$

$\frac{R^2}{I} = 0.30396$

$\frac{R^2}{A} = 18.9125$

$\bar{H} = -0.9193 H_1 - 1.0494 H_2 + 10.6687 P$

$\bar{M} = 11.4785 H_1 - 0.9193(M_1 + M_{T1}) + 16.5357 H_2 + 1.0494 M_2 - 7.2782(T+V) - 6.3351 S + 1.0401 P$

DISPLACEMENTS DUE TO REDUNDANT FORCES:

$E\Delta_{21} = \frac{R^2}{A}\bar{H} - h_j \frac{R^2}{I}\bar{M} = -60.9499 H_1 + 3.4890 M_1 - 82.6039 H_2 - 3.9830 M_2 + 27.6228 V$

$E\Delta^*_{21} = \frac{R^2}{I}\bar{M} = 3.4890 H_1 - 0.2794 M_1 + 5.0262 H_2 + 0.3190 M_2 - 2.2123 V$

$E\Delta_{22} = \frac{R^2}{A}\bar{H} - h_j \frac{R^2}{I}\bar{M} = -72.3659 H_1 + 4.4028 M_1 - 99.0496 H_2 - 5.0268 M_2 + 34.8614 V$

$E\Delta_{22} = \frac{R^2}{I}\bar{M} = 3.4890 H_1 - 0.2794 M_1 + 5.0262 H_2 + 0.3190 M_2 - 2.2123 V$

$EV_{22} = 6.936 \frac{R^2}{I}\bar{M} = 24.1997 H_1 - 1.9379 M_1 + 34.8617 H_2 + 2.2126 M_2 - 17.9267 V$

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.
DESCRIPTION FATIGUE EVALUATION OF HEAD FLANGE, VESSEL FLANGE AND CLOSURE STUDS

NUMBER 5-151-P | A62
SHEET 11 OF 51
DATE 5-12-66 BY COCKRELL
CHECK DATE 5-12-66 BY ALEXANDER

5- DETAILED ANALYSIS:
d- DEVELOPMENT OF CONTINUITY EQUATIONS:
2- MOVEMENTS DUE TO LOADS:

DISPLACEMENTS DUE TO APPLIED FORCES:

$E\delta_{21} = \frac{R^2}{A}\bar{H} - h_i \frac{R^2}{I}\bar{M} = 27.6228T + 24.0433S + 197.8250P$

$E\delta_{21}^{*} = \frac{R^2}{I}\bar{M} = -2.2123T - 1.9256S + 0.3161P$

$E\delta_{22} = \frac{R^2}{A}\bar{H} - h_i \frac{R^2}{I}\bar{M} = 34.8614T + 30.3439S + 196.7907P$

$E\delta_{22}^{*} = \frac{R^2}{I}\bar{M} = -2.2123T - 1.9256S + 0.3161P$

$Ev = 6.936 \frac{R^2}{I}\bar{M} = 96.0232P$

DISPLACEMENTS DUE TO THERMAL EFFECTS:

$E\delta_{21T} = R_{21} E\alpha (T_{a_{eq}} - 70) - h_i \frac{R^2}{I}\bar{M} = 84.049\, E\alpha (T_{a_{eq}} - 70) + 3.4890\, M_{T1}$

$E\delta_{21T}^{*} = R_2 E\alpha \left(\frac{\Delta T}{\Delta X}\right)_{eq} + \frac{R^2}{I}\bar{M} = 91.426\, E\alpha \left(\frac{\Delta T}{\Delta X}\right)_{eq} - 0.2794\, M_{T1}$

$E\delta_{22T} = R_{22} E\alpha (T_{a_{eq}} - 70) - h_i \frac{R^2}{I}\bar{M} = 95.938\, E\alpha (T_{a_{eq}} - 70) + 4.4028\, M_{T1}$

$E\delta_{22T}^{*} = R_2 E\alpha \left(\frac{\Delta T}{\Delta X}\right)_{eq} + \frac{R^2}{I}\bar{M} = 91.426\, E\alpha \left(\frac{\Delta T}{\Delta X}\right)_{eq} - 0.2794\, M_{T1}$

$Ev_{22T} = 6.936 \frac{R^2}{I}\bar{M} + (h_1 + h_2) E\alpha (T_{M_2} - 70) = 634.1307\, E\alpha \left(\frac{\Delta T}{\Delta X}\right)_{eq} - 1.9379\, M_{T1} + 31.030\, E\alpha (T_{M_2} - 70)$

$T_{a_{eq}}$ AND $\left(\frac{\Delta T}{\Delta X}\right)_{eq}$ ARE OBTAINED BY TAKING THE EXISTING AXIAL TEMPERATURE GRADIENT AND TRANSFORMING IT INTO AN EQUIVALENT LINEAR AXIAL GRADIENT BY A COMPUTER PROGRAM. THE COMPUTER OUTPUT VALUES ARE:

T_M (MEAN TEMPERATURE OF RING)
M_T (THERMAL MOMENT OF RING)
$(E\alpha)_i$ (YOUNG'S MODULUS TIMES COEFF. OF THERMAL EXPANSION)

5- Detailed Analysis:
d- Development Of Continuity Equations:
2- Movements Due To Loads:

Such that,

$$M_T = \frac{E\alpha t^2}{12(1-\nu)}(\Delta T) = \frac{E\alpha t^2}{12(1-\nu)}(T_{a eq.} - T_{b eq.})$$

or

$$(T_{a eq.} - T_{b eq.}) = \frac{12(1-\nu) M_T}{E\alpha t^2}$$

$T_{a eq}$ = Equivalent Temp. At Top Of Ring
$T_{b eq}$ = Equivalent Temp. At Bottom Of Ring

$$T_M = \frac{T_{a eq.} + T_{b eq}}{2}$$

$$T_{a eq.} = T_M - \frac{\Delta T}{2}$$

$$T_{b eq.} = T_M + \frac{\Delta T}{2} \quad \text{And} \quad -\frac{\Delta T}{\Delta X} = \frac{T_{a eq.} - T_{b eq.}}{t}$$

With The Above Expressions For $T_{a eq.}$, $T_{b eq.}$ And $\frac{\Delta T}{\Delta X}$ And The Following Values Of T_M, M_T And $(E\alpha)_i$, We Get The Following:

Transient		T_m	M_T	$(E\alpha)_i$	$T_{a eq.}$	$T_{b eq.}$	$\frac{\Delta T}{\Delta X}$
Heatup	4.00 Hrs.	324	151185	198	319.6	328.4	0.326
	4.25	343	152244	200	338.6	347.4	0.325
	4.35	350	152678	200	345.6	354.4	0.326
	4.47	361	153303	201	356.6	365.4	0.325
	5.00	391	147106	203	386.8	395.2	0.309
Steady State		543	-100647	213	545.7	540.3	-0.202
Cooldown	4.00 Hrs.	319	-215210	198	325.3	312.7	-0.464
	4.25	300	-216752	197	306.3	293.7	-0.469
	4.35	292	-217459	196	298.4	285.6	-0.473
	4.47	281	-218439	195	287.5	274.5	-0.478
	5.00	251	-221317	195	257.6	244.4	-0.489

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO.

NUMBER 5-151-P | A 64
SHEET 13 OF 51
DATE 5-12-66 BY COCKRELL
CHECK DATE 5-12-66 BY ALEXANDER

DESCRIPTION: FATIGUE EVALUATION OF HEAD FLANGE, VESSEL FLANGE AND CLOSURE STUDS

5. DETAILED ANALYSIS:

d. DEVELOPMENT OF CONTINUITY EQUATIONS:

2. MOVEMENTS DUE TO LOADS:

SUBSTITUTING VALUES INTO EQUATIONS AS GIVEN ON SHEET-11, WE GET THE FOLLOWING VALUES FOR DISPLACEMENTS AND ROTATIONS OF BODY-2 AT CUTS-1 & 2.

	TRANSIENT	$(T_a - 70)$	$(T_{m_i} - 70)$	$(E\alpha)_m$	$\left(\frac{\Delta T}{\Delta X}\right)_{avg}$	M_{T1}	$E\delta_{21T}$	$E\delta^*_{21T}$	$E\delta_{22T}$	$E\delta^*_{22T}$	$E\psi_{22T}$
HEATUP	4.00 HRS	249.6	254	184	0.326	332271	5019360	-87360	5869010	-87360	844350
	4.25	268.6	273	184	0.325	334688	5321630	-88040	6215050	-88040	948030
	4.35	275.6	280	184	0.326	335628	5433170	-88290	6342750	-88290	986300
	4.47	286.6	291	184	0.325	336991	5608030	-88690	6542930	-88690	1046340
	5.00	316.8	321	185	0.309	259934	5832850	-67400	6767170	-67400	1375240
STEADY STATE		475.7	473	186	-0.202	3217	7447890	-4340	8502770	-4340	2699900
COOLDOWN	4.00 HRS	255.3	247	184	-0.464	-289429	2938400	73060	3232410	73060	1928410
	4.25	236.3	230	183	-0.469	-292207	2615010	73790	2862110	73790	1817890
	4.35	228.4	222	183	-0.473	-293446	2499180	74080	2717960	74080	1774400
	4.47	217.5	211	182	-0.478	-295197	2297140	74530	2498010	74530	1708500
	5.00	187.6	181	182	-0.489	-225118	2084260	54760	2284480	54760	1402010