VOLUME II ATTACHMENTS

ATTACHMENT 1 Part I

LIGO-C960964-01-V

1. STRUCTURAL CALCULATIONS

TITLE	DOCUMENT NO.	REVISION
I. LIGO Vacuum Equipment Structural Design Criteria	V049-1-066	0
II. Beam Splitter Chamber		
Finite Element Analysis of Upper Section	V049-1-014	0
ASME Code Calculations for Upper Section	V049-1-015	0
FE Analysis of 1 in. Thick Flange for Bolt Preload	V049-1-016	0
FE Analysis of 1 in. Thick Flange for Bolt Preload & Pos. Pressure	V049-1-017	0
FE Analysis of 3/4 in Thick Flange	V049-1-018	0
Flange Design for Internal Pressure	V049-1-019	0
Design of Removable Work Floor	V049-1-020	0
Design of Flange Welds	V049-1-021	1
Finite Element Analysis of Lower Section	V049-1-022	0
Design of 60 in Access Covers	V049-1-023	0
Design of Support Legs & Base Plates	V049-1-024	0
Temporary Cover for 60 in Nozzle	V049-1-025	0
Nozzle to Shell Welds	V049-1-026	0
Lifting Lugs	V049-1-027	0
Shipping Loads	V049-1-028	0
BSC Deflections	V049-1-029	1
NASTRAN Buckling Analysis	V049-1-040	0
BSC Support Clevis	V049-1-069	0
Nozzle D Analysis For 200 kg Force	V049-1-104	0
BSC Point Loads	V049-1-111	0

TITLE	DOCUMENT NO.	REVISION
III. Vacuum Equipment General Structural Calculations		
Vacuum Equipment Seismic Acceleration	V049-1-031	0
Component Interface Loads	V049-1-032	0
Bolted Flange Analysis for Tensile Forces	V049-1-042	0
Design of Pipe Bridge (Corner Station)	V049-1-071	0
Welds For 60.5 In. Nozzle To Shell Joints	V049-1-107	0
44" Gate Valve Weld Stub/Beamtube Buckling Stress Evaluation	V049-1-122	0
IV. Horizontal Access Module		
Ham Vessel Stress Analysis (Finite Element Analysis)	V049-1-039	0
Ham Vessel Buckling Analysis	V049-1-041	0
Design of Nozzle Jacking Bolt Assembly	V049-1-048	0
Design of 60 inch Cover	V049-1-073	0
Design of 84 inch Cover	V049-1-074	0

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-066 PAGE 1 OF 3
REV.	DEO #	DATE	BY:	CHECK	TITLE: LIGO Vacuum Equipment Structural Design Criteria	
0	0136	4/24/96	RDC	DMW		
					BY: R. D. Ciatto	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE: Establish structural design criteria for LIGO vacuum vessels and supports.

METHOD: Review LIGO vacuum equipment specification and design codes to determine requirements for structural integrity.

ASSUMPTIONS: N/A

INPUTS: LIGO project drawings and sketches.
LIGO E940002-02-V, Vacuum Equipment Specification

REFERENCES: See INPUTS

CALCULATIONS: (SEE ATTACHED)

CONCLUSIONS: These LIGO vacuum equipment structural design criteria conform to project and code requirements.

NOTES:

LIGO VACUUM EQUIPMENT STRUCTURAL DESIGN CRITERIA

Design Code for Vessels: ASME Boiler & Pressure Vessel Code, Section VIII, Pressure Vessels, Division 1, 1992 Edition through 1994 Addenda

Alternate Design Code for Vessels: ASME Boiler & Pressure Vessel Code, Section VIII, Pressure Vessels, Division 2, 1992 Edition through 1994 Addenda

Internal Vacuum = -14.7 psig

Positive Internal Pressure = 2 psig for vacuum chambers. Special conditions for 80K cryopumps (see Doc. No. V049-I-056).

Mechanical Loads

- Unbalanced Vacuum Loads - These loads are imposed at end components, and at branches in the system.
- Component Interface Loads - Equipment shall be designed for loads from adjacent components that are imposed when gate valves are closed and portions of the system are vented. See Doc. No. V049-1-032.

Design Temperature = 400° F

Seismic Acceleration = .05625 G (ASCE 7-88) lateral only, single direction

Pressure Boundary Material = SA240, Type 304/304L stainless steel for shells and SA182, Grade F Type 304L for flange forging. Other specification numbers and product forms of Type 304/304L may be used for miscellaneous pressure parts. Flange Bolting Material = SA193, Gr B7.

Basic Stress Limits (Section VIII, Div. 1):

Type 304L
S = 14.7 ksi at 400° F for membrane stress
S = 1.5S = 22 ksi at 400° F for membrane + bending stress

Type 304
S = 16.2 ksi at 400° F for membrane stress
S = 1.5S = 24.3 ksi at 400° F for membrane + bending stress

Basic Stress Limits (Section VIII, Div. 2):

Type 304L

Sm = 15.8 ksi at 400° F for Primary Membrane Stress Intensity
1.5Sm = 23.7 ksi at 400° F for Primary Bending Stress Intensity
3Sm = 47.4 ksi at 400° F for Secondary Membrane + Bending

Type 304

Sm = 18.7 ksi at 400° F for Primary Membrane Stress Intensity
1.5Sm = 28.0 ksi at 400° F for Primary Bending Stress Intensity
3Sm = 56.1 ksi at 400° F for Secondary Membrane + Bending

O-Ring Seal Material = Viton (1/4 in)

Maximum Durometer = 90
Compression Range = .058 in to .080 in (21% to 29 %)
Maximum Compression Force = 160 lb/in

Shipping Acceleration

Vertical Accel = 1G
Horizontal Accel = .5G

Design Code for Supports: AISC Manual of Steel Construction, Allowable Stress Design, Ninth Edition.

Material for Supports: A36 shapes and plate, A500 Gr B tube steel. See AISC Code for allowable stresses.

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-014 PAGE 1 OF 16
REV.	DEO #	DATE	BY:	CHECK	TITLE: Beam Splitter Chamber - Finite Element Analysis of Upper Section	
0	0024	12/6/95	RDC	WDB		
					BY: R.D.Ciarlo	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE: To evaluate the upper section of the beam splitter chamber for vacuum pressure and lifting loads.

METHOD: Finite element analysis of vessel upper section including cylinder, head and lifting lugs. 563 nodes and 537 elements.

ASSUMPTIONS: Ring stiffener and nozzle omitted from analysis since area of interest is shell near lug.

INPUTS: LIGO project sketches and drawings. The thickness of the cylinder is .25 in and the thickness head is .375 in.

REFERENCES: 1. ASME Boiler and Pressure Vessel Code, Section VIII, Pressure Vessels, Division 1.
2. Images - 3D, Version 3.0, R.L. Cloud & Associates.
3. Doc. No V049-1-066, LIGO Vacuum Equip., [illegible]

CALCULATIONS: (See Attachment)

CONCLUSIONS: Stresses in head and shell in the area of the lifting lugs are within limits of ASME Code, Section VIII, Div. 1.
A spreader beam should be used when lifting vessel.

NOTES: See stress contour plots attached. See V049-1-015 for buckling analysis per the ASME Code.

Computer File: [illegible]

FINITE ELEMENT ANALYSIS OF BEAM SPLITTER CHAMBER UPPER SECTION

THE PURPOSE OF THIS ANALYSIS IS TO EVALUATE THE UPPER HEAD AND CYLINDER OF THE BEAM SPLITTER CHAMBER FOR THE EFFECTS OF VACUUM PRESSURE AND LIFTING LUG LOADS. IT IS NOTED THAT THE UPPER HEAD IS A FLANGED AND DISHED HEAD WITH A CROWN RADIUS OF 105 IN AND A THICKNESS OF 3/8 IN. THE THICKNESS OF THE STRAIGHT CYLINDER IS 1/4 IN.

TO TAKE ADVANTAGE OF SYMMETRY, THE MODEL EXTENDS 180° AROUND ITS CIRCUMFERENCE.

IMAGES WAS USED FOR THE MODEL WHICH IS FILED AS

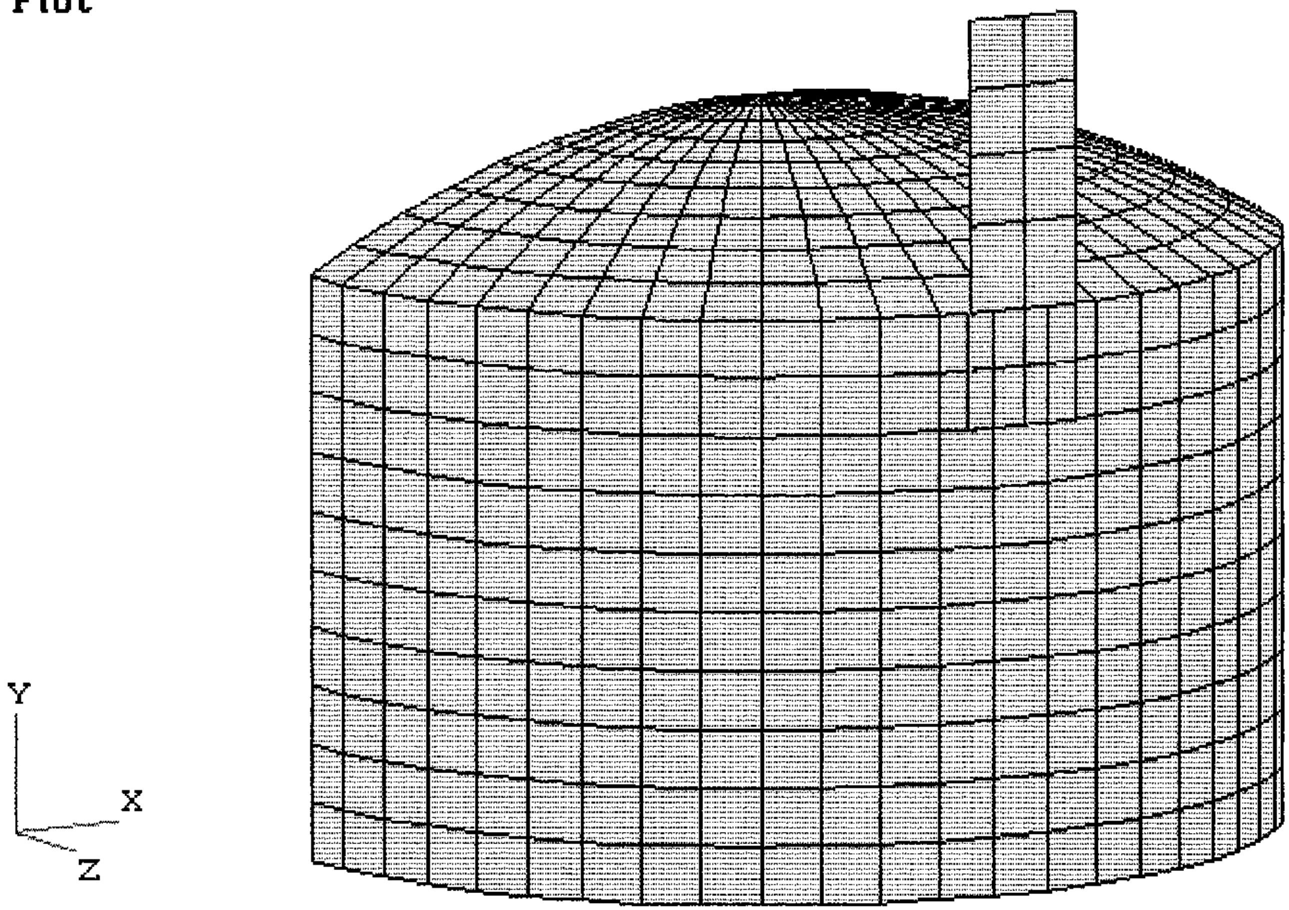
FILE NAME: BSCUPPER

THE STRESS INTENSITY PLOTS SHOWN IN THIS SECTION CONFIRM THAT STRESSES ARE LOW AND MEET THE REQUIREMENTS OF ASME SECT. VIII, DIV. 1, PAR UG-23(c)

PRIMARY MEMBRANE STRESS < S_{ALL} = 14.7 KSI @ 400°F

PRIMARY MEMBRANE + BENDING STRESS < 1.5 S_{ALL} = 22 KSI @ 400°F

IMAGES-3D
Ver. 3.0
Geometry Plot



BEAM SPLITTER CHAMBER - UPPER SECTION CYLINDER & HEAD   10/25/95
Hidden Line Removal   13:53:32

MATERIAL PROPERTIES OF SA240 TYPE 304L AT 400 °F

MODULUS OF ELASTICITY

26.5×10^6 PSI

POISSONS RATIO

$\nu = .3$

COEF. OF THERMAL EXPANSION

$\alpha = 9.19 \times 10^{-6}$

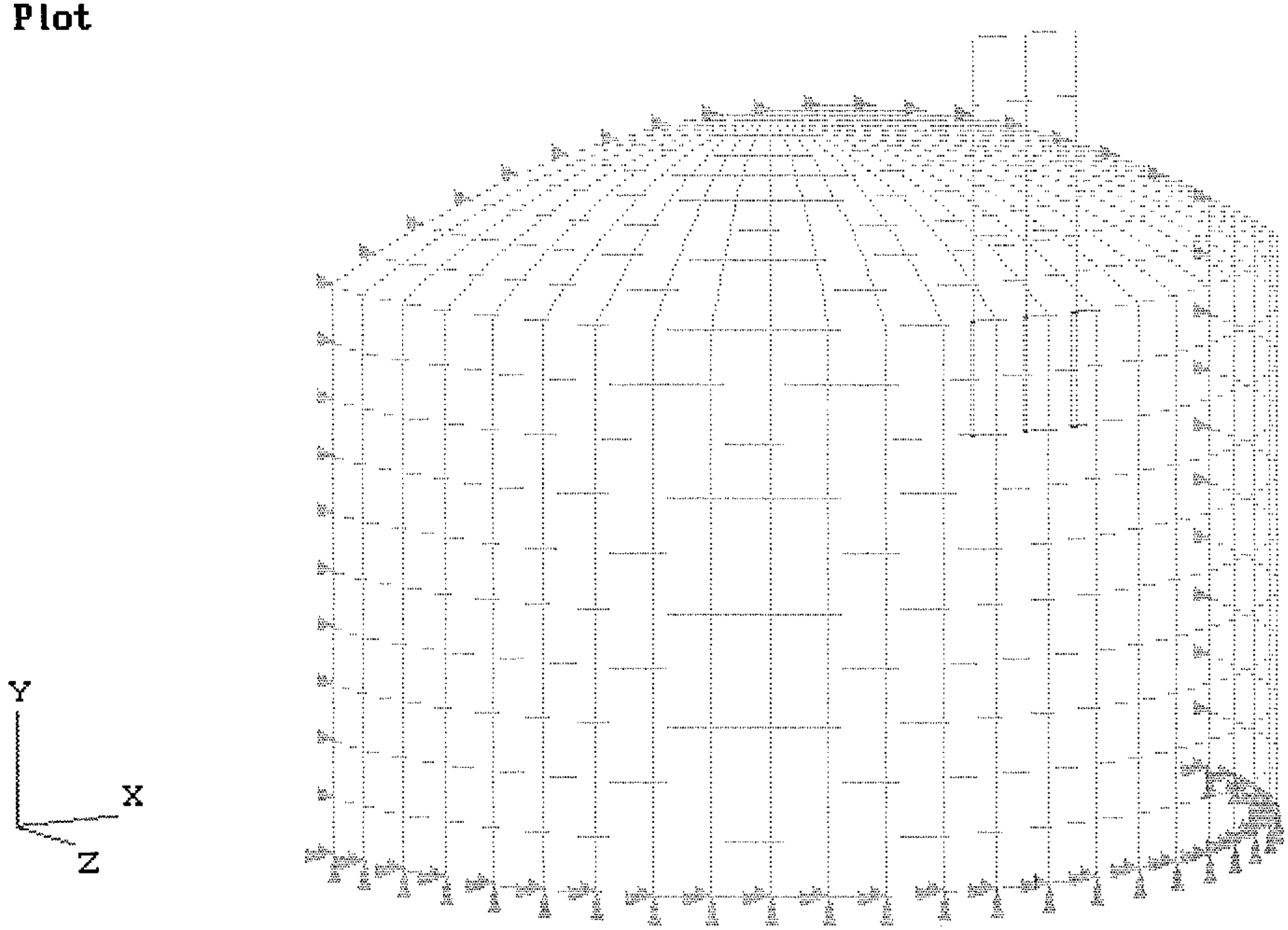
DENSITY

$\rho = .290$ LB/IN^3

THESE PROPERTIES ARE USED IN IMAGES MODELS FOR TYPE 304L SS.

Boundary Condition Shown

IMAGES-3D
Ver. 3.0
Geometry Plot

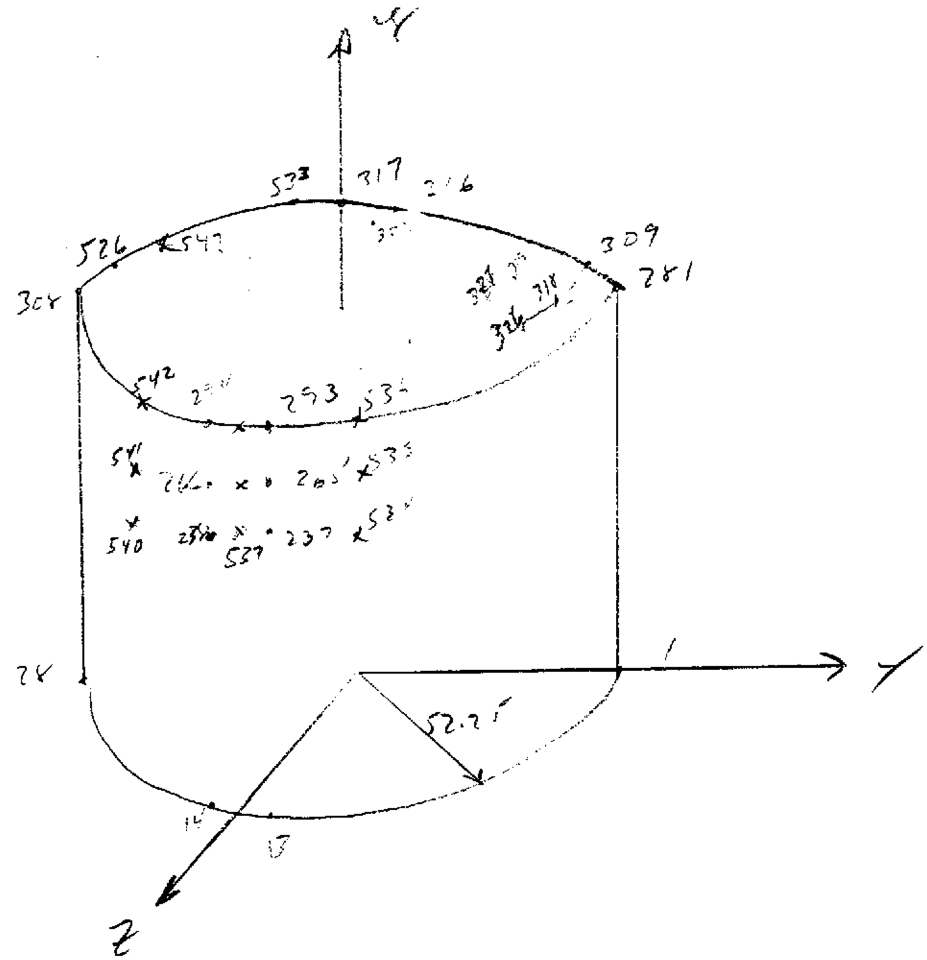


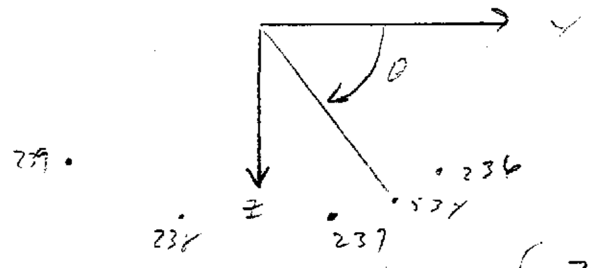
?

Auto Node | Blowup Node# | Elem# Plot | Exit Range | Full Rest | Help Rotate | Local Shrink | Move Slice

IMAGES FE MODEL

KEY NODE NUMBERS

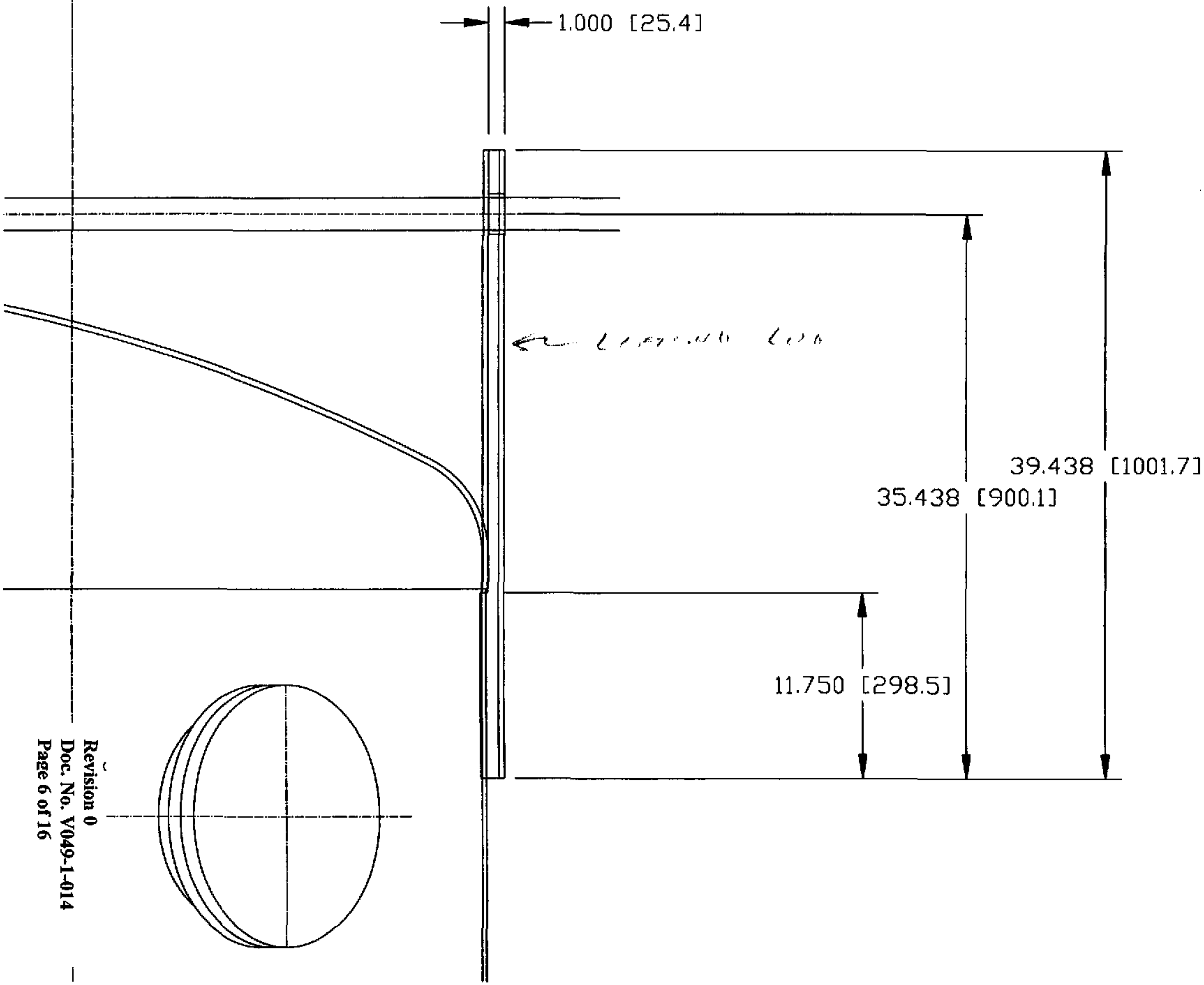


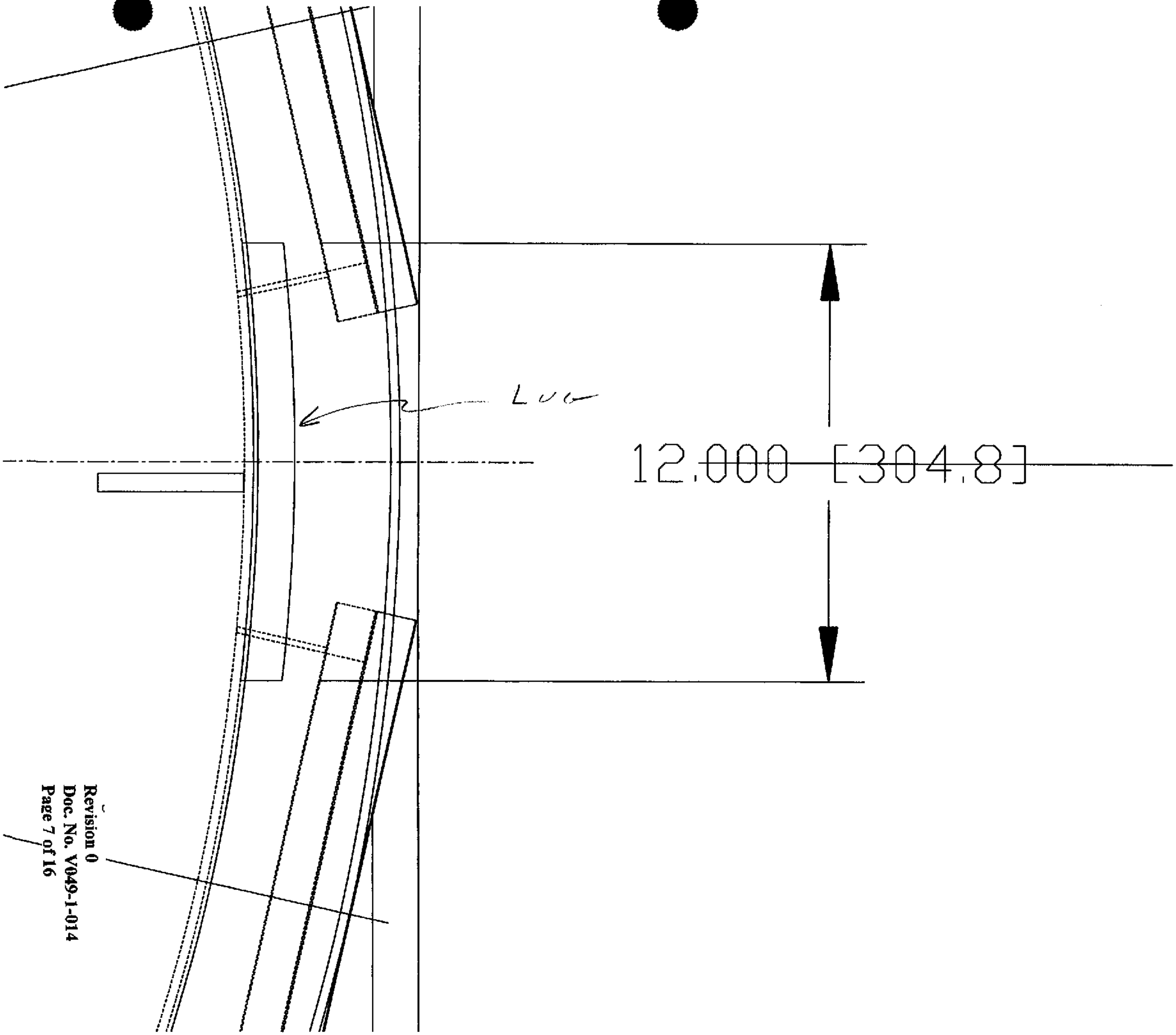


$\theta = 90 - \frac{180}{27}$

$= 83.3333°$

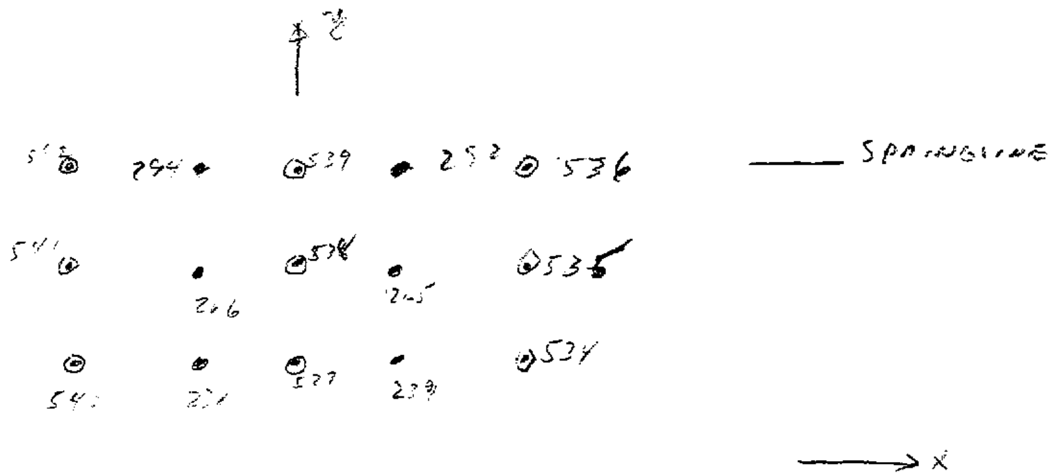
534 $\begin{cases} Y = 52.25 \cos 83.3333 \\ \quad = 6.0655 \\ y = 48 \\ Z = 52.25 \sin 83.333 \\ \quad = 51.8967 \end{cases}$


1.000 [25.4]
39.438 [1001.7]
35.438 [900.1]
11.750 [298.5]


LUG
12.000 [304.8]

LUG

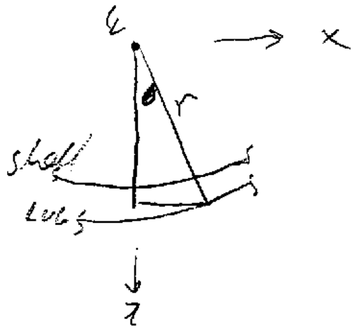
NODES 534 TO 541 ARE ADDED AT MIDSURFACE OF PLATE



ELEMENTS 229 TO 231, 254 TO 258 WERE DELETED AND REPLACED WITH ELEMENTS WITH ABOVE NODES

ADDITIONAL NODES WILL BE OFFSET FROM NODES 534 TO 541 WITH RIGID ELEMENTS. THESE NEW ELEMENTS WILL REPRESENT THE LIFTING LUG

ASSUME LUG IS 1 IN THICK. RADIUS TO MIDSURFACE IS



$$r = 52.25 + .125 + .5 = 52.875 \text{ in}$$

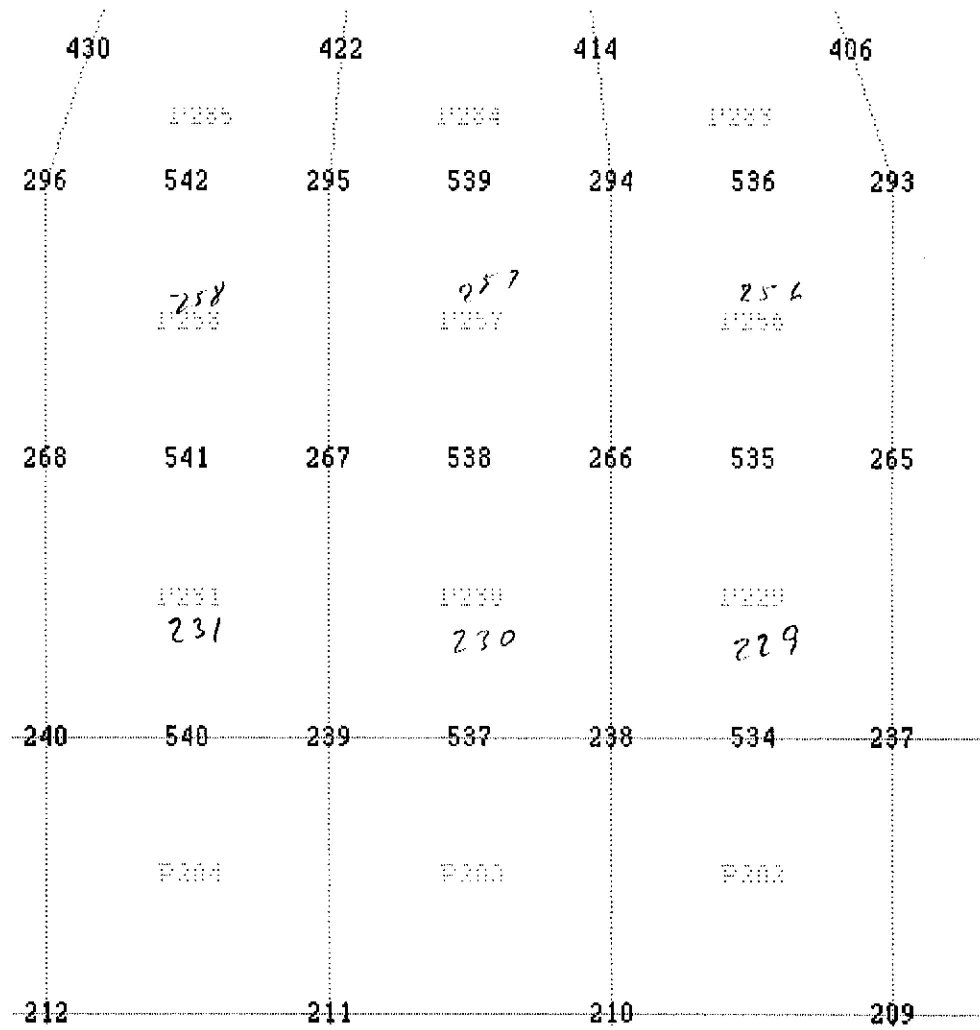
$$\theta = \tan^{-1}\left(\frac{6.0679}{51.8967}\right) = 6.6667^\circ$$

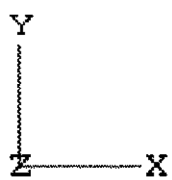
$$y = r \sin \theta$$
$$= 52.875 \sin 6.6667 = 6.1385 \text{ in}$$

$$z = r \cos \theta$$
$$= 52.5175 \text{ in}$$

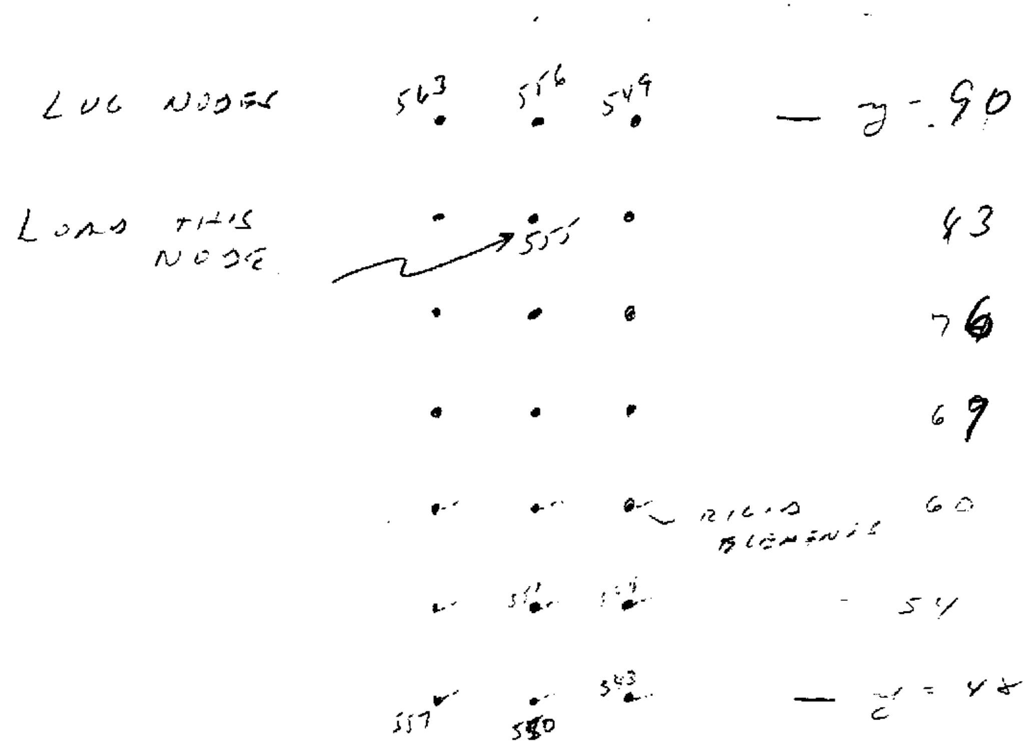
IMAGES-3D
Ver. 3.0
Geometry Plot

5-Solid or 5,F,T
6-Spring or 6,F,T
7-Rigid or 7,F,T
8-No element numbers
9-All elemt. numbers
10-Exit or 10,Size
?9
?#
?h,n
?n
?n
?n
?





Auto Node
Blowup Node#
Elem# Plot
Exit Range
Full Rest
Help Rotate
Local Shrink
Move Slice



RIGID ELEMENTS

I	J	TYPE	RELEASE
534	543	1	0
535	544		
536	545		
537	540		
538	551		
539	552		
540	557		
541	558		
542	559		

PLATES 1 TO 525 HAVE EXT. PRESSURE.

LUG LOADS (NODE 555)

VERTICAL (LOAD CASE 2)

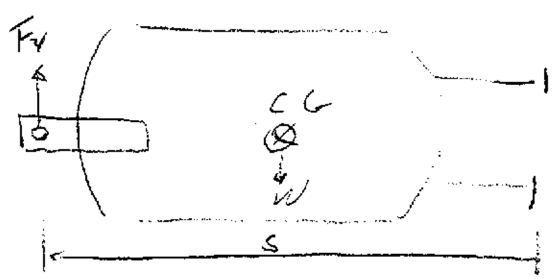
TOTAL ESTIMATED WEIGHT

$W = 12500$ LB

$F_{Y_{555}} = \frac{12500}{2}$

$= 6250$ LB

FOR LIFT TO VERTICAL POSITION FROM HORIZONTAL CONSTRUCTION, SHIPPING POSITION, LOAD IS IN X-DIRECTION. ASSUME VESSEL PIVOTS ABOUT BASE PLATE & CG IS MIDWAY BETWEEN BASE PLATE AND EYE OF LUG

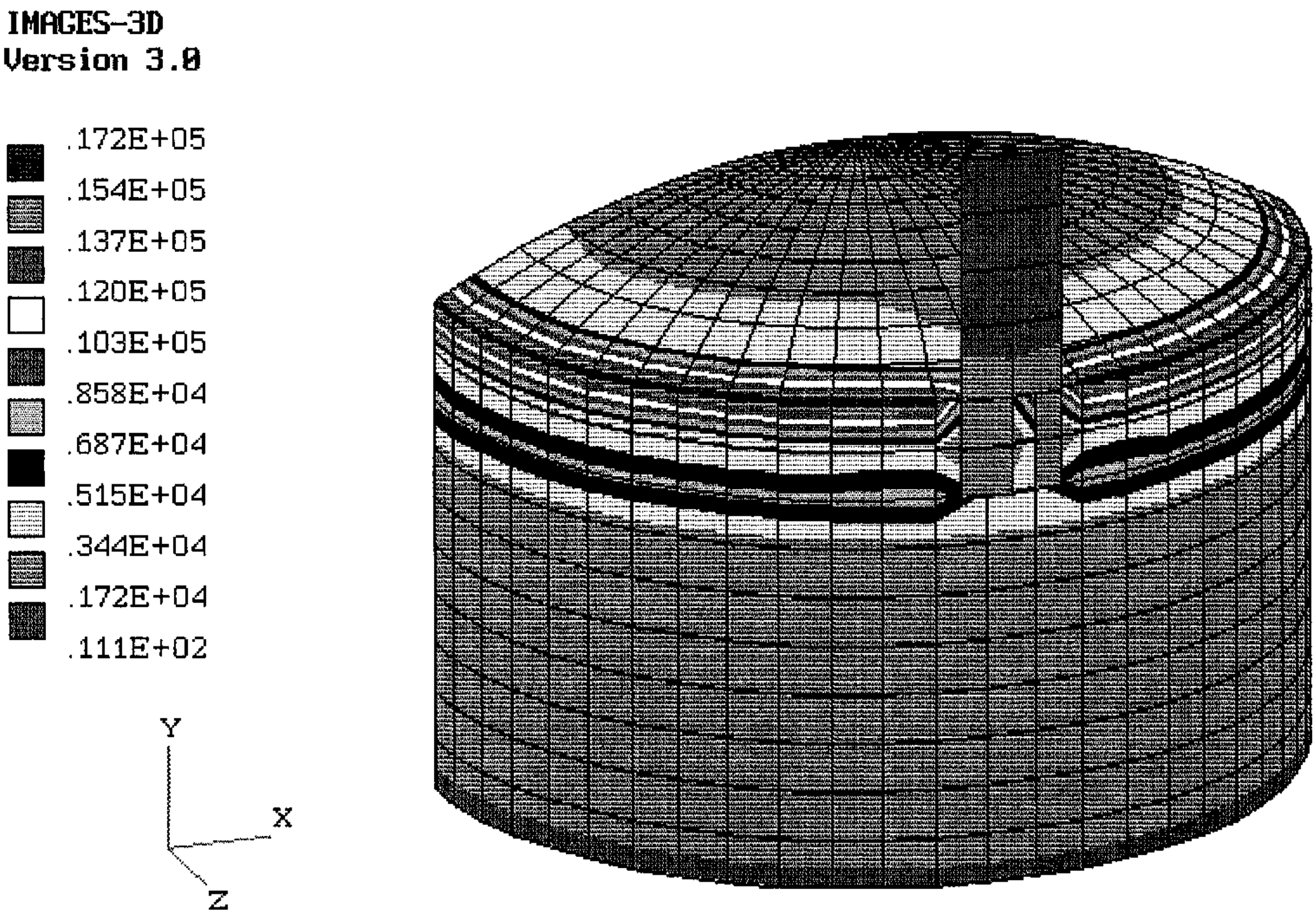


$F_X = \frac{W}{2} \times \frac{S}{2} \div S = \frac{W}{4}$

$= \frac{12500}{4} = 3125$ LB

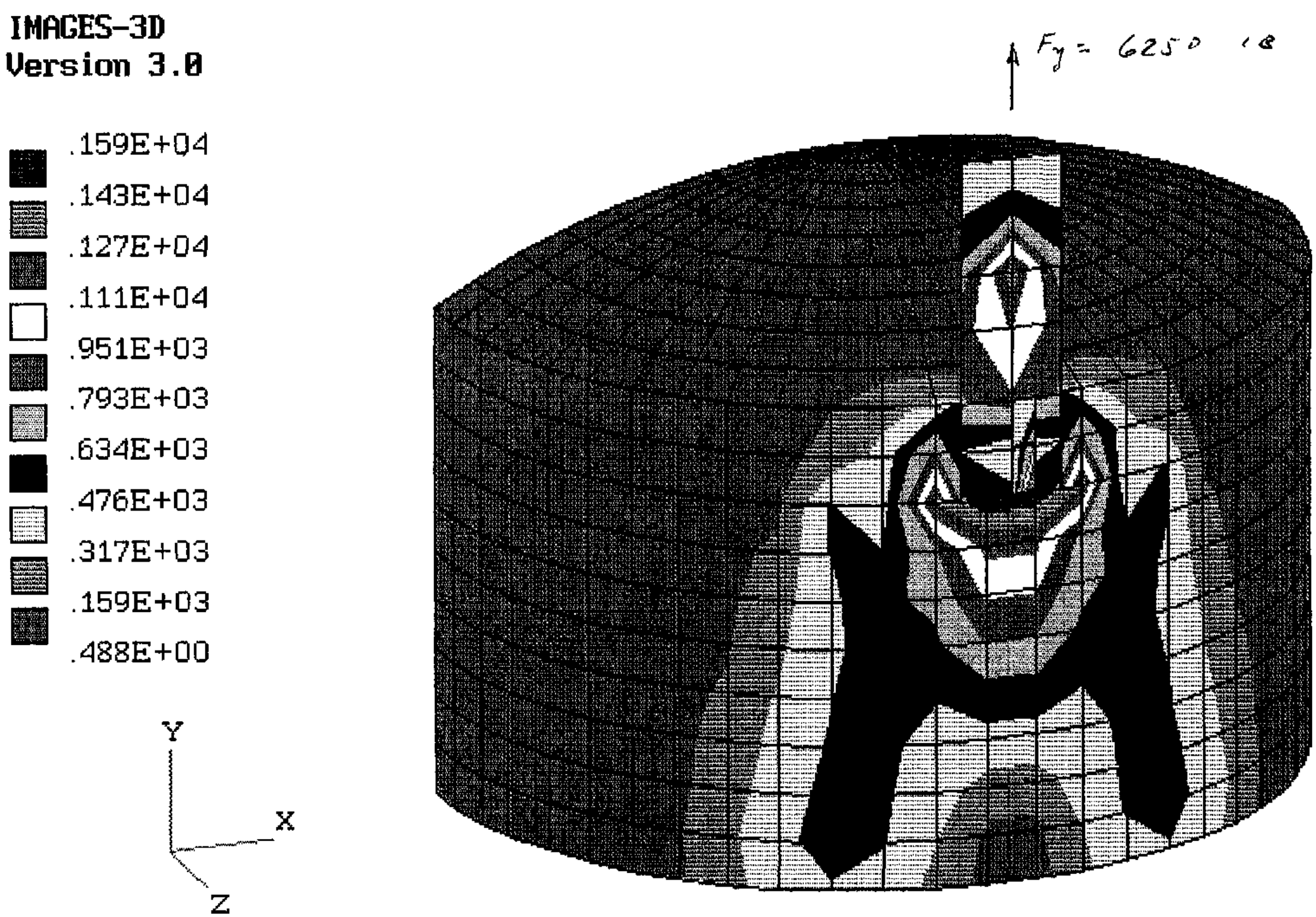
LOAD CASE 3

$F_{Y_{555}} = 3125$



Load Case 1 — Stress Contour Plot — Surf: Bottom — Stress Intensity — 10/25/95 10:46:21

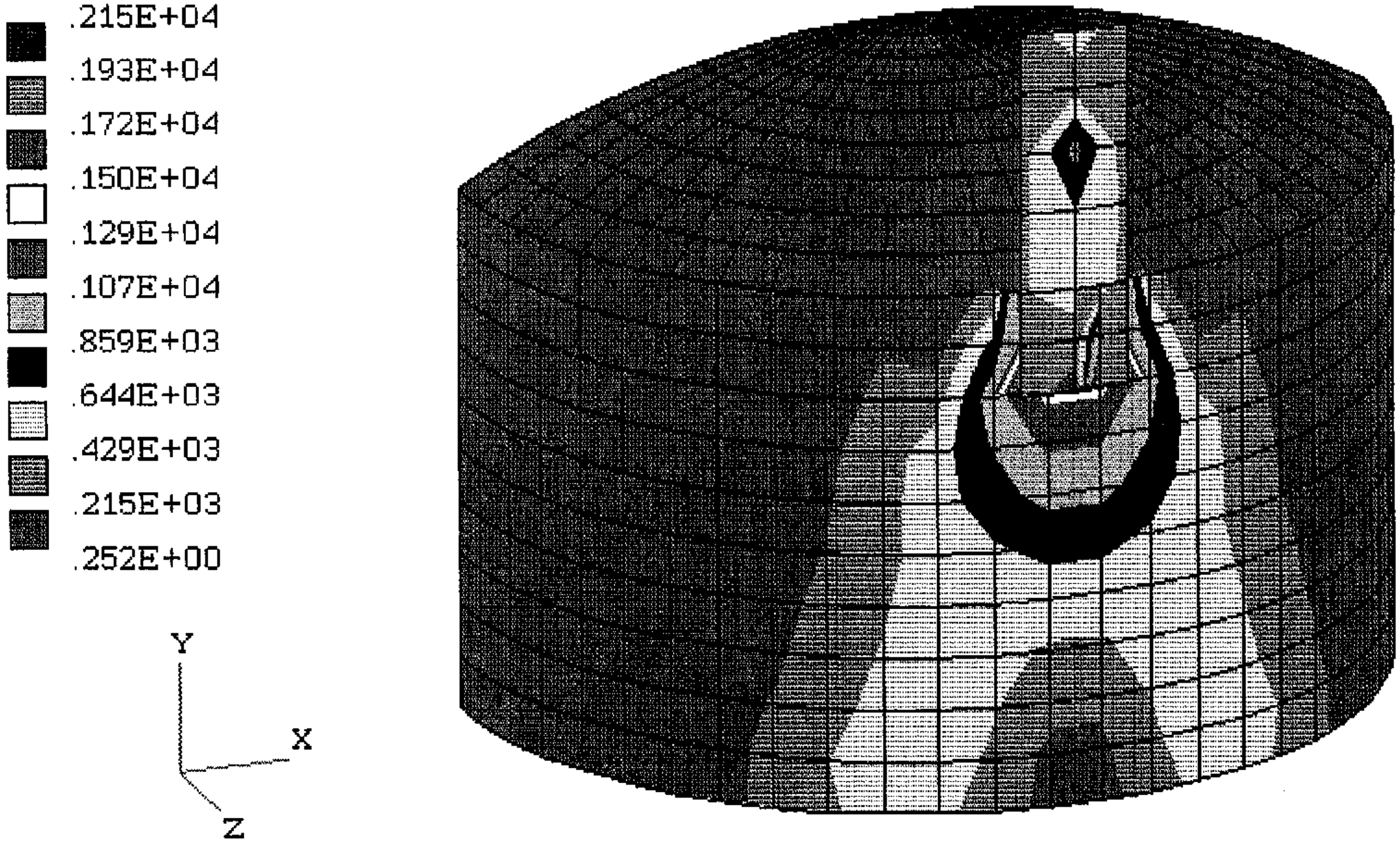
Vacuum Pressure



Load Case 2 | Stress Contour Plot | Surf: Top | Stress Intensity | 10/25/95 10:48:56

Vertical Lug Load

IMAGES-3D
Version 3.0



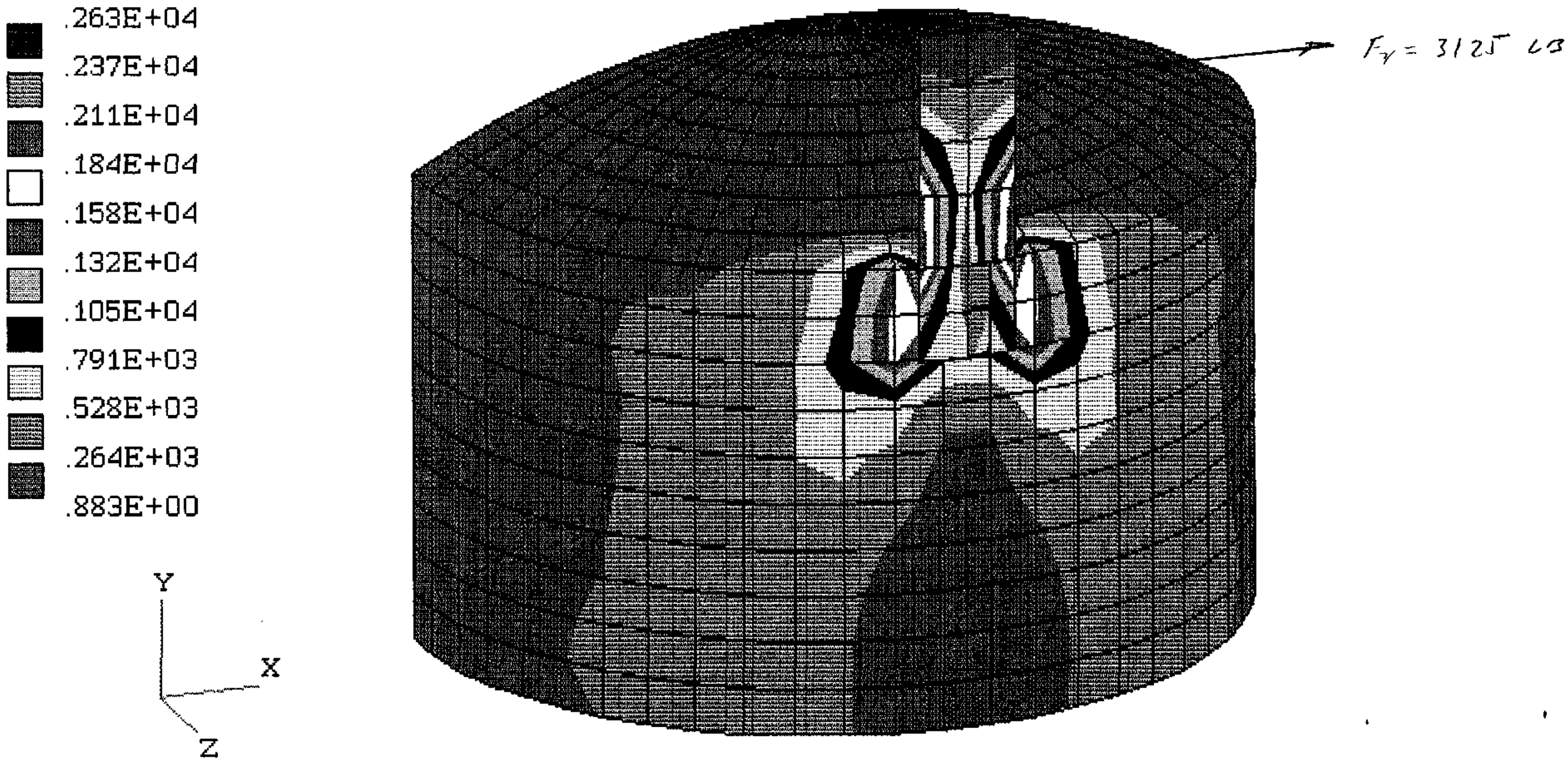
Load Case 2

Stress Contour Plot

Surf: Middle   Stress Intensity

10/25/95
10:52:10

IMAGES-3D
Version 3.0

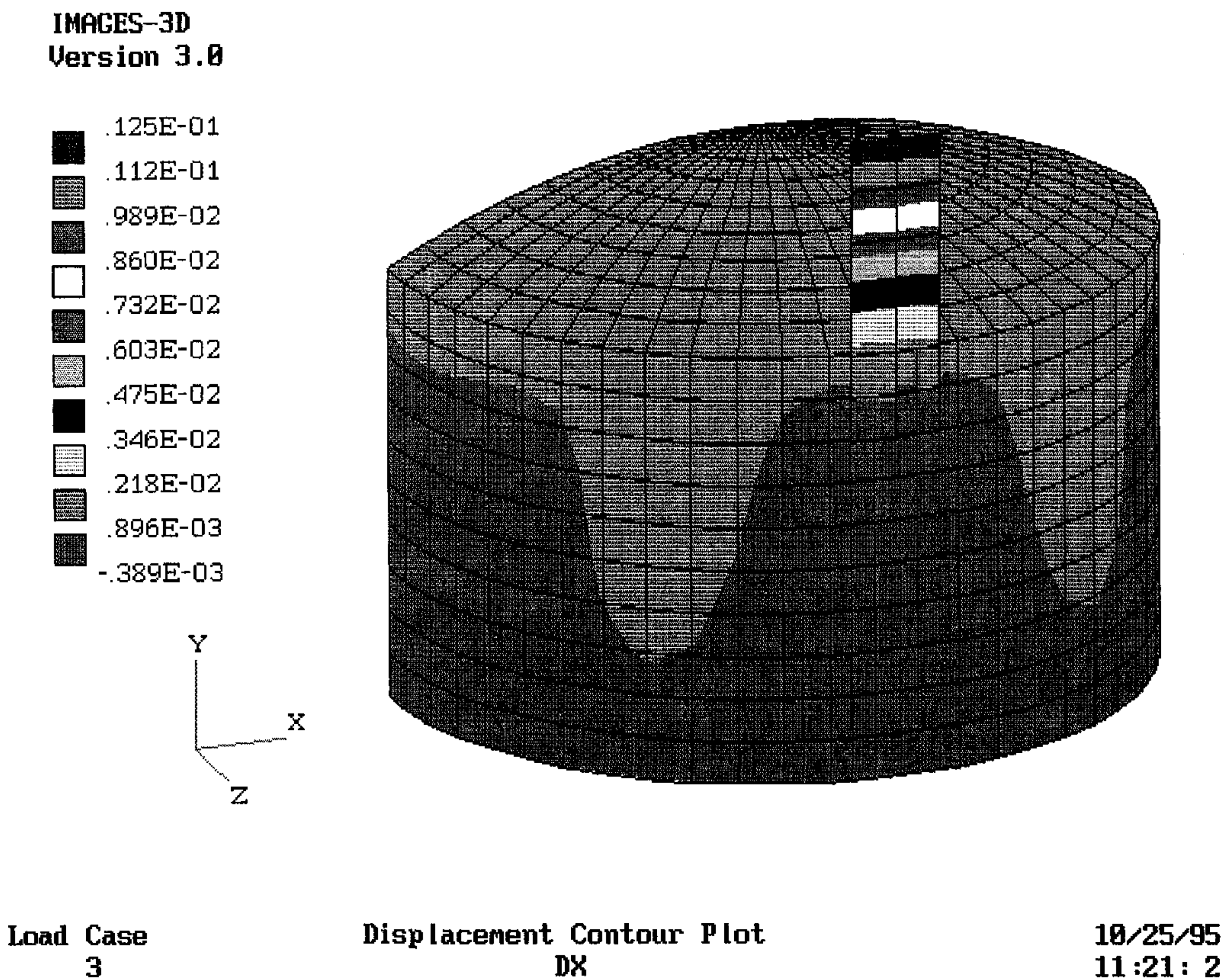


Load Case 3

Stress Contour Plot
Surf: Middle
Stress Intensity

10/25/95
10:54:34

HORIZONTAL LUG LOAD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Version 3.0
.125E-01
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.732E-02
.603E-02
.475E-02
.346E-02
.218E-02
.896E-03
-.389E-03
Y
X
Z
Load Case
3
Displacement Contour Plot
DX
10/25/95
11:21: 2

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-015 PAGE 1 OF 57
REV.	DEO #	DATE	BY:	CHECK	TITLE: Beam Splitter Chamber - ASME Code Calculations for Upper Section.	
0	002U	12/6/95	RDC	P.W.Y.		
					BY: R. D. Ciatto	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE: To determine if the upper section of the beam splitter chamber meets Section VIII, Div. 1 of the ASME Code for vacuum pressure.

METHOD: Equations for thickness requirements and pressure rating in the ASME Code, Section VIII, Div.1 are evaluated using the COMPRESS computer program, Version 5.31.

ASSUMPTIONS:

INPUTS: Vacuum pressure = 14.7 psi
Design temperature = 400^0F

REFERENCES: 1. ASME Boiler & Pressure Vessel Code, Section VIII, Pressure Vessels, Division 1.
2. COMPRESS 5.31, Computer Aided Pressure Vessel Design, Codeware Computer Systems, Inc.
3. Doc. No. V049-1-066, LIGO Vacuum Equip., Structural Design Criteria

CALCULATIONS: (See Attached)

CONCLUSIONS: Requirements of the ASME Code, Section VIII, Div. 1 are met for the BSC upper section head and cylinder.

NOTES: Nozzle G in lower section is included in this analysis. This nozzle penetrates the 1/2 in. shell. No reinforcing pad is required by code.

THE COMPRESS PROGRAM IS USED TO EVALUATE THE UPPER SECTION OF THE BSC. THIS INCLUDES:

TOP HEAD

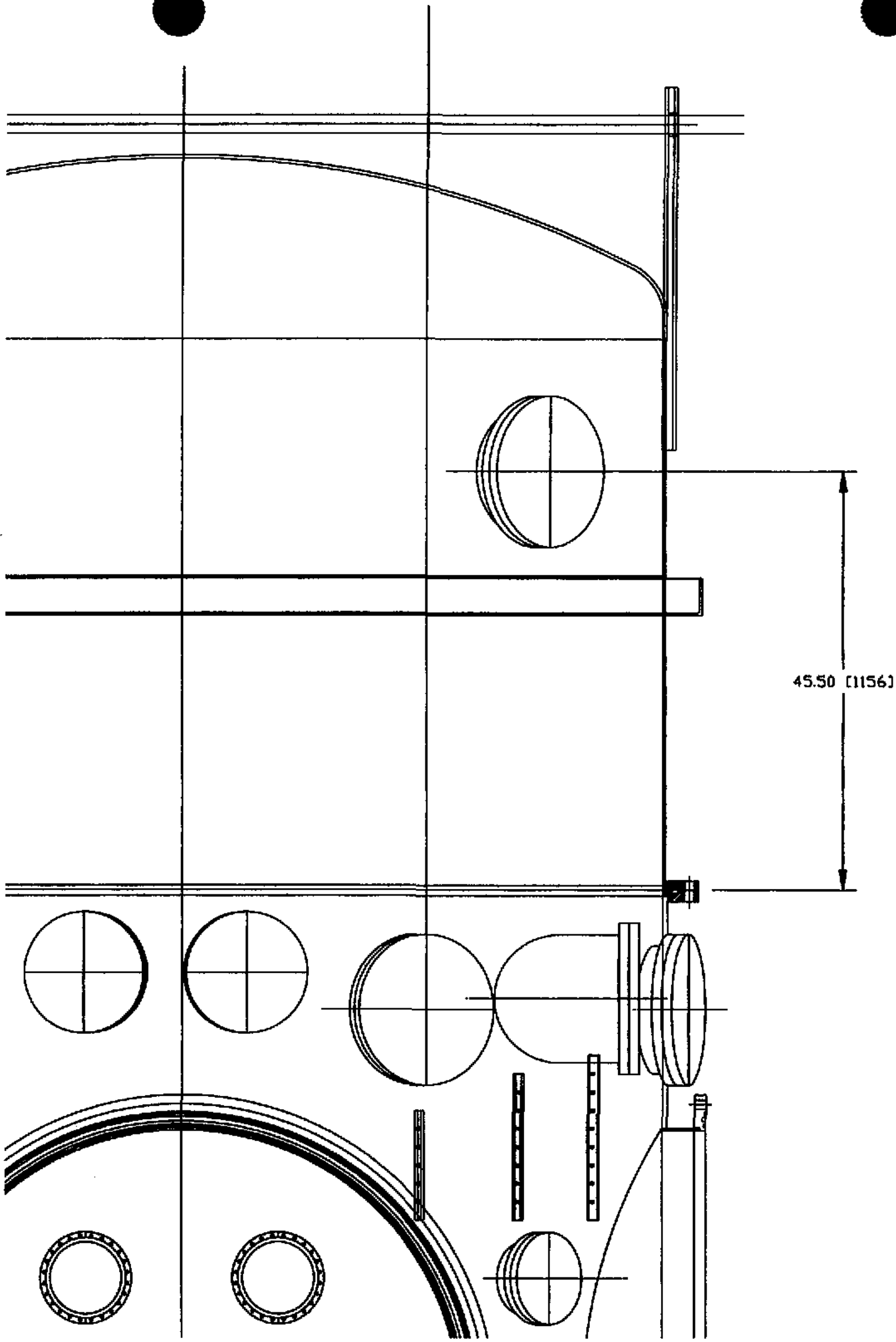
UPPER CYLINDRICAL SHELL (ABOVE FLANGE)

SHELL STIFFENER

MAIN CYLINDRICAL SHELL (BELOW FLANGE) DOES NOT INCLUDE NOZZLES

COMPRESS FILENAME: BSCUPPER, USL

NOTE: THE 1/2 IN LOWER SHELL INCLUDING NOZZLE G IS ALSO INCLUDED IN THE COMPRESS MODEL.


45.50 [1156]

NOZZLE E (LOCATED ON 11/4 IN SHELL)

RADIAL LOAD FROM VACUUM PRES.

$$F_R = \pi r^2 p$$

r $\frac{13.75}{2} = 6.875$ in INSIDE RADIUS

$p = 14.7$ psi

$$F_R = \pi (6.875)^2 (14.7)$$

$= 2183$ LB

WRC 107 ANAL BY COMPRESS

MAX STRESS IS

$f = 13222$ psi ok
AT EDGE OF REINF. PAD

$f = 4686$ psi
AT NOZZLE O.D.

PAD REQ'D (COMPRESS) FOR NOZ E ON UPPER SECTION

OD = 20" x t = .25 in *

WELDS FROM COMPRESS*

INNER FILLET = .125

OUTER " = .25

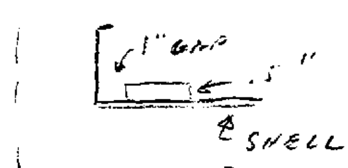
UPPER GROOVE = 0

LOWER GROOVE = .25

* WELDS WERE MODIFIED FOR VACUUM SERVICE SEE V049-1-026

PAD FOR VACUUM SERVICE

*.5 IN USED TO COMPENSATE FOR GAP



NOZZLE G (LOCATED ON 1/2 ID SHELL)

RADIAL LOAD FROM VACUUM PRESSURE

$F_R = \pi r^2 p$

$r = \frac{7.75}{2} = 3.875$

$p = 14.7$ psi

$F_R = \pi (3.875)^2 (14.7)$

$= 693.4$ LB

MAX STRESS FROM WRC 107

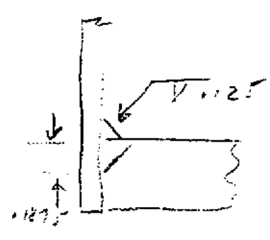
$f = 2378$ psi

NO REINF PAD REQ'D FOR THIS NOZZLE SINCE IT IS ATTACHED TO 1/2" VESSEL

REQUIRED WELDS FROM COMPRESS *

FILLET WELD = .125"

GROOVE WELD = .1875



* WELDS WERE MODIFIED FOR VACUUM VESSEL SERVICE

Top Head - BSC

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Component:	F&D head			
Material specification:	SA 240 304L HIGH			
Internal design pressure:	P = 2	psi @	400	deg F
External design pressure:	Pe= 14.7	psi @	400	deg F
Corrosion allowance: Inner C = 0		Outer= 0		in

PWHT is not performed

Radiography: Category A joints - Spot UW-11(b) type 1
Head to shell seam - Spot UW-11(b) type 1

Estimated weight:	new = 1232.9	corr = 1232.9	lb
capacity:	new = 468.96	corr = 468.96	US ga

ID = 104.25 crown L = 105 knuckle r = 6.5 t = .375 in (min)

Straight flange = 2 forming allowance = 0 in

Design thickness: (At 400 deg F) Appendix 1-4(d) Eq 3

M = .25*(3 + (L/r)^.5)
= .25*(3 + (105/6.5)^.5)
= 1.7548

t = P*L*M/(2*S*E - 0.2*P) + Corrosion + fa
= 2*105*1.7548/(2*14700*0.85 - 0.2*2) + 0 + 0
= 0.0147 in

MAP: (New & at 0 deg F) Appendix 1-4(d) Eq 3

P = 2*S*E*t/(L*M + 0.2*t) - Ps
= 2*16700*0.85*0.375/(105*1.7548 + 0.2*0.375) - 0
= 57.75679 psi

MAWP: (Corroded & at 400 deg F) Appendix 1-4(d) Eq 3

P = 2*S*E*t/(L*M + 0.2*t) - Ps
= 2*14700*0.85*0.375/(105*1.7548 + 0.2*0.375) - 0
= 50.83981 psi

External Pressure: (Corroded & at 400 deg F) UG-33(e)

A = .125/(Ro/t)
= .125/(105.375/0.32589)
= 0.000387

From table HA-3: B = 4755.5

Top Head - BSC

Pa= B/(Ro/t)
= 4755.5/(105.375/0.32589)
= 14.7072 psi

Check the external pressure per UG-33(a)(1)

t = 1.67*Pa*Lo*M/(2*S*E + 1.67*Pa*(M-0.2))
= 1.67*14.7072*105.375*1.7548/(2*14700*1 + 1.67*14.7072*(1.7548-0.2))
= 0.154277 in

Design thickness for external pressure Pa = 14.7072 psi:

= t + Corrosion + fa
= 0.32589 + 0 + 0
= 0.32589 in

Maximum Allowable External Pressure: (Corroded @ 400 deg F)

A = .125/(Ro/t)
= .125/(105.375/0.375)
= 0.000445

From table HA-3: B = 4873.8

Pa= B/(Ro/t)
= 4873.8/(105.375/0.375)
= 17.3445 psi

Check the Maximum External Pressure: UG-33(a)(1) & App. 1-4(d)

Pe= 2*S*E*t/((M*Lo - t*(M-0.2))*1.67)
= 2*14700*1*0.375/((1.7548*105.375 - 0.375*(1.7548-0.2))*1.67)
= 35.81529 psi

The maximum allowable external pressure is 17.3445 psi.

Rev 0
Doc. No. V049-1-015

Top Head - BSC

Analysis for thickness reduced by 1/32"

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Component: F&D head
Material specification: SA 240 304L HIGH

Internal design pressure: P = 2 psi @ 400 deg F
External design pressure: Pe= 14.7 psi @ 400 deg F

Corrosion allowance: Inner C = 0 Outer= 0 in

PWHT is not performed

Radiography: Category A joints - Spot UW-11(b) type 1
Head to shell seam - Spot UW-11(b) type 1

Estimated weight: new = 1128.8 corr = 1128.8 lb
capacity: new = 468.96 corr = 468.96 US ga

ID = 104.25 crown L = 105 knuckle r = 6.5 t = .34375 in (min)
= 11/32

Straight flange = 2 forming allowance = 0 in

WARNING! F&D head geometry is not per UG-32(j).

Crown rad = 105 > OD = 104 5/16
HR

Design thickness: (At 400 deg F) Appendix 1-4(d) Eq 3

M = .25*(3 + (L/r)^.5)
= .25*(3 + (105/6.5)^.5)
= 1.7548

t = P*L*M/(2*S*E - 0.2*P) + Corrosion + fa
= 2*105*1.7548/(2*14700*0.85 - 0.2*2) + 0 + 0
= 0.0147 in

MAP: (New & at 0 deg F) Appendix 1-4(d) Eq 3

P = 2*S*E*t/(L*M + 0.2*t) - Ps
= 2*16700*0.85*0.34375/(105*1.7548 + 0.2*0.34375) - 0
= 52.94551 psi

MAWP: (Corroded & at 400 deg F) Appendix 1-4(d) Eq 3

P = 2*S*E*t/(L*M + 0.2*t) - Ps
= 2*14700*0.85*0.34375/(105*1.7548 + 0.2*0.34375) - 0
= 46.60474 psi

External Pressure: (Corroded & at 400 deg F) UG-33(e)

A = .125/(Ro/t)
= .125/(105.3438/0.32589)

Top Head - BSC

= 0.000387

From table HA-3: B = 4754.8

Pa= B/(Ro/t)
= 4754.8/(105.3438/0.32589)
= 14.7094 psi

Check the external pressure per UG-33(a)(1)

t = 1.67*Pa*Lo*M/(2*S*E + 1.67*Pa*(M-0.2))
= 1.67*14.7094*105.3438*1.7548/(2*14700*1 + 1.67*14.7094*(1.7548-0.2))
= 0.154254 in

Design thickness for external pressure Pa = 14.7094 psi:

= t + Corrosion + fa
= 0.32589 + 0 + 0
= 0.32589 in

Maximum Allowable External Pressure: (Corroded @ 400 deg F)

A = .125/(Ro/t)
= .125/(105.3438/0.34375)
= 0.000408

From table HA-3: B = 4797.2

Pa= B/(Ro/t)
= 4797.2/(105.3438/0.34375)
= 15.6539 psi

Check the Maximum External Pressure: UG-33(a)(1) & App. 1-4(d)

Pe= 2*S*E*t/((M*Lo - t*(M-0.2))*1.67)
= 2*14700*1*0.34375/((1.7548*105.3438 - 0.34375*(1.7548-0.2))*1.67)
= 32.83179 psi

The maximum allowable external pressure is 15.6539 psi.

Upper Cylindrical Shell - BSC

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Component:	Cylinder			
Material specification:	SA 240 304L HIGH			
Internal design pressure:	P = 2	psi @	400	deg F
External design pressure:	Pe= 14.7	psi @	400	deg F
Corrosion allowance: Inner C = 0		Outer= 0		in

PWHT is not performed

Radiography:	Category A joints -	Spot UW-11(b) type 1
	Category B joints -	Spot UW-11(b) type 1

Estimated weight:	new = 1428.1	corr = 1428.1	lb
capacity:	new = 2217.078	corr = 2217.078	US ga

ID = 104.25 length Lc= 60 t = 0.25 in (new)

Design thickness: UG-27(c)(1) Circ. stress

t = P*R/(S*E - 0.6*P) + Corrosion
= 2*52.125/(14700*0.85 - 0.6*2) + 0
= 0.0083 in

MAP: (New & at 0 deg F) UG-27(c)(1)

P = S*E*t/(R + 0.6*t) - Ps
= 16700*0.85*0.25/(52.125 + 0.6*0.25) - 0
= 67.88618 psi

MAWP: (Corroded & at 400 deg F) UG-27(c)(1)

P = S*E*t/(R + 0.6*t) - Ps
= 14700*0.85*0.25/(52.125 + 0.6*0.25) - 0
= 59.7561 psi

External Pressure: (Corroded & at 400 deg F) UG-28

L/Do = 37.90131/104.75 =0.3618 Do/t = 104.75/0.23897 = 438.3395
From table G: A = 0.000425
From table HA-3: B = 4834.6

Pa= 4*B/(3*Do/t)
= 4*4834.6/(3*104.75/0.23897)
= 14.7058 psi

Design thickness for external pressure Pa = 14.7058 psi:

= t + Corrosion
= 0.23897 + 0

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Upper Cylindrical Shell - BSC

= 0.23897 in

Maximum Allowable External Pressure: (Corroded @ 400 deg F)

L/Do = 37.90131/104.75 =0.3618 Do/t = 104.75/0.25 = 419
From table G: A = 0.000452
From table HA-3: B = 4887.2

Pa= 4*B/(3*Do/t)
= 4*4887.2/(3*104.75/0.25)
= 15.5519 psi

Shell Stiffener

Stiffening Ring Calculations Per UG-29

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Identifier:		Shell Stiffener
Ring material specification:		SA 240 304L HIGH
Number of rings in this group:		1
Distance first ring to datum line:		110 in
Ring spacing:		0 in
Ring description:		4x4x1/4 Equal Angle
Ring is rolled:		leg out (easy way)
Ring cross sectional area:	As =	1.94 in^2
Ring moment of inertia:	Ir =	3.04 in^4

Calculations for ring 110 in from datum

Shell material specification:		SA 240 304L HIGH
Required shell thickness:	t =	0.23897 in
Corroded shell thickness:	ts =	0.25 in
Shell outer diameter:	Do =	104.75 in
Design temperature:	=	400 deg F
External design pressure:	P =	14.7 psi
Stiffener supported length:	Ls =	33.95065 in

B = .75*(P*Do/(t + As/Ls))
= .75*(14.7*104.75/(0.23897 + 1.94/33.95065))
= 3900.111

From table HA-3 (ring) A = 2.949634E-04

Required moment of inertia of the combined ring-shell section

Is= (Do^2*Ls*(t + As/Ls)*A)/10.9
= (104.75^2*33.95065*(0.23897 + 1.94/33.95065)*2.949634E-04)/10.9
= 2.985063 in^4

Available moment of inertia of the combined ring-shell section

Shell width contributing smaller of = 5.62911

W = 1.1*Sqr(Do*ts)
= 1.1*Sqr(104.75*0.25)
= 5.62911 in

W = Ls = 33.95065 in

Shell area A1 = W*ts = 1.407277 in^2

Distance to the ring neutral axis

Shell Stiffener

Y2 = Ring NA + ts/2
= 1.09 + 0.25/2
= 1.215 in

Neutral axis of combined section

NA = As*Y2/(A1 + As)
= 1.94*1.215/(1.407277 + 1.94)
= .7041842 in

Inertia of the shell about the combined section NA

I1 = W*ts^3/12 + A1*NA^2
= 5.62911*0.25^3/12 + 1.407277*0.7041842^2
= .7051639 in^4

Inertia of the ring about the combined section NA

I2 = Ir + As*(NA - Y2)^2
= 3.04 + 1.94*(0.7041842 - 1.215)^2
= 3.54621 in^4

Total available I = I1 + I2 = 4.251373 in^4

The 4x4x1/4 Equal Angle vacuum stiffener is satisfactory.

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RING STIFFENER CONNECTION WELD - PER ASME VIII -1, UG-30

USE INTERMITTENT WELD STAGGERED, 2 IN @ 4 IN

MAX CLEAR SPACING

$S = 2\ IN = 8t$ FOR EXT. RING - OK

$t = .25\ IN$ SHELL THICKNESS

1) RADIAL PRESS LOAD

$$PL_s = 14.7(30) = 441\ LB/IN$$

2) RADIAL SHEAR LOAD

$$.01\ PL_s\ D_o = .01(441)(105)$$
$$= 463\ LB$$

FOR A 4 IN SPACING THE TOTAL RADIAL PRESS LOAD IS

$$F = 4\ PL_s = 4(441) = 1764\ LB$$

WELD AREA FOR $t = 3/16 =$

$$A_w = 2 \times 2 \times .707(3/16) = .53\ IN^2$$

WELD SHEAR STRESS

$$f_v = \frac{F}{A_w} = \frac{1764}{.53} = 3300\ psi\ \text{OK}$$
$$< .55(14700) = 8085\ psi$$

MIN WELD SIZE IS 1/4" (UG 30f)

∴ USE 1/4 IN 2 IN @ 4 IN STAGGERED

Main Section Cylinder

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Component:	Cylinder			
Material specification:	SA 240 304L HIGH			
Internal design pressure:	P = 2	psi @	400	deg F
External design pressure:	Pe= 14.7	psi @	400	deg F

Corrosion allowance: Inner C = 0 Outer= 0 in

PWHT is not performed

Radiography:	Category A joints -	Spot UW-11(b) type 1
	Category B joints -	Spot UW-11(b) type 1

Estimated weight:	new = 3817.3	corr = 3817.3	lb
capacity:	new = 2956.104	corr = 2956.104	US ga

ID = 104.25 length Lc= 80 t = 0.5 in (new)

Design thickness: UG-27(c)(1) Circ. stress

t = P*R/(S*E - 0.6*P) + Corrosion
= 2*52.125/(14700*0.85 - 0.6*2) + 0
= 0.0083 in

MAP: (New & at 0 deg F) UG-27(c)(1)

P = S*E*t/(R + 0.6*t) - Ps
= 16700*0.85*0.5/(52.125 + 0.6*0.5) - 0
= 135.3839 psi

MAWP: (Corroded & at 400 deg F) UG-27(c)(1)

P = S*E*t/(R + 0.6*t) - Ps
= 14700*0.85*0.5/(52.125 + 0.6*0.5) - 0
= 119.1702 psi

External Pressure: (Corroded & at 400 deg F) UG-28

L/Do = 80/105.25 =0.7601 Do/t = 105.25/0.30747 = 342.3098
From table G: A = 0.000288
From table HA-3: B = 3807.2

Pa= 4*B/(3*Do/t)
= 4*3807.2/(3*105.25/0.30747)
= 14.8295 psi

Design thickness for external pressure Pa = 14.8295 psi:

= t + Corrosion
= 0.30747 + 0

Main Section Cylinder

= 0.30747 in

Maximum Allowable External Pressure: (Corroded @ 400 deg F)

L/Do = 80/105.25 =0.7601 Do/t = 105.25/0.5 = 210.5
From table G: A = 0.000587
From table HA-3: B = 5117.2

Pa= 4*B/(3*Do/t)
= 4*5117.2/(3*105.25/0.5)
= 32.413 psi

Nozzle E

Opening N-E Reinforcement Calculations Per UG-37

Located on:	Upper Cylindrical Shell - BSC
Local vessel thickness:	.25 in
Liquid static head included:	0 psi
Flange description:	14 inch 75# SO A240304L
ANSI B16.47 flange rating MAP:	115 psi
ANSI B16.47 flange rating MAWP:	80 psi

Nozzle material specification:	SA 240 304L HIGH
Pad material specification:	SA 240 304L HIGH

Nozzle orientation:	45 degrees
End of nozzle to shell center:	55.875 in
Nozzle offset from center Lo:	0 in
Projection outside vessel Lpr:	3.5 in

```
     tn ->| |<-
dp->|     _        _  |<-    corrosion allow = 0 in
    |    | |<- d ->| |twl|   noz thick new tn= .12 in
->| |<tw2|  |      |>| |<-|  nozzle id. new d= 13.75 in
  |  ___/|  |      | |\___   pad diameter dp = 20 in
__|      |  |      | |    |\ pad thickness te= .25 in
te|/|    |  |      | |       fillet weld twl = .125 in
_________|_|_v     | |______ fillet weld tw2 = .25 in
________/|_|_tw3   |_|\_____ groove weld tw3 = .25 in
             ^
<---- L ---->|               To datum L= 124 in
```

Reinforcement Calculations For Nozzle MAWP

Limits of reinforcement UG-40

Parallel to the vessel wall d = 13.75 in
Normal to the vessel wall outside 2.5*(tn-Cn) + te = .55 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .3 in

Nozzle required thickness

trn = P*Rn/(Sn*E - 0.6*P)
= 2*6.875/(14700*1 - 0.6*2)
= 0.0009 in

Required thickness tr from UG-37(a)

tr= P*R/(S*E - 0.6*P)
= 2*52.125/(14700*1 - 0.6*2)
= 0.0071 in

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Area required

Nozzle E

Allowable stresses: Sn = 14700, Sv = 14700, Sp = 14700 psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1
fr2 = lesser of 1 or Sn/Sv so fr2 =1
fr3 = lesser of fr2 or Sp/Sv so fr3 =1
fr4 = lesser of 1 or Sp/Sv so fr4 =1

A = d*tr*F + 2*tn*tr*F*(1 - fr1)
= 13.75*0.0071*1 + 2*0.12*0.0071*1*(1 - 1)
= .0976 in^2

Area available

A1 = larger of the following = 3.34 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 13.75*(1*0.25-1*0.0071) - 2*0.12*(1*0.25-1*0.0071)*(1-1)
= 3.34 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.25+0.12)*(1*0.25-1*0.0071) - 2*0.12*(1*0.25-1*0.0071)*(1-1)
= .18 in^2

A2 = smaller of the following = 0.131 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.12 - 0.0009)*1*0.25
= .149 in^2

= 2*(tn - trn)*(2.5*tn + te)*fr2
= 2*(0.12 - 0.0009)*(2.5*0.12 + 0.25)*1
= .131 in^2

A41 = Leg^2*fr3
= 0.125^2*1 = .016 in^2

A42 = Leg^2*fr4
= 0.25^2*1 = .063 in^2

A5 = (Dp - d - 2*tn)*te*fr4
= (20 - 13.75 - 2*0.12)*0.25*1
= 1.5025 in^2

Area = A1 + A2 + A41 + A42 + A5
= 3.34 + 0.131 + 0.016 + 0.063 + 1.5025
= 5.0525 in^2

As Area > A the reinforcement is adequate for MAWP = 2 at 400 Deg F

Check the welds - From UW-16(c)(2)

Inner Fillet: tmin = lesser of 0.75 or tn or te, tmin = 0.12 in
tw(min) = 0.7*tmin = 0.084 in

Nozzle E

tw(actual) = 0.7*Leg = 0.7*0.125 = 0.0875 in

Outer Fillet: tmin = lesser of 0.75 or te or t, tmin = 0.25 in
tw(min) = 0.5*tmin = 0.125 in
tw(actual) = 0.7*Leg = 0.7*0.25 = 0.175 in

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.0009 in (E = 1)
Wall thickness per UG-45(b)(1):	tr2 = 0.0071 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.328125 in
The greater of tr2 or tr3:	tr5 = 0.0625 in
The lesser of tr4 or tr5:	tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.12 in

The nozzle neck thickness is adequate for MAWP.

Allowable stresses in joints UG-45(c) and UW-15(c)

Groove weld in tension = 0.74*14700 = 10878 psi
Nozzle wall in shear = 0.7*14700 = 10290 psi
Inner fillet weld in shear = 0.49*14700 = 7203 psi
Outer fillet weld in shear = 0.49*14700 = 7203 psi

Strength of welded joints:

(1) Inner fillet weld in shear
(Pi/2)*Nozzle O.D.*Leg*Si = 1.57*13.99*0.125*7203 = 19776.11 lbf

(2) Outer fillet weld in shear
(Pi/2)*Pad O.D.*Leg*So = 1.57*20*0.25*7203 = 56543.55 lbf

(3) Nozzle wall in shear
(Pi/2)*Mean nozzle dia.*tn*Sn = 1.57*13.87*0.12*10290 = 26888.88 lbf

(4) Groove weld in tension
(Pi/2)*Nozzle O.D.*tw*Sg = 1.57*13.99*0.25*10878 = 59731.91 lbf

Loading on welds per UG-41(b)(1)

W = (A - (d - 2*tn)*(E1*t - F*tr))*Sv
= (0.0976 - (13.75 - 2*0.12)*(1*0.25 - 1*0.0071))*14700
=-46804.49 lbf

W1-1 = (A2 + A5 + A41 + A42)*Sv
= (0.131 + 1.5025 + 0.016 + 0.063)*14700
= 25173.75 lbf

W2-2 = (A2 + A3 + A41 + A43 + 2*tn*t*fr1)*Sv

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Nozzle E

= (0.131 + 0 + 0.016 + 0 + 2*0.12*0.25*1)*14700
= 3042.9 lbf

W3-3 = (A2 + A3 + A5 + A41 + A42 + A43 + 2*tn*t*fr1)*Sv
= (0.131 + 0 + 1.5025 + 0.016 + 0.063 + 0 + 2*0.12*0.25*1)*14700
= 26055.75 lbf

Load for path 1-1 lesser of W or W1-1 =-46804.49 lbf
Path 1-1 Thru (2) & (3) = 56543.55 + 26888.88 = 83432.43 lbf
Path 1-1 is stronger than W so it is acceptable per UG-41(b)(2).

Load for path 2-2 lesser of W or W2-2 =-46804.49 lbf
Path 2-2 Thru (1), (4) = 19776.11 + 59731.91 = 79508.02 lbf
Path 2-2 is stronger than W so it is acceptable per UG-41(b)(2).

Load for path 3-3 lesser of W or W3-3 =-46804.49 lbf
Path 3-3 Thru (2), (4) = 56543.55 + 59731.91 = 116275.5 lbf
Path 3-3 is stronger than W so it is acceptable per UG-41(b)(2).

Pad strength = A5*Sp = 22086.75 lbf
Outer fillet weld strength is adequate.

Reinforcement Calculations for External Pressure

Limits of reinforcement UG-40

Parallel to the vessel wall d = 13.75 in
Normal to the vessel wall outside 2.5*(tn-Cn) + te = .55 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .3 in

Nozzle required thickness

L/Do = 3.5/13.99 = .2502 Do/t = 13.99/0.03032 = 461.4116
From table G: A = 0.000587
From table HA-3: B = 5117.2

Pa= 4*B/(3*Do/t)
= 4*5117.2/(3*13.99/0.03032)
= 14.7871 psi

Nozzle required thickness trn = .03032 in

Required thickness tr from UG-37(d)(1) = .239 in

Area required

Allowable stresses: Sn = 14700, Sv = 14700, Sp = 14700 psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1
fr2 = lesser of 1 or Sn/Sv so fr2 =1
fr3 = lesser of fr2 or Sp/Sv so fr3 =1
fr4 = lesser of 1 or Sp/Sv so fr4 =1

Nozzle E

A = 0.5*(d*tr*F + 2*tn*tr*F*(1 - fr1))
= 0.5*(13.75*0.239*1 + 2*0.12*0.239*1*(1 - 1))
= 1.6431 in^2

Area available

A1 = larger of the following = .151 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 13.75*(1*0.25-1*0.239) - 2*0.12*(1*0.25-1*0.239)*(1-1)
= .151 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.25+0.12)*(1*0.25-1*0.239) - 2*0.12*(1*0.25-1*0.239)*(1-1)
= .008 in^2

A2 = smaller of the following = 0.099 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.12 - 0.03032)*1*0.25
= .112 in^2

= 2*(tn - trn)*(2.5*tn + te)*fr2
= 2*(0.12 - 0.03032)*(2.5*0.12 + 0.25)*1
= .099 in^2

A41 = Leg^2*fr3
= 0.125^2*1 = .016 in^2

A42 = Leg^2*fr4
= 0.25^2*1 = .063 in^2

A5 = (Dp - d - 2*tn)*te*fr4
= (20 - 13.75 - 2*0.12)*0.25*1
= 1.5025 in^2

Area = A1 + A2 + A41 + A42 + A5
= 0.151 + 0.099 + 0.016 + 0.063 + 1.5025
= 1.8315 in^2

As Area > A the reinforcement is adequate for Pe = 14.7 at 400 Deg F

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.03032 in (E = 1)
Wall thickness per UG-45(b)(2):	tr2 = 0.0522 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.328125 in
The greater of tr2 or tr3:	tr5 = 0.0625 in
The lesser of tr4 or tr5:	tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

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Nozzle E

Available nozzle wall thickness new, tn = 0.12 in

The nozzle neck thickness is adequate for Pe.

Nozzle E

Applied Loads

Radial load	Pr = 2183 lbf
Circumferential moment	Mc = 0 lbf-ft
Circumferential shear	Vc = 0 lbf
Longitudinal moment	ML = 0 lbf-ft
Longitudinal shear	VL = 0 lbf
Torsion moment	Mt = 0 lbf-ft
Internal pressure	P = 2 psi

Stresses at the nozzle OD per WRC bulletin 107 (psi)

Mean radius Rm = 52.25 in
Rm/t = 104.5

Stress concentration factor Kn (tension) = 1
Stress concentration factor Kb (bending) = 1

Pressure stress intensity factor, Farr equation 11.5

I = .25*(4 + 3*(r/x)^2 + 3*(r/x)^4)
= .25*(4 + 3*(6.875/7.12)^2 + 3*(6.875/7.12)^4)
= 2.351

Local circ. pressure stress = I*P*Rm/t = 491 psi

Local long. pressure stress = P*Rm/2t = 104 psi

Maximum combined stress =-4686 psi
Allowable combined stress = +-3*S = +- 44100 psi

The maximum combined stress is within allowable limits.

Maximum primary membrane stress =-1154 psi
Allowable primary membrane stress = +-1.5*S = +- 22050 psi

The maximum primary membrane stress is within allowable limits.

Nozzle E

From Fig.	Value read	beta	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
3C*	10.677	0.117					-892	-892	-892	-892
4C*	15.057	0.117	-1258	-1258	-1258	-1258				
1C	0.0765	0.117					-4008	4008	-4008	4008
2C-1	0.0440	0.117	-2305	2305	-2305	2305				
3A*	4.1315	0.117								
1A	0.0774	0.117								
3B*	11.036	0.117								
1B-1	0.0281	0.117								
pressure stress*			491	491	491	491	491	491	491	491
Total circ stress			-3072	1538	-3072	1538	-4409	3607	-4409	3607
Primary membrane circ stress*			-767	-767	-767	-767	-401	-401	-401	-401
3C*	10.677	0.117	-892	-892	-892	-892				
4C*	15.057	0.117					-1258	-1258	-1258	-1258
1C-1	0.0744	0.117	-3898	3898	-3898	3898				
2C	0.0467	0.117					-2447	2447	-2447	2447
4A*	7.5665	0.117								
2A	0.0389	0.117								
4B*	4.0872	0.117								
2B-1	0.0385	0.117								
pressure stress*			104	104	104	104	104	104	104	104
Total long stress			-4686	3110	-4686	3110	-3601	1293	-3601	1293
Primary membrane long stress*			-788	-788	-788	-788	-1154	-1154	-1154	-1154
torsion moment Mt										
Circ shear from Vc										
Long shear from VL										
Total Shear stress										
Combined stress			-4686	3110	-4686	3110	-4409	3607	-4409	3607

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Nozzle E

Stresses at the pad edge per WRC bulletin 107 (psi)

Mean radius Rm = 52.25 in
Rm/t = 209

Stress concentration factor Kn (tension) = 1
Stress concentration factor Kb (bending) = 1

Pressure stress intensity factor, Farr equation 11.5

I = .25*(4 + 3*(r/x)^2 + 3*(r/x)^4)
= .25*(4 + 3*(6.875/10.25)^2 + 3*(6.875/10.25)^4)
= 1.489

Local circ. pressure stress = I*P*Rm/t = 622 psi

Local long. pressure stress = P*Rm/2t = 209 psi

Maximum combined stress =-13222 psi
Allowable combined stress = +-3*S = +- 44100 psi

The maximum combined stress is within allowable limits.

Maximum primary membrane stress =-3300 psi
Allowable primary membrane stress = +-1.5*S = +- 22050 psi

The maximum primary membrane stress is within allowable limits.

Nozzle E

From Fig.	Value read	beta	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
3C*	8.2255	0.167					-1375	-1375	-1375	-1375
4C*	20.996	0.167	-3509	-3509	-3509	-3509				
1C	0.0595	0.167					-12469	12469	-12469	12469
2C-1	0.0148	0.167	-3102	3102	-3102	3102				
3A*	6.4789	0.167								
1A	0.0553	0.167								
3B*	13.437	0.167								
1B-1	0.0104	0.167								
pressure stress*			622	622	622	622	622	622	622	622
Total circ stress			-5989	215	-5989	215	-13222	11716	-13222	11716
Primary membrane circ stress*			-2887	-2887	-2887	-2887	-753	-753	-753	-753
3C*	8.2255	0.167	-1375	-1375	-1375	-1375				
4C*	20.996	0.167					-3509	-3509	-3509	-3509
1C-1	0.0390	0.167	-8173	8173	-8173	8173				
2C	0.0353	0.167					-7398	7398	-7398	7398
4A*	17.794	0.167								
2A	0.0238	0.167								
4B*	5.7771	0.167								
2B-1	0.0144	0.167								
pressure stress*			209	209	209	209	209	209	209	209
Total long stress			-9339	7007	-9339	7007	-10698	4098	-10698	4098
Primary membrane long stress*			-1166	-1166	-1166	-1166	-3300	-3300	-3300	-3300
torsion moment Mt										
Circ shear from Vc										
Long shear from VL										
Total Shear stress										
Combined stress			-9339	7007	-9339	7007	-13222	11716	-13222	11716

Nozzle G

Opening N-G Reinforcement Calculations Per UG-37

Located on:	Main Section Cylinder
Local vessel thickness:	.5 in
Liquid static head included:	0 psi
Flange description:	8 inch 75# WN A240304L
ANSI B16.47 flange rating MAP:	115 psi
ANSI B16.47 flange rating MAWP:	80 psi
Nozzle material specification:	SA 240 304L HIGH
Nozzle orientation:	90 degrees
End of nozzle to shell center:	55.625 in
Nozzle offset from center Lo:	0 in
Projection outside vessel Lpr:	3 in

```
   tn ->| |<-
                _            _
        | |<- d ->| |          corrosion allow = 0 in
twl->|  |<        | |          noz thick new tn= .12 in
     |/ |   v     | |\         nozzle id. new d= 7.75 in
_____.-|  |—      | |-.____    fillet weld twl = .125 in
______\|_|_tw2    |_|/______   groove weld tw2 = .1875 in
            ^
<—— L ——>|                     To datum L= 40 in
```

Reinforcement Calculations For Nozzle MAWP

Limits of reinforcement UG-40

Parallel to the vessel wall d = 7.75 in
Normal to the vessel wall outside 2.5*(tn-Cn) + te = .3 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .3 in

Nozzle required thickness

trn = P*Rn/(Sn*E - 0.6*P)
= 2*3.875/(14700*1 - 0.6*2)
= 0.0005 in

Required thickness tr from UG-37(a)

tr= P*R/(S*E - 0.6*P)
= 2*52.125/(14700*1 - 0.6*2)
= 0.0071 in

Area required

Allowable stresses: Sn = 14700, Sv = 14700, psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1

Nozzle G

fr2 = lesser of 1 or Sn/Sv so fr2 = 1

A = d*tr*F + 2*tn*tr*F*(1 - fr1)
= 7.75*0.0071*1 + 2*0.12*0.0071*1*(1 - 1)
= .055 in^2

Area available

A1 = larger of the following = 3.82 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 7.75*(1*0.5-1*0.0071) - 2*0.12*(1*0.5-1*0.0071)*(1-1)
= 3.82 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.5+0.12)*(1*0.5-1*0.0071) - 2*0.12*(1*0.5-1*0.0071)*(1-1)
= .611 in^2

A2 = smaller of the following = 0.072 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.12 - 0.0005)*1*0.5
= .299 in^2

= 5*(tn - trn)*fr2*tn
= 5*(0.12 - 0.0005)*1*0.12
= .072 in^2

A41 = Leg^2*fr2
= 0.125^2*1 = .016 in^2

Area = A1 + A2 + A41
= 3.82 + 0.072 + 0.016
= 3.908 in^2

As Area > A the reinforcement is adequate for MAWP = 2 at 400 Deg F

Check the welds - From UW-16(d):

tmin = lesser of 0.75 or tn or t, tmin = 0.12 in
t1 or t2(min) = lesser of 0.25 or 0.7*tmin, t1(min) = 0.084 in
t1(actual) = 0.7*Leg = 0.7*0.125 = 0.0875 in
t2(actual) = 0.1875 in
t1 + t2 = 0.275 >= 1.25*tmin

The weld sizes for t1 and t2 are satisfactory.

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.0005 in (E = 1)
Wall thickness per UG-45(b)(1):	tr2 = 0.0071 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.28175 in

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Nozzle G

The greater of tr2 or tr3: tr5 = 0.0625 in
The lesser of tr4 or tr5: tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.12 in

The nozzle neck thickness is adequate for MAWP.

Allowable stresses in joints UG-45(c) and UW-15(c)

Groove weld in tension = 0.74*14700 = 10878 psi
Nozzle wall in shear = 0.7*14700 = 10290 psi
Inner fillet weld in shear = 0.49*14700 = 7203 psi

Strength of welded joints:

(1) Inner fillet weld in shear
(Pi/2)*Nozzle O.D.*Leg*Si = 1.57*7.99*0.125*7203 = 11294.57 lbf

(3) Nozzle wall in shear
(Pi/2)*Mean nozzle dia.*tn*Sn = 1.57*7.87*0.12*10290 = 15257.07 lbf

(4) Groove weld in tension
(Pi/2)*Nozzle O.D.*tw*Sg = 1.57*7.99*0.1875*10878 = 25585.67 lbf

Loading on welds per UG-41(b)(1)

W = (A - (d - 2*tn)*(E1*t - F*tr))*Sv
= (0.055 - (7.75 - 2*0.12)*(1*0.5 - 1*0.0071))*14700
=-53606.18 lbf

W1-1 = (A2 + A5 + A41 + A42)*Sv
= (0.072 + 0 + 0.016 + 0)*14700
= 1293.6 lbf

W2-2 = (A2 + A3 + A41 + A43 + 2*tn*t*fr1)*Sv
= (0.072 + 0 + 0.016 + 0 + 2*0.12*0.5*1)*14700
= 3057.6 lbf

Load for path 1-1 lesser of W or W1-1 =-53606.18 lbf
Path 1-1 Thru (1) & (3) = 11294.57 + 15257.07 = 26551.64 lbf
Path 1-1 is stronger than W so it is acceptable per UG-41(b)(2).

Load for path 2-2 lesser of W or W2-2 =-53606.18 lbf
Path 2-2 Thru (1), (4) = 11294.57 + 25585.67 = 36880.24 lbf
Path 2-2 is stronger than W so it is acceptable per UG-41(b)(2).

Reinforcement Calculations for External Pressure

Limits of reinforcement UG-40

Parallel to the vessel wall d = 7.75 in

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Nozzle G

Normal to the vessel wall outside 2.5*(tn-Cn) + te = .3 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .3 in

Nozzle required thickness

L/Do = 3/7.99 = .3755 Do/t = 7.99/0.01833 = 435.8974
From table G: A = 0.000412
From table HA-3: B = 4808.2

Pa= 4*B/(3*Do/t)
= 4*4808.2/(3*7.99/0.01833)
= 14.7074 psi

Nozzle required thickness trn = .01833 in

Required thickness tr from UG-37(d)(1) = .3075 in

Area required

Allowable stresses: Sn = 14700, Sv = 14700, psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1
fr2 = lesser of 1 or Sn/Sv so fr2 =1

A = 0.5*(d*tr*F + 2*tn*tr*F*(1 - fr1))
= 0.5*(7.75*0.3075*1 + 2*0.12*0.3075*1*(1 - 1))
= 1.1916 in^2

Area available

A1 = larger of the following = 1.492 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 7.75*(1*0.5-1*0.3075) - 2*0.12*(1*0.5-1*0.3075)*(1-1)
= 1.492 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.5+0.12)*(1*0.5-1*0.3075) - 2*0.12*(1*0.5-1*0.3075)*(1-1)
= .239 in^2

A2 = smaller of the following = 0.061 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.12 - 0.01833)*1*0.5
= .254 in^2

= 5*(tn - trn)*fr2*tn
= 5*(0.12 - 0.01833)*1*0.12
= .061 in^2

A41 = Leg^2*fr2
= 0.125^2*1 = .016 in^2

Nozzle G

Area = A1 + A2 + A41
= 1.492 + 0.061 + 0.016
= 1.569 in^2

As Area > A the reinforcement is adequate for Pe = 14.7 at 400 Deg F

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.01833 in (E = 1)
Wall thickness per UG-45(b)(2):	tr2 = 0.0522 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.28175 in
The greater of tr2 or tr3:	tr5 = 0.0625 in
The lesser of tr4 or tr5:	tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.12 in

The nozzle neck thickness is adequate for Pe.

Nozzle G

Applied Loads

Radial load	Pr = 693.4 lbf
Circumferential moment	Mc = 0 lbf-ft
Circumferential shear	Vc = 0 lbf
Longitudinal moment	ML =0 lbf-ft
Longitudinal shear	VL = 0 lbf
Torsion moment	Mt = 0 lbf-ft
Internal pressure	P = 2 psi

Stresses at the nozzle OD per WRC bulletin 107 (psi)

Mean radius Rm = 52.375 in
Rm/t = 104.75

Stress concentration factor Kn (tension) = 1
Stress concentration factor Kb (bending) = 1

Pressure stress intensity factor, Farr equation 11.5

I = .25*(4 + 3*(r/x)^2 + 3*(r/x)^4)
= .25*(4 + 3*(3.875/4.12)^2 + 3*(3.875/4.12)^4)
= 2.25

Local circ. pressure stress = I*P*Rm/t = 471 psi

Local long. pressure stress = P*Rm/2t = 105 psi

Maximum combined stress =-2379 psi
Allowable combined stress = +-3*S = +- 44100 psi

The maximum combined stress is within allowable limits.

Maximum primary membrane stress =-364 psi
Allowable primary membrane stress = +-1.5*S = +- 22050 psi

The maximum primary membrane stress is within allowable limits.

Nozzle G

From Fig.	Value read	beta	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
3C*	16.021	0.067					-424	-424	-424	-424
4C*	17.720	0.067	-469	-469	-469	-469				
1C	0.1209	0.067					-2012	2012	-2012	2012
2C-1	0.0837	0.067	-1393	1393	-1393	1393				
3A*	3.2919	0.067								
1A	0.0926	0.067								
3B*	10.724	0.067								
1B-1	0.0440	0.067								
pressure stress*			471	471	471	471	471	471	471	471
Total circ stress			-1391	1395	-1391	1395	-1965	2059	-1965	2059
Primary membrane circ stress*			2	2	2	2	47	47	47	47
3C*	16.021	0.067	-424	-424	-424	-424				
4C*	17.720	0.067					-469	-469	-469	-469
1C-1	0.1238	0.067	-2060	2060	-2060	2060				
2C	0.0837	0.067					-1393	1393	-1393	1393
4A*	4.8183	0.067								
2A	0.0535	0.067								
4B*	3.2756	0.067								
2B-1	0.0696	0.067								
pressure stress*			105	105	105	105	105	105	105	105
Total long stress			-2379	1741	-2379	1741	-1757	1029	-1757	1029
Primary membrane long stress*			-319	-319	-319	-319	-364	-364	-364	-364
torsion moment Mt										
Circ shear from Vc										
Long shear from VL										
Total Shear stress										
Combined stress			-2379	1741	-2379	1741	-1965	2059	-1965	2059

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MAXIMUM ALLOWABLE COMPRESSIVE STRESS IN UPPER SECTION CYLINDRICAL SHELL

REF: UG-23(b)

$$A = \frac{.125}{(R_o/t)}$$

$R_o = 52.375$ IN

$t = .25$ IN

$A = 5.97(10)^{-4}$

$B = 5750$ (FIG HA-1 FOR TP304 @ 400°F SECT. II, P. 680)

LONGITUDINAL STRESS

$$\sigma_\ell = \frac{Pr}{2t}$$

$$= \frac{14.7(52.375)}{2(.25)}$$

$= 1540$ psi $< B$ OK $= 5750$ psi

REQUIREMENT OF UG-23(b) IS MET CYLINDRICAL SHELL

105 IN ID SHELL *

HEAD FOR 105 ID SHELL

DIMENSIONS FROM SECT VIII, DIV 1, UG-32(e)

CROWN RADIUS (INSIDE)

ICR = OD OF SHELL

= 105 + 2t

t = .375

ICR = 105 + 2(.375) = 105.75

KNUCKLE RADIUS = 6% OF ICR

= .06 (105.75) = 6.345 IN

COMPRESS ANALYSIS OF BSC WITH AN
INCREASED ID = 105 IN FOLLOWS

FILENAME: BSC105ID.VSL

RESULTS SHOW THAT CODE REQUIREMENTS ARE
MET FOR THE INCREASED ID.

* FINAL ID IS 104.5 IN

Top Head - BSC

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Component:	F&D head			
Material specification:	SA 240 304L HIGH			
Internal design pressure:	P = 2	psi @	400	deg F
External design pressure:	Pe= 14.7	psi @	400	deg F
Corrosion allowance: Inner C = 0		Outer= 0		in

PWHT is not performed

Radiography:	Category A joints -	Spot UW-11(b) type 1
	Head to shell seam -	Spot UW-11(b) type 1

Estimated weight:	new = 1140.8	corr = 1140.8	lb
capacity:	new = 474.64	corr = 474.64	US ga

ID = 105 crown L = 105.75 knuckle r = 6.345 t = .34375 in (min) 3/8 - 1/32 FOR THINNING

Straight flange = 2 forming allowance = 0 in

WARNING! F&D head geometry is not per UG-32(j).

Design thickness: (At 400 deg F) Appendix 1-4(d) Eq 3

M = .25*(3 + (L/r)^.5)
= .25*(3 + (105.75/6.345)^.5)
= 1.7706

t = P*L*M/(2*S*E - 0.2*P) + Corrosion + fa
= 2*105.75*1.7706/(2*14700*0.85 - 0.2*2) + 0 + 0
= 0.015 in

MAP: (New & at 0 deg F) Appendix 1-4(d) Eq 3

P = 2*S*E*t/(L*M + 0.2*t) - Ps
= 2*16700*0.85*0.34375/(105.75*1.7706 + 0.2*0.34375) - 0
= 52.10122 psi

MAWP: (Corroded & at 400 deg F) Appendix 1-4(d) Eq 3

P = 2*S*E*t/(L*M + 0.2*t) - Ps
= 2*14700*0.85*0.34375/(105.75*1.7706 + 0.2*0.34375) - 0
= 45.86155 psi

External Pressure: (Corroded & at 400 deg F) UG-33(e)

A = .125/(Ro/t)
= .125/(106.0938/0.32873)
= 0.000387

Top Head - BSC

From table HA-3: B = 4755.5

Pa= B/(Ro/t)
= 4755.5/(106.0938/0.32873)
= 14.7348 psi

Check the external pressure per UG-33(a)(1)

t = 1.67*Pa*Lo*M/(2*S*E + 1.67*Pa*(M-0.2))
= 1.67*14.7348*106.0938*1.7706/(2*14700*1 + 1.67*14.7348*(1.7706-0.2))
= 0.15702 in

Design thickness for external pressure Pa = 14.7348 psi:

= t + Corrosion + fa
= 0.32873 + 0 + 0
= 0.32873 in

Maximum Allowable External Pressure: (Corroded @ 400 deg F)

A = .125/(Ro/t)
= .125/(106.0938/0.34375)
= 0.000405

From table HA-3: B = 4793.7

Pa= B/(Ro/t)
= 4793.7/(106.0938/0.34375)
= 15.5319 psi

Check the Maximum External Pressure: UG-33(a)(1) & App. 1-4(d)

Pe= 2*S*E*t/((M*Lo - t*(M-0.2))*1.67)
= 2*14700*1*0.34375/((1.7706*106.0938 - 0.34375*(1.7706-0.2))*1.67)
= 32.30824 psi

The maximum allowable external pressure is 15.5319 psi.

Upper Cylindrical Shell - BSC

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Component:	Cylinder			
Material specification:	SA 240 304L HIGH			
Internal design pressure:	P = 2	psi @	400	deg F
External design pressure:	Pe= 14.7	psi @	400	deg F

Corrosion allowance: Inner C = 0 Outer= 0 in

PWHT is not performed

Radiography: Category A joints - Spot UW-11(b) type 1
Category B joints - Spot UW-11(b) type 1

Estimated weight:	new = 1438.3	corr = 1438.3	lb
capacity:	new = 2249.093	corr = 2249.093	US ga

ID = 105 length Lc= 60 t = 0.25 in (new)

Design thickness: UG-27(c)(1) Circ. stress

t = P*R/(S*E - 0.6*P) + Corrosion
= 2*52.5/(14700*0.85 - 0.6*2) + 0
= 0.0084 in

MAP: (New & at 0 deg F) UG-27(c)(1)

P = S*E*t/(R + 0.6*t) - Ps
= 16700*0.85*0.25/(52.5 + 0.6*0.25) - 0
= 67.40266 psi

MAWP: (Corroded & at 400 deg F) UG-27(c)(1)

P = S*E*t/(R + 0.6*t) - Ps
= 14700*0.85*0.25/(52.5 + 0.6*0.25) - 0
= 59.33049 psi

External Pressure: (Corroded & at 400 deg F) UG-28

L/Do = 37.90327/105.5 =0.3593 Do/t = 105.5/0.24092 = 437.9047
From table G: A = 0.000429
From table HA-3: B = 4842.5

Pa= 4*B/(3*Do/t)
= 4*4842.5/(3*105.5/0.24092)
= 14.7445 psi

Design thickness for external pressure Pa = 14.7445 psi:

= t + Corrosion
= 0.24092 + 0

= 0.24092 in

Maximum Allowable External Pressure: (Corroded @ 400 deg F)

L/Do = 37.90327/105.5 =0.3593 Do/t = 105.5/0.25 = 422
From table G: A = 0.000452
From table HA-3: B = 4887.2

Pa= 4*B/(3*Do/t)
= 4*4887.2/(3*105.5/0.25)
= 15.4414 psi

Shell Stiffener

Stiffening Ring Calculations Per UG-29

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Identifier:		Shell Stiffener
Ring material specification:		SA 240 304L HIGH
Number of rings in this group:		1
Distance first ring to datum line:		110 in
Ring spacing:		0 in
Ring description:		4x4x1/4 Equal Angle
Ring is rolled:		leg out (easy way)
Ring cross sectional area:	As =	1.94 in^2
Ring moment of inertia:	Ir =	3.04 in^4

Calculations for ring 110 in from datum

Shell material specification:		SA 240 304L HIGH
Required shell thickness:	t =	0.24092 in
Corroded shell thickness:	ts =	0.25 in
Shell outer diameter:	Do =	105.5 in
Design temperature:	=	400 deg F
External design pressure:	P =	14.7 psi
Stiffener supported length:	Ls =	33.95164 in

B = .75*(P*Do/(t + As/Ls))
= .75*(14.7*105.5/(0.24092 + 1.94/33.95164))
= 3902.359

From table HA-3 (ring) A = 2.951318E-04

Required moment of inertia of the combined ring-shell section

Is= (Do^2*Ls*(t + As/Ls)*A)/10.9
= (105.5^2*33.95164*(0.24092 + 1.94/33.95164)*2.951318E-04)/10.9
= 3.049713 in^4

Available moment of inertia of the combined ring-shell section

Shell width contributing smaller of = 5.649226

W = 1.1*Sqr(Do*ts)
= 1.1*Sqr(105.5*0.25)
= 5.649226 in

W = Ls = 33.95164 in

Shell area A1 = W*ts = 1.412306 in^2

Distance to the ring neutral axis

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Shell Stiffener

Y2 = Ring NA + ts/2
= 1.09 + 0.25/2
= 1.215 in

Neutral axis of combined section

NA = As*Y2/(A1 + As)
= 1.94*1.215/(1.412306 + 1.94)
= .7031279 in

Inertia of the shell about the combined section NA

I1 = W*ts^3/12 + A1*NA^2
= 5.649226*0.25^3/12 + 1.412306*0.7031279^2
= .7055842 in^4

Inertia of the ring about the combined section NA

I2 = Ir + As*(NA - Y2)^2
= 3.04 + 1.94*(0.7031279 - 1.215)^2
= 3.548306 in^4

Total available I = I1 + I2 = 4.25389 in^4

The 4x4x1/4 Equal Angle vacuum stiffener is satisfactory.

Main Section Cylinder

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Component:	Cylinder			
Material specification:	SA 240 304L HIGH			
Internal design pressure:	P = 2	psi @	400	deg F
External design pressure:	Pe= 14.7	psi @	400	deg F

Corrosion allowance: Inner C = 0 Outer = 0 in

PWHT is not performed

Radiography: Category A joints - Spot UW-11(b) type 1
Category B joints - Spot UW-11(b) type 1

Estimated weight:	new = 3844.7	corr = 3844.7	lb
capacity:	new = 2998.79	corr = 2998.79	US ga

ID = 105 length Lc= 80 t = 0.5 in (new)

Design thickness: UG-27(c)(1) Circ. stress

t = P*R/(S*E - 0.6*P) + Corrosion
= 2*52.5/(14700*0.85 - 0.6*2) + 0
= 0.0084 in

MAP: (New & at 0 deg F) UG-27(c)(1)

P = S*E*t/(R + 0.6*t) - Ps
= 16700*0.85*0.5/(52.5 + 0.6*0.5) - 0
= 134.4223 psi

MAWP: (Corroded & at 400 deg F) UG-27(c)(1)

P = S*E*t/(R + 0.6*t) - Ps
= 14700*0.85*0.5/(52.5 + 0.6*0.5) - 0
= 118.3239 psi

External Pressure: (Corroded & at 400 deg F) UG-28

L/Do = 80/106 =0.7547 Do/t = 106/0.30843 = 343.6761
From table G: A = 0.000288
From table HA-3: B = 3807.2

Pa= 4*B/(3*Do/t)
= 4*3807.2/(3*106/0.30843)
= 14.7705 psi

Design thickness for external pressure Pa = 14.7705 psi:

= t + Corrosion
= 0.30843 + 0

Main Section Cylinder

= 0.30843 in

Maximum Allowable External Pressure: (Corroded @ 400 deg F)

L/Do = 80/106 =0.7547 Do/t = 106/0.5 = 212
From table G: A = 0.000586
From table HA-3: B = 5115.7

Pa= 4*B/(3*Do/t)
= 4*5115.7/(3*106/0.5)
= 32.1742 psi

Nozzle E

Opening N-E Reinforcement Calculations Per UG-37

Located on:	Upper Cylindrical Shell - BSC
Local vessel thickness:	.25 in
Liquid static head included:	0 psi
Flange description:	14 inch 75# SO A240304L
ANSI B16.47 flange rating MAP:	115 psi
ANSI B16.47 flange rating MAWP:	80 psi
Nozzle material specification:	SA 240 304L HIGH
Pad material specification:	SA 240 304L HIGH
Nozzle orientation:	45 degrees
End of nozzle to shell center:	55.875 in
Nozzle offset from center Lo:	0 in
Projection outside vessel Lpr:	3.125 in

```
    tn ->| |<-                       corrosion allow = 0 in
dp->|                        |<-     noz thick new tn= .12 in
    |     _|<- d ->|  _ |twl |       nozzle id. new d= 13.75 in
->| |<tw2|         >| |<-|           pad diameter dp = 20 in
__| ____/ |         | \____          pad thickness te= .25 in
te|/|                |     |\        fillet weld twl = .125 in
________|_|_v        | |________     fillet weld tw2 = .25 in
_______/|_|_tw3     |_|\_______      groove weld tw3 = .25 in
            ^
<--- L --->|                         To datum L= 124 in
```

Reinforcement Calculations For Nozzle MAWP

Limits of reinforcement UG-40

Parallel to the vessel wall d = 13.75 in
Normal to the vessel wall outside 2.5*(tn-Cn) + te = .55 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .3 in

Nozzle required thickness

trn = P*Rn/(Sn*E - 0.6*P)
= 2*6.875/(14700*1 - 0.6*2)
= 0.0009 in

Required thickness tr from UG-37(a)

tr= P*R/(S*E - 0.6*P)
= 2*52.5/(14700*1 - 0.6*2)
= 0.0071 in

Area required

Nozzle E

Allowable stresses: Sn = 14700, Sv = 14700, Sp = 14700 psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1
fr2 = lesser of 1 or Sn/Sv so fr2 =1
fr3 = lesser of fr2 or Sp/Sv so fr3 =1
fr4 = lesser of 1 or Sp/Sv so fr4 =1

A = d*tr*F + 2*tn*tr*F*(1 - fr1)
= 13.75*0.0071*1 + 2*0.12*0.0071*1*(1 - 1)
= .0976 in^2

Area available

A1 = larger of the following = 3.34 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 13.75*(1*0.25-1*0.0071) - 2*0.12*(1*0.25-1*0.0071)*(1-1)
= 3.34 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.25+0.12)*(1*0.25-1*0.0071) - 2*0.12*(1*0.25-1*0.0071)*(1-1)
= .18 in^2

A2 = smaller of the following = 0.131 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.12 - 0.0009)*1*0.25
= .149 in^2

= 2*(tn - trn)*(2.5*tn + te)*fr2
= 2*(0.12 - 0.0009)*(2.5*0.12 + 0.25)*1
= .131 in^2

A41 = Leg^2*fr3
= 0.125^2*1 = .016 in^2

A42 = Leg^2*fr4
= 0.25^2*1 = .063 in^2

A5 = (Dp - d - 2*tn)*te*fr4
= (20 - 13.75 - 2*0.12)*0.25*1
= 1.5025 in^2

Area = A1 + A2 + A41 + A42 + A5
= 3.34 + 0.131 + 0.016 + 0.063 + 1.5025
= 5.0525 in^2

As Area > A the reinforcement is adequate for MAWP = 2 at 400 Deg F

Check the welds - From UW-16(c)(2)

Inner Fillet: tmin = lesser of 0.75 or tn or te, tmin = 0.12 in
tw(min) = 0.7*tmin = 0.084 in

Nozzle E

tw(actual) = 0.7*Leg = 0.7*0.125 = 0.0875 in

Outer Fillet: tmin = lesser of 0.75 or te or t, tmin = 0.25 in
tw(min) = 0.5*tmin = 0.125 in
tw(actual) = 0.7*Leg = 0.7*0.25 = 0.175 in

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.0009 in (E = 1)
Wall thickness per UG-45(b)(1):	tr2 = 0.0071 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.328125 in
The greater of tr2 or tr3:	tr5 = 0.0625 in
The lesser of tr4 or tr5:	tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.12 in

The nozzle neck thickness is adequate for MAWP.

Allowable stresses in joints UG-45(c) and UW-15(c)

Groove weld in tension = 0.74*14700 = 10878 psi
Nozzle wall in shear = 0.7*14700 = 10290 psi
Inner fillet weld in shear = 0.49*14700 = 7203 psi
Outer fillet weld in shear = 0.49*14700 = 7203 psi

Strength of welded joints:

(1) Inner fillet weld in shear
(Pi/2)*Nozzle O.D.*Leg*Si = 1.57*13.99*0.125*7203 = 19776.11 lbf

(2) Outer fillet weld in shear
(Pi/2)*Pad O.D.*Leg*So = 1.57*20*0.25*7203 = 56543.55 lbf

(3) Nozzle wall in shear
(Pi/2)*Mean nozzle dia.*tn*Sn = 1.57*13.87*0.12*10290 = 26888.88 lbf

(4) Groove weld in tension
(Pi/2)*Nozzle O.D.*tw*Sg = 1.57*13.99*0.25*10878 = 59731.91 lbf

Loading on welds per UG-41(b)(1)

W = (A - (d - 2*tn)*(E1*t - F*tr))*Sv
= (0.0976 - (13.75 - 2*0.12)*(1*0.25 - 1*0.0071))*14700
=-46804.49 lbf

W1-1 = (A2 + A5 + A41 + A42)*Sv
= (0.131 + 1.5025 + 0.016 + 0.063)*14700
= 25173.75 lbf

W2-2 = (A2 + A3 + A41 + A43 + 2*tn*t*fr1)*Sv

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Nozzle E

= (0.131 + 0 + 0.016 + 0 + 2*0.12*0.25*1)*14700
= 3042.9 lbf

W3-3 = (A2 + A3 + A5 + A41 + A42 + A43 + 2*tn*t*fr1)*Sv
= (0.131 + 0 + 1.5025 + 0.016 + 0.063 + 0 + 2*0.12*0.25*1)*14700
= 26055.75 lbf

Load for path 1-1 lesser of W or W1-1 =-46804.49 lbf
Path 1-1 Thru (2) & (3) = 56543.55 + 26888.88 = 83432.43 lbf
Path 1-1 is stronger than W so it is acceptable per UG-41(b)(2).

Load for path 2-2 lesser of W or W2-2 =-46804.49 lbf
Path 2-2 Thru (1), (4) = 19776.11 + 59731.91 = 79508.02 lbf
Path 2-2 is stronger than W so it is acceptable per UG-41(b)(2).

Load for path 3-3 lesser of W or W3-3 =-46804.49 lbf
Path 3-3 Thru (2), (4) = 56543.55 + 59731.91 = 116275.5 lbf
Path 3-3 is stronger than W so it is acceptable per UG-41(b)(2).

Pad strength = A5*Sp = 22086.75 lbf
Outer fillet weld strength is adequate.

Reinforcement Calculations for External Pressure

Limits of reinforcement UG-40

Parallel to the vessel wall d = 13.75 in
Normal to the vessel wall outside 2.5*(tn-Cn) + te = .55 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .3 in

Nozzle required thickness

L/Do = 3.125/13.99 = .2234 Do/t = 13.99/0.02972 = 470.7268
From table G: A = 0.000642
From table HA-3: B = 5198.5

Pa= 4*B/(3*Do/t)
= 4*5198.5/(3*13.99/0.02972)
= 14.7247 psi

Nozzle required thickness trn = .02972 in

Required thickness tr from UG-37(d)(1) = .2409 in

Area required

Allowable stresses: Sn = 14700, Sv = 14700, Sp = 14700 psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1
fr2 = lesser of 1 or Sn/Sv so fr2 =1
fr3 = lesser of fr2 or Sp/Sv so fr3 =1
fr4 = lesser of 1 or Sp/Sv so fr4 =1

Nozzle E

A = 0.5*(d*tr*F + 2*tn*tr*F*(1 - fr1))
= 0.5*(13.75*0.2409*1 + 2*0.12*0.2409*1*(1 - 1))
= 1.6562 in^2

Area available

A1 = larger of the following = .125 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 13.75*(1*0.25-1*0.2409) - 2*0.12*(1*0.25-1*0.2409)*(1-1)
= .125 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.25+0.12)*(1*0.25-1*0.2409) - 2*0.12*(1*0.25-1*0.2409)*(1-1)
= .007 in^2

A2 = smaller of the following = 0.099 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.12 - 0.02972)*1*0.25
= .113 in^2

= 2*(tn - trn)*(2.5*tn + te)*fr2
= 2*(0.12 - 0.02972)*(2.5*0.12 + 0.25)*1
= .099 in^2

A41 = Leg^2*fr3
= 0.125^2*1 = .016 in^2

A42 = Leg^2*fr4
= 0.25^2*1 = .063 in^2

A5 = (Dp - d - 2*tn)*te*fr4
= (20 - 13.75 - 2*0.12)*0.25*1
= 1.5025 in^2

Area = A1 + A2 + A41 + A42 + A5
= 0.125 + 0.099 + 0.016 + 0.063 + 1.5025
= 1.8055 in^2

As Area > A the reinforcement is adequate for Pe = 14.7 at 400 Deg F

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.02972 in (E = 1)
Wall thickness per UG-45(b)(2):	tr2 = 0.0525 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.328125 in
The greater of tr2 or tr3:	tr5 = 0.0625 in
The lesser of tr4 or tr5:	tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

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Nozzle E

Available nozzle wall thickness new, tn = 0.12 in

The nozzle neck thickness is adequate for Pe.

Nozzle E

Applied Loads

Radial load	Pr = 2183 lbf
Circumferential moment	Mc = 0 lbf-ft
Circumferential shear	Vc = 0 lbf
Longitudinal moment	ML = 0 lbf-ft
Longitudinal shear	VL = 0 lbf
Torsion moment	Mt = 0 lbf-ft
Internal pressure	P = 2 psi

Stresses at the nozzle OD per WRC bulletin 107 (psi)

Mean radius Rm = 52.625 in
Rm/t = 105.25

Stress concentration factor Kn (tension) = 1
Stress concentration factor Kb (bending) = 1

Pressure stress intensity factor, Farr equation 11.5

I = .25*(4 + 3*(r/x)^2 + 3*(r/x)^4)
= .25*(4 + 3*(6.875/7.12)^2 + 3*(6.875/7.12)^4)
= 2.351

Local circ. pressure stress = I*P*Rm/t = 495 psi

Local long. pressure stress = P*Rm/2t = 105 psi

Maximum combined stress =-4703 psi
Allowable combined stress = +-3*S = +- 44100 psi

The maximum combined stress is within allowable limits.

Maximum primary membrane stress =-1154 psi
Allowable primary membrane stress = +-1.5*S = +- 22050 psi

The maximum primary membrane stress is within allowable limits.

Nozzle E

From Fig.	Value read	beta	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
3C*	10.776	0.116					-894	-894	-894	-894
4C*	15.180	0.116	-1259	-1259	-1259	-1259				
1C	0.0768	0.116					-4024	4024	-4024	4024
2C-1	0.0443	0.116	-2321	2321	-2321	2321				
3A*	4.1565	0.116								
1A	0.0776	0.116								
3B*	11.121	0.116								
1B-1	0.0282	0.116								
pressure stress*			495	495	495	495	495	495	495	495
Total circ stress			-3085	1557	-3085	1557	-4423	3625	-4423	3625
Primary membrane circ stress*			-764	-764	-764	-764	-399	-399	-399	-399
3C*	10.776	0.116	-894	-894	-894	-894				
4C*	15.180	0.116					-1259	-1259	-1259	-1259
1C-1	0.0747	0.116	-3914	3914	-3914	3914				
2C	0.0469	0.116					-2457	2457	-2457	2457
4A*	7.6047	0.116								
2A	0.0390	0.116								
4B*	4.1127	0.116								
2B-1	0.0387	0.116								
pressure stress*			105	105	105	105	105	105	105	105
Total long stress			-4703	3125	-4703	3125	-3611	1303	-3611	1303
Primary membrane long stress*			-789	-789	-789	-789	-1154	-1154	-1154	-1154
torsion moment Mt										
Circ shear from Vc										
Long shear from VL										
Total Shear stress										
Combined stress			-4703	3125	-4703	3125	-4423	3625	-4423	3625

Nozzle E

Stresses at the pad edge per WRC bulletin 107 (psi)

Mean radius Rm = 52.625 in
Rm/t = 210.5

Stress concentration factor Kn (tension) = 1
Stress concentration factor Kb (bending) = 1

Pressure stress intensity factor, Farr equation 11.5

I = .25*(4 + 3*(r/x)^2 + 3*(r/x)^4)
 = .25*(4 + 3*(6.875/10.25)^2 + 3*(6.875/10.25)^4)
 = 1.489

Local circ. pressure stress = I*P*Rm/t = 627 psi

Local long. pressure stress = P*Rm/2t = 210 psi

Maximum combined stress =-13225 psi
Allowable combined stress = +-3*S = +- 44100 psi

The maximum combined stress is within allowable limits.

Maximum primary membrane stress =-3307 psi
Allowable primary membrane stress = +-1.5*S = +- 22050 psi

The maximum primary membrane stress is within allowable limits.

Nozzle E

From Fig.	Value read	beta	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
3C*	8.3321	0.166					-1383	-1383	-1383	-1383
4C*	21.196	0.166	-3517	-3517	-3517	-3517				
1C	0.0595	0.166					-12469	12469	-12469	12469
2C-1	0.0149	0.166	-3123	3123	-3123	3123				
3A*	6.5607	0.166								
1A	0.0553	0.166								
3B*	13.588	0.166								
1B-1	0.0104	0.166								
pressure stress*			627	627	627	627	627	627	627	627
Total circ stress			-6013	233	-6013	233	-13225	11713	-13225	11713
Primary membrane circ stress*			-2890	-2890	-2890	-2890	-756	-756	-756	-756
3C*	8.3321	0.166	-1383	-1383	-1383	-1383				
4C*	21.196	0.166					-3517	-3517	-3517	-3517
1C-1	0.0392	0.166	-8215	8215	-8215	8215				
2C	0.0353	0.166					-7398	7398	-7398	7398
4A*	17.922	0.166								
2A	0.0238	0.166								
4B*	5.8428	0.166								
2B-1	0.0144	0.166								
pressure stress*			210	210	210	210	210	210	210	210
Total long stress			-9388	7042	-9388	7042	-10705	4091	-10705	4091
Primary membrane long stress*			-1173	-1173	-1173	-1173	-3307	-3307	-3307	-3307
torsion moment Mt										
Circ shear from Vc										
Long shear from VL										
Total Shear stress										
Combined stress			-9388	7042	-9388	7042	-13225	11713	-13225	11713

Nozzle G

Opening N-G Reinforcement Calculations Per UG-37

Located on:	Main Section Cylinder
Local vessel thickness:	.5 in
Liquid static head included:	0 psi
Flange description:	8 inch 75# WN A240304L
ANSI B16.47 flange rating MAP:	115 psi
ANSI B16.47 flange rating MAWP:	80 psi

Nozzle material specification:	SA 240 304L HIGH

Nozzle orientation:	90 degrees
End of nozzle to shell center:	55.625 in
Nozzle offset from center Lo:	0 in
Projection outside vessel Lpr:	2.625 in

```
   tn ->| |<-
         _       _
        | |<- d ->| |
 twl->| |<|       | |
      |/| |  v    | |\
 _____.-| | --    | |-.______
 _____\|_|_tw2    |_|/_______
            ^
 <--- L --->|
```

corrosion allow = 0 in
noz thick new tn= .12 in
nozzle id. new d= 7.75 in
fillet weld twl = .125 in
groove weld tw2 = .1875 in

To datum L= 40 in

Reinforcement Calculations For Nozzle MAWP

Limits of reinforcement UG-40

Parallel to the vessel wall d = 7.75 in
Normal to the vessel wall outside 2.5*(tn-Cn) + te = .3 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .3 in

Nozzle required thickness

trn = P*Rn/(Sn*E - 0.6*P)
 = 2*3.875/(14700*1 - 0.6*2)
 = 0.0005 in

Required thickness tr from UG-37(a)

tr= P*R/(S*E - 0.6*P)
 = 2*52.5/(14700*1 - 0.6*2)
 = 0.0071 in

Area required

Allowable stresses: Sn = 14700, Sv = 14700, psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1

Nozzle G

fr2 = lesser of 1 or Sn/Sv so fr2 = 1

A = d*tr*F + 2*tn*tr*F*(1 - fr1)
= 7.75*0.0071*1 + 2*0.12*0.0071*1*(1 - 1)
= .055 in^2

Area available

A1 = larger of the following = 3.82 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 7.75*(1*0.5-1*0.0071) - 2*0.12*(1*0.5-1*0.0071)*(1-1)
= 3.82 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.5+0.12)*(1*0.5-1*0.0071) - 2*0.12*(1*0.5-1*0.0071)*(1-1)
= .611 in^2

A2 = smaller of the following = 0.072 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.12 - 0.0005)*1*0.5
= .299 in^2

= 5*(tn - trn)*fr2*tn
= 5*(0.12 - 0.0005)*1*0.12
= .072 in^2

A41 = Leg^2*fr2
= 0.125^2*1 = .016 in^2

Area = A1 + A2 + A41
= 3.82 + 0.072 + 0.016
= 3.908 in^2

As Area > A the reinforcement is adequate for MAWP = 2 at 400 Deg F

Check the welds - From UW-16(d):

tmin = lesser of 0.75 or tn or t, tmin = 0.12 in
t1 or t2(min) = lesser of 0.25 or 0.7*tmin, t1(min) = 0.084 in
t1(actual) = 0.7*Leg = 0.7*0.125 = 0.0875 in
t2(actual) = 0.1875 in
t1 + t2 = 0.275 >= 1.25*tmin

The weld sizes for t1 and t2 are satisfactory.

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.0005 in (E = 1)
Wall thickness per UG-45(b)(1):	tr2 = 0.0071 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.28175 in

Nozzle G

The greater of tr2 or tr3: tr5 = 0.0625 in
The lesser of tr4 or tr5: tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.12 in

The nozzle neck thickness is adequate for MAWP.

Allowable stresses in joints UG-45(c) and UW-15(c)

Groove weld in tension = 0.74*14700 = 10878 psi
Nozzle wall in shear = 0.7*14700 = 10290 psi
Inner fillet weld in shear = 0.49*14700 = 7203 psi

Strength of welded joints:

(1) Inner fillet weld in shear
(Pi/2)*Nozzle O.D.*Leg*Si = 1.57*7.99*0.125*7203 = 11294.57 lbf

(3) Nozzle wall in shear
(Pi/2)*Mean nozzle dia.*tn*Sn = 1.57*7.87*0.12*10290 = 15257.07 lbf

(4) Groove weld in tension
(Pi/2)*Nozzle O.D.*tw*Sg = 1.57*7.99*0.1875*10878 = 25585.67 lbf

Loading on welds per UG-41(b)(1)

W = (A - (d - 2*tn)*(E1*t - F*tr))*Sv
= (0.055 - (7.75 - 2*0.12)*(1*0.5 - 1*0.0071))*14700
=-53606.18 lbf

W1-1 = (A2 + A5 + A41 + A42)*Sv
= (0.072 + 0 + 0.016 + 0)*14700
= 1293.6 lbf

W2-2 = (A2 + A3 + A41 + A43 + 2*tn*t*fr1)*Sv
= (0.072 + 0 + 0.016 + 0 + 2*0.12*0.5*1)*14700
= 3057.6 lbf

Load for path 1-1 lesser of W or W1-1 =-53606.18 lbf
Path 1-1 Thru (1) & (3) = 11294.57 + 15257.07 = 26551.64 lbf
Path 1-1 is stronger than W so it is acceptable per UG-41(b)(2).

Load for path 2-2 lesser of W or W2-2 =-53606.18 lbf
Path 2-2 Thru (1), (4) = 11294.57 + 25585.67 = 36880.24 lbf
Path 2-2 is stronger than W so it is acceptable per UG-41(b)(2).

Reinforcement Calculations for External Pressure

Limits of reinforcement UG-40

Parallel to the vessel wall d = 7.75 in

Nozzle G

Normal to the vessel wall outside 2.5*(tn-Cn) + te = .3 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .3 in

Nozzle required thickness

L/Do = 2.625/7.99 = .3285 Do/t = 7.99/0.01796 = 444.8775
From table G: A = 0.000462
From table HA-3: B = 4906.1

Pa= 4*B/(3*Do/t)
= 4*4906.1/(3*7.99/0.01796)
= 14.704 psi

Nozzle required thickness trn = .01796 in

Required thickness tr from UG-37(d)(1) = .3084 in

Area required

Allowable stresses: Sn = 14700, Sv = 14700, psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1
fr2 = lesser of 1 or Sn/Sv so fr2 =1

A = 0.5*(d*tr*F + 2*tn*tr*F*(1 - fr1))
= 0.5*(7.75*0.3084*1 + 2*0.12*0.3084*1*(1 - 1))
= 1.1951 in^2

Area available

A1 = larger of the following = 1.485 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 7.75*(1*0.5-1*0.3084) - 2*0.12*(1*0.5-1*0.3084)*(1-1)
= 1.485 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.5+0.12)*(1*0.5-1*0.3084) - 2*0.12*(1*0.5-1*0.3084)*(1-1)
= .238 in^2

A2 = smaller of the following = 0.061 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.12 - 0.01796)*1*0.5
= .255 in^2

= 5*(tn - trn)*fr2*tn
= 5*(0.12 - 0.01796)*1*0.12
= .061 in^2

A41 = Leg^2*fr2
= 0.125^2*1 = .016 in^2

Rev. 0

Nozzle G

Area = A1 + A2 + A41
= 1.485 + 0.061 + 0.016
= 1.562 in^2

As Area > A the reinforcement is adequate for Pe = 14.7 at 400 Deg F

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.01796 in (E = 1)
Wall thickness per UG-45(b)(2):	tr2 = 0.0525 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.28175 in
The greater of tr2 or tr3:	tr5 = 0.0625 in
The lesser of tr4 or tr5:	tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.12 in

The nozzle neck thickness is adequate for Pe.

Nozzle G

Applied Loads

Radial load	Pr = 693.4 lbf
Circumferential moment	Mc = 0 lbf-ft
Circumferential shear	Vc = 0 lbf
Longitudinal moment	ML = 0 lbf-ft
Longitudinal shear	VL = 0 lbf
Torsion moment	Mt = 0 lbf-ft
Internal pressure	P = 2 psi

Stresses at the nozzle OD per WRC bulletin 107 (psi)

Mean radius Rm = 52.75 in
Rm/t = 105.5

Stress concentration factor Kn (tension) = 1
Stress concentration factor Kb (bending) = 1

Pressure stress intensity factor, Farr equation 11.5

I = .25*(4 + 3*(r/x)^2 + 3*(r/x)^4)
= .25*(4 + 3*(3.875/4.12)^2 + 3*(3.875/4.12)^4)
= 2.25

Local circ. pressure stress = I*P*Rm/t = 475 psi

Local long. pressure stress = P*Rm/2t = 106 psi

Maximum combined stress =-2389 psi
Allowable combined stress = +-3*S = +- 44100 psi

The maximum combined stress is within allowable limits.

Maximum primary membrane stress =-363 psi
Allowable primary membrane stress = +-1.5*S = +- 22050 psi

The maximum primary membrane stress is within allowable limits.

Nozzle G

From Fig.	Value read	beta	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
3C*	16.149	0.066					-425	-425	-425	-425
4C*	17.850	0.066	-469	-469	-469	-469				
1C	0.1215	0.066					-2022	2022	-2022	2022
2C-1	0.0842	0.066	-1401	1401	-1401	1401				
3A*	3.3073	0.066								
1A	0.0927	0.066								
3B*	10.765	0.066								
1B-1	0.0441	0.066								
pressure stress*			475	475	475	475	475	475	475	475
Total circ stress			-1395	1407	-1395	1407	-1972	2072	-1972	2072
Primary membrane circ stress*			6	6	6	6	50	50	50	50
3C*	16.149	0.066	-425	-425	-425	-425				
4C*	17.850	0.066					-469	-469	-469	-469
1C-1	0.1244	0.066	-2070	2070	-2070	2070				
2C	0.0842	0.066					-1401	1401	-1401	1401
4A*	4.8372	0.066								
2A	0.0535	0.066								
4B*	3.2886	0.066								
2B-1	0.0699	0.066								
pressure stress*			106	106	106	106	106	106	106	106
Total long stress			-2389	1751	-2389	1751	-1764	1038	-1764	1038
Primary membrane long stress*			-319	-319	-319	-319	-363	-363	-363	-363
torsion moment Mt										
Circ shear from Vc										
Long shear from VL										
Total Shear stress										
Combined stress			-2389	1751	-2389	1751	-1972	2072	-1972	2072

Rev 0

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-016 PAGE 1 OF 33
REV.	DEO #	DATE	BY:	CHECK	TITLE: Beam Splitter Chamber - Finite Element Analysis of 1 in. Thick Flanges for Bolt Preload.	
0	0139	12/6/95	RDC	AGR		
					BY: R.D. Cirrito	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE: To determine if bolt clamping force results in flange deformation that could allow leakage at Viton o-rings.

METHOD: A 3D finite element analysis of a section of the plane flange is performed. Brick elements (3D solids) are used in the IMAGES program. Both a coarse model and an enhanced finer model are analyzed. The enhanced model includes compression only springs at the flange interface to simulate gap opening (nonlinear).

ASSUMPTIONS: Symmetry is used to minimize the model by using appropriate boundary conditions. Curvature of the flange is ignored. O-ring glands and pumping channel are not included.

INPUTS: LIGO project design sketches and drawings. Durometer is 90 for Viton o-rings.

REFERENCES: 1. Parker O-Ring Handbook.
2. IMAGES 3D, Version 3.0, R.L. Cloud & Associates

CALCULATIONS: (See Attachment)

CONCLUSIONS: At a location that is midway between bolts, the gap between flanges opens a negligible amount that does not affect the seal.

NOTES: This analysis was performed for the 104 in diameter flange, but it applies to other diameter as well because of similar dimensions and bolt spacing. Computer file is FLANGE.

104 1/4" ID FLANGES - ANALYSIS FOR VACUUM LOAD

HOLE SPACING

No. OF BOLTS = 48

BOLT CIRCLE DIAM

$C = 110''$

CIRCUM =

$C' = \pi C = 345.6$ in

SPACING

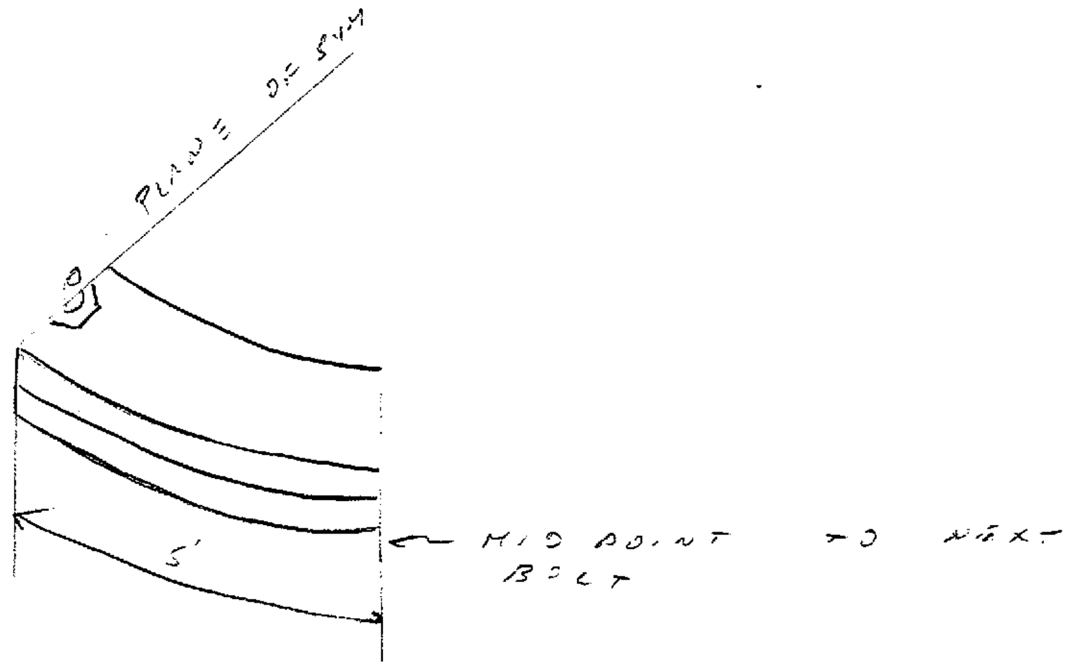
$S = \frac{C'}{N} = \frac{345.6}{48}$

$= 7.20$ in

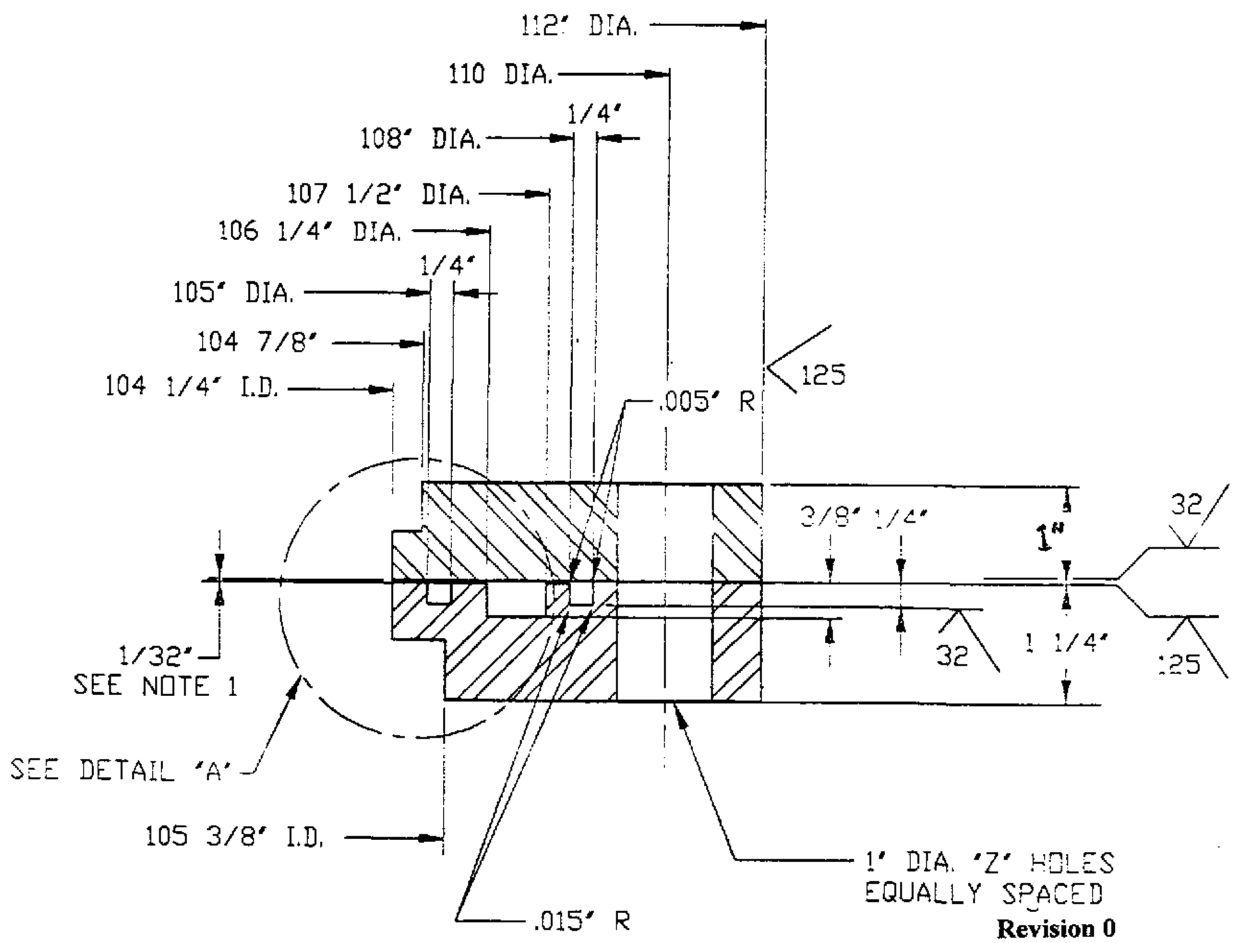
TRY 7/8" Ø BOLTS W/ 1" Ø HOLES

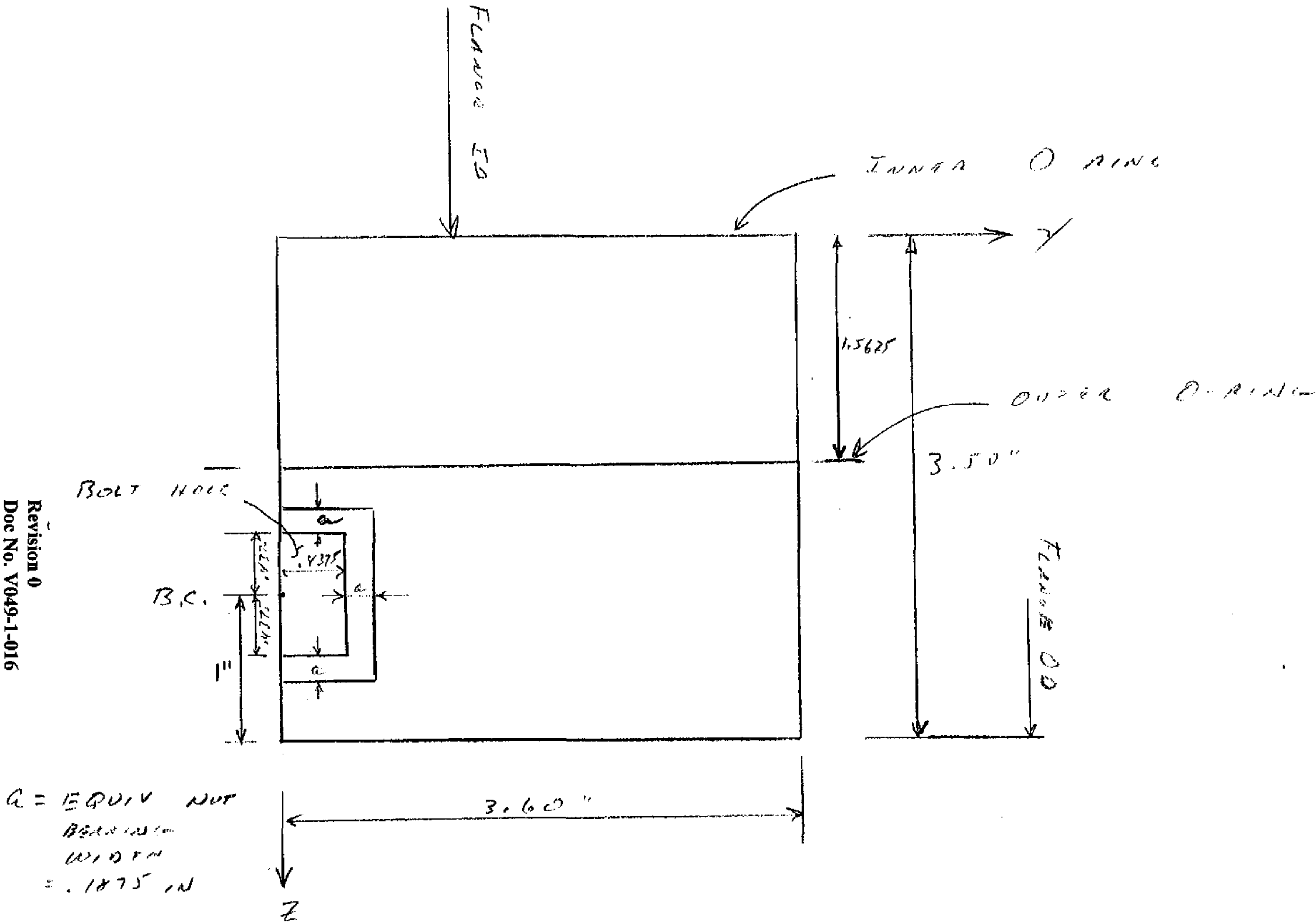
USE FE MODEL TO DETERMINE IF BOLT SPACING & FLANGE THICKNESS WILL BE SUFFICIENT TO PROVIDE CLAMPING FORCE BETWEEN THE BOLTS AT THE O-RING. 3-D BRICK ELEMENTS WILL BE USED. MODEL TOP FLANGE WHICH IS THINNER THAN BOTTOM FLANGE.

THE FE MODEL WILL NEGLECT CURVATURE WHICH HAS NEGLIGIBLE EFFECT ON RESULTS.

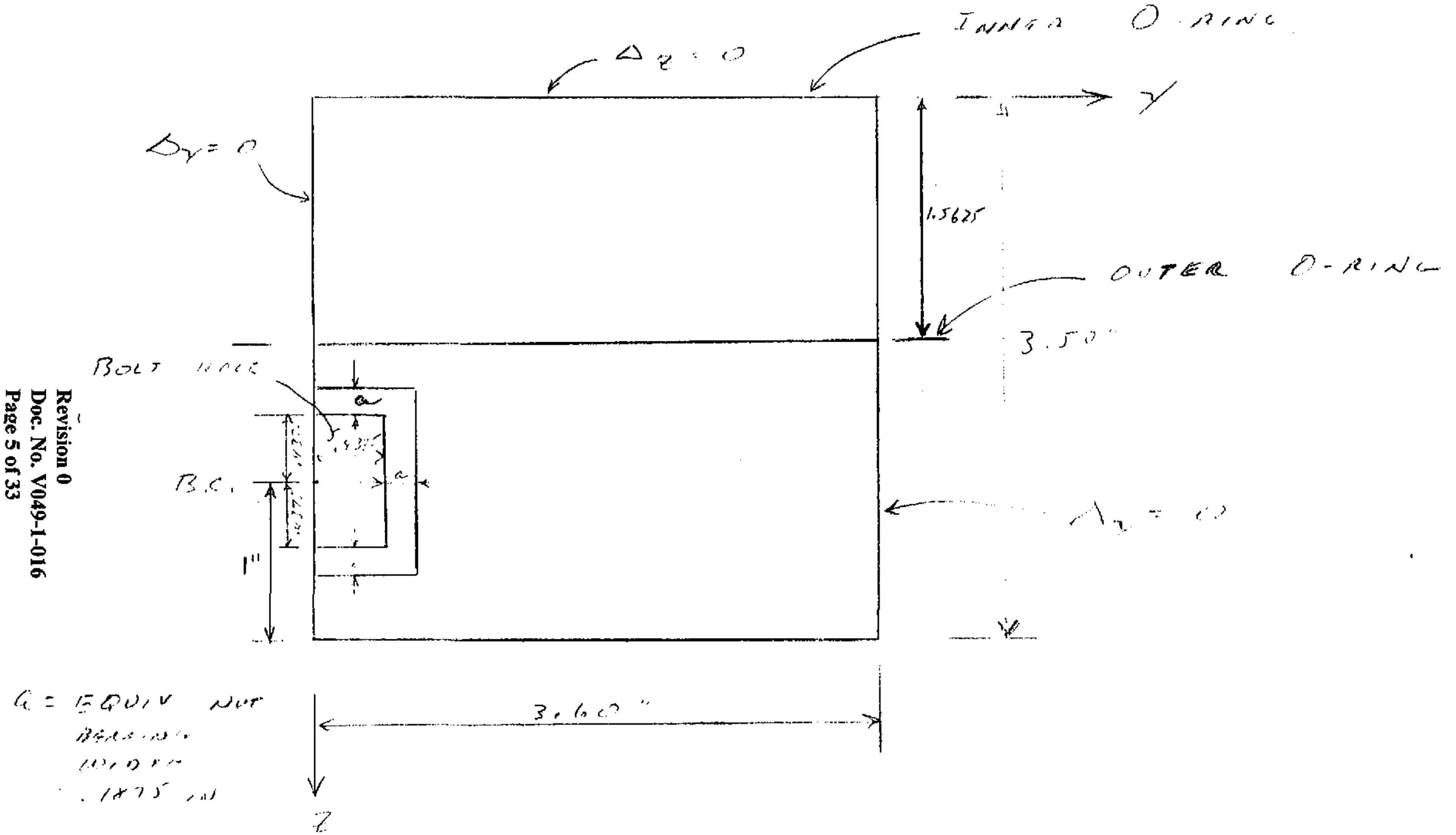


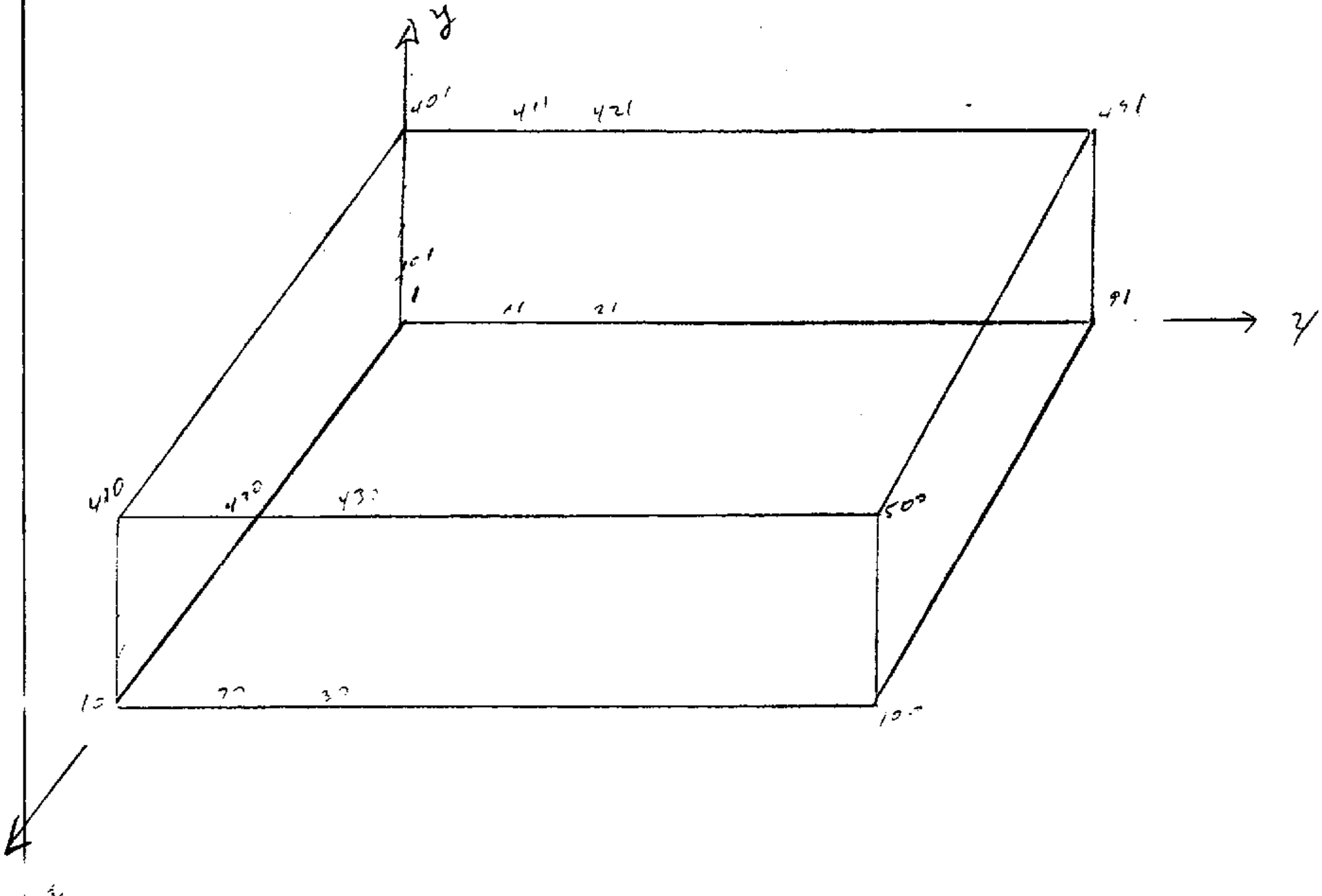
$S' = \frac{S}{2} = 3.60$ in



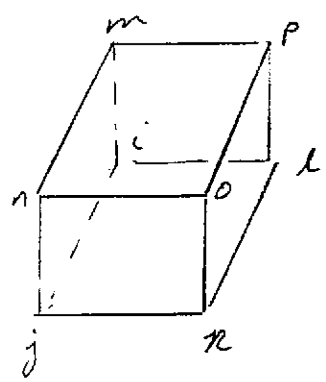

FLANGE ID
INNER O RING
Y
1.5625
OUTER O-RING
3.50"
BOLT HOLE
a
.4375
B.C.
1"
FLANGE OD
3.60"
a = EQUIV NUT BEARING WIDTH = .1875 IN
Z

BOUNDARY CONDITIONS
@ y = 0, Δy = 0





TYPICAL ELEMENT NODE NUMBERING



1ST EL i j k l m n o p

1, 2, 12, 11, 101, 102, 112, 111

DELETE ELEMENTS AT BOLT HOLE
6, 7, 47, 48, 164, 169, 249 250

22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD

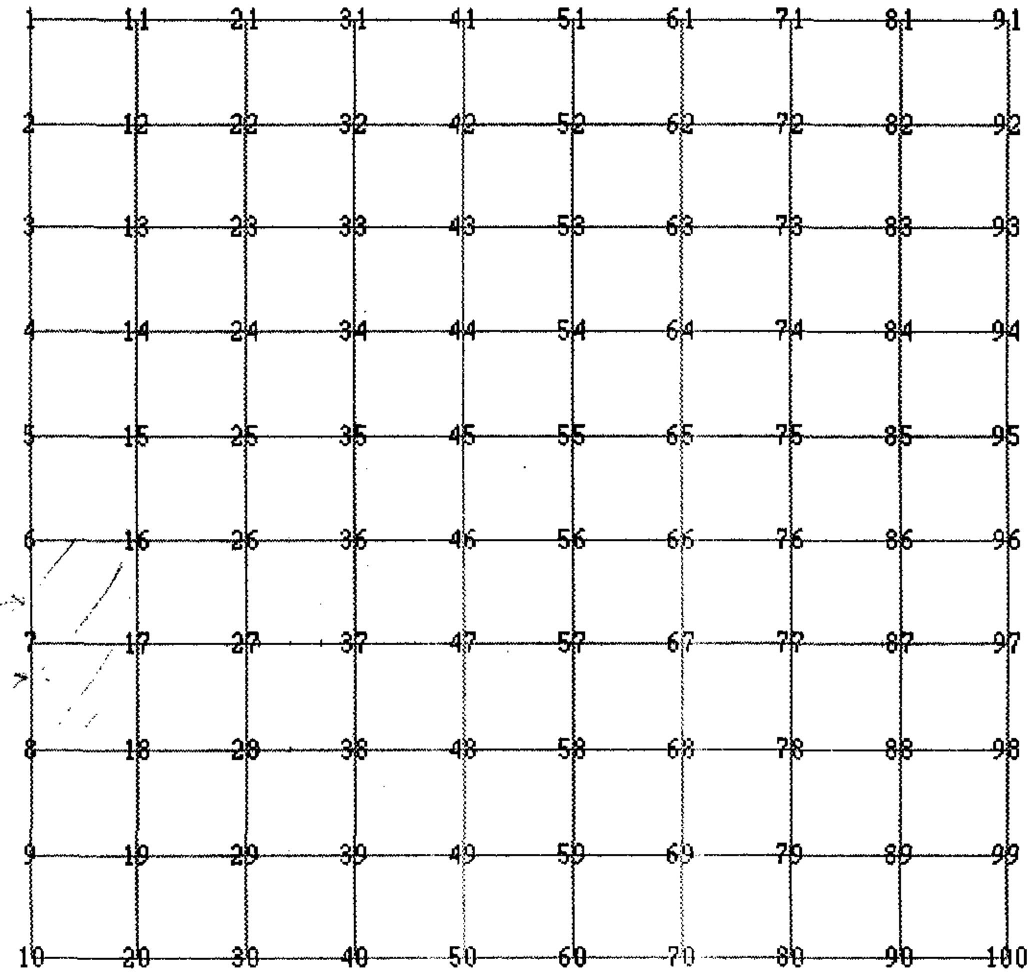
REACTIONS AT BOLT

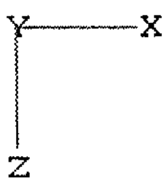
NODE	F_x
6	14.3
8	-25.6
16	16.7
17	-29.7
18	-8.9

REACTIONS AT OUTER O-RING

NODE	F_x
5	49.3
15	65.2
25	50.6
35	43.4
45	41.0
55	27.6
65	46.7
75	57.5
85	-47.0
95	-110

IMAGES-3D
Ver. 3.0
Geometry Plot

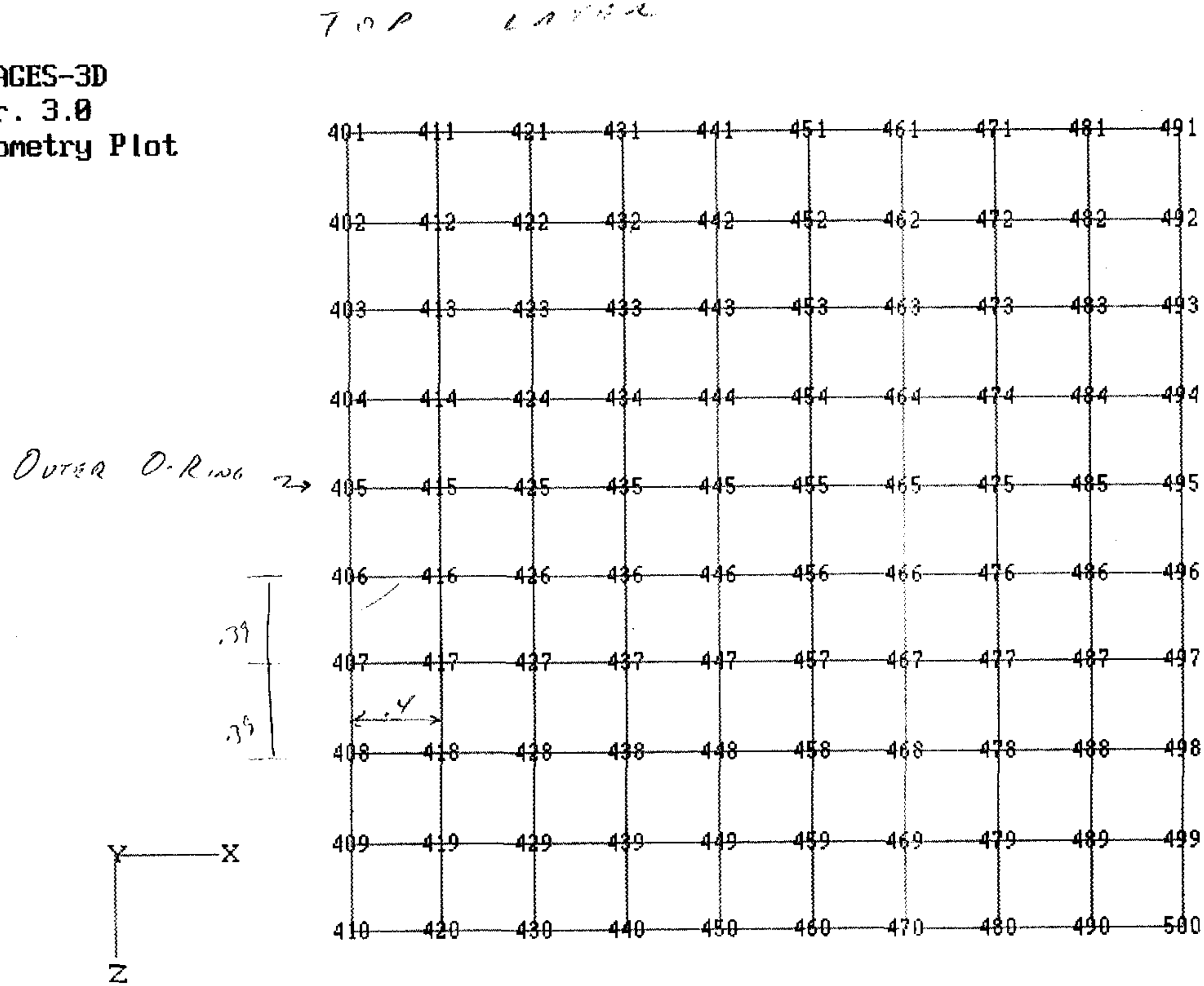




Flange - ID = 104, No. Bolts = 48
Wireframe Plot

9/29/95
15: 4:28

IMAGES-3D
Ver. 3.0
Geometry Plot

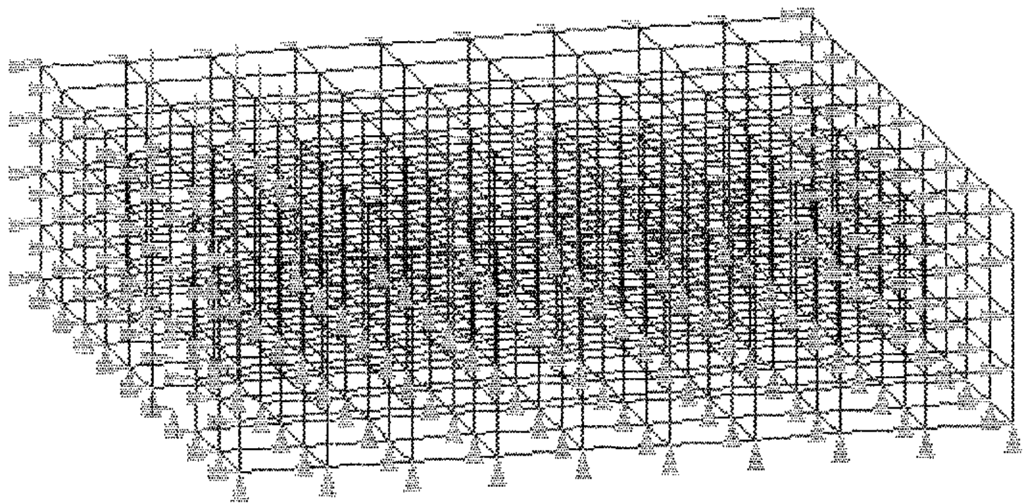


Flange - ID = 104, No. Bolts = 48
Wireframe Plot

9/29/95
14:55:49

IMAGES-3D
Ver. 3.0
L Case 1

Boundary Conditions

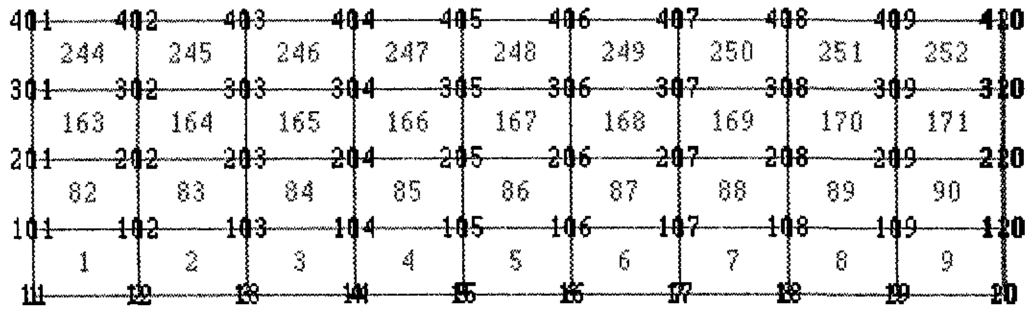


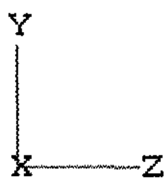
Reading......
Reading......
?t
?r,15,15
?

Auto	Blowup	Disp	Elem#	Exit	Full	Help	Load
Move	Node	Node#	Range	Rest	Rotate	Shrink	Slice

IMAGES-3D
Ver. 3.0
Geometry Plot

8-No element numbers
9-All elemt. numbers
10-Exit or 10,Size
?9
?i
X range
?-1,.6
Y range
?-1,2
Z range
?-1,6
?#
?





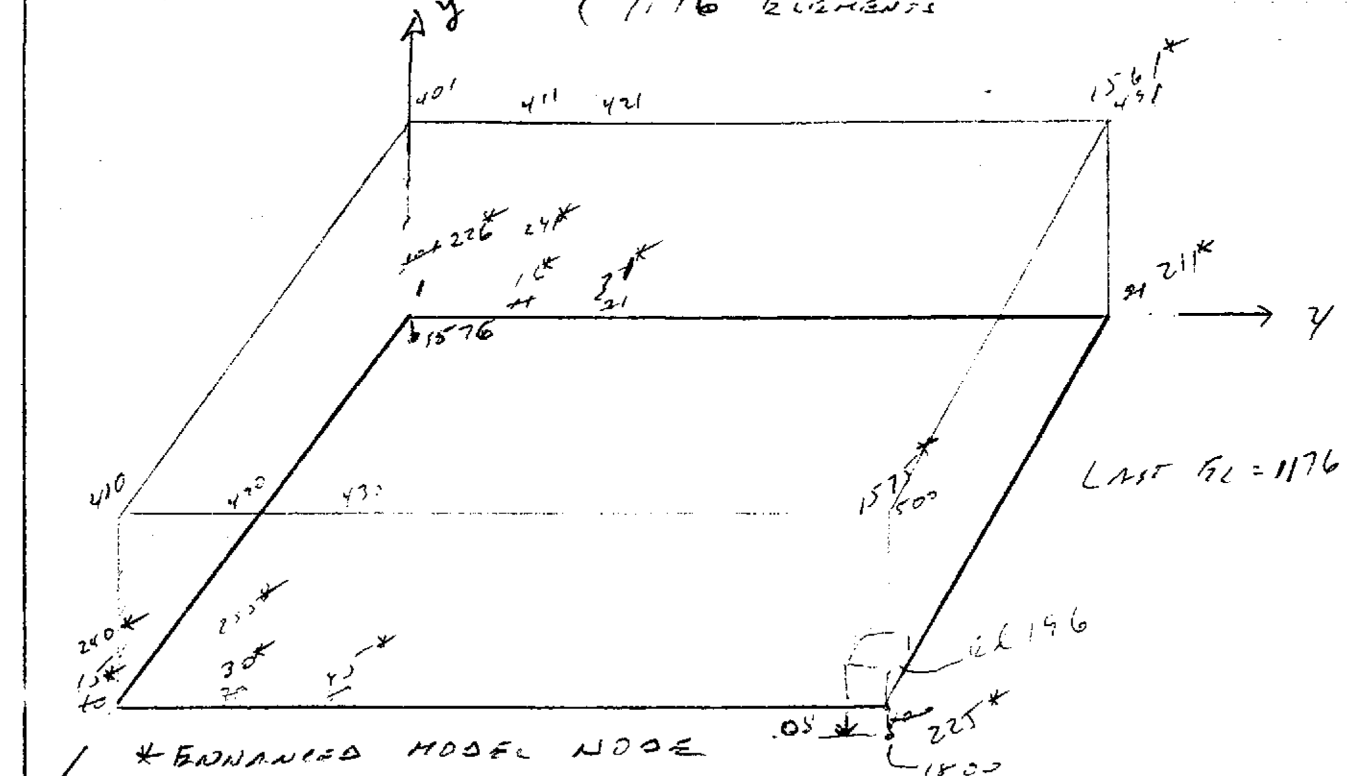
Auto	Blowup	Elem#	Exit	Full	Help	Local	Move
Node	Node#	Plot	Range	Rest	Rotate	Shrink	Slice

ENHANCED FE MODEL

THE MODEL WILL BE ENHANCED BY INCREASING THE NO. OF ELEMENT THROUGH THE FLANGE THICKNESS FROM 4 TO 6 AND BY INCREASING THE NO OF ELEMENTS ALONG EACH SIDE FROM 9 TO 14.

22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD

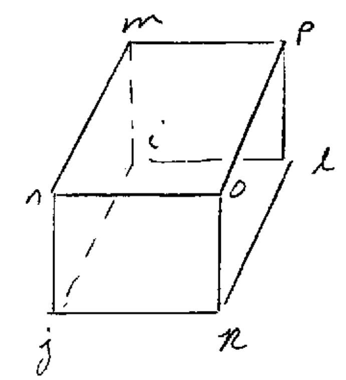
ENHANCED MODEL { 1575 NODES
1176 ELEMENTS }



* ENHANCED MODEL NODE

BOUND COND SAME AS COARSE MODEL EXCEPT THAT $\Delta_z = 0$ ALONG X AXIS ONLY

TYPICAL ELEMENT NODE NUMBERING



	i	j	k	l	m	n	o	p
1ST EL								
INITIAL MODEL →	1,	2,	12,	11,	101,	102,	112,	111
ENHANCED MODEL	1,	2,	17,	16,	226,	227,	242,	241

ENHANCED MODEL:
MAT = 1 (STAINLESS)
KEOPT = 2 / 3RD ORDER INTEGRATION

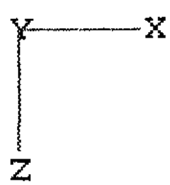
BOTTOM LAYER ELEMENTS

IMAGES-3D
Ver. 3.0
Geometry Plot

Z range
?
Z=Default
?i
X range
?
X=Default
Y range
?-.1,.2
Z range
?
Z=Default
?

DELETED ELEMENTS FOR BOLT HOLE ->

1	15	29	43	57	71	85	99	113	127	141	155	169	183
2	16	30	44	58	72	86	100	114	128	142	156	170	184
3	17	31	45	59	73	87	101	115	129	143	157	171	185
4	18	32	46	60	74	88	102	116	130	144	158	172	186
5	19	33	47	61	75	89	103	117	131	145	159	173	187
6	20	34	48	62	76	90	104	118	132	146	160	174	188
7	21	35	49	63	77	91	105	119	133	147	161	175	189
8	22	36	50	64	78	92	106	120	134	148	162	176	190
9	23	37	51	65	79	93	107	121	135	149	163	177	191
~~10~~	~~24~~	38	52	66	80	94	108	122	136	150	164	178	192
~~11~~	~~25~~	39	53	67	81	95	109	123	137	151	165	179	193
12	26	40	54	68	82	96	110	124	138	152	166	180	194
13	27	41	55	69	83	97	111	125	139	153	167	181	195
14	28	42	56	70	84	98	112	126	140	154	168	182	196



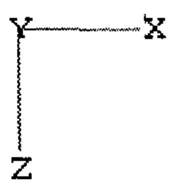
Auto Node   Blowup Node#   Elem# Plot   Exit Range   Full Rest   Help Rotate   Local Shrink   Move Slice

TOP LAYER ELEMENTS

IMAGES-3D
Ver. 3.0
Geometry Plot

9-All elemt. numbers
10-Exit or 10,Size
?9
?i
X range
?
X=Default
Y range
?.8,1.1
Z range
?
Z=Default
?

981	995	1009	1023	1037	1051	1065	1079	1093	1107	1121	1135	1149	1163
982	996	1010	1024	1038	1052	1066	1080	1094	1108	1122	1136	1150	1164
983	997	1011	1025	1039	1053	1067	1081	1095	1109	1123	1137	1151	1165
984	998	1012	1026	1040	1054	1068	1082	1096	1110	1124	1138	1152	1166
985	999	1013	1027	1041	1055	1069	1083	1097	1111	1125	1139	1153	1167
986	1000	1014	1028	1042	1056	1070	1084	1098	1112	1126	1140	1154	1168
987	1001	1015	1029	1043	1057	1071	1085	1099	1113	1127	1141	1155	1169
988	1002	1016	1030	1044	1058	1072	1086	1100	1114	1128	1142	1156	1170
989	1003	1017	1031	1045	1059	1073	1087	1101	1115	1129	1143	1157	1171
990	1004	1018	1032	1046	1060	1074	1088	1102	1116	1130	1144	1158	1172
991	1005	1019	1033	1047	1061	1075	1089	1103	1117	1131	1145	1159	1173
992	1006	1020	1034	1048	1062	1076	1090	1104	1118	1132	1146	1160	1174
993	1007	1021	1035	1049	1063	1077	1091	1105	1119	1133	1147	1161	1175
994	1008	1022	1036	1050	1064	1078	1092	1106	1120	1134	1148	1162	1176

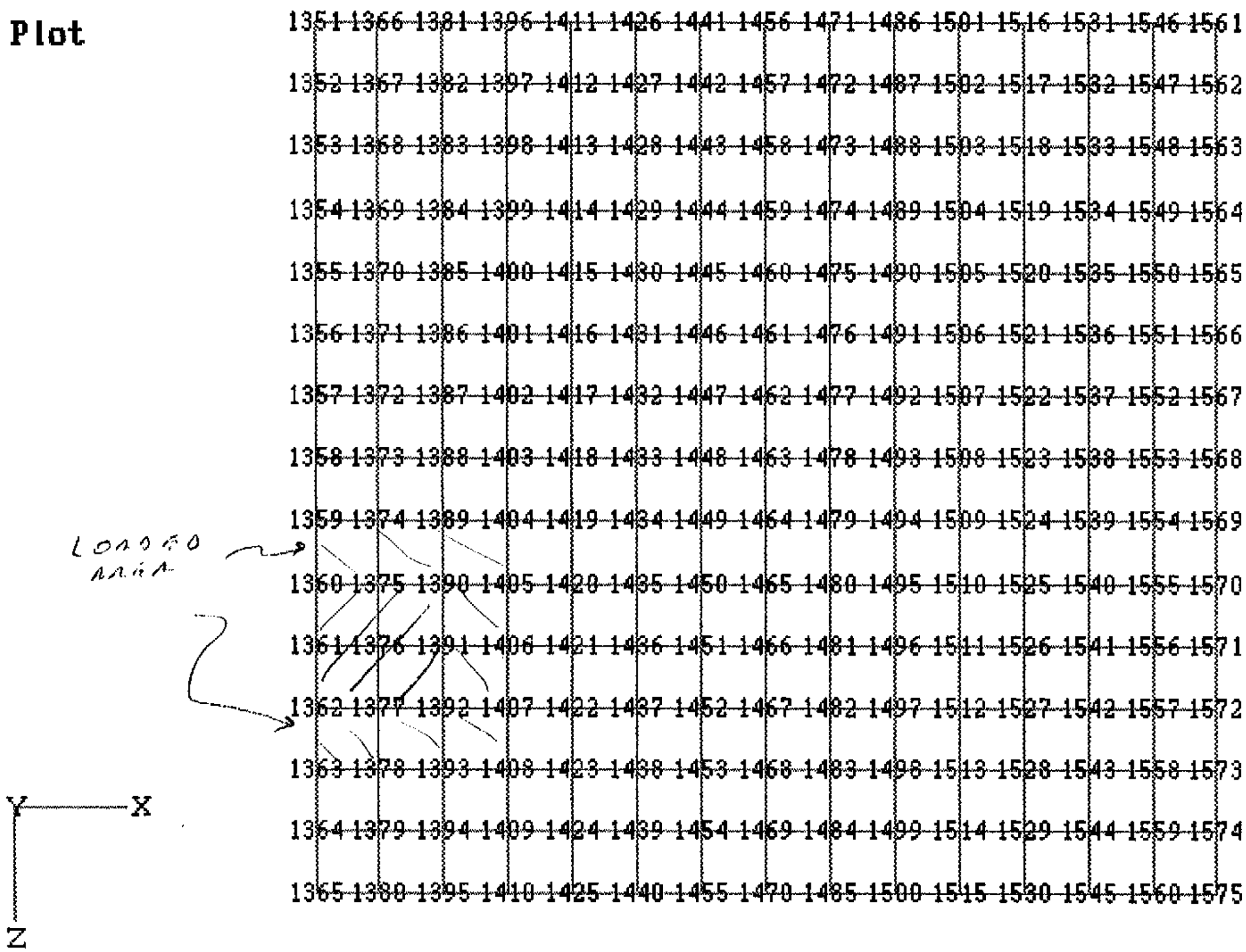


Auto Node   Blowup Node#   Elem# Plot   Exit Range   Full Rest   Help Rotate   Local Shrink   Move Slice

TOP LAYER NODES

IMAGES-3D
Ver. 3.0
Geometry Plot

?
Z=Default
?#
?i
X range
?
X=Default
Y range
?.9,1.1
Z range
?
Z=Default
?



Auto Node
Blowup Node#
Elem# Plot
Exit Range
Full Rest
Help Rotate
Local Shrink
Move Slice

22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD

DELETE ELEMENTS

10, 11, 24, 25 IN BOTTOM LAYER

INCREMENT = 196

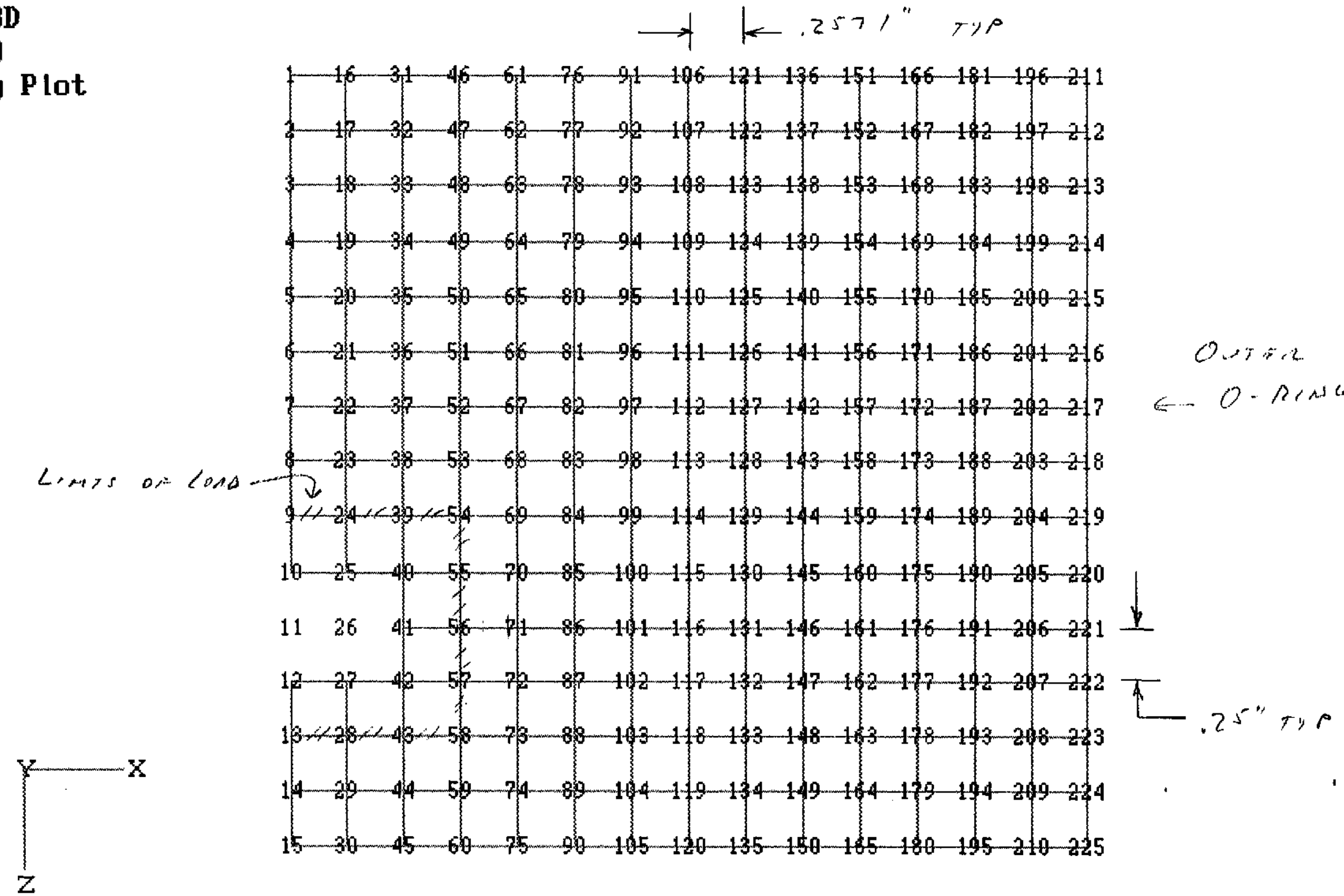
TO
990, 991, 1004, 1005 IN TOP LAYER

APPLY LOADS Fy = -1000* LB AT
TOP LAYER NODES 1359, 1360, 1362, 1363
1374, 1375, 1377, 1378
1389, TO 1393
1404 TO 1408

* UNIT LOAD SELECTED
TO DETERMINE REQUIRED
LOAD FROM OUTPUT

IMAGES-3D
Ver. 3.0
Geometry Plot

?
Z=Default
?i
X range
?
X=Default
Y range
?-.1,.1
Z range
?
Z=Default
?#
?



Auto	Blowup	Elem#	Exit	Full	Help	Local	Move
Node	Node#	Plot	Range	Rest	Rotate	Shrink	Slice

CLAMPING FORCE AT OUTER O-RING

ENHANCED MODEL

FORCE WILL BE NORMALIZED BY NODE SPACING AND TOTAL BOLT LOAD

NODE SPACING

$$S = \frac{3.6}{14} = .2571$$

TOTAL FORCE F_y = 18000 LB, SAY 18^K

$$F_y = \frac{F_{iy}}{.2571(18)} = \frac{F_{iy}}{4.629}$$

WHERE .2571 = NODE SPACING *

NODE	F_{iy}	F_y	$F_y/.25$
7	150.5	32.5	130
22	282.5	61.0	244
37	232.6	50.3	201
52	164.8	35.6	142.4
67	96.3	20.8	83.2
82	42.3	9.14	36.6
97	9.02	1.95	7.8
112	-6.58	-1.42	-5.68
127	-11.1	-2.40	-9.60
142	-10.3	-2.22	-8.88
157	-7.69	-1.66	-6.64
172	-5.14	-1.11	-4.44
187	-3.24	-.70	-2.8
202	-2.12	-.46	-1.84
217	-.88	-.19	-.76

WHERE F_{iy} = OUTPUT FORCE, FILE = FLANGE

$F_y/.25$ NORMALIZED BY NODE SPACING IN BOTH DIRECTIONS

CLAMPING FORCE AT OUTER O-RING

COARSE MODEL

$$S = \frac{3.6}{9} = .40$$

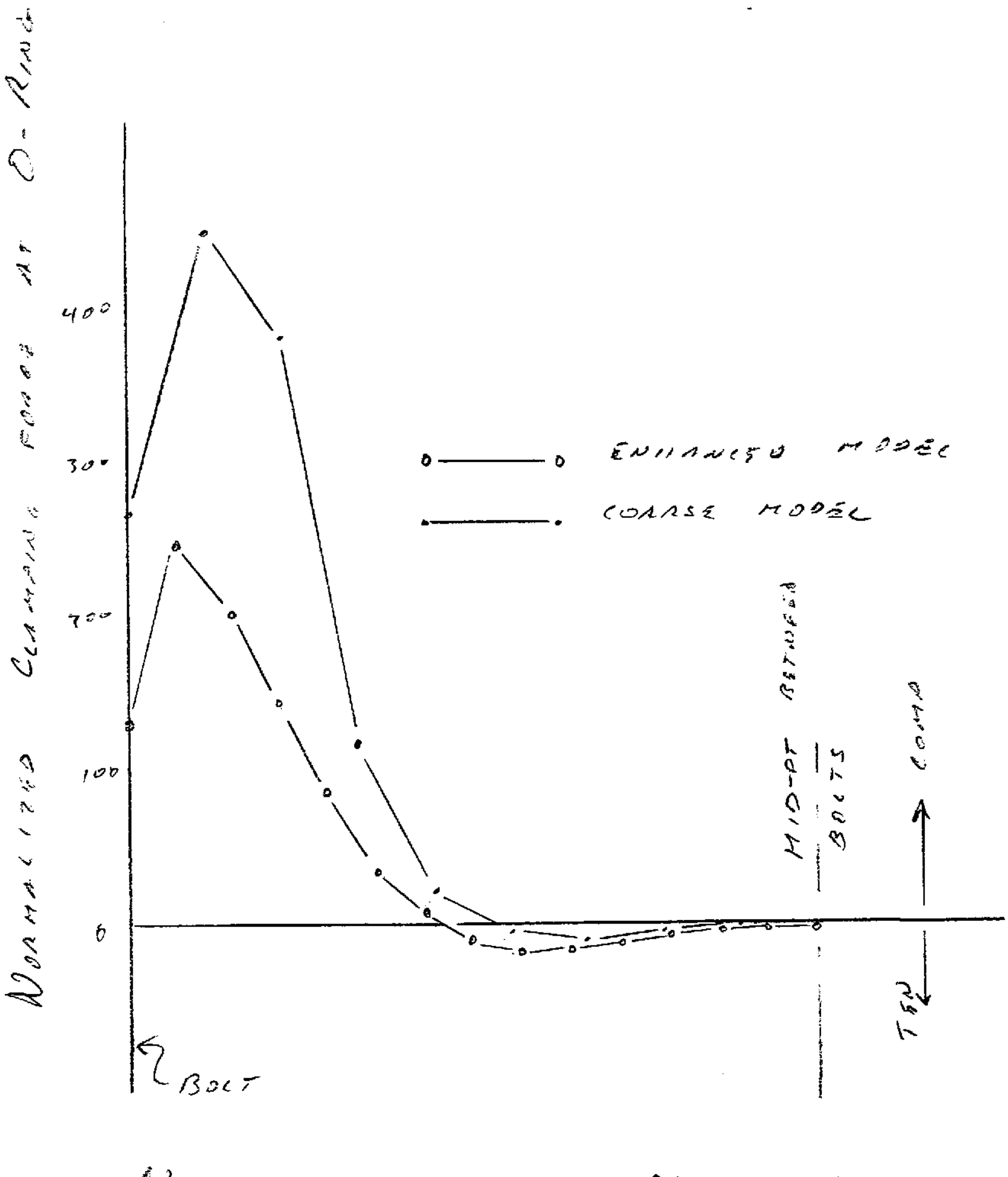
TOTAL FORCE = F_y = 14000 LB = 14K

$$F_y = \frac{F_{iy}}{.40(14)} = \frac{F_{iy}}{5.6}$$

WHERE .40 = NODE SPACING

NODE	F_{iy}	F_y	$F_y/.389$
5	577.4	103.1	265.1
15	984.6	175.8	452.1
25	612.8	109.4	281.3
35	260.7	46.6	119.8
45	54.3	9.69	24.9
55	-16.2	-2.89	-7.4
65	-21.8	-3.89	-10.0
75	-12.6	-2.25	-5.79
85	-5.46	-.975	-2.51
95	-1.64	-.293	-.753

F_{iy} = OUTPUT FORCE, FLGE = FLANGE



NOTE - HIGHER LOAD RESULTS FROM COARSE MODEL BECAUSE APPLIED BOLT FORCE IN MODEL IS CLOSER TO O-RING NODES THAN IT IS IN ENHANCED MODEL

ENHANCED MODEL W/ COMP. ONLY SPRINGS FOR FOUNDATION AND LINEAR SPRINGS FOR VITON

SPRING RATE OF VITON O RING

CROSS SECTION .275 IN

COMPRESSION 20%

= .20 (.275)

= .055 IN

GROOVE DEPTH = .201 TO .211

∴ COMP = .275 - .211

= .064

$= \frac{.064}{.275} = 23\%$ MIN

= .275 - .201

= .074

= .074/.275 = 27% MAX

COMPRESSION LOAD

F ≈ 25 TO 100 LB PER IN OF LENGTH / AT DUROMETER = 70

SEE P. A4-11 OF PARKER O-RING HANDBOOK

SAY F = 60 LB

SPRING RATE

$k = \frac{60}{.064} = 938$ LB/IN MAX PER IN OF CIRCUM

$k = \frac{60}{.074} = 810$ LB/IN MIN PER IN OF CIRCUM

USE

k = 900 LB/IN FOR 1 IN LENGTH

k = 900 (.2571) = 231 LB/IN FOR MODEL

TRY DUROMETER = 90

SQUEEZE (COMPRESSION) VARIES FROM 21 TO 29 % (P A5-13 OF PARKER SEALS HANDBOOK). FROM P. A4-11 OF THE PARKER HANDBOOK, THE COMPRESSIVE FORCE PER INCH OF SEAL RANGES FROM

70 TO 160 LB/IN

FOR .080 IN COMPRESSION (CORRESPONDING TO MAX DEFLECTION) THE STIFFNESS PER IN OF SEAL IS

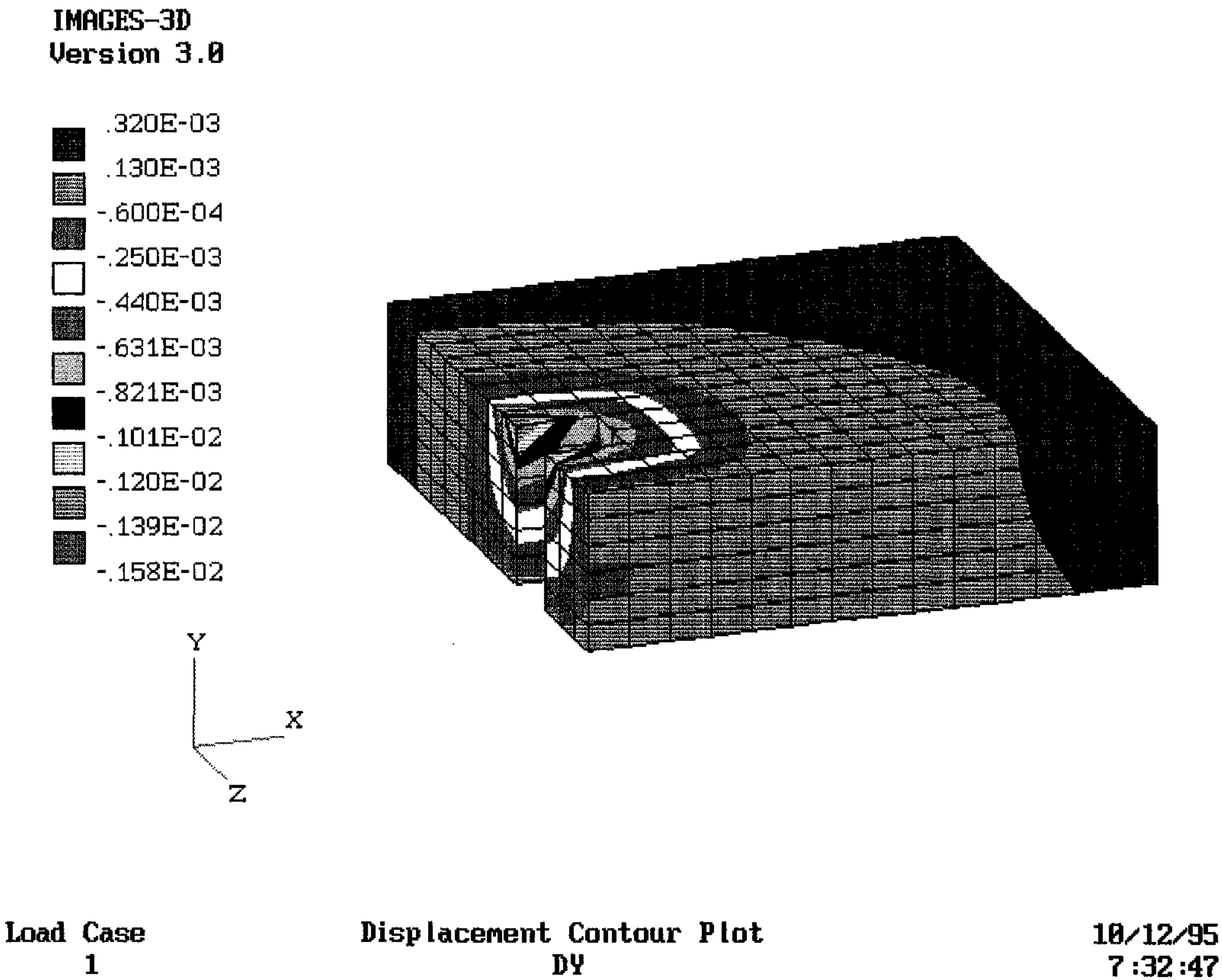
$$k = \frac{160}{.080} = 2000 \text{ LB/IN PER IN OF CIRCUM.}$$

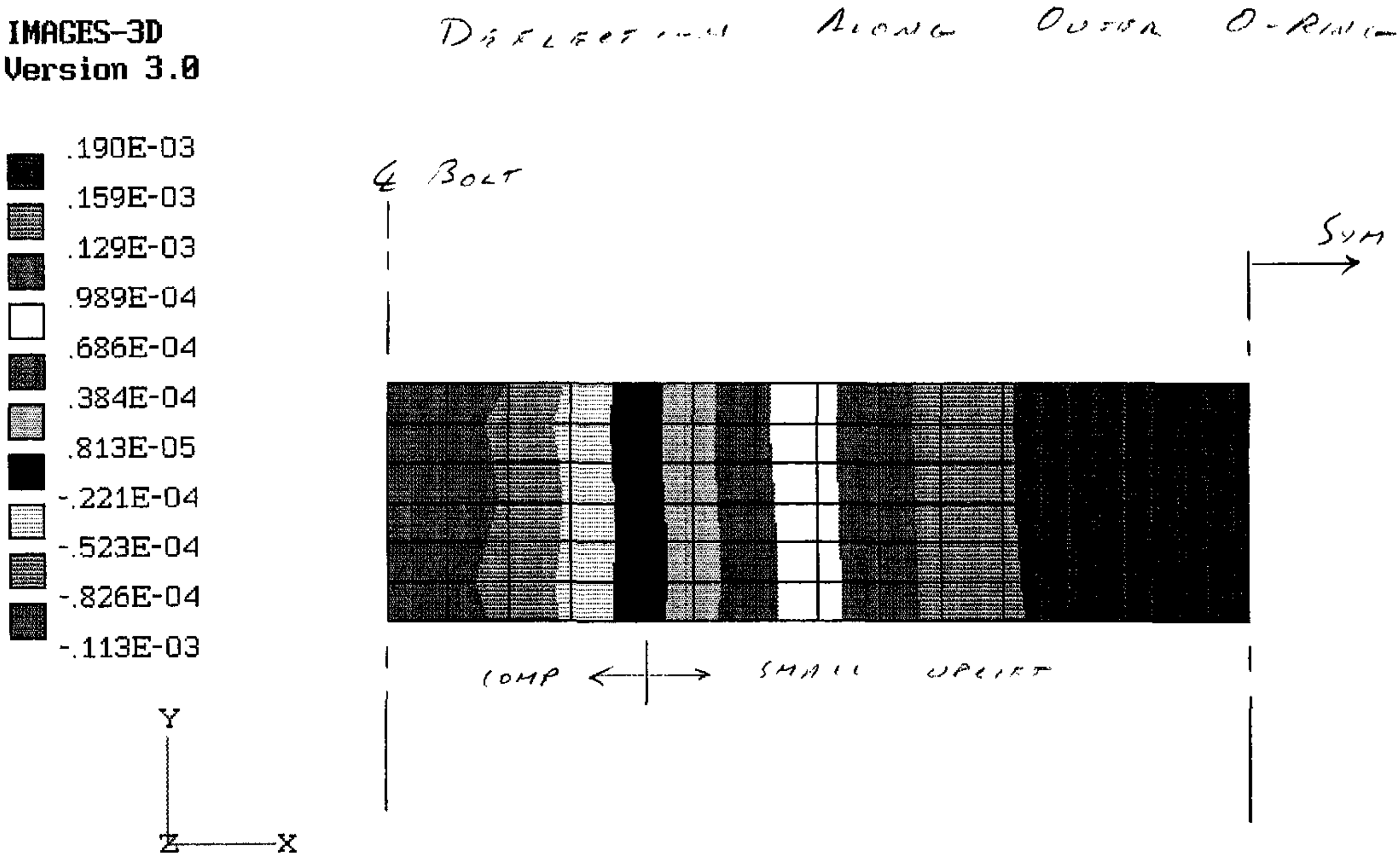
FOR DISCRETE SPRINGS SPACED .2571 IN APART IN MODEL

$$k = 2000\,(.2571)$$
WHERE .2571 = NODE SPACING
$$= 514 \text{ LB/IN} \quad *$$

ALTHOUGH THIS IS HIGHER THAN k 231 LB/IN USED IN FE ANALYSIS, IT IS ORDERS OF MAGNITUDE LESS STIFF THAN FOUNDATION SPRINGS USED TO REPRESENT THE STEEL. ∴ THE MODEL RESULTS WOULD NOT CHANGE SIGNIFICANTLY IF THE HIGHER VALUE OF k IS USED.

* SEE CALC 018, P. 5 FOR UPDATED VALUE OF k. RESULTS WILL NOT BE AFFECTED.


IMAGES-3D
Version 3.0
.320E-03
.130E-03
-.600E-04
-.250E-03
-.440E-03
-.631E-03
-.821E-03
-.101E-02
-.120E-02
-.139E-02
-.158E-02
Y
X
Z
Load Case
1
Displacement Contour Plot
DY
10/12/95
7:32:47



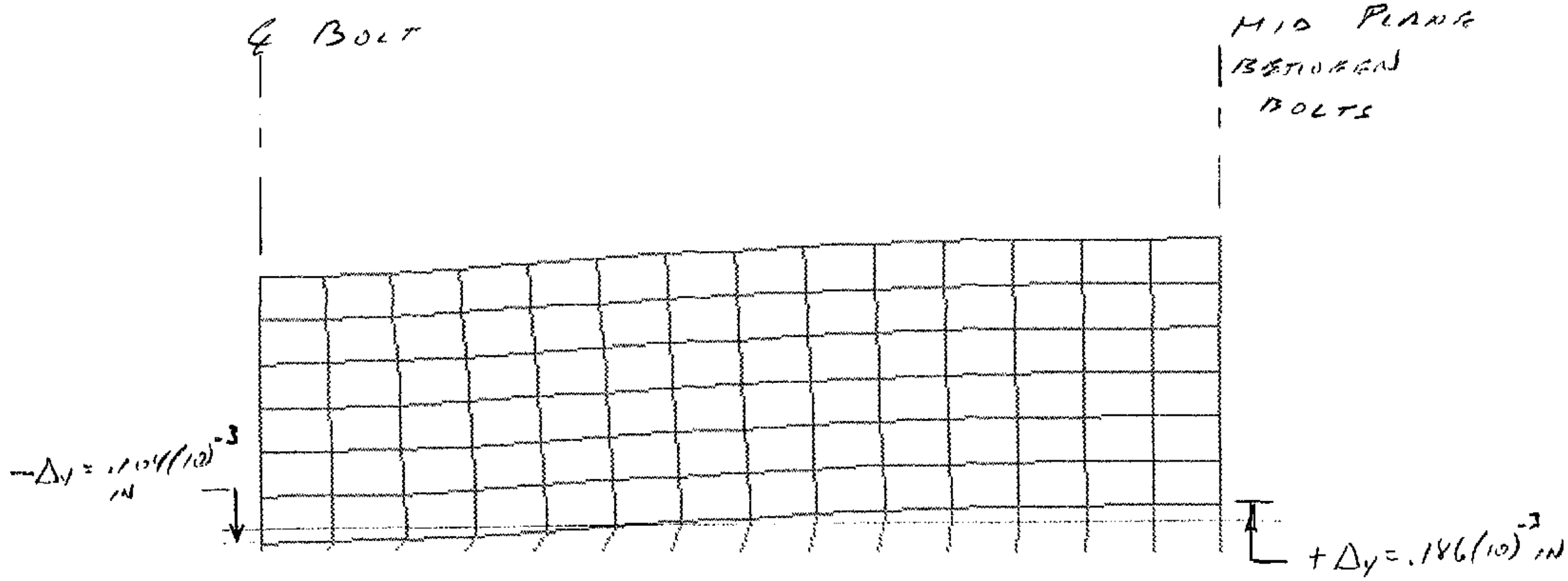
Load Case 1

Displacement Contour Plot
DY

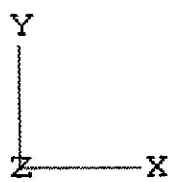
10/12/95
7:40: 0

IMAGES-3D
Version 3.0

DEFLECTION ALONG OUTER O-RING



?c
Scale factor
?500
?

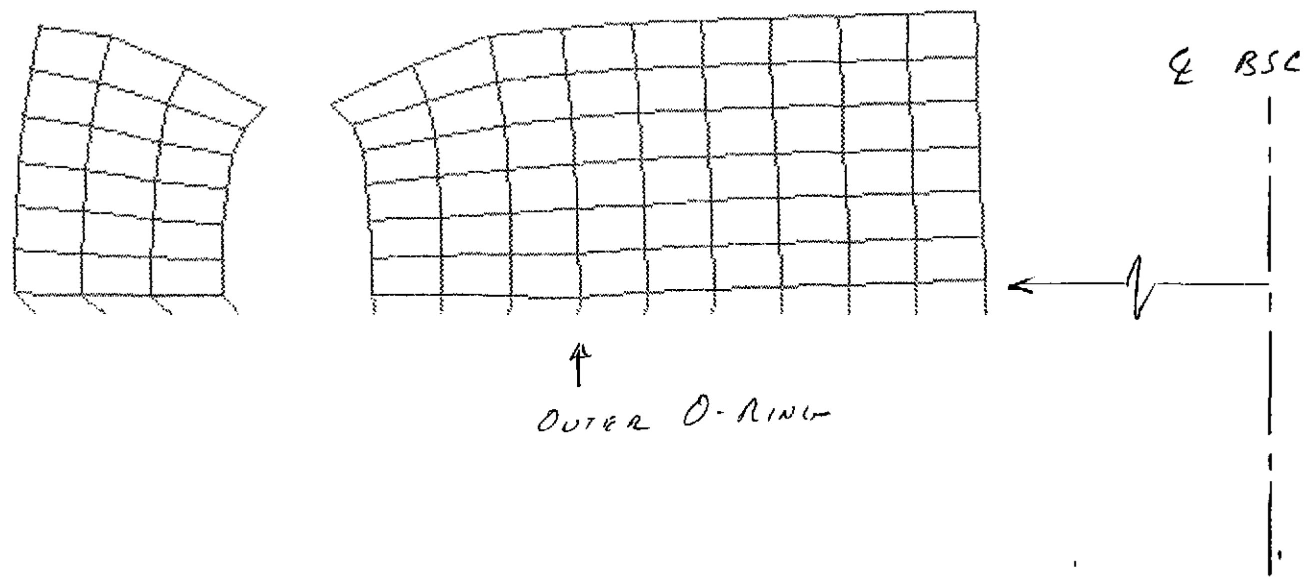


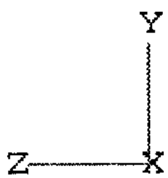
Animate	Auto	Blowup	Elem#	Exit	Full	Help	Lcase
Move	Node#	Origin	Plot	Range	Rotate	Scale	Slice

IMAGES-3D
Version 3.0

Deflection Along Radial Line
Through Bolt Hole

X range
?-.1,.1
Y range
?
Y=Default
Z range
?
Z=Default
?r,,-90
?c
Scale factor
?200
?



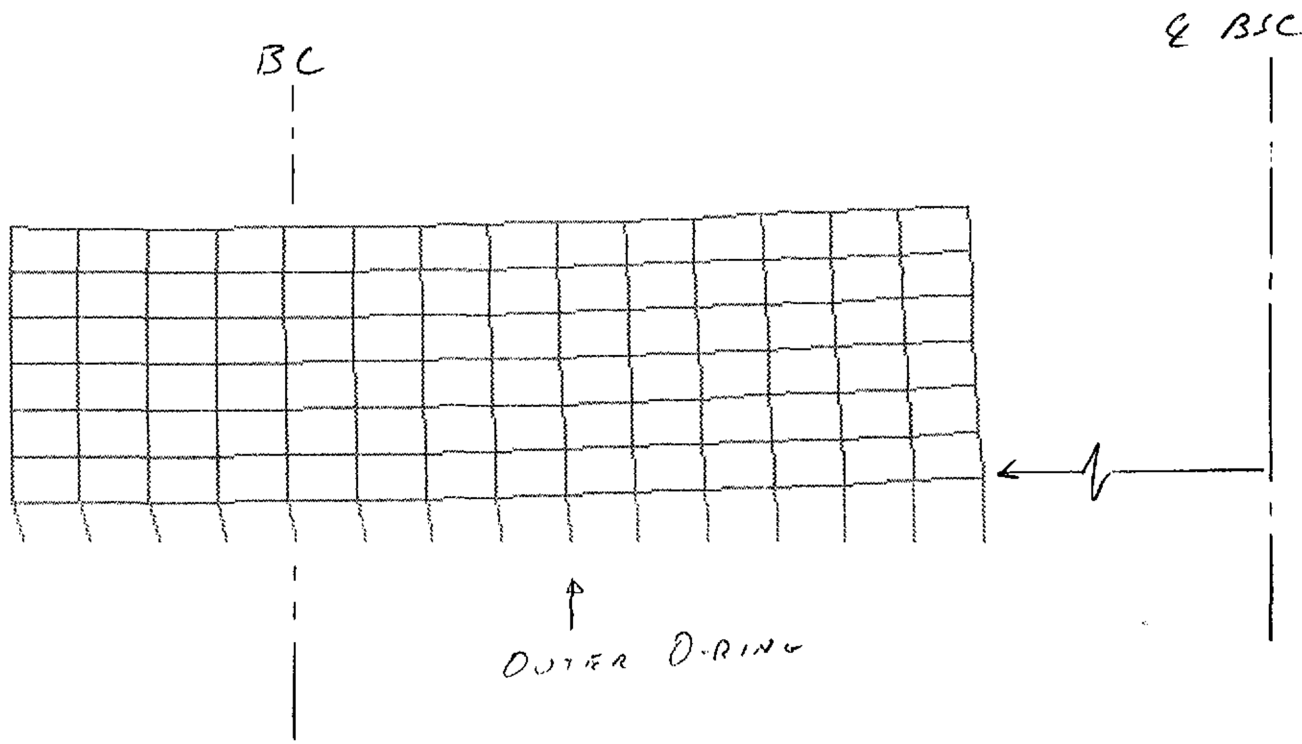


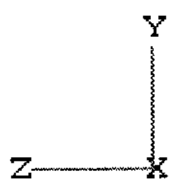
Animate Auto Blowup Elem# Exit Full Help Lcase
Move Node# Origin Plot Range Rotate Scale Slice

IMAGES-3D
Version 3.0

Deflection Along Radial Line
Midway Between Bolts

```
?c
Scale factor
?500
?i
X range
?3.3,3.5
Y range
?
Y=Default
Z range
?
Z=Default
?
```






```
Animate  Auto    Blowup  Elem#  Exit   Full    Help   Lcase
Move     Node#   Origin  Plot   Range  Rotate  Scale  Slice
```

ENHANCED MODEL

FOUNDATION STIFFNESS = 10 x 10^6 LB/IN

OUTER O-RING DEFLECTIONS

NODE	Δy	EXP (10^EXP)
7	-.104	-3
22	-.947	-4
37	-.768	-4
52	-.470	-4
67	-.125	-4
82	.239	-4
97	.572	-4
112	.876	-4
127	.114	-3
142	.136	-3
157	.154	-3
172	.168	-3
187	.178	-3
202	.184	-3
217	.186	-3

FROM COMPUTER OUTPUT
FILE: FLANGE

22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD

DEFLECTIONS AT BOLT

BOTTOM LAYER NODES TOP LAYER NODES

NODE	Δ_Y BOT		NODE	Δ_Y TOP		Δ_{YT}/Δ_{YB}
9	-.540	-4	1359	-.399	-3	14.8
10	-.353	-4	1360	-.153	-2	43
12	-.382	-4	1362	-.158	-2	41
13	-.542	-4	1363	-.822	-3	15
24	-.851	-4	1374	-.576	-3	6.8 *
25	-.592	-4	1375	-.101	-2	17.1
27	-.633	-4	1377	-.105	-2	16.6
28	-.847	-4	1378	-.596	-3	7.0
39	-.702	-4	1389	-.500	-3	7.1
40	-.657	-4	1390	-.648	-3	10.2
41	-.464	-4	1391	-.390	-3	19.2
42	-.664	-4	1392	-.684	-3	10.3
43	-.683	-4	1393	-.514	-3	7.57
54	-.510	-4	1404	-.409	-3	8.0
55	-.622	-4	1405	-.468	-3	7.5
56	-.664	-4	1406	-.484	-3	7.3
57	-.618	-4	1407	-.474	-3	7.7
58	-.488	-4	1408	-.416	-3	8.5

MIN RATIO OF TOP TO BOTTOM DEFLECTION = 6.8
∴ FOUNDATION STIFFNESS, K = 10×10⁶ LB/IN
IS REASONABLE

MAXIMUM LIFTOFF AT OUTER O-RING

IF THE DEFLECTION IN THE NEG DIRECTION IS ACTUALLY 0 AND THE RELATIVE Δy REMAINS CONSTANT, THE MAXIMUM SEPARATION BETWEEN THE FLANGES IS

$$\Delta_y = [.186 - (-.104)] \times 10^{-3}$$
$$= .290(10)^{-3}$$

THIS IS DOUBLED FOR 2 FLANGES

$$\Delta_y = 2(.290)(10)^{-3}$$
$$= .58(10)^{-3}$$
$$= .00058 \text{ IN}$$

THIS COMPARES TO THE MINIMUM O-RING COMPRESSION OF .064 IN *

THE COMPRESSION IS MORE THAN 100 TIMES THE FLANGE SEPARATION.

$$\frac{.064}{.00058} = 110$$

* CALCULATED VALUE, FROM PARKER HANDBOOK, MAXIMUM COMPRESSION IS .080 IN FOR .25 IN O-RING.

FOR THE ANALYSIS, THE BOLT LOAD IS 18000 LB FOR THE HALF MODEL OR 36000 LB PER BOLT. THE TOTAL VACUUM THRUST ON THE FLANGE IS

$$F_y = \pi r^2 (14.7)$$

$$r = \frac{108}{2} = 54$$

= RADIUS TO OUTER O-RING

$$F_y = \pi (54)^2 (14.7)$$

$$= 134,700 \text{ LB}$$

FOR 48 BOLTS THE FORCE PER BOLT IS

$$F_{BOLT} = \frac{134700}{48}$$

$$= 2800 \text{ LB} < 36000$$

∴ BOLT PRELOAD CAN BE REDUCED TO, SAY, 6000 LB, AND THE BOLTS WILL HAVE A NET TENSION UNDER THE VACUUM LOAD

$$F_T = 6000 - 2800 = 3200 \text{ LB}$$

FOR A PRELOAD OF 6000 LB THE UNIFORM LOAD ON THE INNER AND OUTER O-RINGS IS

$$w = \frac{48(6000)}{\pi(108) + \pi(105)}$$ (ASSUME RIGID FLGS)

$$= 430 \text{ LB/IN} \gg 160 \text{ LB/IN}$$

MAX LOAD REQUIRED TO COMPRESS 1/4 VITON O-RING FOR DUROMETER 90

∴ VITON WILL BE COMPRESSED INTO GLAND

FOR FLANGES OF OTHER DIAMETERS, MAINTAIN BOLT SPACING AT 7.2 IN

FLANGE ID	BC DIAM D_{BC}	BC CIRCUM $\pi D_{BC} = C_{BC}$	NO. OF BOLTS $n = C_{BC}/7.2$	USE N=
104 1/4	110	345.6"	48	48
84 1/4	90	282.7	39	40
72 1/4	78	245.0	34	36
60 1/4	66	207.3	29	30
48 1/4	54	169.6	24	24
44 1/4	50	157.1	22	24
30 1/4	36	113	16	16

BOLT DIAMETER = 7/8"
WITH 1" HOLE

TOP FLANGE = 1" THICK
BOTTOM " = 1 1/4" THICK

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA	ENGINEERING CALCULATIONS	NO: V049-1-017
		PAGE 1 OF 10

REV.	DEO #	DATE	BY:	CHECK
0	0/35	12/6/95	RDC	AGR

TITLE: Beam Splitter Chamber Finite Element Analysis of 1 in Thick Flange for Bolt Preload plus Positive Pressure

BY: R. D. Ciatto | DEPT.: 744

PROJECT: LIGO Vacuum Equipment | PROJECT NO: V59049

PURPOSE: To determine if bolt clamping force combined with a positive internal pressure of 2 psi results in flange deformation that could allow leakage at o-rings.

METHOD: The same method used for the preload - only analysis (V049-1-016) is used in this analysis. Pressure load is added to the finite elements model.

ASSUMPTIONS: Same as preload-only analysis (calc. V049-1-016). The fine model is used with compression-only springs at the surface between flanges.

INPUTS: LIGO project design sketches and drawings. Durometer of 90 for Viton o-rings.

REFERENCES: 1. Parker O-Ring Handbook.
2. IMAGES-3D, Version 3.0, R.L. Cloud and Associates

CALCULATIONS: (SEE ATTACHED)

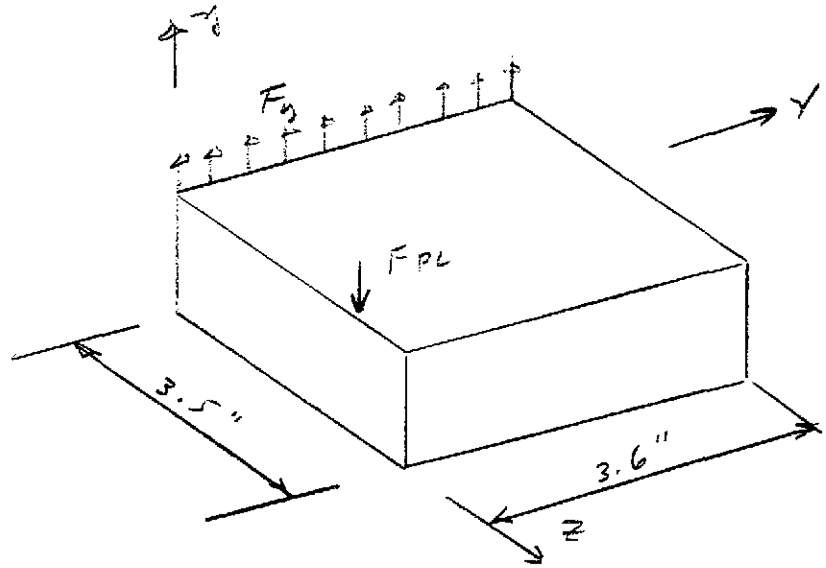
CONCLUSIONS: The gap opening between flanges is negligible and does not permit leakage of o-ring seals.

NOTES: Computer file is FLANGE A.

FLANGES - 3D FE ANALYSIS FOR BOLT PRELOAD PLUS POS PRESSURE

ANALYSIS FOR POSITIVE PRESSURE USING 3D FE MODEL - 104 ID FLANGE USED

PRESSURE LOAD FROM ATTACHED CYLINDER



$$F_y = \frac{p\,r}{2}$$

$$p = 2 \text{ psi}$$

$$r = \frac{104.25}{2}$$

$$F_y = \frac{2\left(\frac{104.25}{2}\right)}{2}$$

$$= 52.1 \text{ LB/IN}$$

NODE SPACING ALONG Y-AXIS IS .2571

$F_{yi} = 52.1(.2571)$
$= 13.4$ LB NODES 1366 TO 1546, INC 15

$= \frac{13.4}{2} = 6.7$ AT $Y = 0, 3.6$
END NODES 1351 & 1561

PRELOAD FORCE

$F_{PL} = 6000^{*}$ LB FOR 7/8" ∅ BOLTS

14 NODES ARE LOADED, 4 NODES ARE AT PLANE OF SYMMETRY (Y = 0)

$$F_{PL} = 14 F_{yi} + 4 \frac{F_{yi}}{2}$$

$$= 16 F_{yi} \qquad @\ Y = 0$$

$$F_{yi} = \frac{6000}{16} = -375 \text{ LB}$$

NODES 1374, 1375, 1377, 1378
1389 TO 1393
1404 TO 1408

$$F_y = \frac{375}{2} = -188 \text{ LB}$$ NODES 1359, 1360, 1362, 1363

THE TOTAL PRESSURE FORCE AT THE CYLINDER IS

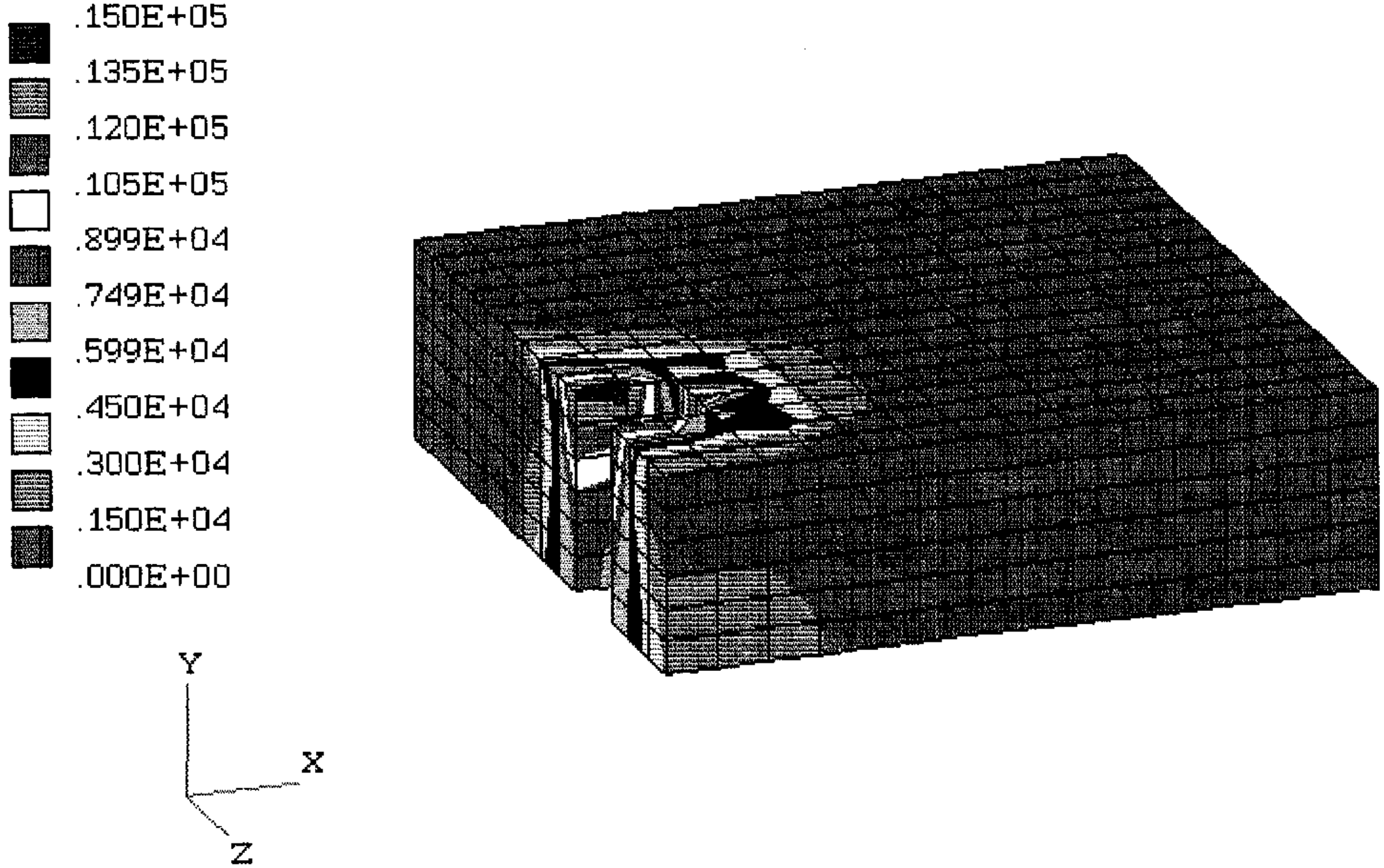
$$F_y = 52.1 \times 3.60 = 188 \text{ LB}$$

$$<< F_{PL} = 6000 \text{ LB}$$

∴ IT WILL BE ASSUMED THAT PRESSURE DOES NOT CHANGE THE PRELOAD FORCE

* ACTUAL LOAD IS 3000 LB FOR 1/2 BOLT. THIS ANALYSIS IS CONSERVATIVE FOR EVALUATION OF FLANGE SEPARATION.

IMAGES-3D
Version 3.0



Load Case
1

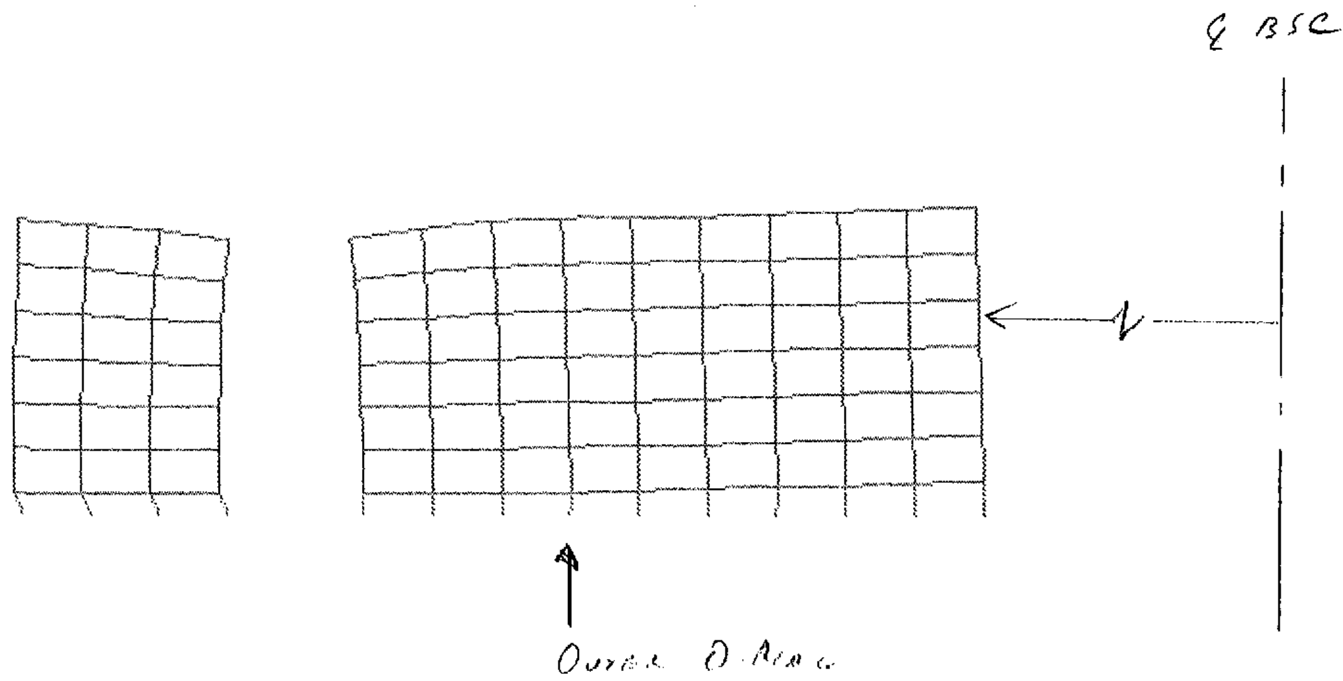
Stress Contour Plot
Stress Intensity

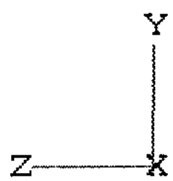
10/18/95
7:48:50

IMAGES-3D
Version 3.0

Internal Pressure
Deflection Along Radial Line
Through Bolt Hole

?C
Scale factor
?200
?I
X range
?-.1,.1
Y range
?
Y=Default
Z range
?
Z=Default
?



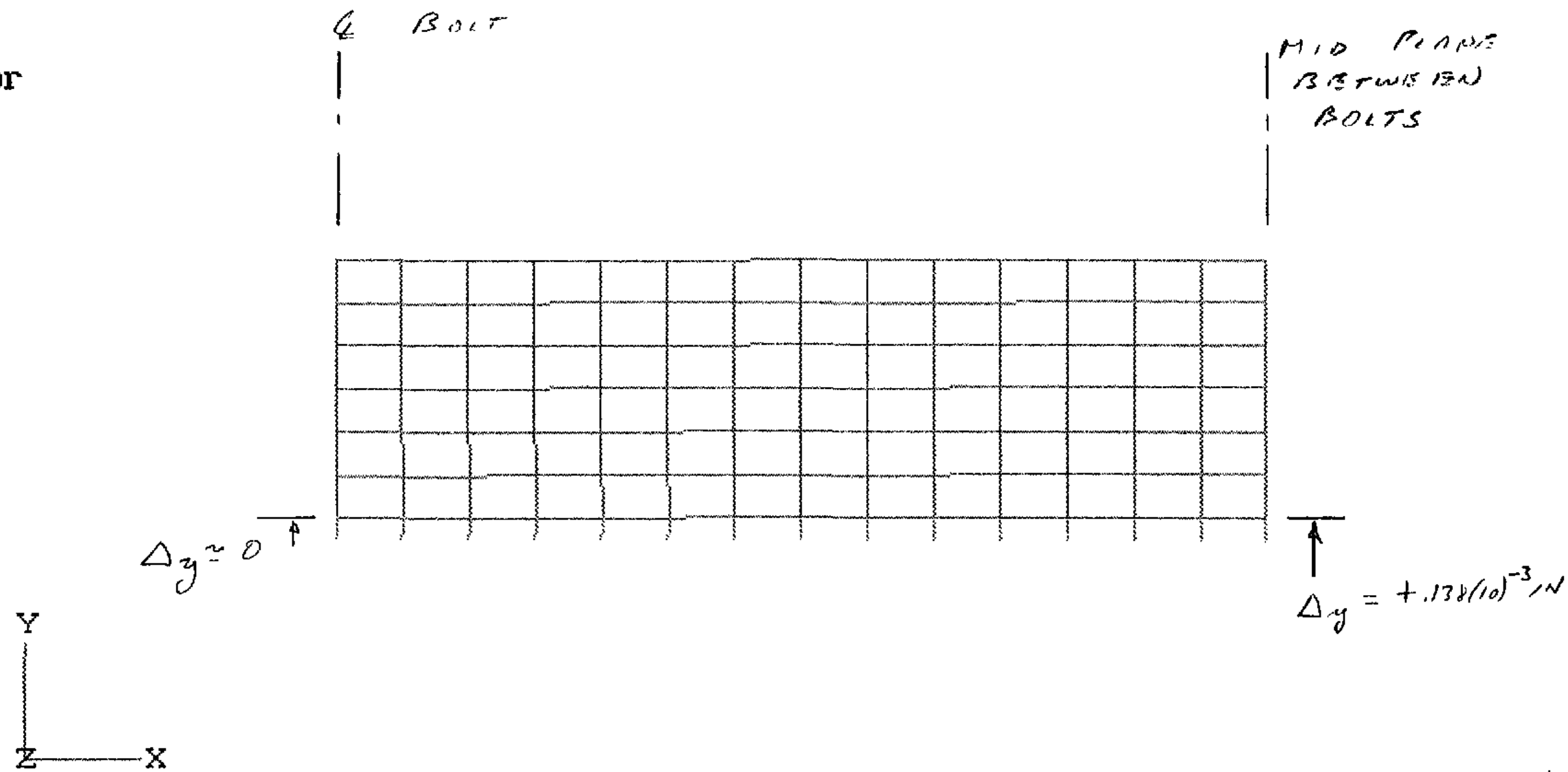


Animate	Auto	Blowup	Elem#	Exit	Full	Help	Lcase
Move	Node#	Origin	Plot	Range	Rotate	Scale	Slice

IMAGES-3D
Version 3.0

Internal Pressure
Deflection Along Outer O-ring

?C
Scale factor
?100
?I
X range
?
X=Default
Y range
?
Y=Default
Z range
?1.4,1.6
?



Animate Auto Blowup Elem# Exit Full Help Lcase
Move Node# Origin Plot Range Rotate Scale Slice

DEFLECTIONS AT OUTER O-RING

NODE	Δy
7	$.256 \times 10^{-5}$
217	$.138 \times 10^{-3}$

SPRING LOADS ALONG OUTER O-RING

NODE	F_y *	EXPONENT
7	.152	-2
22	.271	-2
37	.615	-2
52	.120	-1
67	.197	-1
82	.281	-1
97	.366	-1
112	.445	-1
127	.515	-1
142	.574	-1
157	.624	-1
172	.662	-1
187	.690	-1
202	.706	-1
217	.711	-1

* OUTPUT FROM FILE: FLANGE A

MAXIMUM SEPARATION OF FLANGES

AT NODE 717 WHICH IS MIDWAY BETWEEN 2 BOLTS ALONG THE OUTER O-RING, THE DEFLECTION IS

$$\Delta_y = .138(10)^{-3} \text{ IN}$$

WHICH INDICATES A VERY SLIGHT OPENING OF THE GAP UNDER INTERNAL PRESSURE OF 2 PSI. FOR THE 2 FLANGES, THIS IS DOUBLED

$$\Delta_y = 2(.138)(10)^{-3}$$

$$= .276(10)^{-3} \text{ IN}$$

THIS COMPARES TO THE MINIMUM COMPRESSION OF .058 IN FROM PARKER HANDBOOK (P. A5-13)

$$\frac{58(10)^{-3}}{.276(10)^{-3}} = 210$$

∴ OPENING OF GAP << INITIAL COMPRESSION OF O-RING AND O-RING REMAINS COMPRESSED DURING INTERNAL PRESS = 2 PSIG.

MAX TENSILE LOAD IS $T = \frac{.711(10)^{-1} \text{ LB}}{.257 \text{ IN}}$

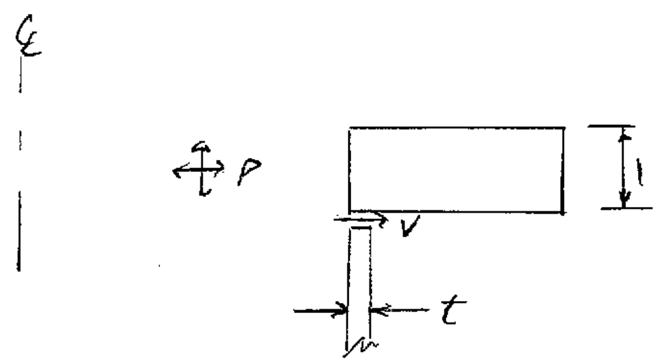
$$T = .280 \text{ LB/IN}$$

THIS COMPARES TO MIN SEALING LOAD OF 70 LB/IN FOR DUROMETER = 60 (PARKER HANDBOOK P A4-11)

$$C - T = 70 - .28 = 69.7 \text{ LB/IN}$$

HOOP STRESS IN FLANGE RING DUE TO INTERNAL PRESSURE

3D FE MODEL DOES NOT COMPUTE HOOP STRESS



$p \approx 2$ psi

$V = .660\ p \sqrt{rt}$

$r = \frac{ID}{2} + \frac{t}{2}$

$ID = 104.75$ (LARGEST)

$t = .5$ (MAX THICKNESS)

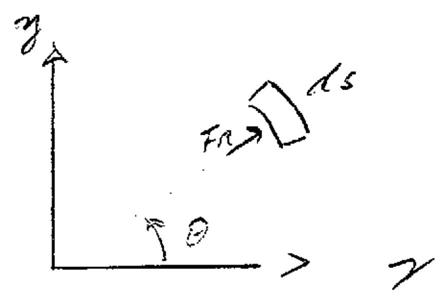
$r = \frac{104.25}{2} + \frac{.5}{2} = 52.4$

$V = .660\ (2) \sqrt{52.4(.5)}$

$= 6.8$ LB/IN

TOTAL RADIAL LOAD

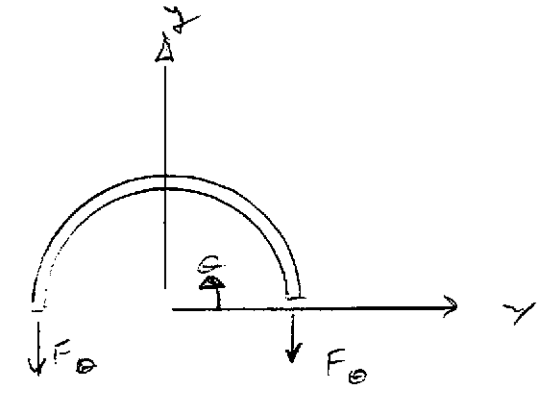
$F_R = p(1) + V = 2 + 6.8 = 8.8$ LB/IN



$dF_y = F_R\, ds \sin\theta$

$ds = r\, d\theta$

$dF_y = F_R\, r \sin\theta\, d\theta$



$$F_\theta = -\int_0^{\frac{\pi}{2}} dF_y =$$

$$= -\int_0^{\frac{\pi}{2}} F_R \, r \sin\theta \, d\theta$$

$$= -F_R \, r \int_0^{\frac{\pi}{2}} \sin\theta \, d\theta$$

$$= + F_R \, r \cos\theta \Big|_0^{\frac{\pi}{2}}$$

$$= + F_R \, r \, (1)$$

$$= + F_R \, r$$

$$= 8.8(52.4) = 461 \text{ LB}$$

FLANGE CROSS SECTION

$$A = 3.5(1) = 3.5 \text{ IN}^2$$

HOOP STRESS

$$f_\theta = \frac{461}{3.5} = 132 \text{ psi NEG}$$

THIS STRESS & STRESS INTENSITY PLOT CONFIRMS THAT STRESS IN FLANGE ARE LOW, EXCEPT AT LOCAL AREA UNDER BOLT

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-018 PAGE 1 OF 18
REV.	DEO #	DATE	BY:	CHECK	TITLE: Beam Splitter Chamber - Finite Element Analysis of 3/4 in. Thick Flange	
0	139	12/6/95	RDC	AGR		
					BY: R. D. Ciatto	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE: To determine if thin flange (3/4 in) can withstand clamping force due to bolt preload plus 2 psi internal pressure without leakage at o-ring.

METHOD: The 3-D finite element model used for the 1 in. thick flange is modified to obtain a 3/4 thick flange model.

ASSUMPTIONS: Same as 1 in. thick flange analysis.

INPUTS: LIGO project design sketches and drawings.
Durometer of 90 for Viton o-rings.

REFERENCES: 1. Parker O-Ring Handbook.
2. IMAGES 3D, Version 3.0, R.L. Cloud & Associates.

CALCULATIONS: (SEE ATTACHED)

CONCLUSIONS: Although the gap opening between flanges is greater for the 3/4 in. thickness than it is for the 1 in. thickness, it still does not permit leakage. If a 1 in. or 1 1/4 in. flange is damaged by gouging, its thickness may be reduced by as much as 1/4 in by machining.

NOTES: If a flange requires machining, it location should be evaluated for tensile force.
Computer file is FLANGE B.

FLANGE ANALYSIS USING 3D MODEL FOR THICKNESS = 3/4 IN.

THE FOLLOWING ANALYSIS USES THE SAME MODEL AS THE ENHANCED MODEL BUT THERE ARE 5 ELEMENTS THROUGH THE THICKNESS RATHER THAN 6 BECAUSE THE FLANGE IS THINNER, 3/4" VS. 1".

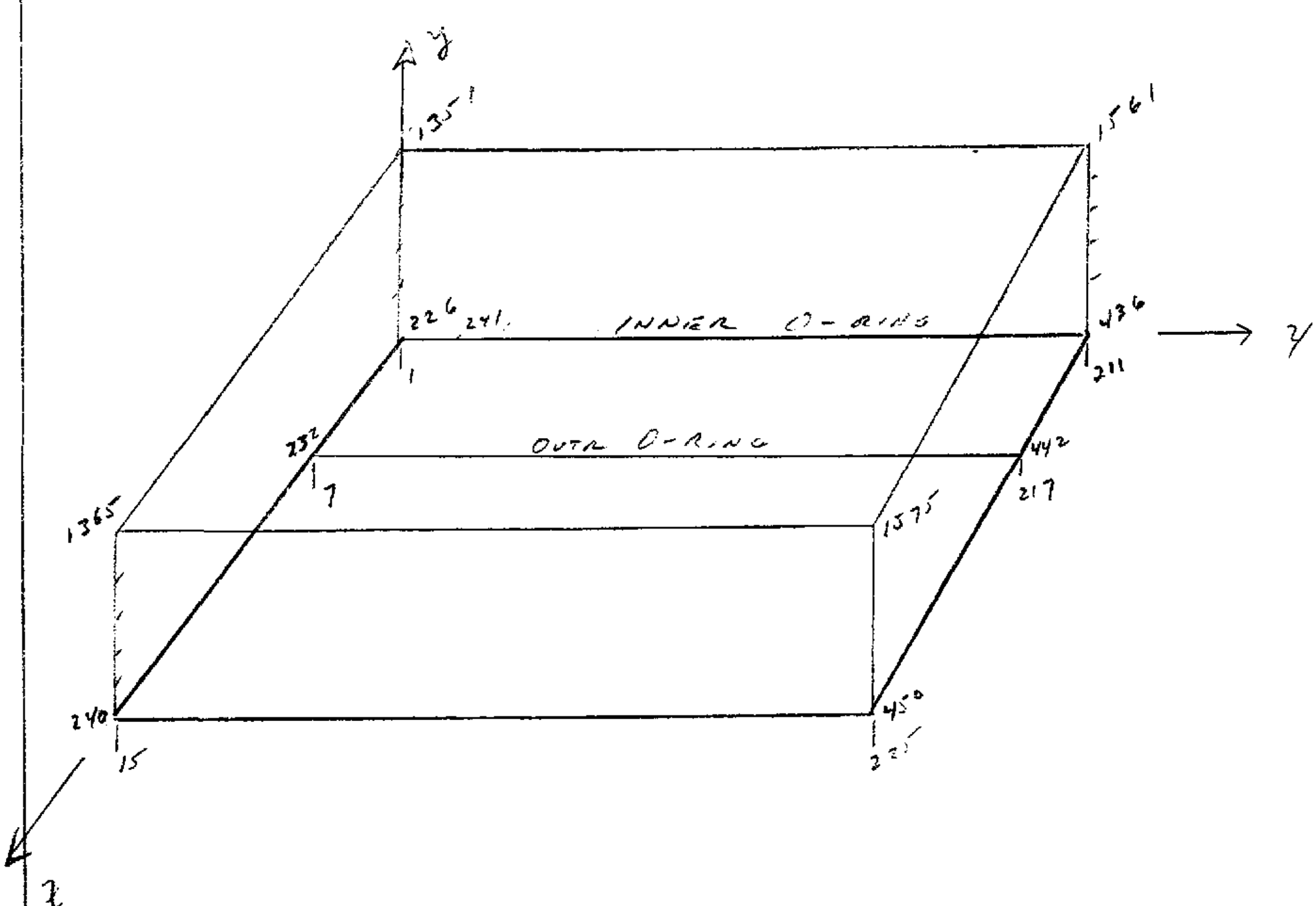
LOADS INCLUDE THE 8500* LB BOLT PRELOAD AND THE 2 PSI INTERNAL PRESSURE WHICH IS A WORST CASE FOR OPENING OF THE GAP BETWEEN THE FLANGES.

NONLINEAR GAP ELEMENTS ARE USED AT THE FOUNDATION OF THE MODEL TO SIMULATE THE OPENING OF THE GAP.

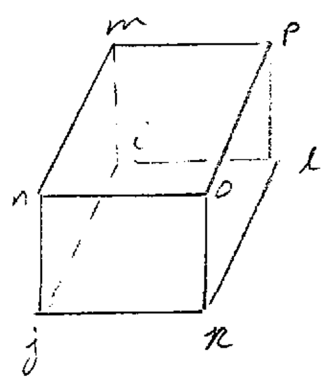
BOTH THE INNER AND OUTER O-RINGS ARE INCLUDED. THESE ARE MODELED AS LINEAR SPRINGS USING THE MAX STIFFNESS POSSIBLE FOR THE 1/4 IN VITON MATERIAL.

* THE MAXIMUM PRELOAD WILL BE 10 000 LB (REF. CALC V049-1-042, P. 26). THIS WILL NOT AFFECT THE RESULTS OF THIS CALC SIGNIFICANTLY.

22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD



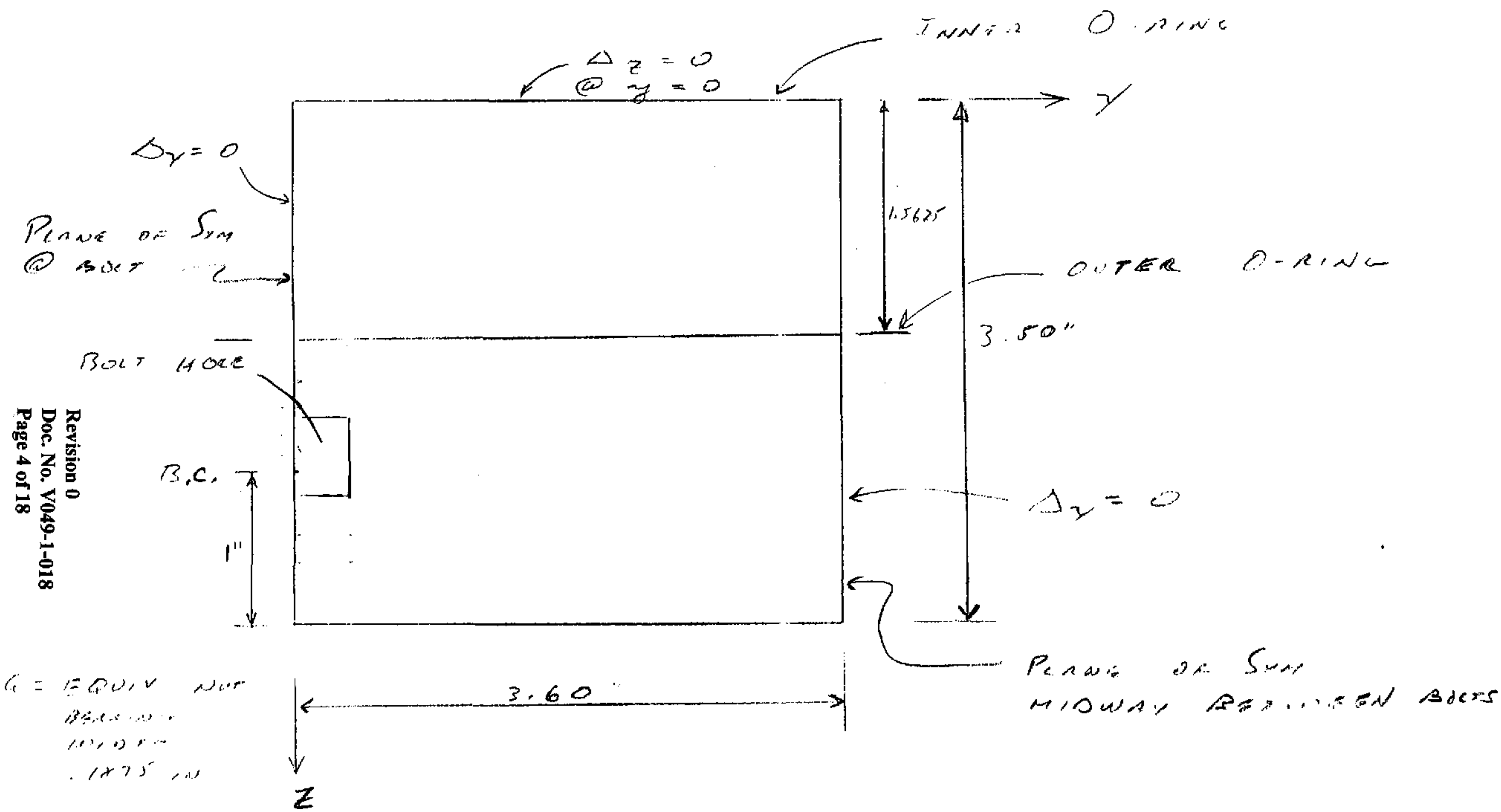
TYPICAL ELEMENT NODE NUMBERING



1ST EL i j k l m n o p

226, 227, 242, 241, 451, 452, 467, 466

Boundary Conditions
@ $y = 0$, $\Delta y = 0$



SPRING RATE FOR SPRINGS REPRESENTING
VITON O RING

DUROMETER = 90

OBTAIN MAX SPRING RATE

MAX COMP IS 25% (P A5-13 OF
PARKER HANDBOOK). FROM P A4-11
THE MAX LINEAR FORCE IS

160 LB/IN

MIN DEFLECTION (SQUEEZE) IS .058
(P. A5-13)

$$R = \frac{160}{.058} = 2759 \text{ LB/IN}$$
PER IN OF CIRCUMFERENCE

FOR DISCRETE SPRINGS AT .2571*
IN SPACING

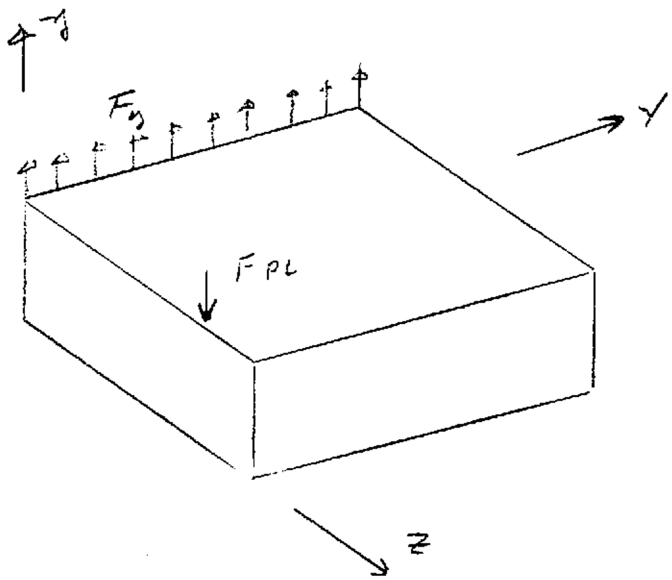
$$R_s = 2759 \ (.2571)$$
$$= 709 \text{ LB/IN}$$

* REF CALC V049-1-016, P. 18 FOR
NODE SPACING

FLANGES - 3D FE ANALYSIS FOR BOLT PRELOAD PLUS POS PRESSURE

ANALYSIS FOR POSITIVE PRESSURE USING 3D FE MODEL - 104 ID FLANGE USED

PRESSURE LOAD FROM ATTACHED CYLINDER



$$F_y = \frac{p\,r}{2}$$

$$p = 2 \text{ psi}$$

$$r = \frac{104.25}{2}$$

$$F_y = \frac{2\left(\frac{104.25}{2}\right)}{2}$$

$$= 52.1 \text{ lb/in}$$

NODE SPACING ALONG x-AXIS IS .2571

$$F_{yi} = 52.1(.2571)$$
$$= 13.4 \text{ lb}$$

$$= \frac{13.4}{2} = 6.7 \quad \text{AT } x = 0,\ 7.6$$

DELETE SPRINGS

151, 152

DELETE SOLIDS

10, 11, 24, 25

206, 207, 220, 221

402, 403, 416, 417

598, 599, 612, 613

794, 795, 808, 809

BOLT LOADS

$F_{PL} = 8500$ LB

FOR 1/2 BOLT

$F_{PL} = -4250$ LB

$F_{PL} = 14 F_{yi} + 4 \frac{F_{yi}}{2}$ ← Y = 0

$-4250 = 16 F_{yi}$

$F_{yi} = -266$ LB INTERIOR NODES

$F_{yi} = \frac{-266}{2} = -133$ LB @ X = 0 AT END NODES

PRESSURE LOAD

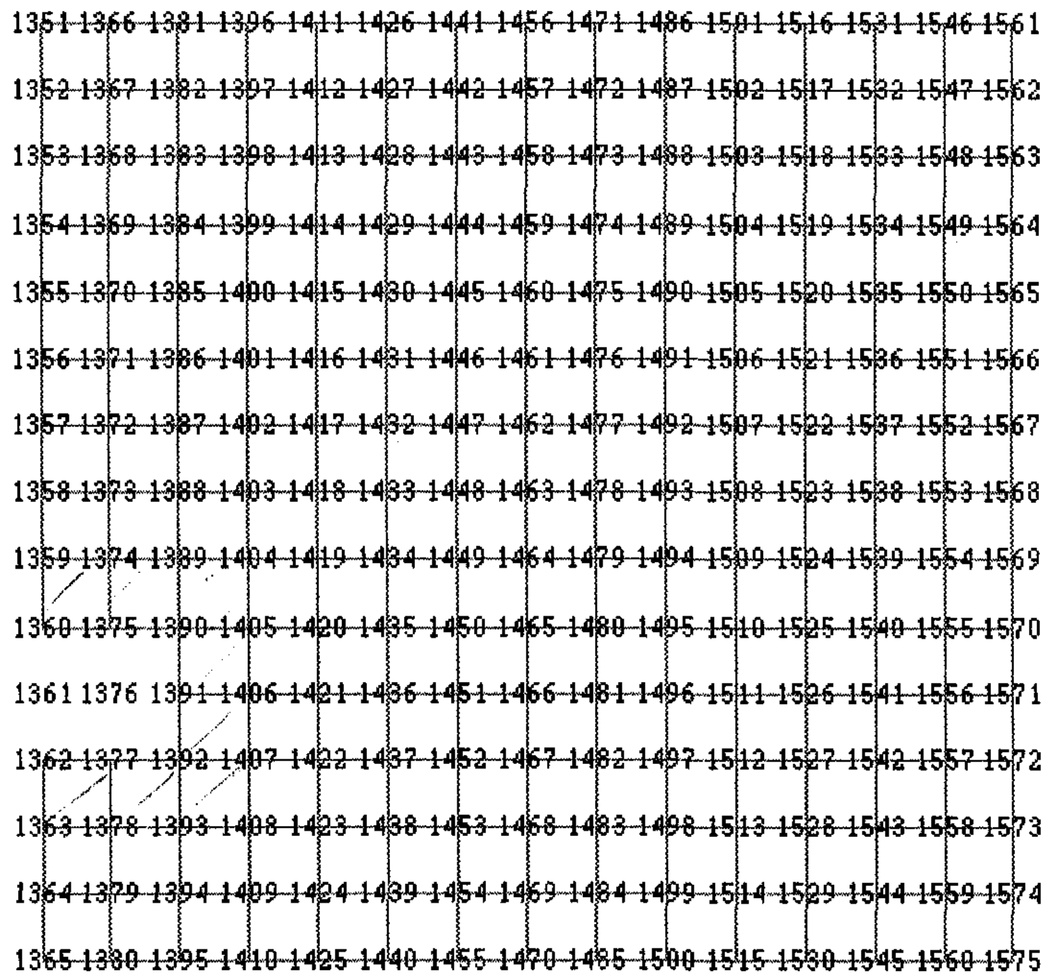
$F_{yi} = +13.4$ LB AT INTERIOR NODES @ Z = 0

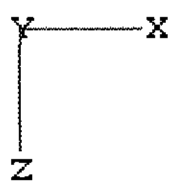
$F_{yi} = +6.7$ LB AT END NODES @ Z = 0

IMAGES-3D
Ver. 3.0
Geometry Plot

Top Layer Nodes

?i
X range
?
X=Default
Y range
?.7,.75
Z range
?
Z=Default
?n
?n
?#
?



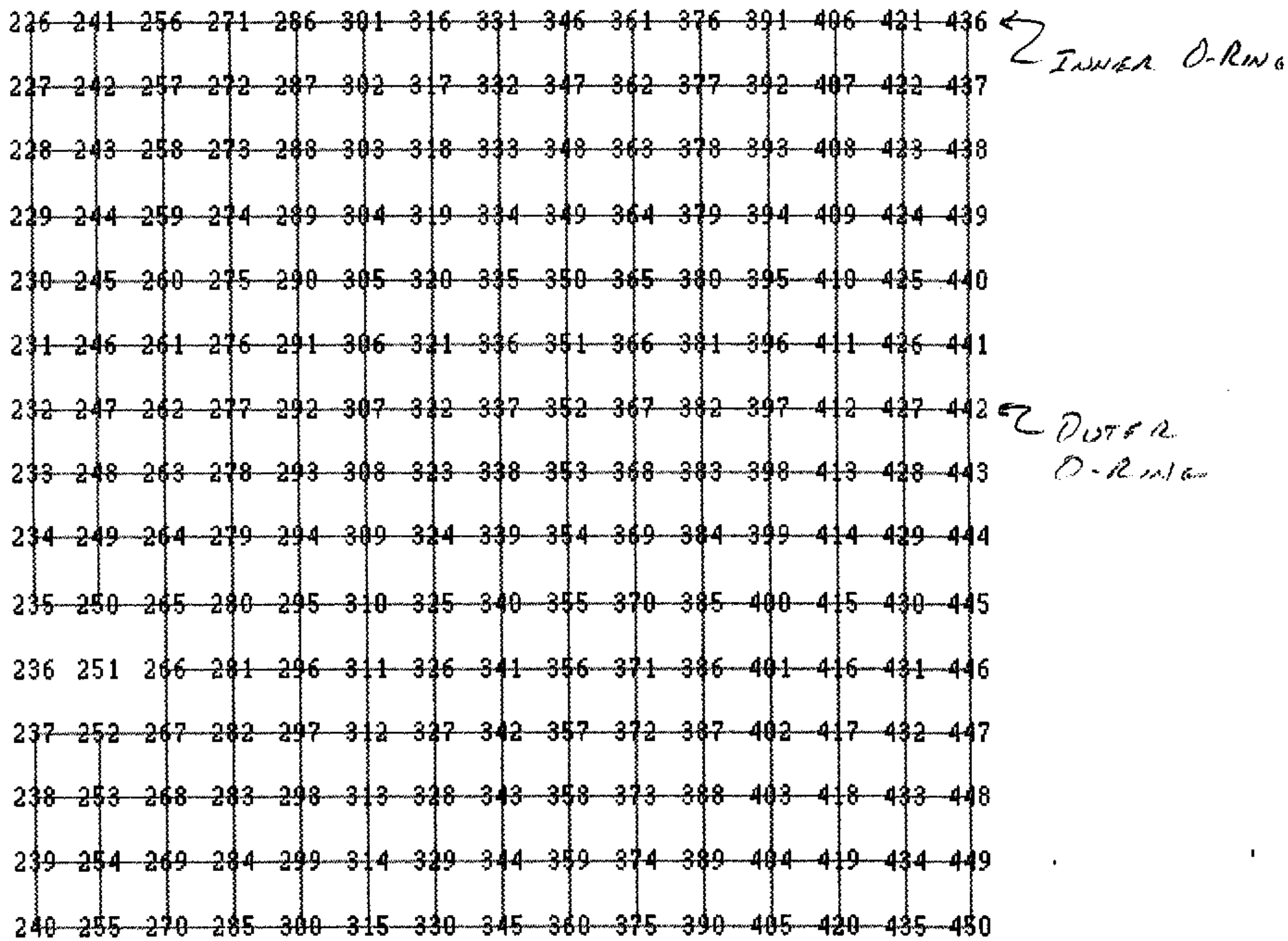


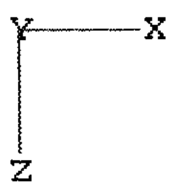
Auto Blowup Elem# Exit Full Help Local Move
Node Node# Plot Range Rest Rotate Shrink Slice

IMAGES-3D
Ver. 3.0
Geometry Plot

Nodes at Y = 0 (at interface between flanges)

X range
?
X=Default
Y range
?-.04,.04
Z range
?
Z=Default
?r,90
?n
?n
?#
?



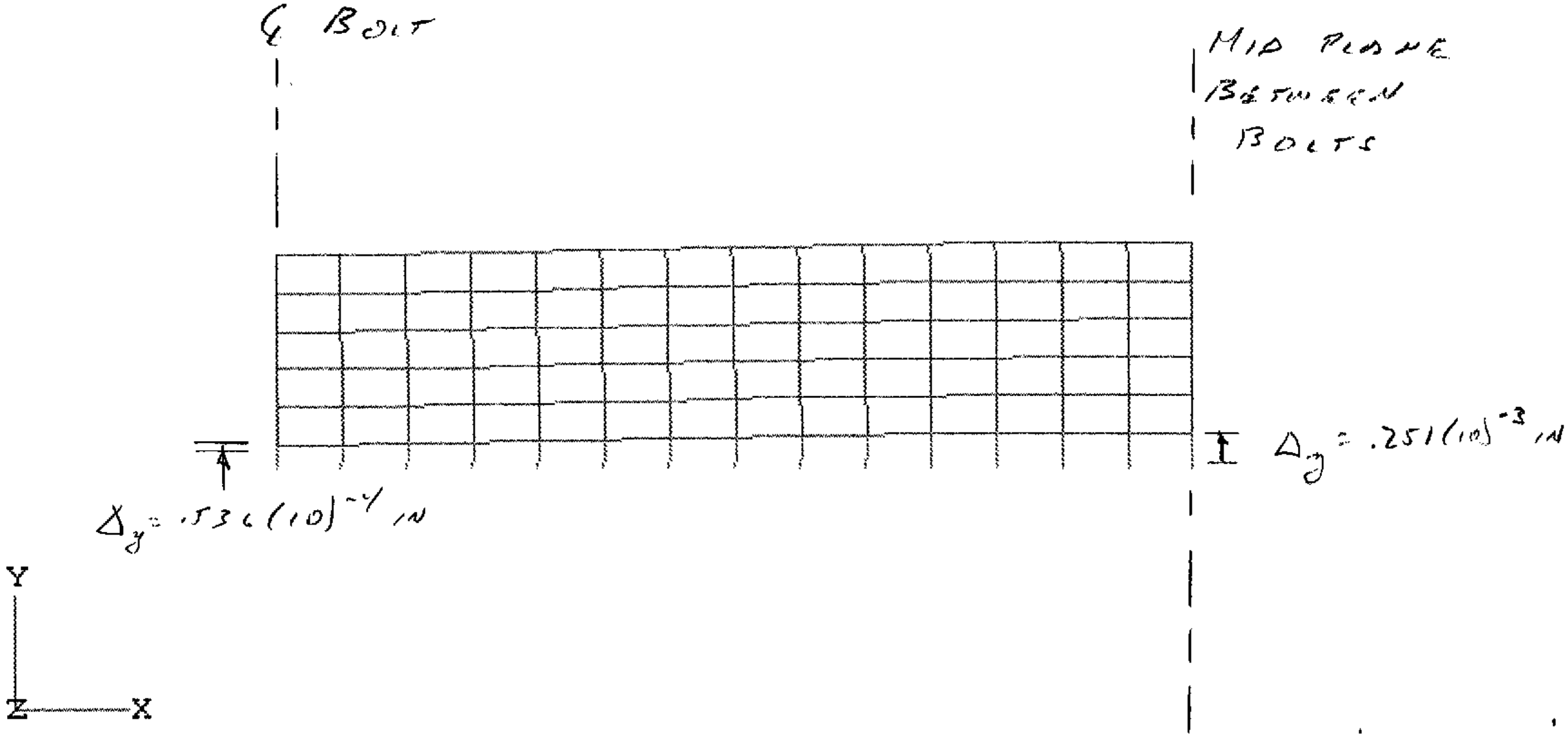


Auto Node | Blowup Node# | Elem# Plot | Exit Range | Full Rest | Help Rotate | Local Shrink | Move Slice

9

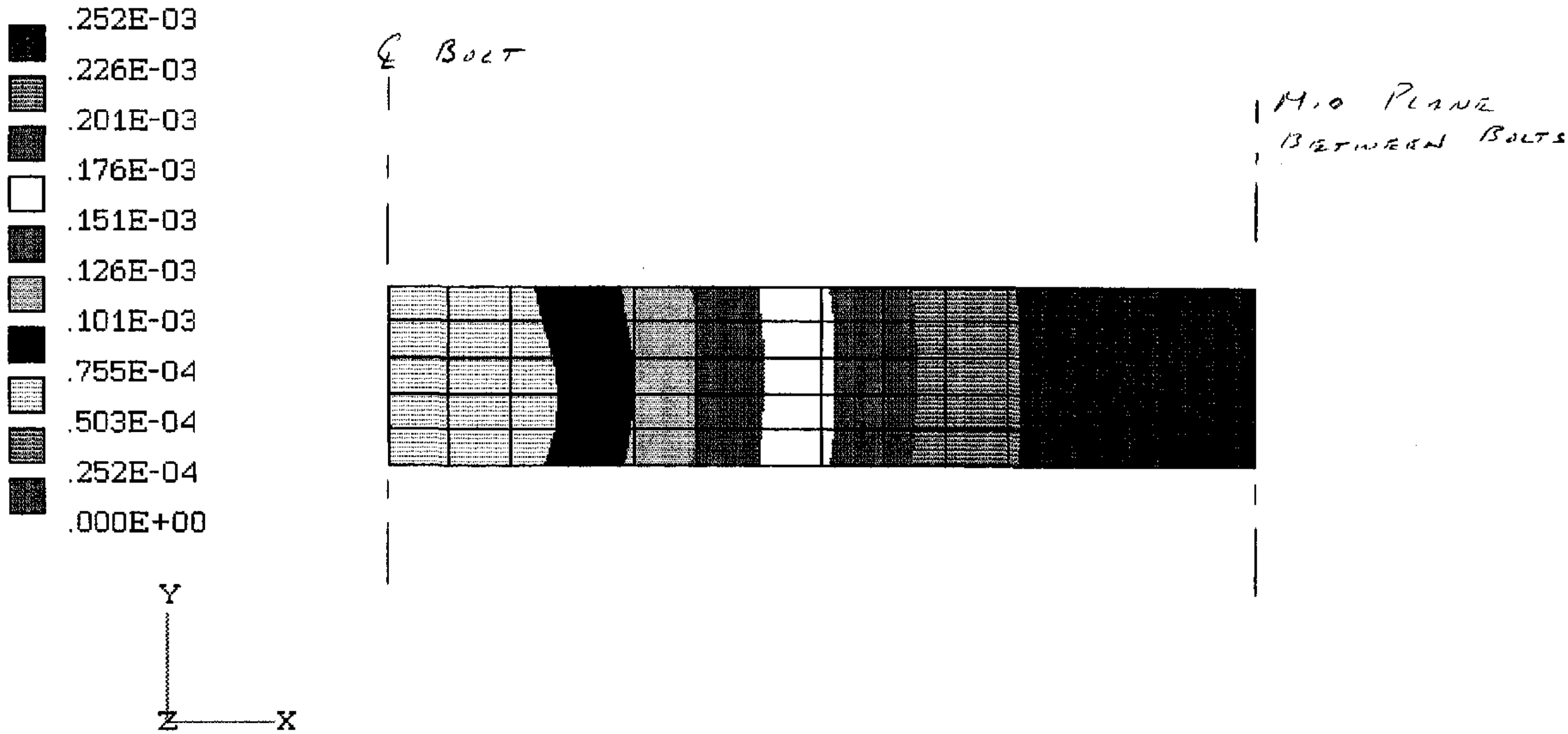
IMAGES-3D
Version 3.0

DEFLECTION ALONG OUTER O-RING



IMAGES-3D
Version 3.0

Y- Displacement along Outer O-Ring



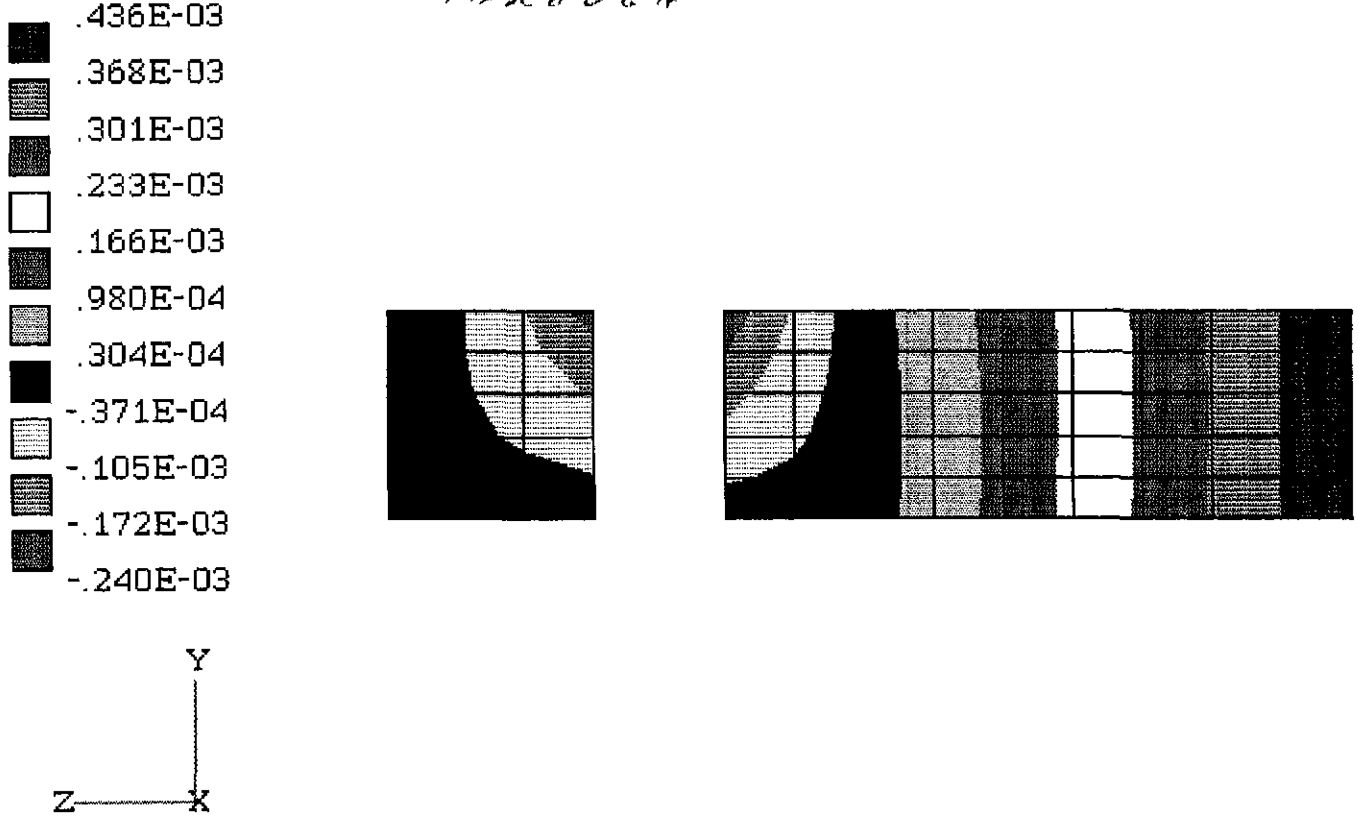
Load Case
1

Displacement Contour Plot
DY

10/20/95
6:53:23

IMAGES-3D
Version 3.0

Y - DISPLACEMENT ALONG RADIAL PLANE PASSING THROUGH BOLT



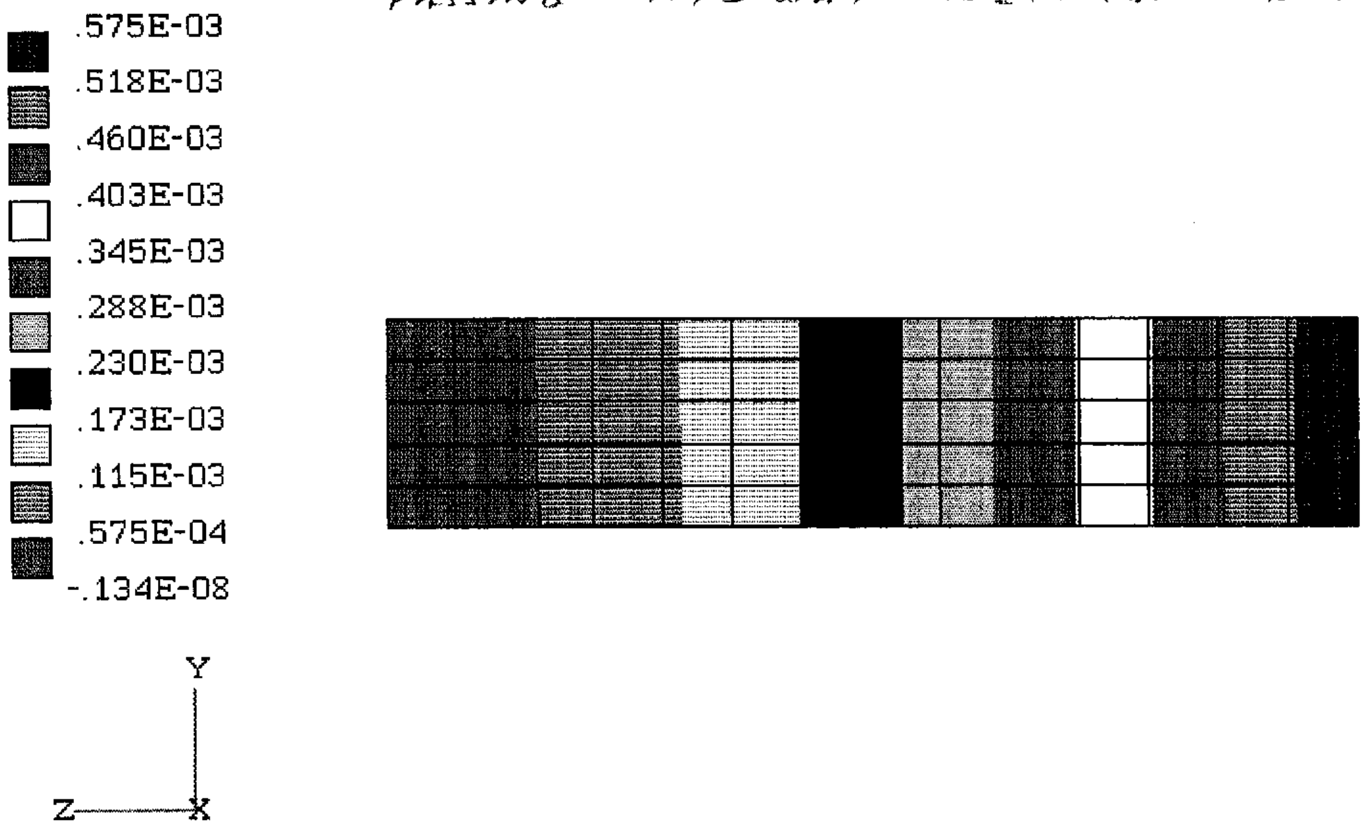
Load Case
1

Displacement Contour Plot
DY

10/20/95
6:57: 6

IMAGES-3D
Version 3.0

Y-DISPLACEMENT ALONG RADIAL PLANE
PASSING MIDWAY BETWEEN BOLTS



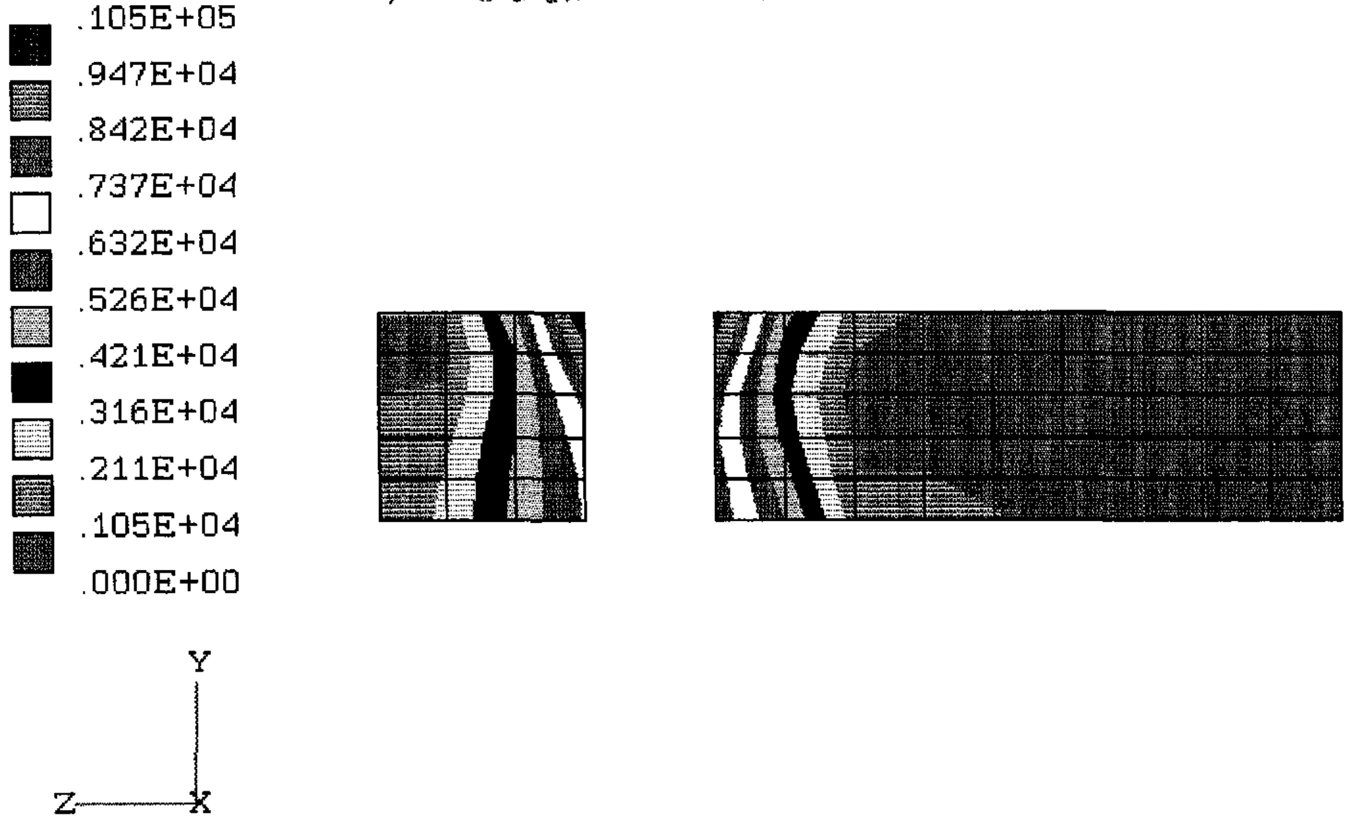
Load Case
1

Displacement Contour Plot
DY

10/20/95
7: 2:23

IMAGES-3D
Version 3.0

Stress Intensity Along Radial Plane Passing Through Bolt



Load Case
1

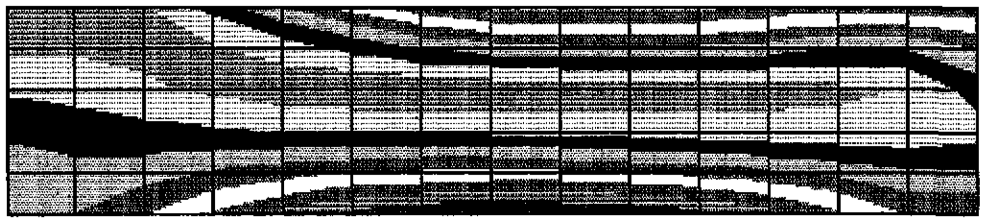
Stress Contour Plot
Stress Intensity

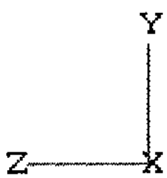
10/20/95
7:17:13

IMAGES-3D
Version 3.0

Stress Intensity Along Radial Plane Passing Midway Between Bolts

.911E+03
.820E+03
.729E+03
.638E+03
.547E+03
.456E+03
.365E+03
.273E+03
.182E+03
.911E+02
.000E+00





Load Case
1

Stress Contour Plot
Stress Intensity

10/20/95
7:22:44

Y - DISPLACEMENTS

	NODE	Δ_y	EXP		
INNER O-RING	226	.427	-3	IN	FROM OUTPUT FILE: FLANGE B
	331	.501	-3		
	436	.568	-3		
OUTER O-RING	232	.536	-4		SEE P. 10
	337	.172	-3		
	442	.251	-3		

RESIDUAL NORM = $.184 (10)^{-10}$

$< 1 (10)^{-5}$

∴ RUN CONVERGED BEFORE 10 ITERATIONS

FOUNDATION SPRING LOADS

	NODE	F_y	
INNER O-RING	226	.303	LB
	331	.355	
	436	.403	
OUTER O-RING	232	$.380 (10)^{-1}$	ok
	337	.122	
	442	.178	

CHECK Y-DIRECTION REACTION.

BOLT + PRESS

$-4250 + 52.1 \text{ lb/in} \times 3.6 \text{ in}$

$= -4062 \#$

REACTION FROM OUTPUT =

ΣF_y 4068 OK

MAXIMUM SEPARATION OF FLANGES

MAX DEFLECTION IS AT NODE 436 (AT INNER O-RING MIDWAY BETWEEN BOLTS)

$$\Delta_y = .568(10)^{-3}$$

FOR BOTH BOLT PRELOAD AND INTERNAL PRESSURE, DOUBLE THIS FOR 2 FLANGES

$$\Delta_y = 2(.568)(10)^{-3}$$
$$= 1.137(10)^{-3} \text{ IN}$$

COMPARE TO MINIMUM COMPRESSION OF .058 IN FROM PARKER HANDBOOK (P A5-13)

$$\frac{58(10)^{-3}}{1.137} = 51$$

∴ OPENING OF GAP << INITIAL COMPRESSION OF O-RING AND O-RING REMAINS COMPRESSED DURING INTERNAL PRESS = 2 PSIG.

MAXIMUM TENSILE LOAD AT VITON O-RING

T = .403 LB AT NODE 436

ALONG THE O-RING, THE LINEAR FORCE IS

$T = \frac{.403}{.257}$ ← NODE SPACING

= 1.57 LB/IN

THIS COMPARES TO MINIMUM SEALING LOAD OF 70 LB/IN (PARKER HANDBOOK, P. A4-11)

C - T = 70 - 1.57 = 68.4 LB/IN

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-019 PAGE 1 OF 24
REV.	DEO #	DATE	BY:	CHECK	Flange Design for Internal Pressure	
0	0024	12/6/95	RDC	P.W.Y.		
					BY: R.D.Cinatto	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE: To evaluate bolted flanges ranging in diameter from 30 in. to 104 in. for positive internal pressure in accordance with the ASME Code.

METHOD: The calculation method provided in Appendix Y of the ASME Code, Section VIII, Division 1, is used. This method applies to flat face flanges with metal-to-metal contact. It has been programmed on the Quatro-Pro spread sheet program.

ASSUMPTIONS:

INPUTS: LIGO project sketches and drawings.

REFERENCES: 1. ASME Boiler & Pressure Vessel Code, Section VIII, Pressure Vessels, Div. 1, Appendix Y.
2. Quattro-Pro Version 5.00
3. Doc. No. V049-1-066, LIGO Vacuum Equip., Structural Design Criteria

CALCULATIONS: (SEE ATTACHED)

CONCLUSIONS: Using a nominal bolt spacing of 7.2 in. for 7/8 in. diameter bolts, the flanges meet App. Y of Section Viii, Div. 1 for 2 psi internal pressure. Computer file names are shown in the lower left corner of the output in this calculation.

NOTES: In this calculation, flange thickness varies from 3/4 in. for small diameter to 1 in. for large diameter flanges.

FILED:\F:\TEMPLATE\ENGFORMS\ENGCALC

FLANGE DESIGN FOR INTERNAL PRESSURE

THE FLANGES ARE EVALUATED IN ACCORDANCE WITH SECT VIII, DIV 1 APP Y FOR FLAT FACE FLANGES. THE DESIGN INTERNAL PRESSURE IS 2.00 PSIG

THE NOMINAL FLANGE THICKNESS IS 1.00 IN

QUATTRO PRO IS USED FOR THE FLANGE EVALUATION. IT SHOWS THAT 7/8" BOLTS SPACED AT APPROX 7.2 IN ARE ADEQUATE.

FOR THE MAXIMUM DIAMETER OF 9D, THE MAXIMUM O-RING COMPRESSION LOAD IS 160 LB/IN. SINCE THERE IS A DOUBLE O-RING, THE LOAD IS MULTIPLIED BY 2 FOR THE ANALYSIS.

$$H_g = 2(160) = 320 \text{ LB/IN}$$

THE ROOT DIAMETER OF THE 7/8" BOLT IS

$$d_b = .755 \text{ IN} \quad (\text{AISC p 4-147})$$

WHICH IS USED TO COMPUTE THE TENSILE AREA

THIS PROGRAM SOLVES APPENDIX Y (Y-6.1) FLAT FACE FLANGES FOR CLASS 1, CAT 3 (LOOSE TYPE) ASSEMBLIES WITH OR WITHOUT SELF ENERGIZING GASKETS FOR POSITIVE PRESSURE OR VACUUM PRESSURE.

ANALYZE CLASS 1 CATEGORY 3 LOOSE TYPE FLANGES ONLY.

ASSUMPTIONS:

1. g1=g0=thickness of vessel wall
2. B substituted for B1
3. conditions at bolt up and operating evaluated separately
4. F'=0 for loose type flanges implies Ms is zero

REQUIRED INPUT:

C = BOLT CIRCLE DIAM in.
PRES = Internal Pressure for this case
B=ID OF FLANGE
A=OD OF FLANGE
g1=THICKNESS OF HUB AT LARGE END
g0=THICK @ SMALL END
rb=SEE FIG Y3.1
V=SEE FIG 2-7.3
F=SEE FIG 2-7.2
t=FLANGE THICKNESS ASSUMED VALUE
G=DIAM OF GASKET
n=NUMBER OF BOLTS
db=DIAM OF BOLTS
re=RATIO MOD E (flange)/ MOD E (bolts)
Z=SEE FIG 2-7.1

INPUT VALUES............ 104 1/4 in Flange

C	PRES	B	A	g1	go	V
110.00	2.00	105.00	112.00	0.000	0.000	0.11
				g1 and g2=0 for loose type flanges		fig 2-7.3 required for integral only

F	t	G	n	db	rE	fhub
0.75	1.00	108.25	48	0.755	0.96	1.00
fig 2-7.2 required for integral only						fig 2-7.6 fhub= 1 for loose type flanges

Sbb	Sfb	Snb	Sbo	Sfo	Sno	Hg(unit)
25000	14700	14700	25000	14700	14700	320
						unit gasket seating and sealing forces

RESULTS/OUTPUT:

Si = DESIGN PRESTRESS IN BOLTS
Sr@BC = RAD. FLANGE STRESS @ BOLT CIRCLE
Sr@ID = RAD FLG STRESS @ ID
St = TANGENTIAL FLANGE STRESS @ ID
Sh = LONGITUDINAL HUB STRESS (= 0 FOR CL1, CAT3 LOOSE TYPE FLANGES)

CALCULATED VALUES.............

R	hd	Js	Jp	B1	BETA	a
2.50	2.50	0.0525	0.0292	105.00	1.02	1.0571
	for loose type flgs					

hc	ho	F'	Hd	H	Ht	Hg
0.50	0.00	0.00	17318.0	18406.7	1088.66	108824.8

hg	ht	l	Mpo	Mpb	Ms	E0bo
0.88	1.688	2.755	140353.861	95221.7	0.00	7129.60
			operating moment	bolt up moment		operating slope

E0bb	Hco	Hcb	Wm1o	Wm1b	Ab	D
4837.01	2.807E+05	190443.3	407939.2	299268.1	21.49	0.818
bolt up slope	operating contact	bolt up contact	operating bolt load	bolt up bolt load		

AR	rB	nrB	K	Z
0.113550	0.000467	0.022424	1.066667	15.52

SIGMAbo	SIGMAbb	Sio	Sib	Sr@BCo	Sr@BCb
18983.24	13926.29	18844.7	13832.3	2749.0	1865.0
25000.00	25000.00	25000.00	25000.00	14700.00	14700.00
ok	ok	ok	ok	ok	ok

St@IDo	St@IDb
67.9	46.1
14700.00	14700.00
ok	ok

THIS PROGRAM SOLVES APPENDIX Y (Y-6.1) FLAT FACE FLANGES FOR CLASS 1, CAT 3 (LOOSE TYPE) ASSEMBLIES WITH OR WITHOUT SELF ENERGIZING GASKETS FOR POSITIVE PRESSURE OR VACUUM PRESSURE.

ANALYZE CLASS 1 CATEGORY 3 LOOSE TYPE FLANGES ONLY.

ASSUMPTIONS:

1. g1=g0=thickness of vessel wall
2. B substituted for B1
3. conditions at bolt up and operating evaluated separately
4. F'=0 for loose type flanges implies Ms is zero

REQUIRED INPUT:

C = BOLT CIRCLE DIAM in.
PRES = Internal Pressure for this case
B=ID OF FLANGE
A=OD OF FLANGE
g1=THICKNESS OF HUB AT LARGE END
g0=THICK @ SMALL END
rb=SEE FIG Y3.1
V=SEE FIG 2-7.3
F=SEE FIG 2-7.2
t=FLANGE THICKNESS ASSUMED VALUE
G=DIAM OF GASKET
n=NUMBER OF BOLTS
db=DIAM OF BOLTS
re=RATIO MOD E (flange)/ MOD E (bolts)
Z=SEE FIG 2-7.1

INPUT VALUES............84 1/4 in Flange

C	PRES	B	A	g1	go	V
89.75	2.00	84.75	91.75	0.000	0.000	0.11
				g1 and g2=0 for loose type flanges		fig 2-7.3 required for integral only

F	t	G	n	db	rE	fhub
	1.00	88.00	40	0.755	0.96	1.00
fig 2-7.2 required for integral only						fig 2-7.6 fhub= 1 for loose type flanges

Sbb	Sfb	Snb	Sbo	Sfo	Sno	Hg(unit)
25000	14700	14700	25000	14700	14700	320
						unit gasket seating and sealing forces

RESULTS/OUTPUT:

Si = DESIGN PRESTRESS IN BOLTS
Sr@BC = RAD. FLANGE STRESS @ BOLT CIRCLE
Sr@ID = RAD FLG STRESS @ ID
St = TANGENTIAL FLANGE STRESS @ ID
Sh = LONGITUDINAL HUB STRESS (= 0 FOR CL1, CAT3 LOOSE TYPE FLANGES)

CALCULATED VALUES.............

R	hd	Js	Jp	B1	BETA	a
2.50	2.50	0.0650	0.0363	84.75	1.03	1.0708
	for loose type flgs					

hc	ho	F'	Hd	H	Ht	Hg
0.50	0.00	0.00	11282.3	12164.2	881.90	88467.2

hg	ht	I	Mpo	Mpb	Ms	E0bo
0.88	1.688	2.755	107102.913	77408.8	0.00	6759.70
			operating moment	bolt up moment		operating slope

E0bb	Hco	Hcb	Wm1o	Wm1b	Ab	D
4885.59	2.142E+05	154817.7	314837.3	243284.9	17.91	0.880
bolt up slope	operating contact	bolt up contact	operating bolt load	bolt up bolt load		

AR	rB	nrB	K	Z
0.124841	0.000685	0.027398	1.082596	12.63

SIGMAbo	SIGMAbb	Sio	Sib	Sr@BCo	Sr@BCb
17580.95	13585.37	17451.7	13491.9	2604.2	1882.2
25000.00	25000.00	25000.00	25000.00	14700.00	14700.00
ok	ok	ok	ok	ok	ok

St@IDo	St@IDb
79.8	57.6
14700.00	14700.00
ok	ok

THIS PROGRAM SOLVES APPENDIX Y (Y-6.1) FLAT FACE FLANGES FOR CLASS 1, CAT 3 (LOOSE TYPE) ASSEMBLIES WITH OR WITHOUT SELF ENERGIZING GASKETS FOR POSITIVE PRESSURE OR VACUUM PRESSURE.

ANALYZE CLASS 1 CATEGORY 3 LOOSE TYPE FLANGES ONLY.

ASSUMPTIONS:

1. g1=g0=thickness of vessel wall
2. B substituted for B1
3. conditions at bolt up and operating evaluated separately
4. F'=0 for loose type flanges implies Ms is zero

REQUIRED INPUT:

C = BOLT CIRCLE DIAM in.
PRES = Internal Pressure for this case
B=ID OF FLANGE
A=OD OF FLANGE
g1=THICKNESS OF HUB AT LARGE END
g0=THICK @ SMALL END
rb=SEE FIG Y3.1
V=SEE FIG 2-7.3
F=SEE FIG 2-7.2
t=FLANGE THICKNESS ASSUMED VALUE
G=DIAM OF GASKET
n=NUMBER OF BOLTS
db=DIAM OF BOLTS
re=RATIO MOD E (flange)/ MOD E (bolts)
Z=SEE FIG 2-7.1

INPUT VALUES............72 1/4 in Flange

C	PRES	B	A	g1	go	V
77.75	2.00	72.75	79.75	0.000	0.000	0.11
				g1 and g2=0 for loose type flanges		fig 2-7.3 required for integral only

F	t	G	n	db	rE	fhub
	1.00	76.00	36	0.755	0.96	1.00
fig 2-7.2 required for integral only						fig 2-7.6 fhub= 1 for loose type flanges

Sbb	Sfb	Snb	Sbo	Sfo	Sno	Hg(unit)
25000	14700	14700	25000	14700	14700	320
						unit gasket seating and sealing forces

RESULTS/OUTPUT:

Si = DESIGN PRESTRESS IN BOLTS
Sr@BC = RAD. FLANGE STRESS @ BOLT CIRCLE
Sr@ID = RAD FLG STRESS @ ID
St = TANGENTIAL FLANGE STRESS @ ID
Sh = LONGITUDINAL HUB STRESS (= 0 FOR CL1, CAT3 LOOSE TYPE FLANGES)

CALCULATED VALUES.............

R	hd	Js	Jp	B1	BETA	a
2.50	2.50	0.0754	0.0422	72.75	1.03	1.0825
	for loose type flgs					

hc	ho	F'	Hd	H	Ht	Hg
0.50	0.00	0.00	8313.5	9072.9	759.38	76403.5

hg	ht	l	Mpo	Mpb	Ms	E0bo
0.88	1.688	2.755	88918.3929	66853.1	0.00	6516.01
			operating moment	bolt up moment		operating slope

E0bb	Hco	Hcb	Wm1o	Wm1b	Ab	D
4899.04	1.778E+05	133706.2	263313.2	210109.7	16.12	0.880
bolt up slope	operating contact	bolt up contact	operating bolt load	bolt up bolt load		

AR	rB	nrB	K	Z
0.129698	0.000825	0.029710	1.096220	10.92

SIGMAbo	SIGMAbb	Sio	Sib	Sr@BCo	Sr@BCb
16337.53	13036.47	16213.8	12943.5	2509.7	1886.9
25000.00	25000.00	25000.00	25000.00	14700.00	14700.00
ok	ok	ok	ok	ok	ok

St@IDo	St@IDb
89.6	67.3
14700.00	14700.00
ok	ok

THIS PROGRAM SOLVES APPENDIX Y (Y-6.1) FLAT FACE FLANGES FOR CLASS 1, CAT 3 (LOOSE TYPE) ASSEMBLIES WITH OR WITHOUT SELF ENERGIZING GASKETS FOR POSITIVE PRESSURE OR VACUUM PRESSURE.

ANALYZE CLASS 1 CATEGORY 3 LOOSE TYPE FLANGES ONLY.

ASSUMPTIONS:

1. g1=g0=thickness of vessel wall
2. B substituted for B1
3. conditions at bolt up and operating evaluated separately
4. F'=0 for loose type flanges implies Ms is zero

REQUIRED INPUT:

C = BOLT CIRCLE DIAM in.
PRES = Internal Pressure for this case
B=ID OF FLANGE
A=OD OF FLANGE
g1=THICKNESS OF HUB AT LARGE END
g0=THICK @ SMALL END
rb=SEE FIG Y3.1
V=SEE FIG 2-7.3
F=SEE FIG 2-7.2
t=FLANGE THICKNESS ASSUMED VALUE
G=DIAM OF GASKET
n=NUMBER OF BOLTS
db=DIAM OF BOLTS
re=RATIO MOD E (flange)/ MOD E (bolts)
Z=SEE FIG 2-7.1

INPUT VALUES............60 1/4 in Flange Thickness = 7/8 in

C	PRES	B	A	g1	go	V
65.75	2.00	60.75	67.75	0.000	0.000	0.11
				g1 and g2=0 for loose type flanges		fig 2-7.3 required for integral only

F	t	G	n	db	rE	fhub
	0.875	64.00	30	0.755	0.96	1.00
fig 2-7.2 required for integral only						fig 2-7.6 fhub= 1 for loose type flanges

Sbb	Sfb	Snb	Sbo	Sfo	Sno	Hg(unit)
25000	14700	14700	25000	14700	14700	320
						unit gasket seating and sealing forces

RESULTS/OUTPUT:

Si = DESIGN PRESTRESS IN BOLTS
Sr@BC = RAD. FLANGE STRESS @ BOLT CIRCLE
Sr@ID = RAD FLG STRESS @ ID
St = TANGENTIAL FLANGE STRESS @ ID
Sh = LONGITUDINAL HUB STRESS (= 0 FOR CL1, CAT3 LOOSE TYPE FLANGES)

CALCULATED VALUES.............

R	hd	Js	Jp	B1	BETA	a
2.50	2.50	0.0896	0.0500	60.75	1.04	1.0988
	for loose type flgs					

hc	ho	F'	Hd	H	Ht	Hg
0.50	0.00	0.00	5797.1	6434.0	636.86	64339.8

hg	ht	I	Mpo	Mpb	Ms	E0bo
0.88	1.688	2.505	71864.8462 operating moment	56297.3 bolt up moment	0.00	9327.89 operating slope

E0bb	Hco	Hcb	Wm1o	Wm1b	Ab	D
7307.26 bolt up slope	1.437E+05 operating contact	112594.7 bolt up contact	214503.5 operating bolt load	176934.5 bolt up bolt load	13.43	0.880

AR	rB	nrB	K	Z
0.127808	0.000960	0.028798	1.115226	9.21

SIGMAbo	SIGMAbb	Sio	Sib	Sr@BCo	Sr@BCb
15970.90	13173.69	15777.3	13022.0	3126.0	2448.9
25000.00	25000.00	25000.00	25000.00	14700.00	14700.00
ok	ok	ok	ok	ok	ok

St@IDo	St@IDb
134.4	105.2
14700.00	14700.00
ok	ok

THIS PROGRAM SOLVES APPENDIX Y (Y-6.1) FLAT FACE FLANGES FOR CLASS 1, CAT 3 (LOOSE TYPE) ASSEMBLIES WITH OR WITHOUT SELF ENERGIZING GASKETS FOR POSITIVE PRESSURE OR VACUUM PRESSURE.

ANALYZE CLASS 1 CATEGORY 3 LOOSE TYPE FLANGES ONLY.

ASSUMPTIONS:

1. g1=g0=thickness of vessel wall
2. B substituted for B1
3. conditions at bolt up and operating evaluated separately
4. F'=0 for loose type flanges implies Ms is zero

REQUIRED INPUT:

C = BOLT CIRCLE DIAM in.
PRES = Internal Pressure for this case
B=ID OF FLANGE
A=OD OF FLANGE
g1=THICKNESS OF HUB AT LARGE END
g0=THICK @ SMALL END
rb=SEE FIG Y3.1
V=SEE FIG 2-7.3
F=SEE FIG 2-7.2
t=FLANGE THICKNESS ASSUMED VALUE
G=DIAM OF GASKET
n=NUMBER OF BOLTS
db=DIAM OF BOLTS
re=RATIO MOD E (flange)/ MOD E (bolts)
Z=SEE FIG 2-7.1

INPUT VALUES............48 1/4 in Flange   Thickness = 7/8 in

C	PRES	B	A	g1	go	V
53.75	2.00	48.75	55.75	0.000	0.000	0.11
				g1 and g2=0 for loose type flanges		fig 2-7.3 required for integral only

F	t	G	n	db	rE	fhub
	0.875	52.00	24	0.755	0.96	1.00
fig 2-7.2 required for integral only						fig 2-7.6 fhub= 1 for loose type flanges

Sbb	Sfb	Snb	Sbo	Sfo	Sno	Hg(unit)
25000	14700	14700	25000	14700	14700	320
						unit gasket seating and sealing forces

RESULTS/OUTPUT:

Si = DESIGN PRESTRESS IN BOLTS
Sr@BC = RAD. FLANGE STRESS @ BOLT CIRCLE
Sr@ID = RAD FLG STRESS @ ID
St = TANGENTIAL FLANGE STRESS @ ID
Sh = LONGITUDINAL HUB STRESS (= 0 FOR CL1, CAT3 LOOSE TYPE FLANGES)

CALCULATED VALUES.............

R	hd	Js	Jp	B1	BETA	a
2.50	2.50	0.1103	0.0615	48.75	1.05	1.1231
	for loose type flgs					

hc	ho	F'	Hd	H	Ht	Hg
0.50	0.00	0.00	3733.1	4247.4	514.34	52276.1

hg	ht	l	Mpo	Mpb	Ms	E0bo
0.88	1.688	2.505	55942.2729	45741.6	0.00	8927.49
			operating moment	bolt up moment		operating slope

E0bb	Hco	Hcb	Wm1o	Wm1b	Ab	D
7299.62	1.119E+05	91483.2	168408.1	143759.3	10.74	0.880
bolt up slope	operating contact	bolt up contact	operating bolt load	bolt up bolt load		

AR	rB	nrB	K	Z
0.125074	0.001146	0.027507	1.143590	7.50

SIGMAbo	SIGMAbb	Sio	Sib	Sr@BCo	Sr@BCb
15673.57	13379.53	15489.8	13229.3	2967.4	2426.3
25000.00	25000.00	25000.00	25000.00	14700.00	14700.00
ok	ok	ok	ok	ok	ok

St@IDo	St@IDb
160.2	131.0
14700.00	14700.00
ok	ok

THIS PROGRAM SOLVES APPENDIX Y (Y-6.1) FLAT FACE FLANGES FOR CLASS 1, CAT 3 (LOOSE TYPE) ASSEMBLIES WITH OR WITHOUT SELF ENERGIZING GASKETS FOR POSITIVE PRESSURE OR VACUUM PRESSURE.

ANALYZE CLASS 1 CATEGORY 3 LOOSE TYPE FLANGES ONLY.

ASSUMPTIONS:

1. g1=g0=thickness of vessel wall
2. B substituted for B1
3. conditions at bolt up and operating evaluated separately
4. F'=0 for loose type flanges implies Ms is zero

REQUIRED INPUT:

C = BOLT CIRCLE DIAM in.
PRES = Internal Pressure for this case
B=ID OF FLANGE
A=OD OF FLANGE
g1=THICKNESS OF HUB AT LARGE END
g0=THICK @ SMALL END
rb=SEE FIG Y3.1
V=SEE FIG 2-7.3
F=SEE FIG 2-7.2
t=FLANGE THICKNESS ASSUMED VALUE
G=DIAM OF GASKET
n=NUMBER OF BOLTS
db=DIAM OF BOLTS
re=RATIO MOD E (flange)/ MOD E (bolts)
Z=SEE FIG 2-7.1

INPUT VALUES............44 1/4 in Flange Thickness = 3/4 in

C	PRES	B	A	g1	go	V
49.75	2.00	44.75	51.75	0.000	0.000	0.11
				g1 and g2=0 for loose type flanges		fig 2-7.3 required for integral only

F	t	G	n	db	rE	fhub
	0.750	48.00	24	0.755	0.96	1.00
fig 2-7.2 required for integral only						fig 2-7.6 fhub= 1 for loose type flanges

Sbb	Sfb	Snb	Sbo	Sfo	Sno	Hg(unit)
25000	14700	14700	25000	14700	14700	320
						unit gasket seating and sealing forces

RESULTS/OUTPUT:

Si = DESIGN PRESTRESS IN BOLTS
Sr@BC = RAD. FLANGE STRESS @ BOLT CIRCLE
Sr@ID = RAD FLG STRESS @ ID
St = TANGENTIAL FLANGE STRESS @ ID
Sh = LONGITUDINAL HUB STRESS (= 0 FOR CL1, CAT3 LOOSE TYPE FLANGES)

CALCULATED VALUES.............

R	hd	Js	Jp	B1	BETA	a
2.50	2.50	0.1199	0.0670	44.75	1.06	1.1341
	for loose type flgs					

hc	ho	F'	Hd	H	Ht	Hg
0.50	0.00	0.00	3145.6	3619.1	473.50	48254.9

hg	ht	l	Mpo	Mpb	Ms	E0bo
0.88	1.688	2.255	50886.0759 operating moment	42223.0 bolt up moment	0.00	14046.64 operating slope

E0bb	Hco	Hcb	Wm1o	Wm1b	Ab	D
11655.27 bolt up slope	1.018E+05 operating contact	84446.0 bolt up contact	153646.1 operating bolt load	132700.9 bolt up bolt load	10.74	0.880

AR	rB	nrB	K	Z
0.135130	0.001351	0.032420	1.156425	6.93

SIGMAbo	SIGMAbb	Sio	Sib	Sr@BCo	Sr@BCb
14299.69	12350.34	13981.6	12086.4	4015.4	3331.8
25000.00	25000.00	25000.00	25000.00	14700.00	14700.00
ok	ok	ok	ok	ok	ok

St@IDo	St@IDb
235.4	195.3
14700.00	14700.00
ok	ok

THIS PROGRAM SOLVES APPENDIX Y (Y-6.1) FLAT FACE FLANGES FOR CLASS 1, CAT 3 (LOOSE TYPE) ASSEMBLIES WITH OR WITHOUT SELF ENERGIZING GASKETS FOR POSITIVE PRESSURE OR VACUUM PRESSURE.

ANALYZE CLASS 1 CATEGORY 3 LOOSE TYPE FLANGES ONLY.

ASSUMPTIONS:

1. g1=g0=thickness of vessel wall
2. B substituted for B1
3. conditions at bolt up and operating evaluated separately
4. F'=0 for loose type flanges implies Ms is zero

REQUIRED INPUT:

C = BOLT CIRCLE DIAM in.
PRES = Internal Pressure for this case
B=ID OF FLANGE
A=OD OF FLANGE
g1=THICKNESS OF HUB AT LARGE END
g0=THICK @ SMALL END
rb=SEE FIG Y3.1
V=SEE FIG 2-7.3
F=SEE FIG 2-7.2
t=FLANGE THICKNESS ASSUMED VALUE
G=DIAM OF GASKET
n=NUMBER OF BOLTS
db=DIAM OF BOLTS
re=RATIO MOD E (flange)/ MOD E (bolts)
Z=SEE FIG 2-7.1

INPUT VALUES............30 1/4 in Flange Thickness = 3/4 in

C	PRES	B	A	g1	go	V
35.75	2.00	30.75	37.75	0.000	0.000	0.11
				g1 and g2=0 for loose type flanges		fig 2-7.3 required for integral only

F	t	G	n	db	rE	fhub
	0.750	34.00	16	0.755	0.96	1.00
fig 2-7.2 required for integral only						fig 2-7.6 fhub= 1 for loose type flanges

Sbb	Sfb	Snb	Sbo	Sfo	Sno	Hg(unit)
25000	14700	14700	25000	14700	14700	320
						unit gasket seating and sealing forces

RESULTS/OUTPUT:

Si = DESIGN PRESTRESS IN BOLTS
Sr@BC = RAD. FLANGE STRESS @ BOLT CIRCLE
Sr@ID = RAD FLG STRESS @ ID
St = TANGENTIAL FLANGE STRESS @ ID
Sh = LONGITUDINAL HUB STRESS (= 0 FOR CL1, CAT3 LOOSE TYPE FLANGES)

CALCULATED VALUES..............

R	hd	Js	Jp	B1	BETA	a
2.50	2.50	0.1694	0.0942	30.75	1.08	1.1951
	for loose type flgs					

hc	ho	F'	Hd	H	Ht	Hg
0.50	0.00	0.00	1485.3	1815.8	330.55	34180.5

hg	ht	l	Mpo	Mpb	Ms	E0bo
0.88	1.688	2.255	34178.988	29908.0	0.00	13266.80
			operating moment	bolt up moment		operating slope

E0bb	Hco	Hcb	Wm1o	Wm1b	Ab	D
11608.97	6.836E+04	59815.9	104354.3	93996.5	7.16	0.880
bolt up slope	operating contact	bolt up contact	operating bolt load	bolt up bolt load		

AR	rB	nrB	K	Z
0.125365	0.001728	0.027643	1.227642	4.94

SIGMAbo	SIGMAbb	Sio	Sib	Sr@BCo	Sr@BCb
14568.23	13122.23	14273.2	12864.0	3711.4	3247.6
25000.00	25000.00	25000.00	25000.00	14700.00	14700.00
ok	ok	ok	ok	ok	ok

St@IDo	St@IDb
323.6	283.1
14700.00	14700.00
ok	ok

BOLT PRELOAD

PER APP Y, THE DESIGN BOLT PRELOAD IS WM10 (CONSIDERS INTERNAL PRESSURE AT OPERATING CONDITION)

FLANGE SIZE	N	WM10 LB	FPL = WM10/N	USE
104	48	407939	8498	8500 LB
84	40	314837	7870	↓
72	36	263313	7314	↓
60	30	214504	7150	↓
48	24	168404	7017	↓
44	24	153646	6402	↓
30	16*	104354	6522	↓

NOTE: BOLT PRELOAD IS INCREASED TO 10K. BOLT STRESS IS STILL WITHIN LIMIT REF CALC 042, P. 20.

* NO. OF BOLTS WAS INCREASED TO 24 FOR THE 30 IN FLANGE. REF CALC 042, P. 26

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Doc. No V049-1-019
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PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-020 PAGE 1 OF 13
REV.	DEO #	DATE	BY:	CHECK	Beam Splitter Chamber - Design of Removable Work Floor	
0	0024	4/12/96	RDC	AGR		
					BY: R.D. Ciatto	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE: To design removable floor and support members to be installed in lower section of BSC.

METHOD: Hand calculations using standard beam design method for support beams and removable floor sections.

ASSUMPTIONS:

INPUTS: See Doc. No. LIGO-C951078-00-4 which provides response to V049-NL-PL-9 (requesting clarifications of BSC internal floor requirements). Maximum load is 500 Kg during maintenance. This occurs at room temperature.

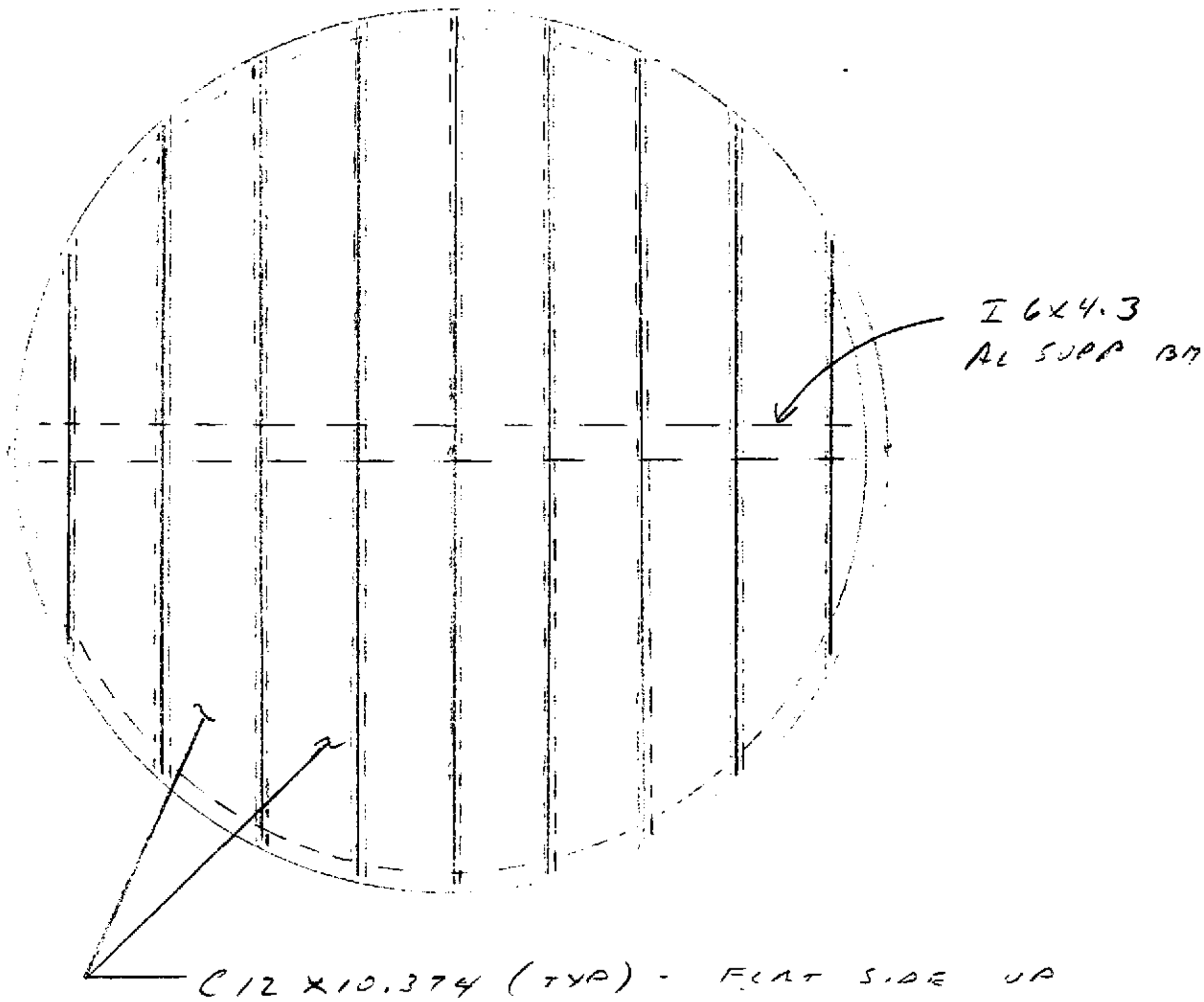
REFERENCES:
1. ASME Boiler & Pressure Vessel Code, Section VIII, Pressure Vessels, Division 1.
2. Aluminum Association Hankbook
3. Doc. No. V049-1-066, LIGO Vac. Equip., Stress Design Crit.

CALCULATIONS: (SEE ATTACHED)

CONCLUSIONS: The design of the aluminum floor meets LIGO requirements and the requirements of the applicable design codes.

NOTES:

BSC FLOOR 2-23-96



MATERIAL

(B 221) - 6061 - T6

REV 0
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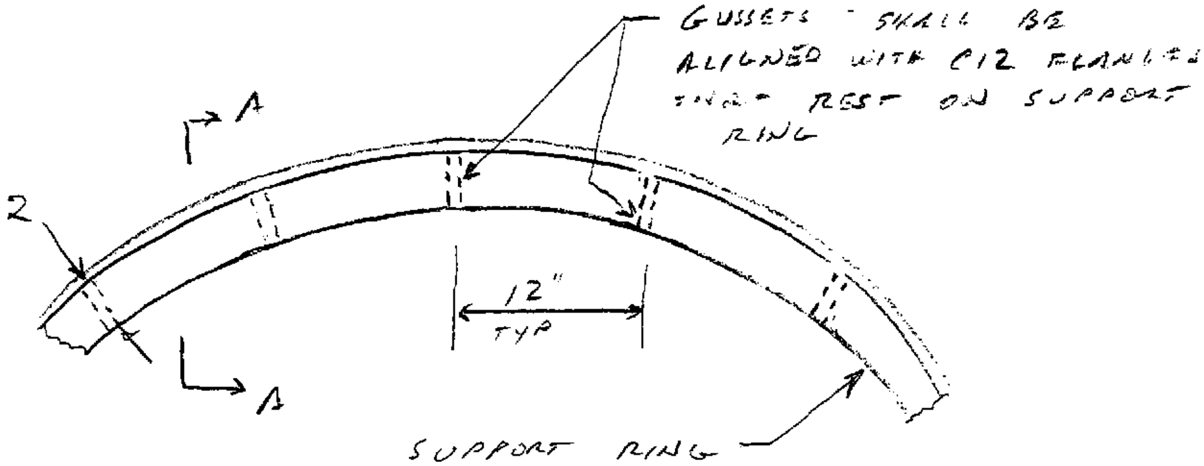
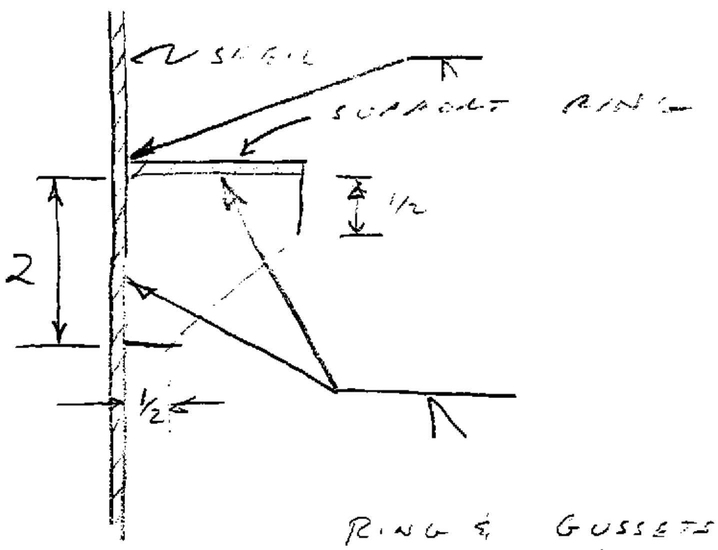


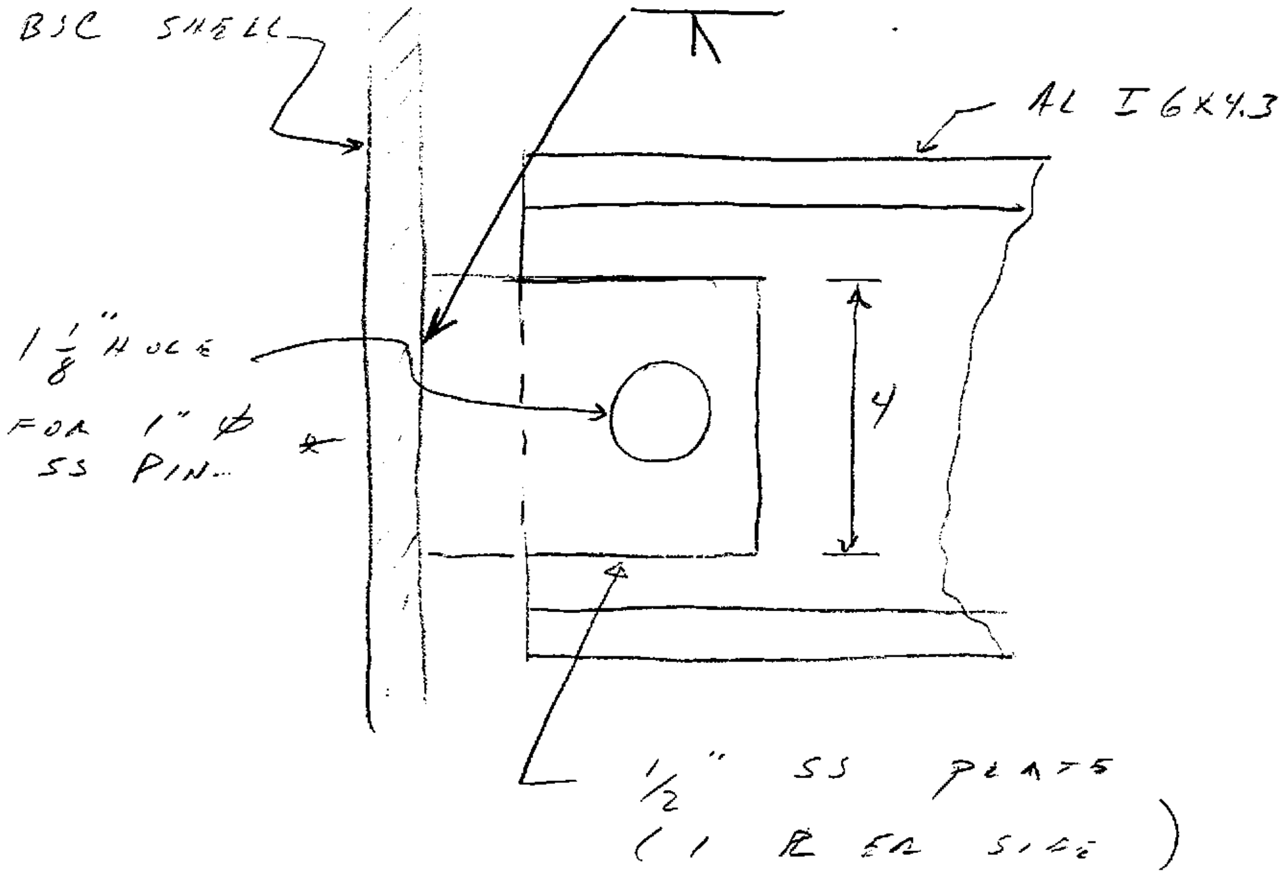


A - A

RING THICKNESS = 1/2"
GUSSET THICKNESS = 3/8"

2-23-96

FLOOR SUPPORT TO SHELL CONNECTION



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Doc. No V0491-020
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22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD

FLOOR MATERIAL

SB 221 ALLOY 6061
TEMPER T6

YIELD F_y = 35 KSI @ ROOM TEMP

ULTIMATE F_u = 38 KSI @ ROOM TEMP
REF ASME B&PV CODE SECT II, TABLE 1B

ALLOWABLE BENDING STRESS

FOR WEAK AXIS BENDING USE AISC RULE

$$F_b = .75 F_y$$

$$= .75(35) = 26.25 \text{ KSI}$$

NOTE: THIS IS LESS THAN ALUMINUM ASSOC. ALLOWABLE USING FS = 1.5

FOR STRONG AXIS BENDING STRESS WILL BE KEPT REASONABLY LOW TO AVOID LOCAL & LATERAL TORSIONAL BUCKLING

REV 0
DOC. NO V0490-1-070
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STIFFNESS OF ALUM C12 × 7.41 FOR LOAD AT CENTER BENT ABOUT WEAK AXIS

$$\Delta_y = \frac{PL^3}{48EI_y}$$

$L = 104$

$E = 10(10)^6$ psi

$I_y = 3.59$ IN^4

$$\Delta_y = \frac{P(104)^3}{48(10)^7(3.59)}$$

$$= \frac{P}{1702.6}$$

$$R_1 = 1702.6 = \frac{P}{\Delta_y} \text{ LB/IN}$$

STIFFNESS OF ALUM I6 × 4.3 FOR LOAD AT CENTER BENT ABOUT STRONG AXIS

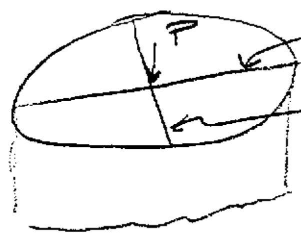
$I_x = 22.08$ IN^4

$$R_2 = \frac{P}{\Delta_y} = \frac{48(10)^7(22.08)}{(104)^3}$$

$$= 9422 \text{ LB/IN}$$

REV 0
DOC. NO. V049-1-020
P. 6 OF 13

THE SUPPORT BEAM AND THE LONGEST CHANNEL ACT AS SPRINGS IN PARALLEL FOR A LOAD AT THE CENTER OF THE VESSEL



$$K = k_1 + k_2$$

$$= 1703 + 5422 = 11125 \text{ LB/IN}$$

MAX DEFL AT CTR (LIVE LOAD)

$$K = \frac{P}{\Delta}$$

$$\Delta = \frac{P}{k} = \frac{1100}{11125} = .10 \text{ IN} \quad \text{ok}$$

STRESS IN INVERTED CHANNEL DUE TO LOAD AT CTR

$$\Delta = \frac{P_1}{k_1}$$

$$P_1 = \Delta k_1 = .10(1703) = 170 \text{ LB}$$

$$M = \frac{PL}{4} + \frac{w}{8}L^2$$

$$= \frac{170(104)}{4} + \frac{\frac{7.41}{12}(104)^2}{8}$$

$$= 4420 + 835$$

$$= 5255 \text{ IN-LB}$$

REV 0
DOC NO V049-1-020
P. 7 OF 13

$S_y = 1.76$ FOR C12 x 7.41 *

* CHANGED TO C10 x 10.374

$f_b = \frac{M}{S} = \frac{5255}{1.76} = 3000$ psi ok

STRESS IN INVERTED CHANNEL W/ LOAD AT 1/4 PT

$M = \frac{1100(52)}{4} + \frac{\frac{7.41}{12}(52)^2}{8}$

$= 14300 + 209$

$= 14500$ IN-LB

$f_b = \frac{M}{S} = \frac{14500}{1.76} = 8250$ psi ok

STRESS IN SUPPORT BEAM DUE TO DW

USE $w = 30 + 4.3 = 34.3$ LB/FT (C12 x)

$\cong 3$ LB/IN

$M = \frac{wL^2}{8} = \frac{3(104)^2}{8}$

$= 4056$

$f_b = \frac{M}{S_y} = \frac{4056}{7.36} = 550$ psi

REV O
DOC. NO. V049-1-020
P. 8 OF 13

STRESS IN SUPPORT BEAM DUE TO LIVE LOAD AT CENTER

$$\Delta = \frac{P_2}{R_2}$$

$$P_2 = R_2 \Delta$$

$$= 9422(.10) = 942 \text{ LB}$$

$$M = \frac{PL}{4} = \frac{942(104)}{4} = 24500$$

$$f_b = \frac{M}{S_x} = \frac{24500}{22.08}$$

$$= 1100 \text{ psi}$$

TOTAL STRESS IN SUPPORT BEAM DUE TO DW & LL

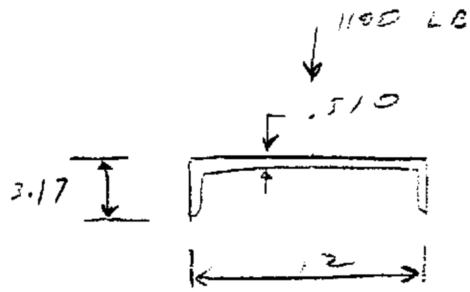
$$f_b = 1100 + 560$$

$$= 1660 \text{ psi} \quad \text{LOW}$$

BENDING STRESS IN WEB OF CHANNEL

TRY C12 X 10.37# AISC STD

WEB THICKNESS $t = .510$



ASSUME THAT LOAD IS 3 IN WIDE

$$M = \frac{PL}{4} = \frac{1100(12)}{4}$$
$$= 3300 \text{ IN-\#}$$

$$f = \frac{6M}{bt^2} = \frac{6(3300)}{3(.510)^2}$$
$$= 25400 \text{ psi}$$
$$< 26250 \text{ psi} \quad \text{OK}$$

FOR BENDING ABOUT WEAK AXIS

$S_y = 2.06$ REF AISC CODE
= SECT. MODULUS

ASSUME SINGLE SUPPORT BEAM AT CENTER OF BSC

$L = 52$

$$M = \frac{1100(52)}{4} = 14300 \text{ in-\#}$$

$$f_b = \frac{14300}{2.06} = 6940 \quad \text{OK}$$

REV 0
DOC. NO V019-1-023
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SUPPORT RING

GUSSETS WILL BE PLACED UNDER SUPPORT RING SO THAT THEY ARE ALIGNED WITH POINT OF CONTACT OF C12 FLANGE

THE MAX REACTION FROM THE C12 IS

$$R = 1100 + 52 \times 10.334$$
$$= 1640 \text{ LB}$$

SINCE THE GUSSET IS 3/8" THICK AND THE WELD BETWEEN THE SHELL & GUSSET IS A FULL PEN., THE WELD AREA IS

$$A_w = 2\left(\tfrac{3}{8}\right) = .75 \text{ IN}^2$$

WELD SHEAR STRESS

$$f_v = \frac{R}{A_w}$$

$$= \frac{1640}{.75} =$$

$$= 2200 \text{ psi} \quad \text{OK}$$

SUPPORT RING & GUSSETS ARE 304/304L

SUPPORT BEAM CONNECTION TO SHELL

THE WEB THICKNESS FOR THE I 6 x 4.3 IS

$$t_w = .23 \text{ IN}$$

FOR A SINGLE 3/4 IN PIN THE BEARING AREA IS

$$A_p = .75(.23)$$
$$= .1725$$

THE MAX REACTION IS

$$R = \frac{104}{2(12)} (34.3 \text{ LB/FT}) + 1100$$

$$= 150 + 1100 = 1250 \text{ LB}$$

BEARING STRESS

$$f_p = \frac{R}{A_p}$$

$$= \frac{1250}{.1725} = 7250 \text{ psi} \quad \text{HR}$$

USE A 1 IN PIN TO REDUCE f_p

$$f_p = 7250 \times .75 = 5400 \text{ psi}$$

IN DOUBLE SHEAR

$$A_s = 2\pi(.5)^2$$

$$= 1.57 \text{ IN}^2$$

$$f_v = \frac{1250}{1.57} = 800 \text{ psi}$$ OK FOR SS PIN

DIFFERENTIAL EXPANSION BETWEEN STEEL AND ALUM SUPPORT BEAM

$$\alpha_{AL} = 13.52\ (10)^{-6}\ \frac{in}{in\ ^\circ F}$$

FOR 6061

$$\alpha_{SS} = 9.19\ (10)^{-6}$$

FOR TEMP RISE TO 400°F

$$\Delta T = 400 - 70 = 330\ ^\circ F$$

$$\delta = (\alpha_{AL} - \alpha_{SS})\ L\ \Delta T$$

$$L = 104\ in$$

$$\delta = (13.52 - 9.19)(10)^{-6}(104)(330)$$

$$= .15\ in$$

USE AN OVERSIZE BOLT HOLE TO TAKE δ. FROM BLC ℄

$$\delta = \frac{.15}{2} = .075$$

HOLE SIZE = 1 1/4 GIVES CLEARANCE

$$.125 > \delta = .075$$ OK

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-021 PAGE 1 OF 7
REV.	DEO #	DATE	BY:	CHECK	TITLE: **Design of Flange Welds**	
0	0136	12/6/95	RDC	WDB		
1	0253	8/22/96	RDC	WDB		
					BY: R. D. Ciatto	DEPT.: 744
PROJECT: LIGO Vacuum Equipment					PROJECT NO: V59049	

PURPOSE:

Design partial penetration seal weld and external stitch weld between flanges and shells

METHOD:

Hand calculations are used to determine the pressure discontinuity moments and forces and resulting weld stresses.

ASSUMPTIONS:

To determine the discontinuity moment and shear it is conservatively assumed that the cylindrical shell is built into a rigid wall.

INPUTS:

1. LIGO project drawings of bolted flanges
2. Calc. No. V049-1-066, LIGO Vacuum Equipment Structural Design Criteria

REFERENCES:

1. ASME Boiler & Pressure Vessel Code, Section VIII, Pressure Vessels, Div. 2.
2. Doc. No. V049-1-042, Bolted Flange Analysis for Tensile Forces.
3. Doc. No. V049-1-107, Welds for 60.5 in Nozzle to Shell Joints.

CALCULATIONS:

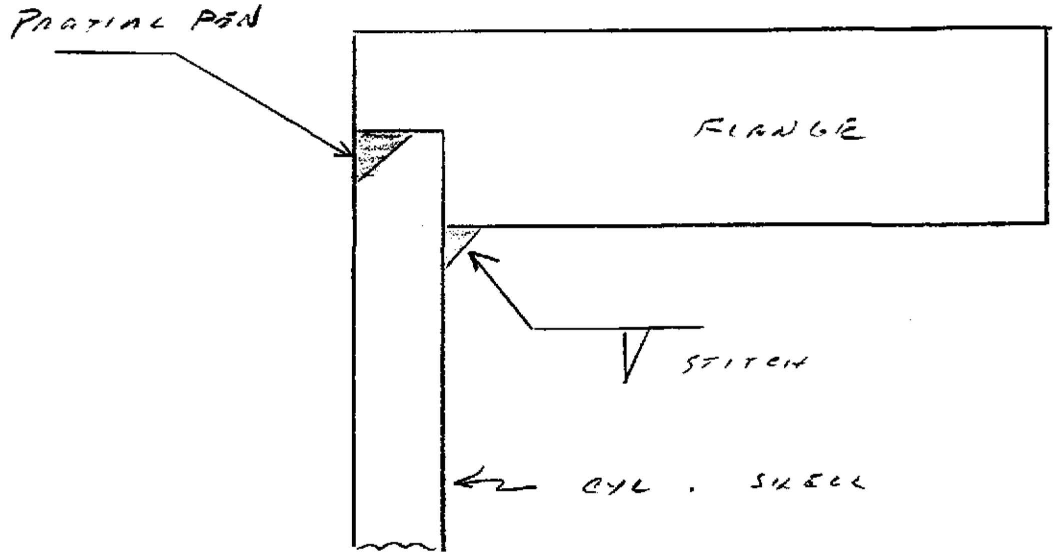
(SEE ATTACHED)

CONCLUSIONS:

For the bounding case flange (104.5 in) the required partial penetration weld is 1/8 in and the required fillet weld is 3/16 (2 in every 6 in).

NOTES:

EACH FLANGE IS CONNECTED TO THE CYLINDRICAL SHELL WITH 2 WELDS AS SHOWN BELOW

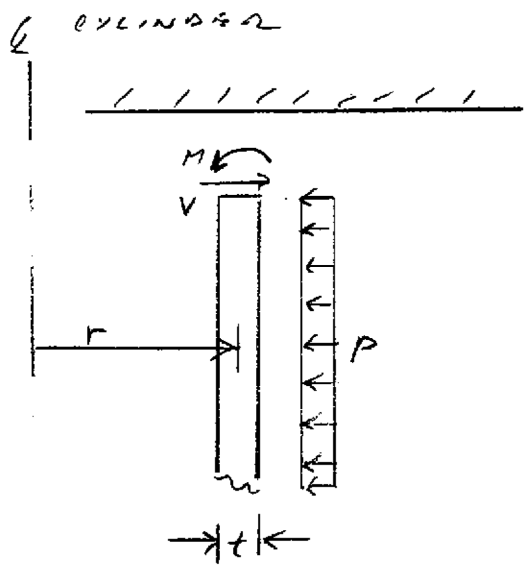


THESE WELDS RESIST DISCONTINUITY MOMENTS AND FORCES BETWEEN THE CYLINDER AND THE FLANGE THAT RESULT FROM POSITIVE OR NEGATIVE PRESSURE. THE FILLET WELD (STITCH) AND THE PARTIAL PEN WELD ARE ASSUMED TO RESIST THE DISCONTINUITY MOMENT WITH A FORCE COUPLE.

ONLY THE PARTIAL PEN WELD IS ASSUMED TO RESIST THE DISCONTINUITY SHEAR FORCE. TO BE CONSERVATIVE, ONLY THE PAR PEN WELD IS ASSUMED TO RESIST THE FLANGE TENSILE FORCE THAT RESULTS FROM POSITIVE PRESSURE AND COMPONENT INTERFACE LOADS (REF CALCS V049-1-032 AND -042).

PRESSURE DISCONTINUITY STRESSES FOR A CYLINDER AT A RIGID WALL.

(FLANGE IS ASSUMED TO ACT AS A RIGID WALL



FOR EXTERNAL PRESSURE, p

SHEAR $V = .660\, p \sqrt{rt}$

MOMENT $M = .257\, p\, r\, t$

STRESSES WILL BE DETERMINED WITHOUT REGARD TO SIGN IN ORDER TO OBTAIN THE MAXIMUM TENSILE STRESS IN THE PP WELD AND THE MAXIMUM SHEAR STRESS IN THE STITCH FILLET WELD.

FROM THE ABOVE EQUATIONS, IT IS APPARENT THAT THE MAX DISCONTINUITY SHEAR FORCE AND MOMENT PER IN OF CIRCUMFERENCE WILL BE IN THE LARGEST DIAMETER FLANGE. THIS IS THE 104 1/2 IN FLANGE

$r = 52.25 + .25 = 52.5$ IN

FOR 104 1/2 IN FLANGE

$t = .5$

$p = 14.7$ psi FOR FULL VACUUM

$V = .660\, p \sqrt{rt}$

$= .660\,(14.7)\,(52.5 \times .5)^{1/2}$

$= 49.71$ LB/IN

$M = .257\, prt =$

$= .257(14.7)(52.5)(.5)$

$= 99.17$ IN-LB/IN

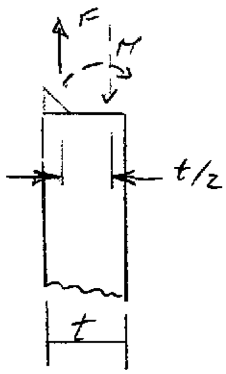
SHEAR STRESS IN PAR. PEN. WELD (TRY 1/8 IN)

$f_v = \dfrac{V}{t_{wp}}$

$t_{wp} = .125$

$f_v = \dfrac{49.71}{.125} = 398$ psi

DIRECT STRESS IN PAR. PEN WELD



$p = 14.7$ psi

$F = \dfrac{pr}{2} + \dfrac{M}{t/2} + 362^{*}$

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Doc. No. V049-1-021
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t/2 IS CONSERVATIVE MOMENT ARM

* REF CALC V049-1-042, P 72 FOR WORST CASE FLANGE TENSILE FORCE. THIS IS FOR 30.25 IN FLANGE.

$$F = \frac{14.7(52.5)}{2} + \frac{99.17}{.25} + 362$$

$$= 386 + 397 + 362 = 1145 \text{ LB/IN}$$

TENSILE STRESS

$$f_t = \frac{1145}{.125} = 9160 \text{ psi}$$

PRINCIPAL STRESS IN PAR PEN WELD

$$\sigma_{MAX}, \sigma_{MIN} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau^2}$$

$$\sigma_x = f_t = 9160$$

$$\sigma_y = 0$$

$$\tau = f_v = 398$$

$$\sigma_{MAX, MIN} = \frac{9160}{2} \pm \sqrt{\left(\frac{9160}{2}\right)^2 + 398^2}$$

$$= 4580 \pm 4597$$

$$= 9177 \text{ OR } -17 \text{ psi}$$

STRESS INTENSITY

$$SI = 9177 - (-17)$$

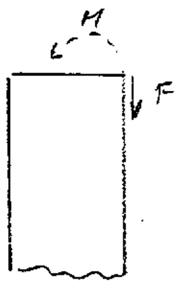
$$= 9195, \text{ SAY } 9200 \text{ psi}$$ ok

FOR P_L CATEGORY, COMPARE TO $1.5 S_M$

$S_M = 15.8$ KSI* => $1.5 S_M = 23.7$ KSI ok

*FOR 304L @ 400°F. FLANGE IS 304L SS.

STITCH FILLET WELD RESISTS MOMENT THROUGH COUPLE, DIRECT LOAD, F, IS ASSUMED TO BE RESISTED BY PAR PEN WELD ONLY



$$M = F\left(\frac{t}{2}\right)$$

$$\Rightarrow F = \frac{2M}{t}$$

$$= \frac{2(99.17)}{.5}$$

$$= 397$$

FOR A 3/16 STITCH FILLET, 2 IN LONG @ 6 IN THE SHEAR STRESS IS

$$f_v = \frac{F}{.707\, t_{ws}} \times \frac{6}{2}$$

$$= \frac{397}{.707(3/16)} \times \frac{6}{2}$$

$$= 9000 \text{ psi}$$

COMPARE TO LIMIT FOR PURE SHEAR PER SECT. VIII, DIV. 2, AD 132.2. THIS IS .6 S_m

$S_m = 15.8$ KSI @ 400°F FOR SA 304L

304L IS USED FOR FLANGES

$$.6 S_m = 9.5 \text{ KSI}$$

$$f_v < .6 S_m \quad \text{ok}$$

TRY THE SAME WELD CONFIGURATION IN A 1/4" THICK SHELL SINCE MANY OF THE TUBES ARE 1/4 IN THICK

$$V = .660\,(14.7)\,(52.5 \times .25)^{\frac{1}{2}}$$

$$= 35.15 \text{ LB/IN}$$

$$M = .257\,(14.7)\,(52.5)(.25)$$

$$= 49.58 \text{ IN-LB/IN}$$

1/8" PAR PEN WELD

$$f_V = \frac{V}{t_{WP}} = \frac{35.15}{.125}$$

$$= 281 \text{ psi}$$

$$F = 386 + \frac{M}{t/2} + 362$$

$$= 386 + \frac{49.58}{\frac{.25}{2}} + 362$$

$$= 386 + 397 + 362 = 1145$$

SAME AS F FOR 1/2 IN SHELL

$$\therefore SI \simeq 9200 \quad \text{ok}$$

FOR 3/16 STITCH FILLET *

$$F = \frac{2M}{t} = \frac{2(49.58)}{.25} = 397$$

SAME AS F FOR 1/2 IN SHELL

$$\therefore f_V \simeq 3000 \text{ psi} \quad \text{ok}$$

* NOTE 1/4" FILLET 2"@6" USED IN FINAL DESIGN OF SOME COMPONENTS.

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-022 PAGE 1 OF 16
REV.	DEO #	DATE	BY:	CHECK	Beam Splitter Chamber, FE Analysis of Lower Section	
0	0128	12/6/95	RDC			
		4/18/96		P.W.Y.		
					BY: R.D. Ciatto	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE:. To evaluate the lower section of the beam splitter chamber for vacuum pressure.

METHOD: A finite element analysis of the lower section shell, head and nozzles is performed.

ASSUMPTIONS:

INPUTS: LIGO project sketches and drawings.
Shell thickness = .5 in.
60 in. nozzle thickness = .5 in.
Head thickness = .375 in.

REFERENCES: 1. ASME Boiler & Pressure Vessel Code, Section VIII, Div. 1
2. IMAGES 3D, Version 3.0, R.L. Cloud & Associates
3. Doc. No. V049-1-066, LIGO Vacuum Equip. Structural Design Criteria

CALCULATIONS: (SEE ATTACHED)

CONCLUSIONS: Stresses are within the limits of the ASME Code, Section VIII. Div. 1.

NOTES: Computer files are BMSPLI10, BMSPLI11, and BMSPLI12.

BSC

END CAP LOADS ON 60 IN. NOZZLES

THERE ARE 32 ELEMENTS AROUND THE 60 IN FLANGE. THE AREA FOR THE MODEL IS 1/2 OF THE CIRCLE

$$A = \frac{\pi r^2}{2}$$

$$r = 30 \text{ in}$$

$$A = 1414 \text{ in}^2$$

THE TOTAL VACUUM LOAD IS

$$F = pA$$

$$p = 14.7 \text{ psi}$$

$$F = 14.7(1414) = 20,782 \text{ LB}$$

FOR EACH FLANGE NODE AT THE NOZZLE INNER SURFACE, EXCEPT THE CORNER NODES THE FORCE IS

$$F_i = \frac{20782}{32} = 649.4 \text{ LB}$$

FOR CORNER NODES

$$F_i = \frac{649.4}{2} = 324.7 \text{ LB}$$

22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD

ROL - 11/16/95

BSC FE ANALYSIS OF LOWER SHELL

FILENAME: BMSPLE12 - FINAL MODEL

LOAD CASES

1. VACUUM PRESS = 14.7 PSI
   END CAP LOAD ON BOTH 60 IN NOZZLES

2. SAME AS 1 BUT 60 IN NOZZLE ON Y AXIS HAS NO END CAP LOAD (SIMULATES BELLOWS)

3. SAME AS 1 BUT SUPPORT LEG REACTIONS ARE APPLIED TO SHELL AT 12 NODES AT WELD PAD

LOAD CASE 3

SUPPORT LEG REACTION ON SHELL.

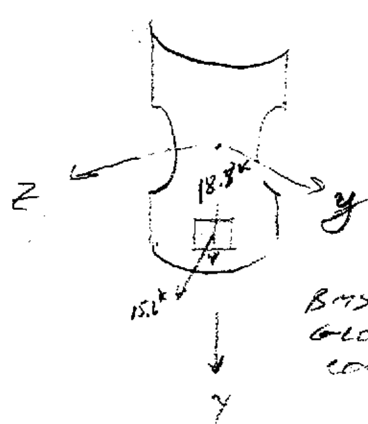
LOCAL MEMBER FORCES (FROM BSCSUPPB FILE*)

MEMBER	NODE	F_x (VERT)	F_y (TANGENTIAL)	F_z (RADIAL)
2	3	-3019	-14900	442 LB
→ 4	6	-18760	125	-15620
6	9	-3281	14900	694
8	12	12460	-125	14490

THE BOUNDING LOADS ARE AT MEMBER 4 WHERE THE VERTICAL LOAD IS 18760 AND THE RADIAL LOAD IS 15620 LB, MOMENTS ARE NOT EVALUATED SINCE THEY ARE ROTATION LIMITED.

* NO PRINTED OUTPUT, VALUES FROM SCREEN

FOR THE BMSPL112 MODEL, THE RADIAL LOAD WILL BE APPLIED OUTWARD AND THE VERTICAL LOAD WILL BE DOWNWARD THE GLOBAL LOAD COMPONENTS IN THE BMSPL112 MODEL ARE



BMSPL112 GLOBAL COORD.

F_x = 18 760 LB

F_y = .707 (15620) = 11045 LB

F_z = F_y = 11045 LB

θ = 45° = ANGULAR LOCATION OF PAD FROM Z-AXIS

APPLY LOADS TO THE FOLLOWING NODES

131
134
138
135
273
275
~~278~~
~~279~~
281
1028
1031
1035
1174
1172

⇒ 12 NODES

NODAL LOADS

$$F_{xi} = \frac{F_x}{12} = \frac{18760}{12} = 1563 \text{ LB}$$

$$F_{yi} = \frac{F_y}{12} = \frac{11045}{12} = 920.4 \text{ LB}$$

$$F_{zi} = \frac{F_z}{12} = 920.4 \text{ LB}$$

NOTE: THE FINAL SUPPORT LEG FORCES ARE GIVEN ON P. 32 OF CALC 024. THE RESULTANT FINAL LOAD IS F = 27.2 K. FOR THE FORCES ANALIZED IN THIS CALC THE RESULTANT IS SLIGHTLY LOWER

$$F = (18.76^2 + 11.0^2 + 11.0^2)^{\frac{1}{2}}$$

$$= 24.4^K$$

SINCE THE SHELL STRESS PLOT ON P. 15 SHOWS THAT THE MAX STRESS INTENSITY IS ONLY ABOUT 8.5 KSI AT THE LEG ATTACHMENT, THE SHELL STRESSES MEET THE CRITERIA FOR THE FINAL LOADS.

LOAD CASE 1 RESULTS SUMMARY FOR 3 LAST MODELS

BEAM SPLI 10

- NO STIFFENERS AT G-NOZZLES
- RING STIFFENER ABOVE PORTS
- ANGLE STIFFENER BELOW "

MAX SE = 18.5 KSI AT MIDDLE G-NOZ

" " = 14 KSI AT UPPER NOZZLES (VERY LOCAL AT RING STIFFENER

MAX SE = 23.8 KSI AT BOTTOM OF 60 IN PORTS

STRESS IN LOWER HEAD ≅ 2.7 KSI
MAX RADIAL DISP = .15" @ G NOZ

BMSPLI11

- SAME AS BMSPLI10 BUT NO STIFFENER BELOW 60" PORT

MAX RADIAL DISP = .19 IN @ G NOZ

AT MIDDLE NOZ G, MAX SI = 21 KSI

AT UPPER NOZZLES, MAX SE = 17 KSI AT RING STIFFENER.

AT BOTTOM OF 60 IN PORTS, SE = 28.4 KSI

LOAD CASE 1 RESULTS CONT.

BMSPLI12 (FINAL MODEL)

- STIFFENERS BETWEEN 6 NOZ*
- RING STIFFENER ABOVE PORTS (4" x 3/8")
- NO STIFFENER BELOW PORTS

AT 6 -NOZZ

MAX SI = 27.9 KSI AT STIFF. JCT W/ 60 IN PORT

MAX Δ = .14" (RADIAL)

AT UPPER NOZZLES

MAX SI = 14 KSI AT TOP OF 60" PORT

AT BOT OF 60 IN PORTS

MAX SI = 27.9 KSI (VERY LOCAL)

NOTE: ALL SIs ARE AT SURFACES

*STIFFENER IS 3" x 3/4"

22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD

THE FOLLOWING PLOTS ARE FROM THE MODEL WITH

FILENAME: BMSPLI12

MAX STRESS INTENSITY

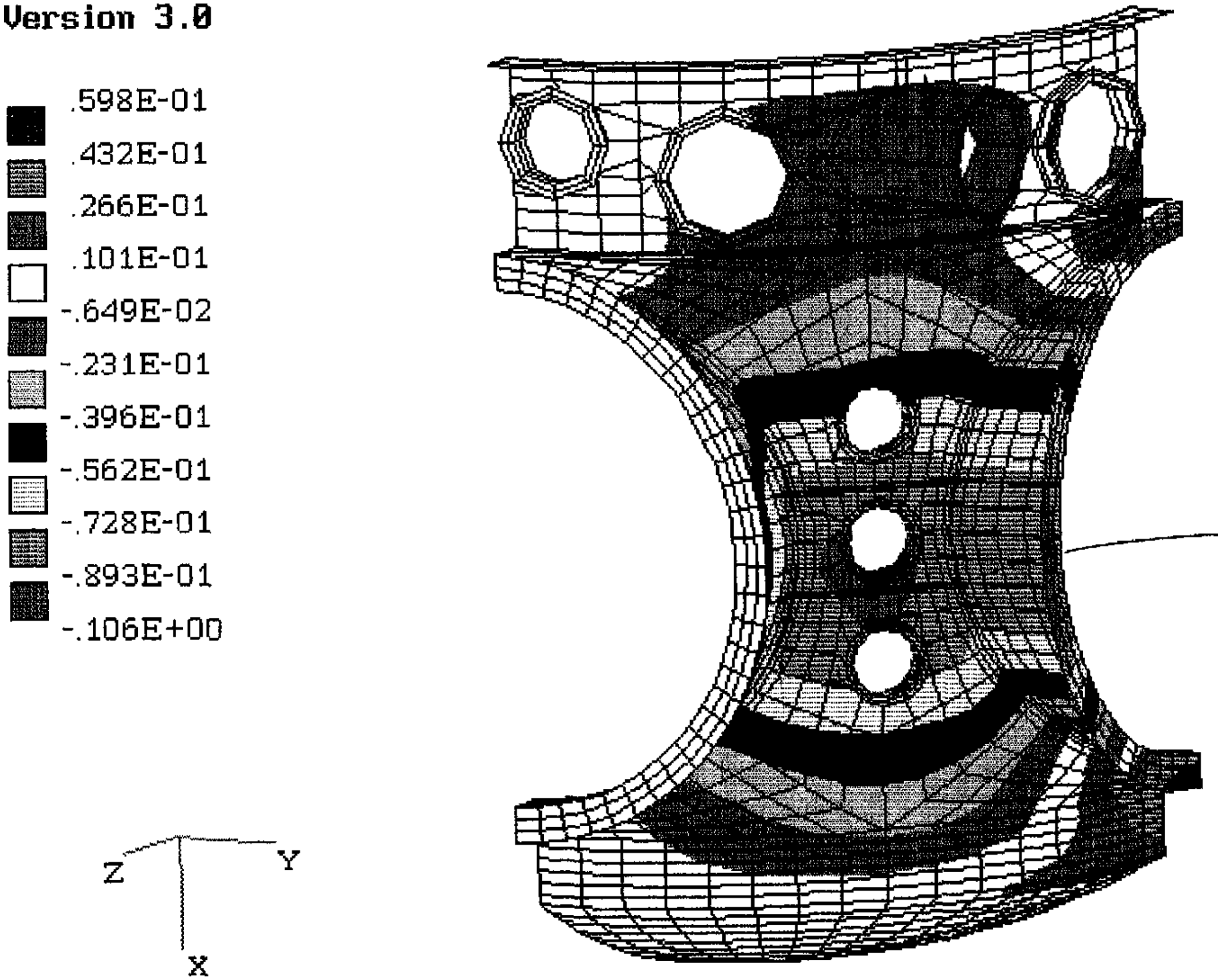
SI_{MAX} = 23.5 KSI AT BOTTOM OF 60 IN NOZZLES

≤ 1.5 S = 24.3 KSI @ 400°F

FOR 304/304L DUAL CERTIFIED MATERIAL

THIS MAX STRESS IS VERY LOCAL

IMAGES-3D
Version 3.0

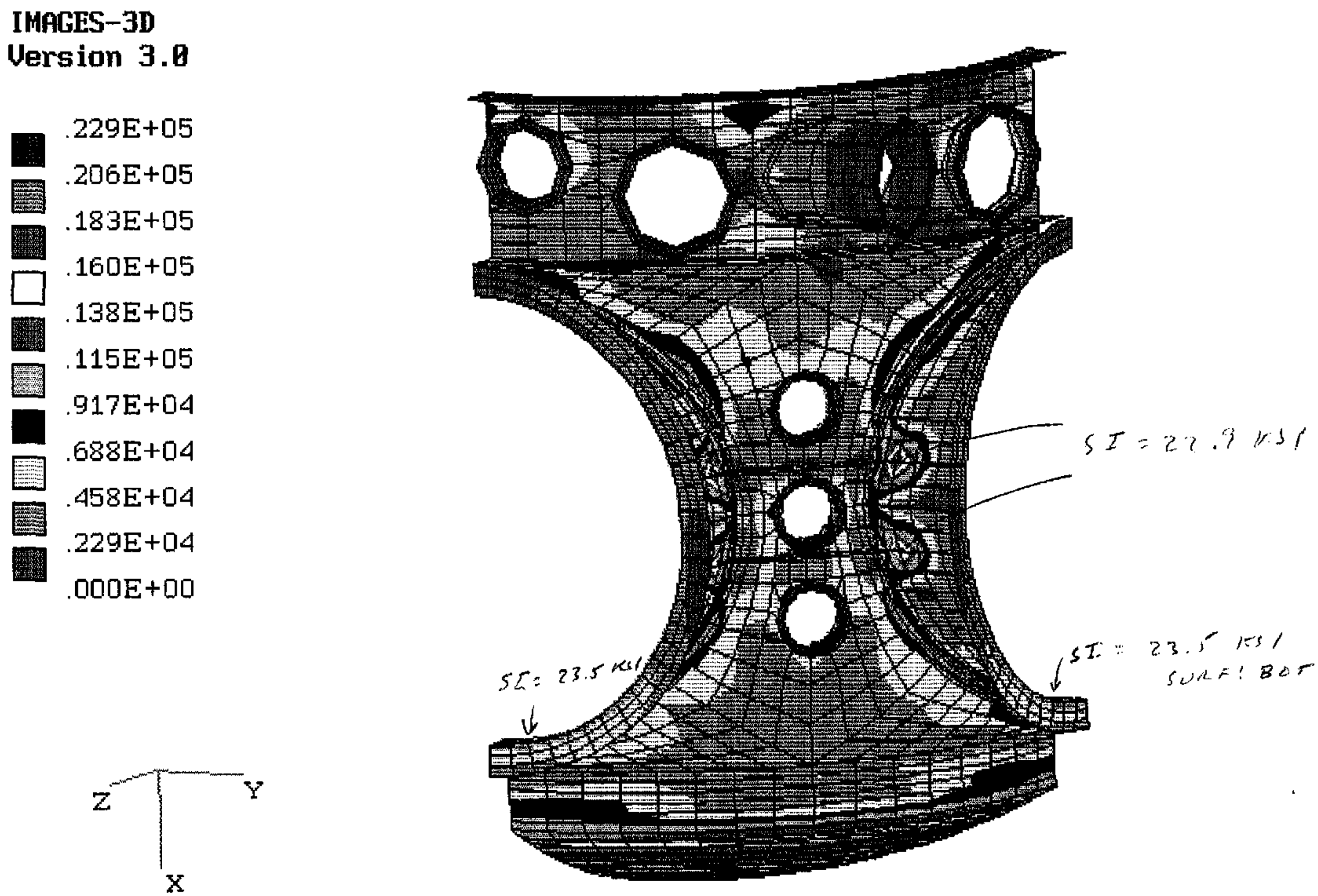


MAX DEFL AT MIDDLE NOZZLE G
$D_x = D_y = .106"$
Radial $\Delta = (.106^2 + .106^2)^{\frac{1}{2}}$
$= .15$ IN INWARD

Load Case 1

Displacement Contour Plot DY

11/16/95
11:53:45



Load Case 1

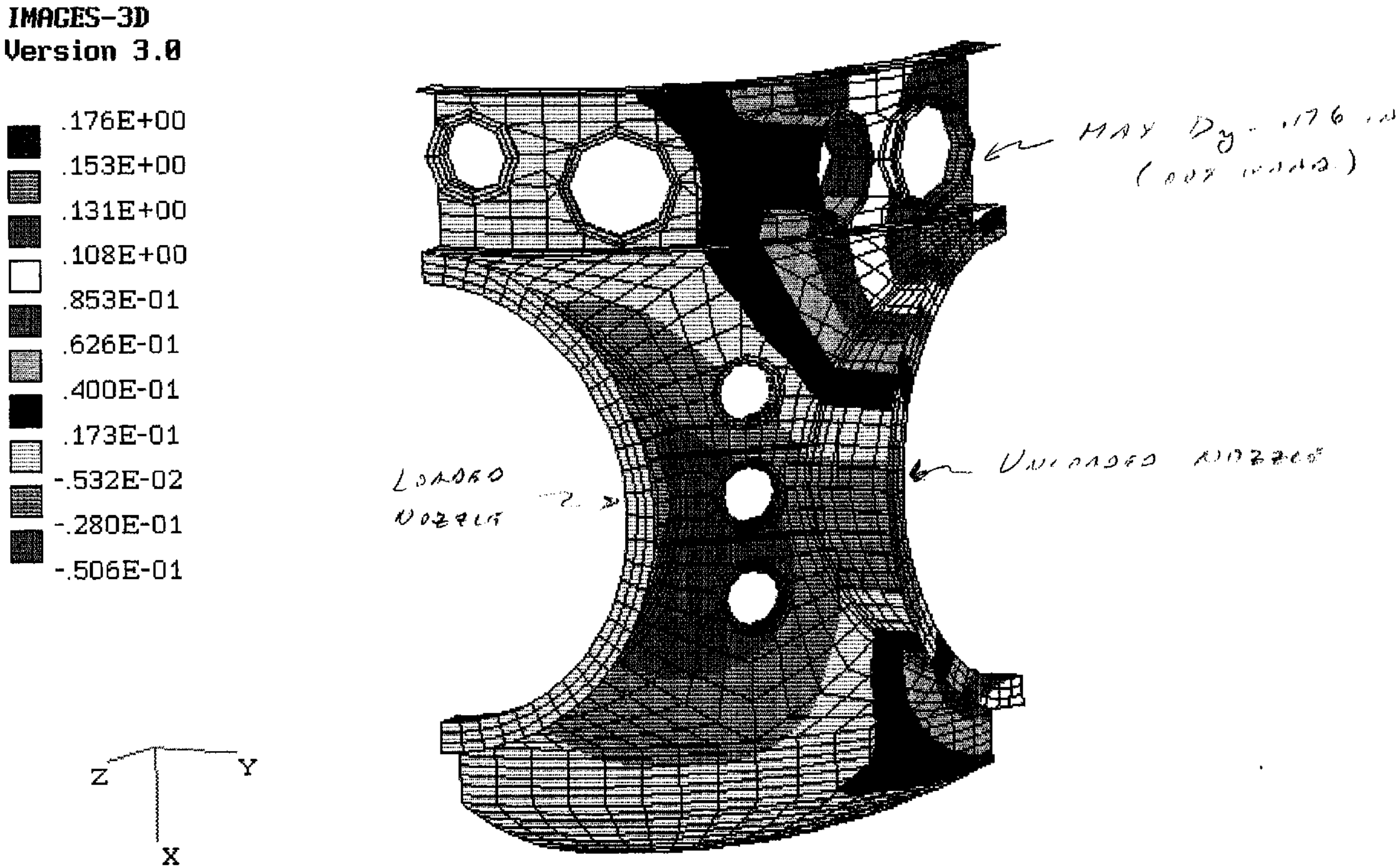
Stress Contour Plot

Surf: Top

Stress Intensity

11/16/95

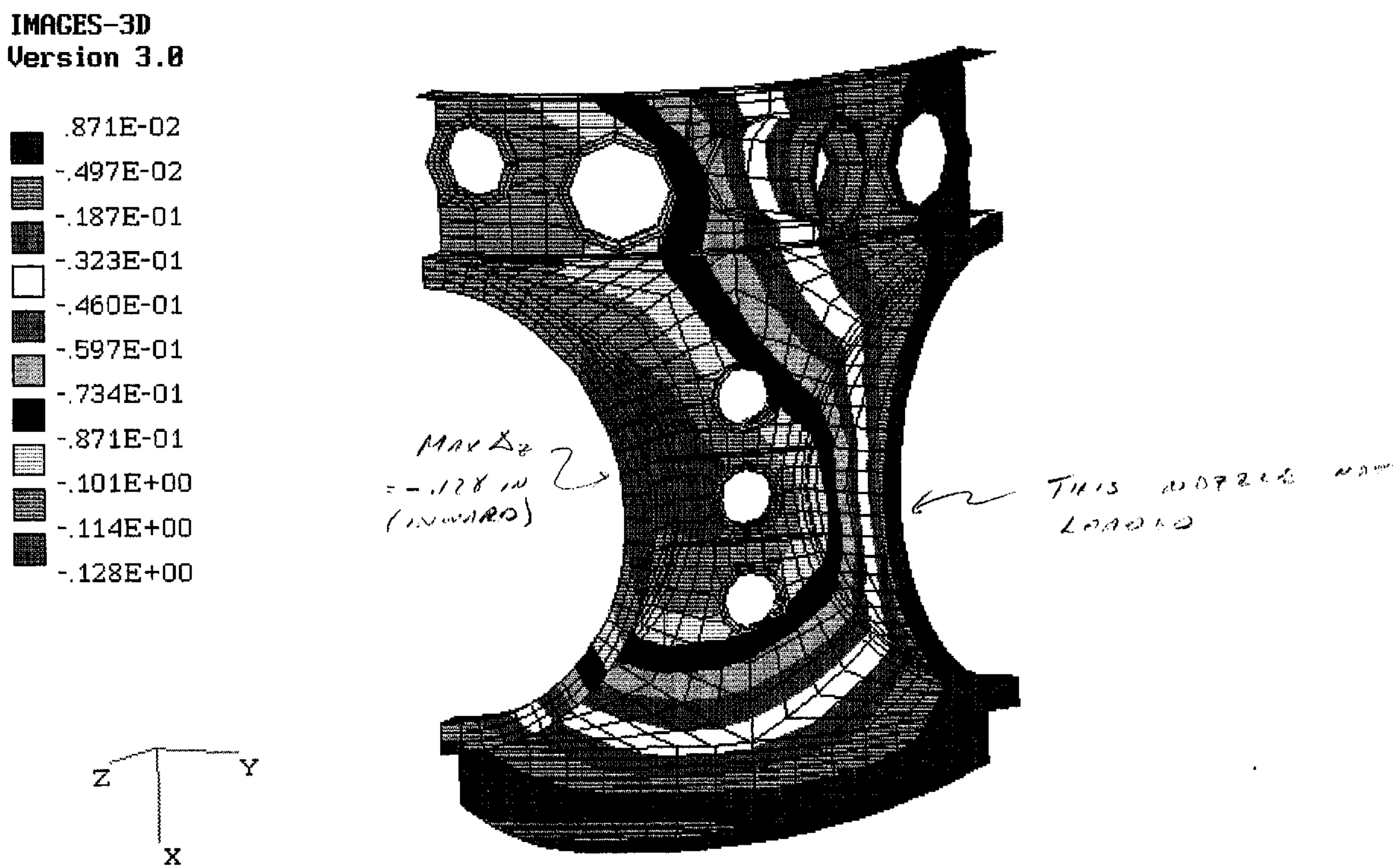
11:50:33



Load Case 2

Displacement Contour Plot DY

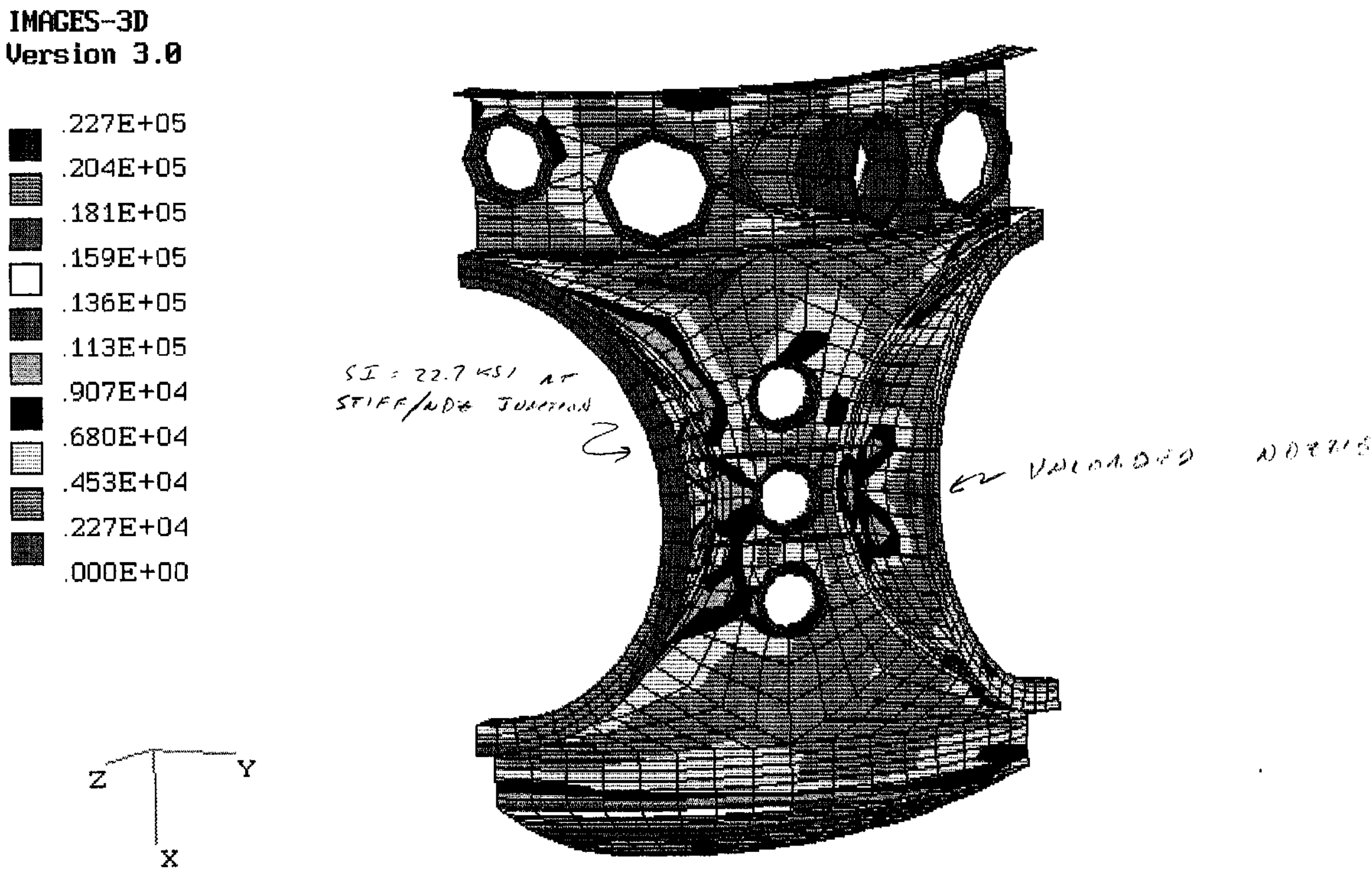
11/16/95
12: 2: 0



Load Case 2

Displacement Contour Plot DZ

11/16/95 11:57:54



Load Case 2

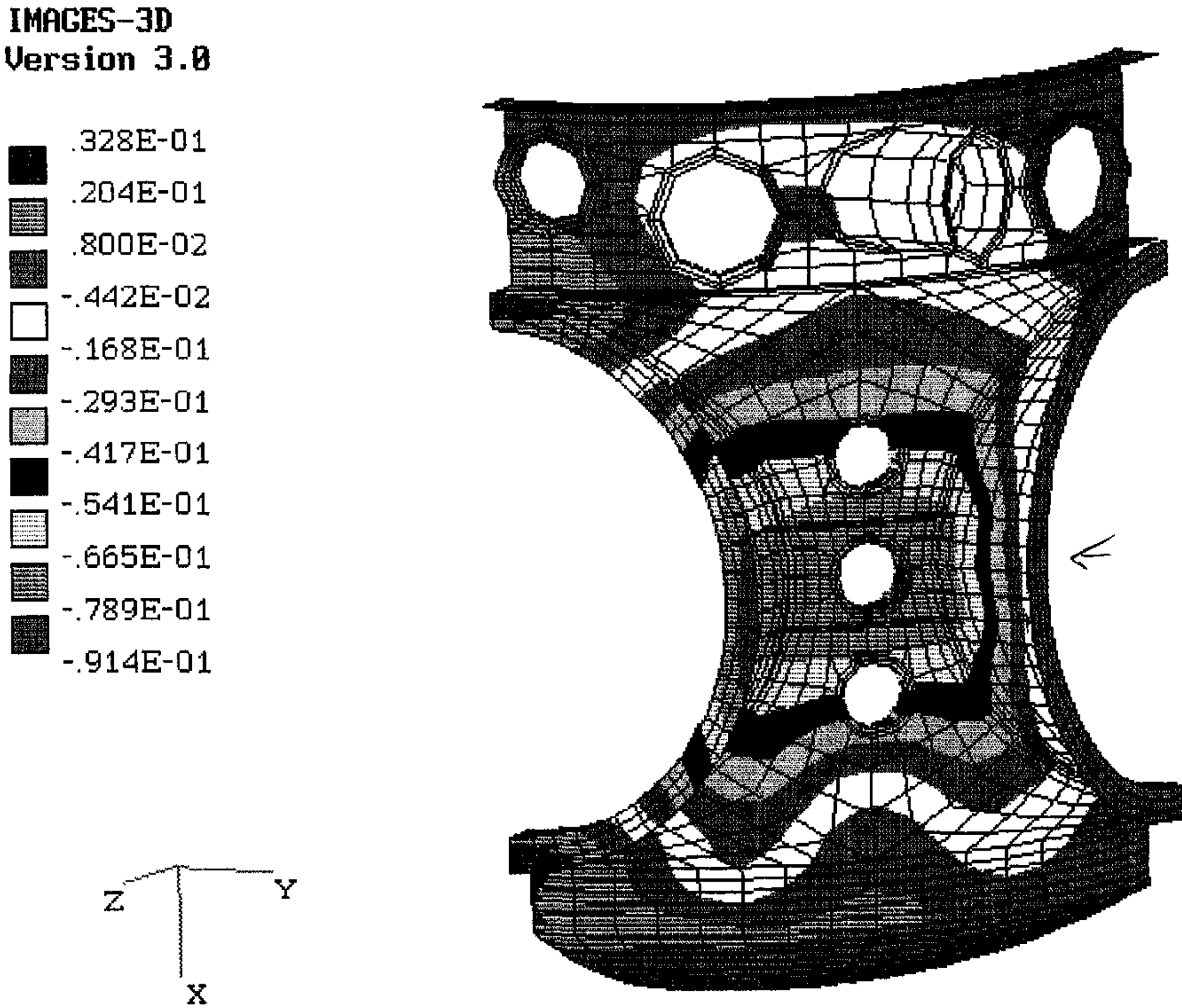
Stress Contour Plot

Surf: Top

Stress Intensity

11/16/95

13: 1:10



MAX INWARD DEFL
AT MIDDLE NODE G

$D_Y = D_Z \approx .0914$ IN

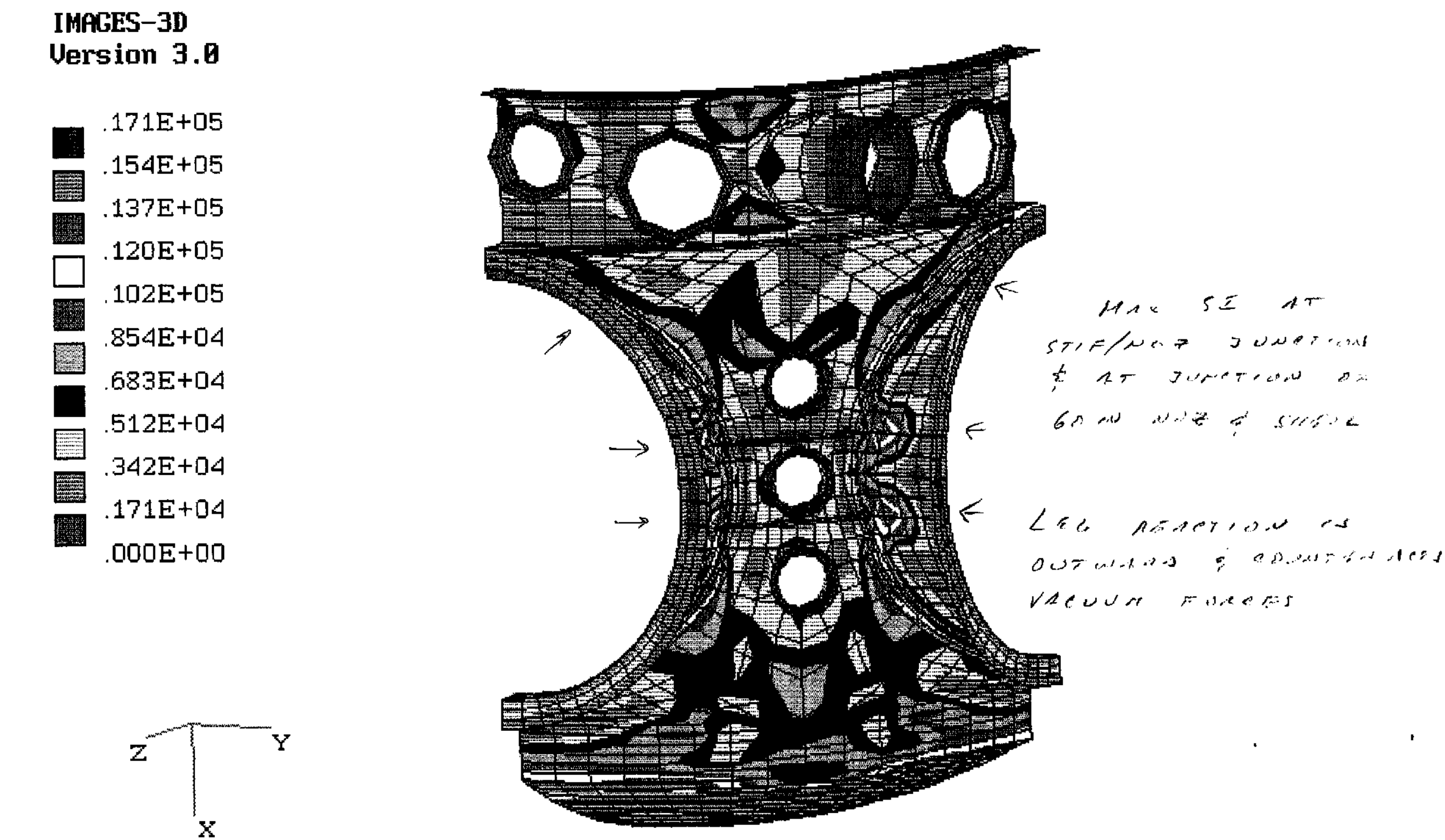
RADIAL $\Delta \approx (.0914^2 + .0914^2)^{\frac{1}{2}}$

$\approx .129$ IN

Load Case 3

Displacement Contour Plot D2

11/16/95
13: 7:39



Load Case 3 | Stress Contour Plot | Surf: Top | Stress Intensity | 11/16/95 13:12:30

CONFIRMATION OF NOZZLE G

NOZ G WRC 107 ANALYSIS FOR RADIAL FORCE

F = 738.5 LB

STRESS

σ = -2646 psi

THE "COMPRESS" PRESSURE VESSEL PROGRAM SHOWS THAT NO REINFORCEMENT IS REQD.

SHELL THICKNESS = .5 IN

NOZZLE THICKNESS = .125 IN

22-141 50 SHEETS
22-142 100 SHEETS
22-144 200 SHEETS
AMPAD

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-023 PAGE 1 OF 19
REV.	DEO #	DATE	BY:	CHECK	Beam Splitter Chamber - Design of 60 in. Access Covers	
0	0029	12/6/95	RDC	P.W.Y.		
					BY: R.D. Ciatto	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO:

PURPOSE:. To design the cover and nozzles for the 60 in. port in accordance with Section VIII, Division 1 of the ASME Code.

METHOD: Equations for thickness and reinforcement requirements of the ASME Code, Section VIII, Division 1, are evaluated using the COMPRESS computer program, Version 5.31.

ASSUMPTIONS:

INPUTS: Vacuum pressure = 14.7 psi
Design temperature = 400^0F

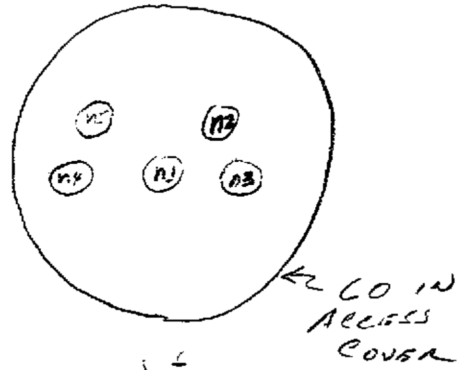
REFERENCES: 1. ASME Boiler & Pressure Vessel Code, Section VIII, Pressure Vessels, Division 1.
.2. COMPRESS 5.31, Computer Aided Pressure Vessel Design, Codeware Computer Systems, Inc.
3. Doc. No. V049-1-066, LIGO Vacuum Equip., Structural Design Criteria

CALCULATIONS: (SEE ATTACHED)

CONCLUSIONS: The requirements of the ASME Code, Section VIII. Division 1, are met for the cover and nozzles for the 60 in. port.

NOTES:

NOZZLE LOCATIONS IN COMPRESS



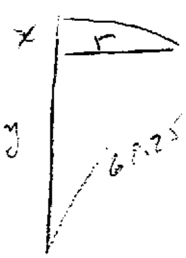
n1 @ CENTER

n2 $\theta = 42.90$

$r = (10.625^2 + 9.875^2)^{\frac{1}{2}}$

$= 14.51$

$L = 14.53 - \gamma$



$y = (60.25^2 - 14.51^2)^{\frac{1}{2}}$

$= 58.48$ IN

$\gamma = 60.25 - 58.48$

$= 1.77$

$L = 14.53 - 1.77$

$= 12.76$

n_3 $\theta = 90$

$r = 15.75$

$y = (60.25^2 - 15.75^2)^{\frac{1}{2}}$

$= 58.15$

$\gamma = 60.25 - 58.15 = 2.10$

$L = 14.53 - 2.10 = 12.43$

DUE TO OVERLAPPING REINFORCEMENT, NOZZLE LOCATIONS WERE MOVED AS SHOWN ON THE FOLLOWING SHEET.

G-NOZZLES IN 60 IN PORT.

ID = 7.5 IN

WALL THICKNESS = .25 IN

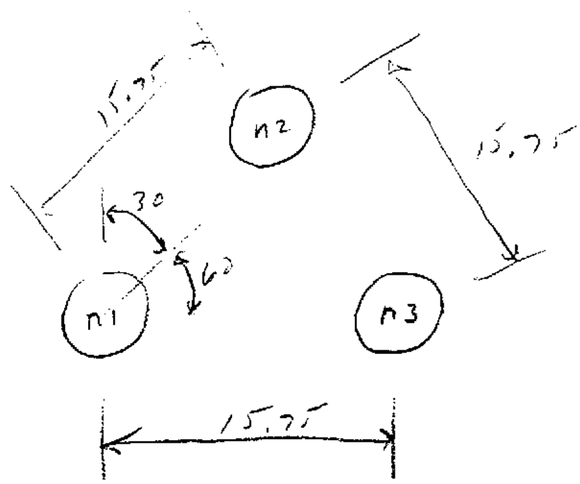
FILLET WELD = .25

PARTIAL PEN GROOVE WELD = .1875

REINF. REQ = .73 IN^2

" AVAIL = .83 IN^2 OK W/O PAD

LOCATE NOZZLES AS FOLLOWS



RADIAL PRESSURE LOAD FOR WRC 107 ANAL

$$F_R = \frac{\pi}{4} (7.5)^2 (14.7)$$
$$= 649.4 \text{ LB}$$

MAX STRESS AT OD OF NOZZLE

f = 2802 psi FROM

WRC 107 ANAL

ASME Section VIII Division 1, 1992 Edition, A93 Addenda

Component: F&D head
Material specification: SA 240 304L HIGH

External design pressure: Pe= 14.7 psi @ 400 deg F

Corrosion allowance: Inner C = 0 Outer= 0 in

PWHT is not performed

Radiography: Category A joints - Spot UW-11(b) type 1
Head to shell seam - Spot UW-11(b) type 1

Estimated weight: new = 272.9 corr = 272.9 lb
capacity: new = 88.89 corr = 88.89 US ga

ID = 60.25 crown L = 60.25 knuckle r = 3.75 t = .25 in (min)

Straight flange = 1 forming allowance = 0 in

MAP: (New & at 0 deg F) Appendix 1-4(d) Eq 3

P = 2*S*E*t/(L*M + 0.2*t) - Ps
= 2*16700*0.85*0.25/(60.25*1.7521 + 0.2*0.25) - 0
= 67.20225 psi

MAWP: (Corroded & at 0 deg F) Appendix 1-4(d) Eq 3

P = 2*S*E*t/(L*M + 0.2*t) - Ps
= 2*16700*0.85*0.25/(60.25*1.7521 + 0.2*0.25) - 0
= 67.20225 psi

External Pressure: (Corroded & at 400 deg F) UG-33(e)

A = .125/(Ro/t)
= .125/(60.5/0.18787)
= 0.000388

From table HA-3: B = 4757.7

Pa= B/(Ro/t)
= 4757.7/(60.5/0.18787)
= 14.774 psi

Check the external pressure per UG-33(a)(1)

t = 1.67*Pa*Lo*M/(2*S*E + 1.67*Pa*(M-0.2))
= 1.67*14.774*60.5*1.7521/(2*14700*1 + 1.67*14.774*(1.7521-0.2))
= 0.088842 in

Design thickness for external pressure Pa = 14.774 psi:

= t + Corrosion + fa
= 0.18787 + 0 + 0
= 0.18787 in

Maximum Allowable External Pressure: (Corroded @ 400 deg F)

A = .125/(Ro/t)
= .125/(60.5/0.25)
= 0.000517

From table HA-3: B = 5004.1

Pa= B/(Ro/t)
= 5004.1/(60.5/0.25)
= 20.6781 psi

Check the Maximum External Pressure: UG-33(a)(1) & App. 1-4(d)

Pe= 2*S*E*t/((M*Lo - t*(M-0.2))*1.67)
= 2*14700*1*0.25/((1.7521*60.5 - 0.25*(1.7521-0.2))*1.67)
= 41.67247 psi

The maximum allowable external pressure is 20.6781 psi.

Center Nozzle n1

Opening n1 Reinforcement Calculations Per UG-37

Located on: BSC 60 in Port
Local vessel thickness: .25 in
Liquid static head included: 0 psi
Flange description: 8 inch 75# SO A240304L
ANSI B16.47 flange rating MAP: 115 psi
ANSI B16.47 flange rating MAWP: 115 psi

Nozzle material specification: SA 240 304L HIGH

Nozzle orientation: 0 degrees
End of nozzle to datum line: 14.53 in
Nozzle calculated as hillside: no
Projection outside vessel Lpr: 2.996 in

```
   tn ->|  |<-
                 _             _
              | |<- d ->|   |  |      corrosion allow = 0 in
              | |             |  |      noz thick new tn= .25 in
twl->|  |<|             |  |      nozzle id. new d= 7.5 in
         |/|  |   v         |  |\       fillet weld twl = .25 in
______.-|  |  —         |  |-.______  groove weld tw2 = .1875 in
______\|_|_tw2       |_|/______
              ^
<—— R ——>|                     To head center R= 0 in
```

Reinforcement Calculations For Nozzle MAWP

Limits of reinforcement UG-40

Parallel to the vessel wall d = 7.5 in
Normal to the vessel wall outside 2.5*(tn-Cn) + te = .625 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .625 in

Nozzle required thickness

trn = P*Rn/(Sn*E - 0.6*P)
= 0*3.75/(16700*1 - 0.6*0)
= 0 in

Required thickness tr from UG-37(a)(1)

tr= P*L*M/(2*S*E - 0.2*P)
= 0*60.25*1/(2*16700*1 - 0.2*0)
= 0 in

Area required

Allowable stresses: Sn = 16700, Sv = 16700, psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1

Center Nozzle n1

fr2 = lesser of 1 or Sn/Sv so fr2 =1

A = d*tr*F + 2*tn*tr*F*(1 - fr1)
= 7.5*0*1 + 2*0.25*0*1*(1 - 1)
= 0 in^2

Area available

A1 = larger of the following = 1.875 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 7.5*(1*0.25-1*0) - 2*0.25*(1*0.25-1*0)*(1-1)
= 1.875 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.25+0.25)*(1*0.25-1*0) - 2*0.25*(1*0.25-1*0)*(1-1)
= .25 in^2

A2 = smaller of the following = 0.313 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.25 - 0)*1*0.25
= .313 in^2

= 5*(tn - trn)*fr2*tn
= 5*(0.25 - 0)*1*0.25
= .313 in^2

A41 = Leg^2*fr2
= 0.25^2*1 = .063 in^2

Area = A1 + A2 + A41
= 1.875 + 0.313 + 0.063
= 2.251 in^2

As Area > A the reinforcement is adequate for MAWP = 0 at 0 Deg F

Check the welds - From UW-16(d):

tmin = lesser of 0.75 or tn or t, tmin = 0.25 in
t1 or t2(min) = lesser of 0.25 or 0.7*tmin, t1(min) = 0.175 in
t1(actual) = 0.7*Leg = 0.7*0.25 = 0.175 in
t2(actual) = 0.1875 in
t1 + t2 = 0.3625 >= 1.25*tmin

The weld sizes for t1 and t2 are satisfactory.

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0 in (E = 1)
Wall thickness per UG-45(b)(1):	tr2 = 0 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.28175 in

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Doc. No. V049-1-023

Center Nozzle n1

The greater of tr2 or tr3: tr5 = 0.0625 in
The lesser of tr4 or tr5: tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.25 in

The nozzle neck thickness is adequate for MAWP.

Allowable stresses in joints UG-45(c) and UW-15(c)

Groove weld in tension = 0.74*16700 = 12358 psi
Nozzle wall in shear = 0.7*16700 = 11690 psi
Inner fillet weld in shear = 0.49*16700 = 8183 psi

Strength of welded joints:

(1) Inner fillet weld in shear
(Pi/2)*Nozzle O.D.*Leg*Si = 1.57*8*0.25*8183 = 25694.62 lbf

(3) Nozzle wall in shear
(Pi/2)*Mean nozzle dia.*tn*Sn = 1.57*7.75*0.25*11690 = 35559.52 lbf

(4) Groove weld in tension
(Pi/2)*Nozzle O.D.*tw*Sg = 1.57*8*0.1875*12358 = 29103.09 lbf

Loading on welds per UG-41(b)(1)

W = (A - (d - 2*tn)*(E1*t - F*tr))*Sv
= (0 - (7.5 - 2*0.25)*(1*0.25 - 1*0))*16700
=-29225 lbf

W1-1 = (A2 + A5 + A41 + A42)*Sv
= (0.313 + 0 + 0.063 + 0)*16700
= 6279.2 lbf

W2-2 = (A2 + A3 + A41 + A43 + 2*tn*t*fr1)*Sv
= (0.313 + 0 + 0.063 + 0 + 2*0.25*0.25*1)*16700
= 8366.7 lbf

Load for path 1-1 lesser of W or W1-1 =-29225 lbf
Path 1-1 Thru (1) & (3) = 25694.62 + 35559.52 = 61254.14 lbf
Path 1-1 is stronger than W so it is acceptable per UG-41(b)(2).

Load for path 2-2 lesser of W or W2-2 =-29225 lbf
Path 2-2 Thru (1), (4) = 25694.62 + 29103.09 = 54797.71 lbf
Path 2-2 is stronger than W so it is acceptable per UG-41(b)(2).

Reinforcement Calculations for External Pressure

Rev 0

Limits of reinforcement UG-40

Doc. No V049-1-023

Parallel to the vessel wall d = 7.5 in

Center Nozzle n1

Normal to the vessel wall outside 2.5*(tn-Cn) + te = .625 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .625 in

Nozzle required thickness

L/Do = 2.996/8 = .3745 Do/t = 8/0.0184 = 434.7826
From table G: A = 0.000415
From table HA-3: B = 4814.3

Pa= 4*B/(3*Do/t)
= 4*4814.3/(3*8/0.0184)
= 14.7639 psi

Nozzle required thickness trn = .0184 in

Required thickness tr from UG-37(d)(1) = .1879 in

Area required

Allowable stresses: Sn = 14700, Sv = 14700, psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1
fr2 = lesser of 1 or Sn/Sv so fr2 =1

A = 0.5*(d*tr*F + 2*tn*tr*F*(1 - fr1))
= 0.5*(7.5*0.1879*1 + 2*0.25*0.1879*1*(1 - 1))
= .7046 in^2

Area available

A1 = larger of the following = .466 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 7.5*(1*0.25-1*0.1879) - 2*0.25*(1*0.25-1*0.1879)*(1-1)
= .466 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.25+0.25)*(1*0.25-1*0.1879) - 2*0.25*(1*0.25-1*0.1879)*(1-1)
= .062 in^2

A2 = smaller of the following = 0.289 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.25 - 0.0184)*1*0.25
= .289 in^2

= 5*(tn - trn)*fr2*tn
= 5*(0.25 - 0.0184)*1*0.25
= .289 in^2

A41 = Leg^2*fr2
= 0.25^2*1 = .063 in^2

Rev 0

V049-1-023

Center Nozzle n1

Area = A1 + A2 + A41
= 0.466 + 0.289 + 0.063
= .818 in^2

As Area > A the reinforcement is adequate for Pe = 14.7 at 400 Deg F

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.0184 in (E = 1)
Wall thickness per UG-45(b)(2):	tr2 = 0.0528 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.28175 in
The greater of tr2 or tr3:	tr5 = 0.0625 in
The lesser of tr4 or tr5:	tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.25 in

The nozzle neck thickness is adequate for Pe.

Center Nozzle n1

Applied Loads

Radial load	Pr = 649.4 lbf
Circumferential moment	M1 = 0 lbf-ft
Circumferential shear	V2 = 0 lbf
Longitudinal moment	M2 = 0 lbf-ft
Longitudinal shear	V1 = 0 lbf
Torsion moment	Mt = 0 lbf-ft
Internal pressure	P = 0 psi

Stresses at the nozzle OD per WRC bulletin 107 (psi)

Mean dish radius Rm = 60.375 in
U = ro/Sqr(Rm*t) = 1.03

Stress concentration factor Kn (tension) = 1
Stress concentration factor Kb (bending) = 1

Pressure stress intensity factor, Farr equation 11.7

I = 1 + (r/x)^2
= 1 + (3.75/4.25)^2
= 1.779

Local pressure stress = I*P*Rm/2*t = 0 psi

Maximum combined stress =-2802 psi
Allowable combined stress = +-3*S = +- 50100 psi

The maximum combined stress is within allowable limits.

Maximum primary membrane stress =-670 psi
Allowable primary membrane stress = +-1.5*S = +- 25050 psi

The maximum primary membrane stress is within allowable limits.

Rev 0

Center Nozzle n1

From Fig.	Value read	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
SR-2*	0.0645	-670	-670	-670	-670	-670	-670	-670	-670
SR-2	0.0342	-2132	2132	-2132	2132	-2132	2132	-2132	2132
SR-3*	0.0750								
SR-3	0.0711								
SR-3*	0.0750								
SR-3	0.0711								
pressure stress*									
Total Ox stress		-2802	1462	-2802	1462	-2802	1462	-2802	1462
Primary membrane Ox stress*		-670	-670	-670	-670	-670	-670	-670	-670
SR-2*	0.0200	-208	-208	-208	-208	-208	-208	-208	-208
SR-2	0.0103	-642	642	-642	642	-642	642	-642	642
SR-3*	0.0226								
SR-3	0.0214								
SR-3*	0.0226								
SR-3	0.0214								
pressure stress*									
Total Oy stress		-850	434	-850	434	-850	434	-850	434
Primary membrane Oy stress*		-208	-208	-208	-208	-208	-208	-208	-208
torsion moment Mt									
Shear from V1									
Shear from V2									
Total Shear stress									
Combined stress		-2802	1462	-2802	1462	-2802	1462	-2802	1462

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Nozzle n5

Opening n5 Reinforcement Calculations Per UG-37

Located on:	BSC 60 in Port
Local vessel thickness:	.25 in
Liquid static head included:	0 psi
Flange description:	8 inch 75# WN A240304L
ANSI B16.47 flange rating MAP:	115 psi
ANSI B16.47 flange rating MAWP:	115 psi

Nozzle material specification: SA 240 304L HIGH

Nozzle orientation:	330 degrees
End of nozzle to datum line:	13.76 in
Nozzle calculated as hillside:	no
Projection outside vessel Lpr:	4.173 in

```
  tn ->| |<-
        _          _
       | |<- d ->| |
twl->| |<|       | |
     |/| |   v   | |\
_____.-| | __    | |-._____
_____\|_|_tw2    |_|/_____
           ^
<---- R ---->|
```

```
corrosion allow = 0 in
noz thick new tn= .25 in
nozzle id. new d= 7.5 in
fillet weld twl = .25 in
groove weld tw2 = .1875 in

To head center R= 15.75 in
```

Reinforcement Calculations For Nozzle MAWP

Limits of reinforcement UG-40

Parallel to the vessel wall d = 7.5 in
Normal to the vessel wall outside 2.5*(tn-Cn) + te = .625 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .625 in

Nozzle required thickness

trn = P*Rn/(Sn*E - 0.6*P)
= 0*3.75/(16700*1 - 0.6*0)
= 0 in

Required thickness tr from UG-37(a)(1)

tr= P*L*M/(2*S*E - 0.2*P)
= 0*60.25*1/(2*16700*1 - 0.2*0)
= 0 in

Area required

Allowable stresses: Sn = 16700, Sv = 16700, psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1

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Nozzle n5

fr2 = lesser of 1 or Sn/Sv so fr2 =1

A = d*tr*F + 2*tn*tr*F*(1 - fr1)
 = 7.5*0*1 + 2*0.25*0*1*(1 - 1)
 = 0 in^2

Area available

A1 = larger of the following = 1.875 in^2

 = d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
 = 7.5*(1*0.25-1*0) - 2*0.25*(1*0.25-1*0)*(1-1)
 = 1.875 in^2

 = 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
 = 2*(0.25+0.25)*(1*0.25-1*0) - 2*0.25*(1*0.25-1*0)*(1-1)
 = .25 in^2

A2 = smaller of the following = 0.313 in^2

 = 5*(tn - trn)*fr2*t
 = 5*(0.25 - 0)*1*0.25
 = .313 in^2

 = 5*(tn - trn)*fr2*tn
 = 5*(0.25 - 0)*1*0.25
 = .313 in^2

A41 = Leg^2*fr2
 = 0.25^2*1 = .063 in^2

Area = A1 + A2 + A41
 = 1.875 + 0.313 + 0.063
 = 2.251 in^2

As Area > A the reinforcement is adequate for MAWP = 0 at 0 Deg F

Check the welds - From UW-16(d):

tmin = lesser of 0.75 or tn or t, tmin = 0.25 in
t1 or t2(min) = lesser of 0.25 or 0.7*tmin, t1(min) = 0.175 in
t1(actual) = 0.7*Leg = 0.7*0.25 = 0.175 in
t2(actual) = 0.1875 in
t1 + t2 = 0.3625 >= 1.25*tmin

The weld sizes for t1 and t2 are satisfactory.

UG-45 Nozzle Neck Thickness Check

Rev 0

Wall thickness per UG-45(a):	tr1 = 0 in (E = 1)
Wall thickness per UG-45(b)(1):	tr2 = 0 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.28175 in

Doc. No V049-1-023

Nozzle n5

The greater of tr2 or tr3: tr5 = 0.0625 in
The lesser of tr4 or tr5: tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.25 in

The nozzle neck thickness is adequate for MAWP.

Allowable stresses in joints UG-45(c) and UW-15(c)

Groove weld in tension = 0.74*16700 = 12358 psi
Nozzle wall in shear = 0.7*16700 = 11690 psi
Inner fillet weld in shear = 0.49*16700 = 8183 psi

Strength of welded joints:

(1) Inner fillet weld in shear
(Pi/2)*Nozzle O.D.*Leg*Si = 1.57*8*0.25*8183 = 25694.62 lbf

(3) Nozzle wall in shear
(Pi/2)*Mean nozzle dia.*tn*Sn = 1.57*7.75*0.25*11690 = 35559.52 lbf

(4) Groove weld in tension
(Pi/2)*Nozzle O.D.*tw*Sg = 1.57*8*0.1875*12358 = 29103.09 lbf

Loading on welds per UG-41(b)(1)

W = (A - (d - 2*tn)*(E1*t - F*tr))*Sv
= (0 - (7.5 - 2*0.25)*(1*0.25 - 1*0))*16700
=-29225 lbf

W1-1 = (A2 + A5 + A41 + A42)*Sv
= (0.313 + 0 + 0.063 + 0)*16700
= 6279.2 lbf

W2-2 = (A2 + A3 + A41 + A43 + 2*tn*t*fr1)*Sv
= (0.313 + 0 + 0.063 + 0 + 2*0.25*0.25*1)*16700
= 8366.7 lbf

Load for path 1-1 lesser of W or W1-1 =-29225 lbf
Path 1-1 Thru (1) & (3) = 25694.62 + 35559.52 = 61254.14 lbf
Path 1-1 is stronger than W so it is acceptable per UG-41(b)(2).

Load for path 2-2 lesser of W or W2-2 =-29225 lbf
Path 2-2 Thru (1), (4) = 25694.62 + 29103.09 = 54797.71 lbf
Path 2-2 is stronger than W so it is acceptable per UG-41(b)(2).

Reinforcement Calculations for External Pressure

Limits of reinforcement UG-40

Parallel to the vessel wall d = 7.5 in

Nozzle n5

Normal to the vessel wall outside 2.5*(tn-Cn) + te = .625 in
Normal to the vessel wall inside 2.5*(tn-Cn-C) = .625 in

Nozzle required thickness

L/Do = 4.173/8 = .5216 Do/t = 8/0.02022 = 395.6479
From table G: A = 0.000332
From table HA-3: B = 4394.8

Pa= 4*B/(3*Do/t)
= 4*4394.8/(3*8/0.02022)
= 14.8105 psi

Nozzle required thickness trn = .02022 in

Required thickness tr from UG-37(d)(1) = .1879 in

Area required

Allowable stresses: Sn = 14700, Sv = 14700, psi

fr1 = lesser of 1 or Sn/Sv so fr1 =1
fr2 = lesser of 1 or Sn/Sv so fr2 =1

A = 0.5*(d*tr*F + 2*tn*tr*F*(1 - fr1))
= 0.5*(7.5*0.1879*1 + 2*0.25*0.1879*1*(1 - 1))
= .7046 in^2

Area available

A1 = larger of the following = .466 in^2

= d*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 7.5*(1*0.25-1*0.1879) - 2*0.25*(1*0.25-1*0.1879)*(1-1)
= .466 in^2

= 2*(t+tn)*(E1*t-F*tr) - 2*tn*(E1*t-F*tr)*(1-fr1)
= 2*(0.25+0.25)*(1*0.25-1*0.1879) - 2*0.25*(1*0.25-1*0.1879)*(1-1)
= .062 in^2

A2 = smaller of the following = 0.287 in^2

= 5*(tn - trn)*fr2*t
= 5*(0.25 - 0.02022)*1*0.25
= .287 in^2

= 5*(tn - trn)*fr2*tn
= 5*(0.25 - 0.02022)*1*0.25
= .287 in^2

A41 = Leg^2*fr2
= 0.25^2*1 = .063 in^2

Nozzle n5

Area = A1 + A2 + A41
= 0.466 + 0.287 + 0.063
= .816 in^2

As Area > A the reinforcement is adequate for Pe = 14.7 at 400 Deg F

UG-45 Nozzle Neck Thickness Check

Wall thickness per UG-45(a):	tr1 = 0.02022 in (E = 1)
Wall thickness per UG-45(b)(2):	tr2 = 0.0528 in
Wall thickness per UG-16(b):	tr3 = 0.0625 in
Std pipe wall per UG-45(b)(4):	tr4 = 0.28175 in
The greater of tr2 or tr3:	tr5 = 0.0625 in
The lesser of tr4 or tr5:	tr6 = 0.0625 in

Req'd per UG-45 is the larger of tr1 or tr6 = 0.0625 in

Available nozzle wall thickness new, tn = 0.25 in

The nozzle neck thickness is adequate for Pe.

Nozzle n5

Applied Loads

Radial load	Pr = 649.4 lbf
Circumferential moment	M1 = 0 lbf-ft
Circumferential shear	V2 = 0 lbf
Longitudinal moment	M2 = 0 lbf-ft
Longitudinal shear	V1 = 0 lbf
Torsion moment	Mt = 0 lbf-ft
Internal pressure	P = 0 psi

Stresses at the nozzle OD per WRC bulletin 107 (psi)

Mean dish radius Rm = 60.375 in
U = ro/Sqr(Rm*t) = 1.03

Stress concentration factor Kn (tension) = 1
Stress concentration factor Kb (bending) = 1

Pressure stress intensity factor, Farr equation 11.7

I = 1 + (r/x)^2
= 1 + (3.75/4.25)^2
= 1.779

Local pressure stress = I*P*Rm/2*t = 0 psi

Maximum combined stress =-2802 psi
Allowable combined stress = +-3*S = +- 50100 psi

The maximum combined stress is within allowable limits.

Maximum primary membrane stress =-670 psi
Allowable primary membrane stress = +-1.5*S = +- 25050 psi

The maximum primary membrane stress is within allowable limits.

Nozzle n5

From Fig.	Value read	Au	Al	Bu	Bl	Cu	Cl	Du	Dl
SR-2*	0.0645	-670	-670	-670	-670	-670	-670	-670	-670
SR-2	0.0342	-2132	2132	-2132	2132	-2132	2132	-2132	2132
SR-3*	0.0750								
SR-3	0.0711								
SR-3*	0.0750								
SR-3	0.0711								
pressure stress*									
Total Ox stress		-2802	1462	-2802	1462	-2802	1462	-2802	1462
Primary membrane Ox stress*		-670	-670	-670	-670	-670	-670	-670	-670
SR-2*	0.0200	-208	-208	-208	-208	-208	-208	-208	-208
SR-2	0.0103	-642	642	-642	642	-642	642	-642	642
SR-3*	0.0226								
SR-3	0.0214								
SR-3*	0.0226								
SR-3	0.0214								
pressure stress*									
Total Oy stress		-850	434	-850	434	-850	434	-850	434
Primary membrane Oy stress*		-208	-208	-208	-208	-208	-208	-208	-208
torsion moment Mt									
Shear from V1									
Shear from V2									
Total Shear stress									
Combined stress		-2802	1462	-2802	1462	-2802	1462	-2802	1462

Rev 0
V0V9-1-023

PROCESS SYSTEMS INTERNATIONAL, INC. WESTBOROUGH, MA					ENGINEERING CALCULATIONS	NO: V049-1-024 / PAGE 1 OF 35
REV.	DEO #	DATE	BY:	CHECK	Beam Splitter Chamber - Design of Support Legs and Base Plates	
0	6128	12/29/97	RDC	AGR		
					BY: R.D. CIATTO	DEPT.: 744

PROJECT: LIGO Vacuum Equipment

PROJECT NO: V59049

PURPOSE:. To design BSC support structure to meet requirements of AISC Code for unbalanced vacuum loads.

METHOD: A stiffness analysis of the frame is performed using the IMAGES program.

ASSUMPTIONS:

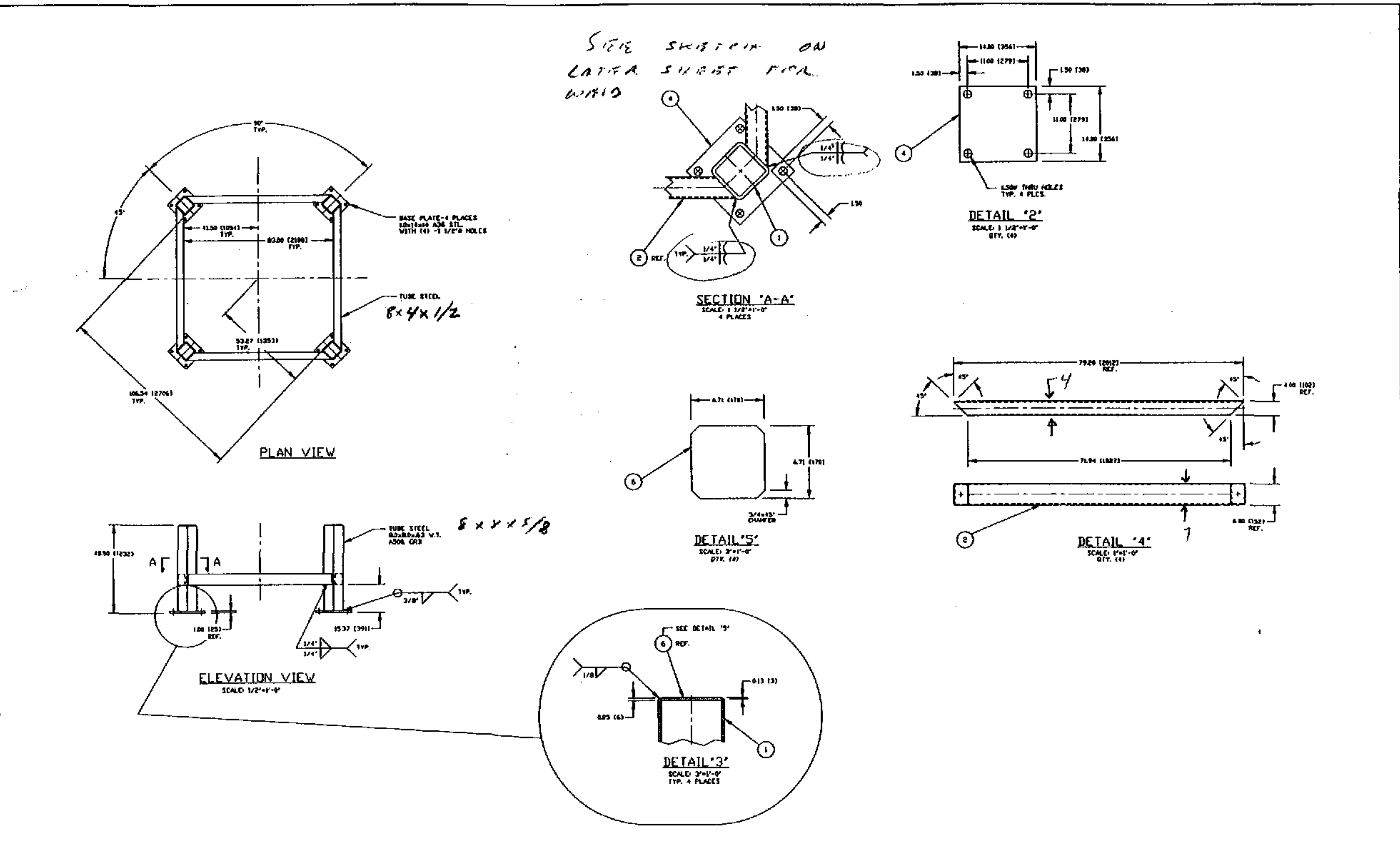
INPUTS: 1. LIGO project design sketches & drawings.
2. Vacuum pressure = 14.7 psi.
3. Seismic acceleration = .05625 g.
4. Forces from V049-1-032

REFERENCES: 1. AISC Code, Ninth Edition, Allowable Stress Design.
2. ASCE 7-88, Minimum Design Loads
3. IMAGES 3D, Version 3.0, R.L. Cloud & Associates
4. Calc. No. V049-1-032 "Component Interface Loads"
5. V049-1-066, LIGO VAC. EQUIP STRUCT. DES. CRITERIA

CALCULATIONS: (SEE ATTACHED)

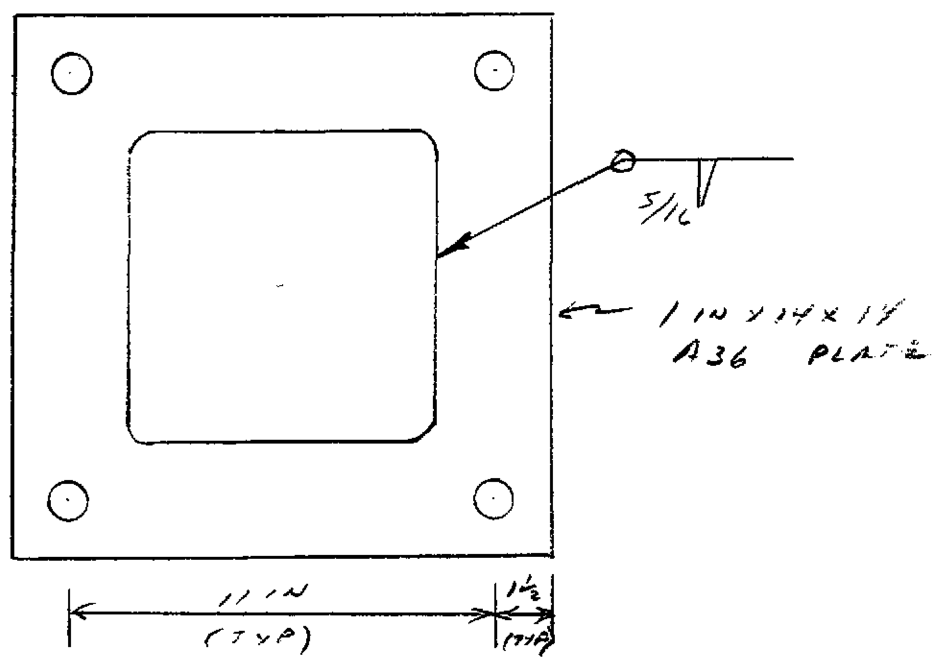
CONCLUSIONS: Support legs and other frame members meet the AISC Code.

NOTES: Computer file is BSCSUPP. ✓



PROCESS SYSTEMS INTERNATIONAL, INC.	
CHAMBER SUPPORT ASSEMBLY BSC LIGO VACUUM EQUIPMENT	
DWG NO.	B049-4-023
REV.	P3
SIZE	D
SCALE	1/2"=1'-0"
SHEET	1 OF 1
P1	PRELIMINARY FOR QUOTES
DO NOT SCALE THIS DRAWING	

BASE PLATE



HILTI HVA ADHESIVE ANCHORS

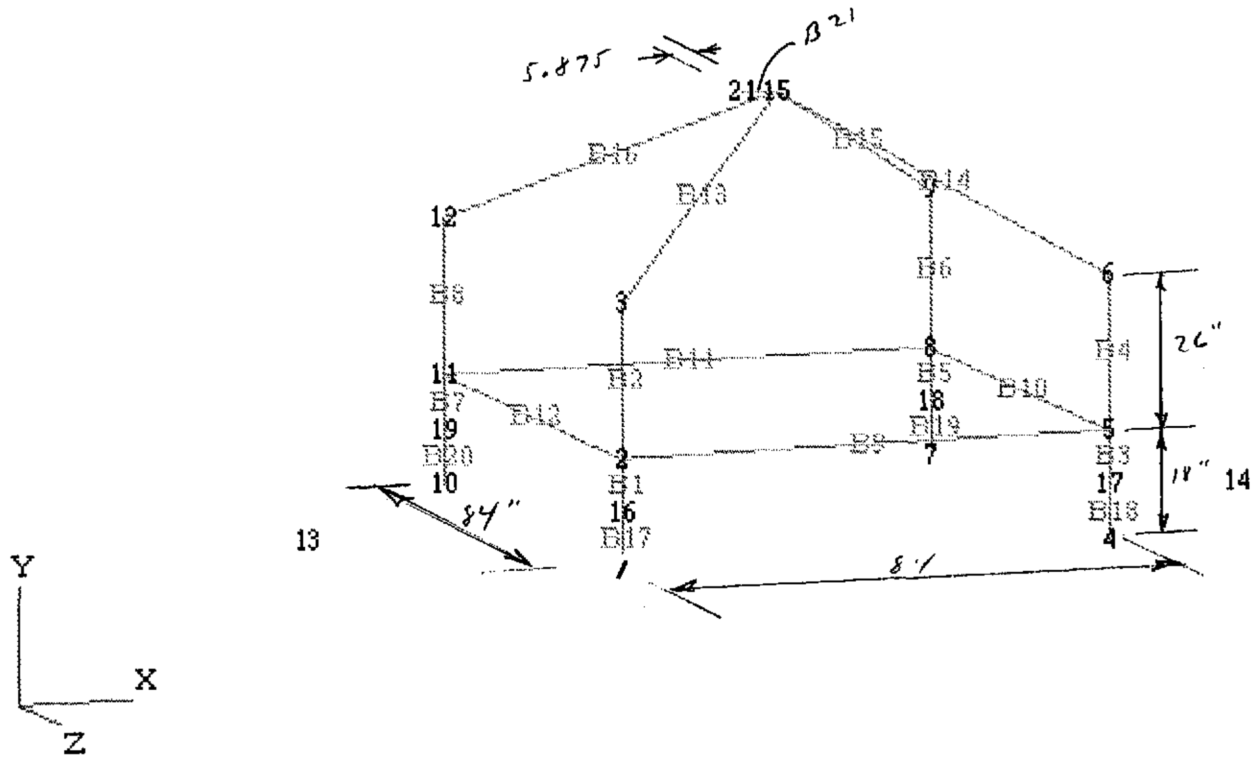
1" Ø , 8 1/4" EMBEDMENT

A307 (OR EQUIV) THREADED ROD

IMAGES-3D
Ver. 3.0
Geometry Plot

Images 3D Model

BSC Support Structure



Beam Splitter Chamber Support Structure
Wireframe Plot

12/28/95
12:58:58

================ I M A G E S - 3 D =================

===

SOLVE BEAM LOADS/STRESSES Version 3.0 12/31/93

Filename=BSCSUPP
Title =Beam Splitter Chamber Support Structure

Load Case 1:MAXIMUM UNBALANCED LOADS COMBINED WITH DEAD WT + EQ(.056G)

BEAM LOADS AND/OR STRESSES

GLoads LLoads Stress	Node Node Node	Fx Axial Axial Corner1	Fy Y-Shear Y-Shear Corner2	Fz Z-Shear Z-Shear Corner3	Mx Torsion Torsion Corner4	My Y-Bending Y-Bending Maximum	Mz Z-Bending Z-Bending Minimum
			BEAM NO. 1				
GLoads	16	-.1840E+05	-.3504E+04	-.1547E+05	.1393E+06	.4803E-11	-.1656E+06
GLoads	2	.1840E+05	.3504E+04	.1547E+05	-.2785E+06	-.4803E-11	.3313E+06
LLoads	16	-.3504E+04	.2395E+05	.2072E+04	.4803E-11	.1865E+05	-.2156E+06
LLoads	2	.3504E+04	-.2395E+05	-.2072E+04	-.4803E-11	-.3730E+05	.4312E+06
Stress	16	.2014E+03	.3201E+04	.2769E+03	-.7064E-13	.4876E+03	.5636E+04
		-.4947E+04	.6325E+04	.5350E+04	-.5923E+04	.6325E+04	-.5923E+04
Stress	2	.2014E+03	.3201E+04	.2769E+03	-.7064E-13	.9752E+03	.1127E+05
		-.1010E+05	.1245E+05	.1050E+05	-.1205E+05	.1245E+05	-.1205E+05
			BEAM NO. 2				
GLoads	2	-.1016E+05	-.2198E+03	-.6706E+04	.4805E+05	.0000E+00	.4180E+05
GLoads	3	.1016E+05	.2198E+03	.6706E+04	-.2224E+06	.0000E+00	.2224E+06
LLoads	2	-.2198E+03	.1193E+05	.2444E+04	.0000E+00	-.6353E+05	-.4415E+04
LLoads	3	.2198E+03	-.1193E+05	-.2444E+04	.0000E+00	.0000E+00	.3145E+06
Stress	2	.1263E+02	.1594E+04	.3265E+03	.0000E+00	-.1661E+04	.1154E+03
		-.1764E+04	-.1533E+04	.1789E+04	.1558E+04	.1789E+04	-.1764E+04
Stress	3	.1263E+02	.1594E+04	.3265E+03	.0000E+00	.0000E+00	.8223E+04
		-.8211E+04	.8236E+04	.8236E+04	-.8211E+04	.8236E+04	-.8211E+04
			BEAM NO. 3				
GLoads	17	-.5751E+04	.3504E+05	.2574E+04	-.2317E+05	.9770E-13	-.5176E+05
GLoads	5	.5751E+04	-.3504E+05	-.2574E+04	.4633E+05	-.9770E-13	.1035E+06
LLoads	17	.3504E+05	-.5886E+04	.2246E+04	.9770E-13	.2022E+05	.5298E+05
LLoads	5	-.3504E+05	.5886E+04	-.2246E+04	-.9770E-13	-.4043E+05	-.1060E+06
Stress	17	-.2014E+04	-.7866E+03	.3002E+03	-.1437E-14	.5285E+03	-.1385E+04
		-.1001E+03	-.2870E+04	-.3927E+04	-.1157E+04	-.1001E+03	-.3927E+04
Stress	5	-.2014E+04	-.7866E+03	.3002E+03	-.1437E-14	.1057E+04	-.2770E+04
		.1813E+04	-.3727E+04	-.5841E+04	-.3006E+03	.1813E+04	-.5841E+04
			BEAM NO. 4				
GLoads	5	-.1399E+05	.2025E+05	-.9445E+04	-.3035E+06	.0000E+00	.3059E+06
GLoads	6	.1399E+05	-.2025E+05	.9445E+04	.5790E+05	.0000E+00	.5790E+05
LLoads	5	.2025E+05	-.3215E+04	.1657E+05	.0000E+00	-.4309E+06	-.1715E+04
LLoads	6	-.2025E+05	.3215E+04	-.1657E+05	.0000E+00	.0000E+00	-.8188E+05
Stress	5	-.1164E+04	-.4297E+03	.2215E+04	.0000E+00	-.1127E+05	.4484E+02
		-.1247E+05	-.1238E+05	.1015E+05	.1006E+05	.1015E+05	-.1247E+05
Stress	6	-.1164E+04	-.4297E+03	.2215E+04	.0000E+00	.0000E+00	-.2141E+04
		.9770E+03	-.3304E+04	-.3304E+04	.9770E+03	.9770E+03	-.3304E+04

LOADS

ESTIMATED WEIGHT

$W = 14000$ LB

SEISMIC FORCE

$F_{EQ} = a W$

$a = .05625\ g$ - REF CALC V049-1-031

$F_{EQ} = .05625(14000)$

$= 788$ LB

UNBALANCED VACUUM LOADS

FROM CALC V049-1-032, BSCs 7 & 8 IN THE WACS HAVE THE MAX UNBALANCED FORCES. LET $F_x = 45460$ LB

$F_z = 29440$ LB

COMINE SEISMIC w/ F_x UNBALANCED LOAD

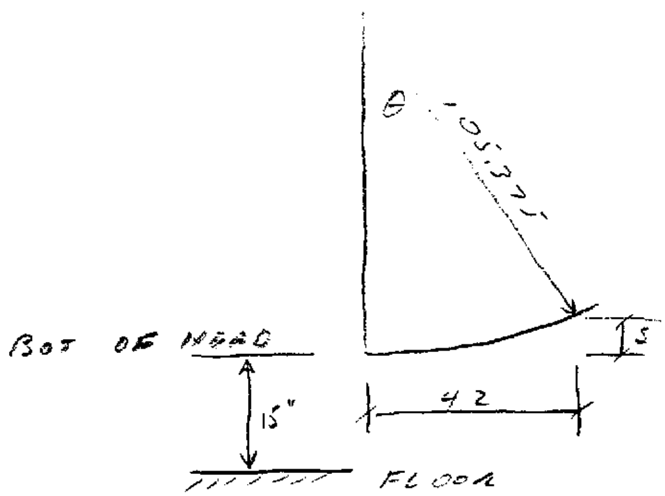
NODE 15: $F_x = 45460 + 788 = 46248$ LB

NODE 15: $F_y = -W = -14000$ LB

NODE 21: $F_z = 29440$ LB

DETERMINE IF CROSS MEMBER WILL CLEAR HEAD

LOWEST PT AT CENTER OF BOTTOM HEAD IS 15 IN ABOVE FLOOR, ℄ OF COLUMN IS 42 IN FROM BEAM TUBE AXIS. CROWN RADIUS OF LOWER HEAD IS 105.375



$$\sin\theta = \frac{42}{105.375}$$

$$\theta = 23.49^\circ$$

$$S = 105.375 - 105.375\cos\theta$$

$$= 8.73 \text{ IN}$$

OR FOR 6 IN DEEP MEMBER

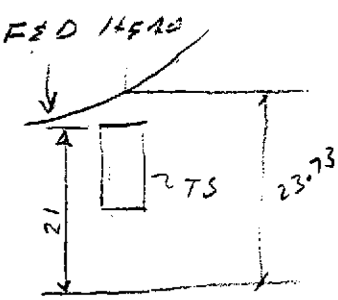
HEIGHT OF CROSS MEMBER ABOVE FLOOR

$$h = 15 + \frac{6}{2}$$

$= 18$ IN TO ℄ OF TS 6x4x1/4

USE TS 7x4x3/8

TOTAL CLEARANCE FROM FLOOR = $15 + 8.73 = 23.73$ IN



MEMBERS

1 TO 8 } TS 8x8x5/8
17 TO 20 }

9 TO 12 TS 8x4x1/2

13 TO 16 DUMMY

PIN TYPE SUPPORTS AT NODES 1, 4, 7, & 10

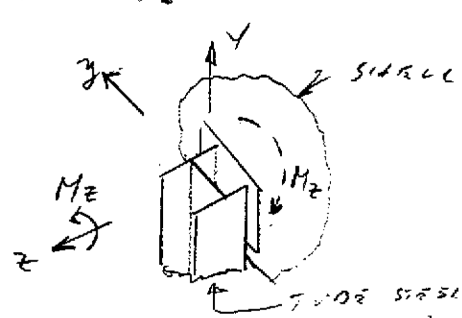
PIN RELEASE AT TOP OF 8x8x5/8 TUBE STL SUPPORT LEG

SINCE THE SHELL IS FLEXIBLE FOR LOCAL BENDING AT SUPPORT LEG ATTACHMENT, THE FOLLOWING LOCAL MOMENTS ARE RELEASED:

M_Y (PIN RELEASE 4 - TORSION IN W8x58)

M_y (" " 5 - WEAK AXIS BENDING IN W8x58)

M_z IS RETAINED SINCE IT TRANSFERS MEMBRANE FORCES INTO SHELL



Y, y, z ARE LOCAL AXES OF SUPPORT LEG

```
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Interactive Microcomputer Analysis & Graphics of Engineering Systems

IMAGES-3D              Version 3.0                12/31/93

Filename=BSCSUPP                              RUN ID=XX11111

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=                        94704                        =
=                                                     =
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=======================================================
```

CHECK GEOMETRY Version 3.0 12/31/93

Filename=BSCSUPP
Title =Beam Splitter Chamber Support Structure

Acceleration of Gravity = 3.864E+02

Force Multiplier	Length Multiplier	Temp. Multiplier
1.0000E+00	1.0000E+00	1.0000E+00

(For database properties only)

MATERIAL PROPERTIES

Mat No.	N a m e
1	STEEL
2	STEEL
3	STEEL

Mat No	Ex Gxy	Ey Gyz	Ez Gzx	nxy axy	nyz ayz	nzx azx	Density
1	2.900E+07	2.900E+07	2.900E+07	3.000E-01	3.000E-01	3.000E-01	2.830E-01
	1.120E+07	1.120E+07	1.120E+07	6.500E-06	6.500E-06	6.500E-06	
2	2.900E+07	2.900E+07	2.900E+07	3.000E-01	3.000E-01	3.000E-01	2.830E-01
	1.120E+07	1.120E+07	1.120E+07	6.500E-06	6.500E-06	6.500E-06	
3	2.900E+07	2.900E+07	2.900E+07	3.000E-01	3.000E-01	3.000E-01	2.830E-01
	1.120E+07	1.120E+07	1.120E+07	6.500E-06	6.500E-06	6.500E-06	

BEAM CROSS SECTION PROPERTIES

Prop No.	N a m e
1	W8X58
2	TB4X4X4
4	TB8X4X8
5	TB8X8X10

Pr. No.	X-Section Area	Moment of Inertia Iy	/ Iz	Torsional Const.- J	Iyz	Shear Shape Fact. SFy	/ SFz
1	1.710E+01	7.510E+01	2.280E+02	3.340E+00	0.000E+00	3.83E+00	1.28E+00
2	3.590E+00	8.220E+00	8.220E+00	1.350E+01	0.000E+00	1.79E+00	1.79E+00
3	1.000E+02	1.000E+03	1.000E+03	2.000E+03	0.000E+00	0.00E+00	0.00E+00
4	1.040E+01	2.460E+01	7.510E+01	6.410E+01	0.000E+00	1.30E+00	2.60E+00
5	1.740E+01	1.530E+02	1.530E+02	2.580E+02	0.000E+00	1.74E+00	1.74E+00