New Technologies Engineering Division

Mechanical Engineering Safety Note

Time Projection Chamber

MESN99-020-OA

April 26, 1999

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Contents

A. Description

This safety note covers the design of time projection chambers (TPC) used in a full volume imaging detector. The chambers are used in building 132N, room 2723. There are three parts to the full volume imaging detector system. The first part is the gas purification subsystem that is used to purify and deliver electronegative free (99.9999999%) gas. This part of the system is being built commercially by Insync Systems. The second part of the system, designed and built at LLNL, includes the time projection chambers (TPC) where the experiments will be performed. Gas, from the purification panel, feeds the TPC's that will nominally operate at 300 psig but are being designed for 350 psig maximum operating pressure (MOP). It will be necessary to work around the TPC's with radioactive sealed sources for testing and calibration; thus this is a manned operation. The third part of the system uses cylinders to reclaim the purified gas. These cylinders have been fabricated by ACME CRYOGENICS INC. and are rated by them at 3000 psig MAWP. Gas will be transferred in the TPC system by thermal cycles, using LN2 to create the temperature gradient inside the chamber via conduction through the walls of the cryogenic thimble. A certain percentage of alcohol may be used in the LN2 bath to move the temperature of the bath above 73K.

The TPC's are the experimental chambers designed at LLNL. These chambers are used for two purposes but were mechanically designed to be identical. The first chamber will be used as an ionization chamber where electron drift will be used as a measure of gas purity. The second chamber is the actual TPC itself, which is used for position sensitive readout of electron clouds and hence gamma ray imaging. Figure 1 depicts a TPC with its associated hardware. In the experimental setup, the chambers are connected together with high pressure tubing. The chambers have been designed to allow a 400 keV gamma ray to penetrate the chamber wall in well-defined places, specifically in the center of the 2 3/4 inch conflat flange and in a linear series of VCR blanks on the side of the chamber. It will be necessary to use radioactive sources in conjunction with these windows to probe the capabilities of the chamber. The 1 3/4 inch conflat flanges has been outfitted with a high voltage (20 kV) ceramic feedthrough from Ceramaseal. Many of the penetrations into the chamber and the internals of the chamber are attached to the conflat gasketed chamber head to allow easy removal from the chamber body. The chambers will be filled with a gas (Ar, Xe, along with at least one the following: CH_4, CO_2, and P10) using the 135psi gas purification system and then condensed by cooling the chamber using a cryogenic thimble. LN2 will envelope the outside of the thimble creating non-uniform thermal stresses along with membrane stress throughout the vessel.

This ME Safety Note is required because the TPC of the system contains compressed gas at pressures exceeding 150 psig or 100kJ of stored energy. This Safety Note covers the vessel depicted in Figure 1 up to and including the output connections. If required, a separate safety note will cover the remaining parts of full volume imaging detector system less the TCP's.

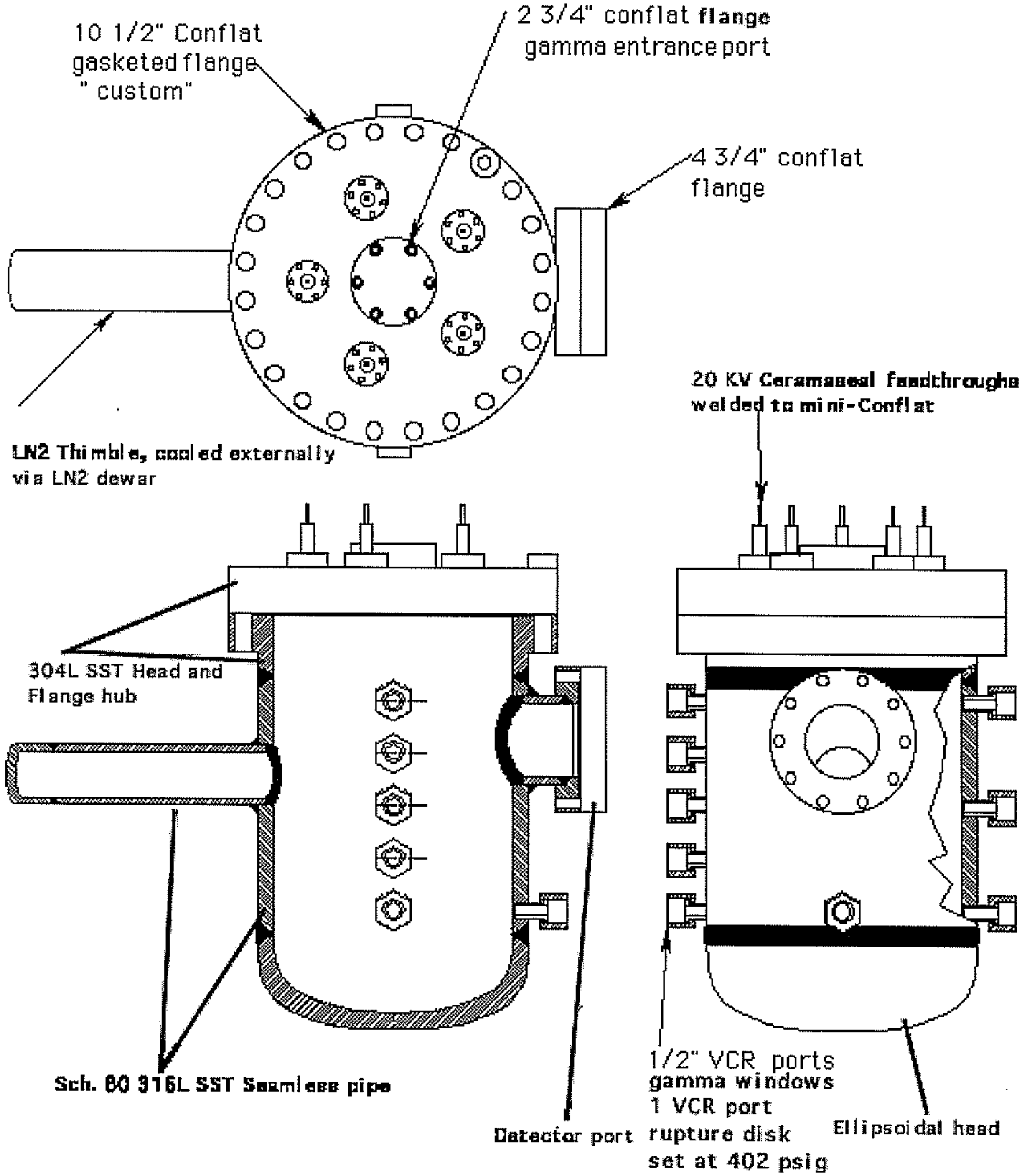

Figure 1 – Diagram of the Time Projection Chambers (TPC)

B. Operational Hazards

Associated hazards are those typical of any high pressure gas system. Failure of a vessel or component could result in either shrapnel or a blast overpressure to the body. Since the gases involved are not air, there is also the potential concern of asphyxiation. Other hazards include physical exposure to the radioactive sealed source and cold temperatures. The hazards other than those associated with the pressure vessel will be addressed by the FSP (if applicable) or separate OSP for this experiment.

C. Procedures

Design safety factors are robust for all intended pressures. The system is adequately protected by a pressure relief device at a VCR port so that components cannot be over-pressurized. This document also specifies shielding requirements for personnel protection from shrapnel in the event of an accident. However, an OSP for this experiment will address associated interlocks and operational steps required during pressurization.

D. Calculations

The following will certify the TPC for this system:

[1] Hardware and Fabrication
The vessel is fabricated using commercially purchased metals. Fabrication and joining techniques are also standard technology. Welding was performed by LLNL ASME certified welders experienced in pressure systems.

[2] Engineered Design
The system design has relief devices at strategic locations (a VCR fitting) to insure that the MAWP's are never exceeded.

An evaluation of high risk pressure components indicated that a Ceramaseal feed-through may fail if improperly handled. Specifically, the weld joint at the Conflat is susceptible to bending and fracture. To minimize this risk, a fragment deflector/stop fixture was designed and will be mounted in front of the head where the Ceramaseal is mounted. A Kevlar drape will also be employed if this device fails to capture all fragments. This stop and Kevlar drape will be interlocked during pressure vessel operation.

[3] Testing
Detailed proof testing procedures at 1.5 times MAWP and at the working temperature, induced by LN2 cooling, have been developed and are enclosed as Appendix A. Successful completion of these procedures by a LLNL pressure inspector will complete the certification of the TPC's. Proof testing is the crux of pressure vessel qualification for fracture critical components and is best stated from literature[5] as follows:

"The critical flaw size associated with proof test conditions can also be used for life expectance considerations. Specifically, if a pressure vessel survives a given proof test it can be concluded that the largest defect present in the structure is smaller than the critical flaw size at the proof test conditions. Therefore, in the absence of non-destructive inspection, this flaw size can be considered the existing flaw size at the beginning of life at the operating conditions and would, in turn, serve as the basis for further crack growth consideration"[5] (also see fracture analysis below).

The vessel has been designed to meet ASME Boiler and Pressure Vessel Code design guidelines. Stresses are low enough to eliminate the need for impact testing of the material in the heat effected zones created by the butt welds, UHA-51 (g) (see misc.nb calculations in Appendix C). The ASME Code also exempt austenitic, chromium-nickel stainless steels from impact testing, UHA-51(d)(1)(a). Thus, the base materials 304L and 316L are exempt.

[4] Calculations

Most calculations were done using ASME Pressure Vessel Code, Section VIII, Division 1 guidelines. The TPC has a MAWP of 978 psig when using the C-Ring type head (no openings) and 402 psig for the Conflat type head(s) (with and without openings). A future addendum to this safety note will cover a head (with openings) to be used at 978 psig MAWP. The allowable stresses used in all calculations are based on values found in the ASME Pressure Vessel Code, Section II. *For both 316L and 304L the allowable stress is 16,700 psi which provides a* ***nominal Safety Factor of ~5*** *in all Pressure Vessel Code calculations (i.e., head thickness, maximum vessel pressure, minimum wall thickness, etc).* The following tables are summaries of the detailed calculations found in Appendix A.

Vessel

The energy in each pressure vessel was calculated to be 55, 852 ft-lb. or 16.4 g TNT at the MAWP of 978 psig. The following table summarizes the analytical results for the main 8 inch schedule 80 pressure vessel, the detector pipe, the VCR "Cajon" fittings/ pipes, and the LN_2 pipe connected to the main vessel. All tubing is 316L. Calculations were made at a MAWP of 978 psig. The last column refers to the ratio of yield stress (37ksi) to Von Mises stress at the test pressure of 1.5 x MAWP. Values must be greater than 1.0 for a safe proof test.

	S1 (psi)	S2 (psi)	S3 (psi)	Von Mises (psi)	Required wall thickness (in)	Actual wall thickness (in)	If ≥ 1.0 stress less than yield for 1.5xMAWP
Main 8" vessel	3499	7976	-978	7755	0.336	0.500	3.2
Detector pipe 2.87 OD	1839	4657	-978	4880	0.102	0.275	5.1

VCR pipe 0.5" OD	1739	4455	-978	4705	0.012	0.050	5.2
LN2 pipe 1.9" OD	1618	4214	-978	4496	0.066	0.200	5.5

Analytical results for welds, area reinforcement, and their related loads that attach the detector pipe, LN_2 pipe, and VCR pipe to the main vessel shell are detailed in the table below at a MAWP of 978 psig. Generally, if the nozzle and fillet weld load paths are greater than the total weld load, then the strengths are sufficient. The total weld load (W~(Area required – Area available)*Allowable stress)) for the VCR pipe is less than 0 because the vessel wall is 0.160" thicker than required creating much more area available than required. Thus, the area available is greater than the area removed and a negative number results.

	Area of mat'l. required (in^2)	Area of mat'l. avail. (in^2)	Total weld load (lb)	Nozzle wall load path (lb)	Fillet weld load path (lb)
Detector pipe	0.780	0.800	8172	13172	12749
LN_2 pipe	0.504	0.508	5396	6243	6106
VCR pipe	0.134	0.194	< 0	413	402

The butt welds connecting the hub to the main vessel and the ellipsiodal head to the main vessel, the ellipsiodal head on liquid nitrogen pipe, and the hub to the detector pipe, reduced the allowable working pressure in the vessel they are connected to by 'E' (butt weld efficiency). An 'E' of 0.7 was used for these welds which reduced their associated allowable working pressures to 1421 psig, 6979 psig, and 6139 psig for the of the main vessel, LN pipe, and detector pipe respectively. Again, all of these calculated pressures use an allowable stress of 16,700 psi which has a nominal SF = 5.0 so an additional SF of 1.5 (1421 / 978) is obtained. Using a butt weld efficiency of 0.7 allows no radiography to be performed on the welds according to the ASME Boiler and Pressure Codes.

The VCR, LN_2, and detector port openings in the vessel shell are mounted 90° to each other. The radial distance between hole centers is approximately 6.0 inches. ASME Boiler Code requires that all openings be less than the sum of their respective diameters. The maximum sum of the diameters is 3.37 inches between LN_2 and the detector port.

Holes that do not penetrate the vessel shell may be required to horizontally mount the vessel. The depth of tapped 1/4-20 holes and 3/8-16 holes shall be ≤ 0.25 inches. Holes can not be placed near other openings or reinforcements.

Head / Flange Calculations

The following table summarizes the analytical results for the integral flange butt welded to the main access port and the small flange butt welded on the side of the vessel (detector port). Again, the allowable stress in 16,700 psi for the base material. Also, the ASME allowable hub stress is 1.5 time the allowable stress.

Flange	MOP	Longitudinal hub stress (psi)	Radial flange stress (psi)	Tangential flange stress (psi)
Main 10.5" OD	850	16973	6327	4173
Main 10.5" OD	350	6388	2381	1571
Detector 4.625" OD	850	13998	2794	7367

The head for operating at 850 psig, uses a C-Ring type metal seal and is made from 304L stainless steel. The (24) required bolts for this flange are Unbrako KS 1216 1/2"-13 SHCS with a tensile strength of 160,000 psi (or 304 Stainless Steel with a 81 ksi tensile strength). The main flange for operating at 350 psig is a Conflat (CF) type (304L), sealed with a soft copper flat gasket to a knife edge. The (24) required bolts for this flange are Unbrako KS 1216 1/2"-13 SHCS with a tensile strength of 160,000 psi (or 304 Stainless Steel with a 81 ksi tensile strength). All other CF flanges (1 1/3 and 2 3/4 inch) shall be bolted to the 350 MOP head using Unbrako KS 1216 psi (or 304 Stainless Steel with a 81 ksi tensile strength), 8-32 or 1/4-28 SHCS as required.

The smaller 4 5/8" CF type flange for the detector port requires 10 bolts, Unbrako KS 1216 5/16"-24 SHCS with a tensile strength of 160,000 psi (or 304 Stainless Steel with a 81 ksi tensile strength) and is made from 304L stainless steel. The following table summarizes the fastener calculations.

Flange	MOP	No. of bolts	Bolt	Torque (in-lb)	Flange design bolt load, operating. (lb)	Flange design bolt load, gasket seal. (lb)	Max. allowable bolt load (SF 4 applied)
Main, C-ring 10.5" OD	850	24	1/2-13	1140	58850	96913	134976
Main, CF 10.5" OD	350	24	1/2-13	1140	24677	79827	134976
Detector 4.625" OD	850	10	5/16-24	347	10144	15452	20760
1 1/3" CF	350	6	#8-32	51	231	230	2936
2 3/4" CF	350	6	1/4-28	152	1015	527	7937

Analytical results for the commercially purchased SA316 ellipsiodal head on main vessel, nominal wall thickness 0.5 inches and the SA316 ellipsiodal head on liquid nitrogen pipe, nominal wall thickness 0.2 inches follows. The head thickness calculations were done at the MAWP of 978 psig and allow for strength reduction due to the butt weld connecting them to the vessel.

	MOP	Max. pressure (psig)	Required head thickness (in)	Actual head thickness (in)	If ≥ 1.0 stress less than yield for 1.5x MAWP
Main vessel	850	1513	0.320	0.500	1.03
LN_2 pipe	850	3036	0.063	0.200	2.07

Results of the unstayed flat heads are presented in the table below. Two head types are planned for the main vessel, one CF type for low pressures at 350 MOP that has instrumentation ports, and one C-Ring type for high pressure (850 MOP) for vessel pressure testing and to be modified for a future head design (and subsequently proof tested along with a Safety Note Addendum). Stress concentration factors for the circular holes in a plate with internal pressure were used from empirical data in Wiley[2]. Although not a perfectly matching model to Wiley, the concentration factors used are conservative. The stress concentration factor (2.278) reduced the allowable stress to 7,331 psi from 16,700 psi. Hole reinforcement requirements were also calculated using the ASME Codes. These results confirmed the thickness requirements using Wiley stress concentration factors.

Results of two types of Conflat feedthrough heads mounted to the 10.5 inch CF flange are also presented below. All head thickness calculations use the ASME head equation involving bending with the exception of the 2 3/4 inch CF where both bending and no bending cases were used. This flange was bored out to leave a head depth of 0.125" by 1.5" in diameter. The flange thickness around its mounting holes and under its knife edge remains at the nominal flange thickness of 0.5 inches. Thus, calculations were made for both and summarized below. A minimum thickness for the 1/2 inch VCR plug is calculated. The pressure side of a VCR plug is bored out 1/4 inch in diameter to this minimum thickness to be used as a gamma port.

Flange type	MOP	Required head thickness (in)	Actual head thickness (in)	Required hub thickness (in)	Actual hub thickness (in)
Conflat flange, Cu seal AAA99-104240	350	1.261	1.5	0.624	1.250
C-Ring type metal seal. AAA99-104243	850	1.247	1.980	0.661	1.250
Conflat flange, 4 5/8" Ø, x 0.750" thick. Commercial product	850	0.613	0.750	0.423	0.810

1 1/3" CF	350	0.100	0.300	N/A	N/A
2 3/4" CF	350	0.178 / 0.092	0.5 / 0.125	N/A	N/A
VCR plug	850	0.052	0.052	N/A	N/A

Blind holes in the unstayed flat head were analyzed on the basis of area replacement. If the actual cross-sectional area available was greater than the cross-sectional area required, reinforcement was not required. The following table summarizes the results for the 350 MOP flat head. These calculations can also apply to blind mounting holes of the same dimension for mounting and handling the head with the caveat that hole can not be placed near other openings or reinforcements.

Hole type:	Area available (in^2)	Area required (in^2)
8-32 mini conflat holes	0.195	0.051
8-32 mounting bracket holes (internal)	0.205	0.041
1/4-28 medium conflat holes	0.250	0.125

Conflat (CF) flanges are used as connecting members and instrumentation feedthroughs in this pressure vessel design. Five 1 1/3 inch on a 5.5 inch bolt circle pattern and one 2 3/4 inch centrally located CF flanges are used on the 350 MOP head. A 4 5/8 inch CF flange is used on the detector port (850 MOP).

CF flanges were pressure tested in 1992 under the safety note END 92-072. The 1 1/3 inch nominally sized CF flanges with stainless steel bolts started leaking at ~15,000 psi. The 4 5/8 inch CF flange had no leakage with water as the pressure medium up to 1200 psi and minor (10^{-6} Torr-L/s) leaking with helium from 500 psi to 930 psi. All tests were done <u>without</u> catastrophic failure. Leakage occurred around the copper seal. A blank 2 3/4 inch was not proof tested.

For operation, the mating 1 1/3 inch CF flange to the CF port on the 350 MOP head has a high voltage feedthrough that is not rated by the manufacturer (Ceramaseal) because it is a special order. The manufacturer welded the high voltage feedthrough to an opening in the flange. LLNL has proof tested this component to burst (5850 psi). There is a concern for brittle fracture or weld failure due to cracking by mishandling that is addressed in the Fragment Hazard Mitigation paragraph below. The mating 2 3/4 inch CF flange will be proof tested at 604 psig along with the rest of the head. The mating 4 5/8 inch CF flange will be blanked off for pressure testing and initial operational tests. An addendum to this note will follow at a later date to address the attachment method of the detector to the mating flange. It will then be proof tested at 1467 psig.

Fracture Critical Components

This vessel is considered a Category IV risk according to MEDSS. Its failure has the potential for moderate injury and material testing is recommended.

The material used in this vessel is standard ASTM 304L and 316L stainless steel. Material testing was not done for the following reasons:

(1) SA316L and SA304L are standard materials with strict manufacturing requirements.
(2) ASME Boiler and Pressure Vessel Code does not require testing for austenitic stainless steels.
(3) the large critical crack depths (a_{cr}) and lengths calculated using conservative stress intensity factors (K_{Ic}) from literature.
(4) the number of cycles to failure were > 10^5; far larger than the ≤ 10^2 cycles expected using crack growth rates[7] from literature.
(5) The leak-before-break criterion is satisfied by a factor of ~10 or greater (136740 / 13824). Also, the CF type flanges used in the TPC design practically guarantee a leak before failure as demonstrated by earlier proof testing.
(6) 316 and 304 stainless steel both have excellent toughness properties at cryogenic temperatures. Sharpy V-notch impact test data[6,8] on 304 stainless steel indicates a slightly lowered toughness from room temperature to –196°C (150 to 124 ft-lb). For 316, the toughness lowered 13% from 141 to 122 ft-lb.

The table below summarizes the fracture toughness calculations in Appendix C.

	K_{Ic} (psi in^1/2)	K_I (psi in^1/2)	a_{cr} surface flaw (in)	a_{cr} sub-surface flaw (in)	2c length of surface flaw (in)	2c length of sub-surface flaw (in)
Main vessel	136740	10115	77.3	93.6	309.3	374.2
Ellipsoidal head	136740	13824	42.2	51.1	168.9	204.4
Flat head	136740	6528	183.1	221.6	732.5	886.3

The Unbrako bolts recommended above in the Head / Flange Calculations section are rated at their maximum tensile strength at –400°F. The alternative, 304 stainless steel fasteners have the same safe fracture critical properties as the vessel. No fracture critical calculations were performed for fasteners.

A proof test at 1.5xMAWP and at the working cryogenic temperature is planned for this vessel. Proof testing is the crux of pressure vessel qualification and is best stated from literature[5] as follows:

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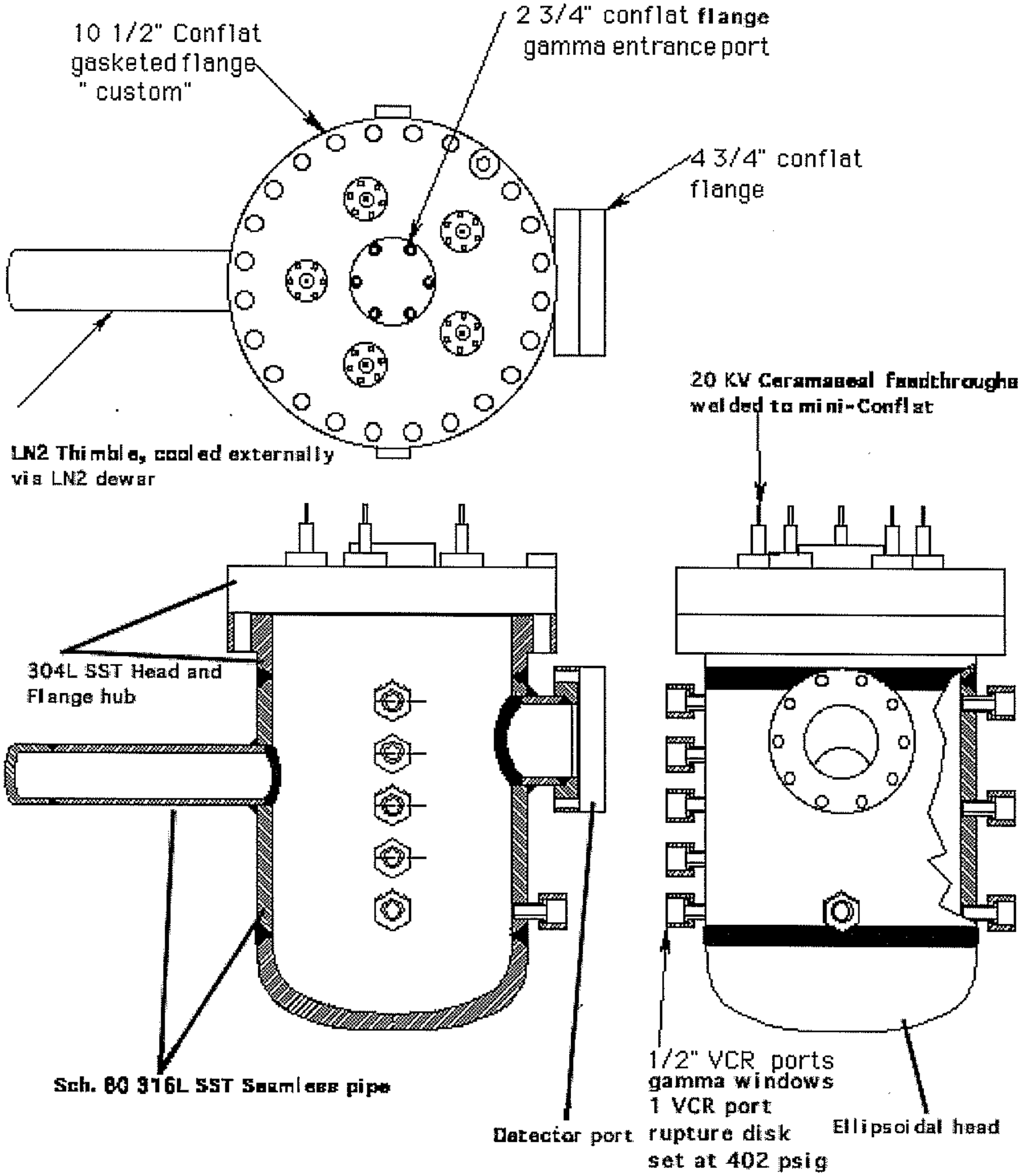

Figure 1 – Diagram of the Time Projection Chambers (TPC)

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The vessel has been designed to meet ASME Boiler and Pressure Vessel Code design guidelines. Stresses are low enough to eliminate the need for impact testing of the material in the heat effected zones created by the butt welds, UHA-51 (g) (see misc.nb calculations in Appendix C). The ASME Code also exempt austenitic, chromium-nickel stainless steels from impact testing, UHA-51(d)(1)(a). Thus, the base materials 304L and 316L are exempt.

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LN_2 pipe	0.504	0.508	5396	6243	6106
VCR pipe	0.134	0.194	< 0	413	402

The butt welds connecting the hub to the main vessel and the ellipsiodal head to the main vessel, the ellipsiodal head on liquid nitrogen pipe, and the hub to the detector pipe, reduced the allowable working pressure in the vessel they are connected to by 'E' (butt weld efficiency). An 'E' of 0.7 was used for these welds which reduced their associated allowable working pressures to 1421 psig, 6979 psig, and 6139 psig for the of the main vessel, LN pipe, and detector pipe respectively. Again, all of these calculated pressures use an allowable stress of 16,700 psi which has a nominal SF = 5.0 so an additional SF of 1.5 (1421 / 978) is obtained. Using a butt weld efficiency of 0.7 allows no radiography to be performed on the welds according to the ASME Boiler and Pressure Codes.

The VCR, LN_2, and detector port openings in the vessel shell are mounted 90° to each other. The radial distance between hole centers is approximately 6.0 inches. ASME Boiler Code requires that all openings be less than the sum of their respective diameters. The maximum sum of the diameters is 3.37 inches between LN_2 and the detector port.

Holes that do not penetrate the vessel shell may be required to horizontally mount the vessel. The depth of tapped 1/4-20 holes and 3/8-16 holes shall be ≤ 0.25 inches. Holes can not be placed near other openings or reinforcements.

Head / Flange Calculations

The following table summarizes the analytical results for the integral flange butt welded to the main access port and the small flange butt welded on the side of the vessel (detector port). Again, the allowable stress in 16,700 psi for the base material. Also, the ASME allowable hub stress is 1.5 time the allowable stress.

Flange	MOP	Longitudinal hub stress (psi)	Radial flange stress (psi)	Tangential flange stress (psi)
Main 10.5" OD	850	16973	6327	4173
Main 10.5" OD	350	6388	2381	1571
Detector 4.625" OD	850	13998	2794	7367

The head for operating at 850 psig, uses a C-Ring type metal seal and is made from 304L stainless steel. The (24) required bolts for this flange are Unbrako KS 1216 1/2"-13 SHCS with a tensile strength of 160,000 psi (or 304 Stainless Steel with a 81 ksi tensile strength). The main flange for operating at 350 psig is a Conflat (CF) type (304L), sealed with a soft copper flat gasket to a knife edge. The (24) required bolts for this flange are Unbrako KS 1216 1/2"-13 SHCS with a tensile strength of 160,000 psi (or 304 Stainless Steel with a 81 ksi tensile strength). All other CF flanges (1 1/3 and 2 3/4 inch) shall be bolted to the 350 MOP head using Unbrako KS 1216 psi (or 304 Stainless Steel with a 81 ksi tensile strength), 8-32 or 1/4-28 SHCS as required.

The smaller 4 5/8" CF type flange for the detector port requires 10 bolts, Unbrako KS 1216 5/16"-24 SHCS with a tensile strength of 160,000 psi (or 304 Stainless Steel with a 81 ksi tensile strength) and is made from 304L stainless steel. The following table summarizes the fastener calculations.

Flange	MOP	No. of bolts	Bolt	Torque (in-lb)	Flange design bolt load, operating. (lb)	Flange design bolt load, gasket seal. (lb)	Max. allowable bolt load (SF 4 applied)
Main, C-ring 10.5" OD	850	24	1/2-13	1140	58850	96913	134976
Main, CF 10.5" OD	350	24	1/2-13	1140	24677	79827	134976
Detector 4.625" OD	850	10	5/16-24	347	10144	15452	20760
1 1/3" CF	350	6	#8-32	51	231	230	2936
2 3/4" CF	350	6	1/4-28	152	1015	527	7937

Analytical results for the commercially purchased SA316 ellipsiodal head on main vessel, nominal wall thickness 0.5 inches and the SA316 ellipsiodal head on liquid nitrogen pipe, nominal wall thickness 0.2 inches follows. The head thickness calculations were done at the MAWP of 978 psig and allow for strength reduction due to the butt weld connecting them to the vessel.

	MOP	Max. pressure (psig)	Required head thickness (in)	Actual head thickness (in)	If ≥ 1.0 stress less than yield for 1.5x MAWP
Main vessel	850	1513	0.320	0.500	1.03
LN_2 pipe	850	3036	0.063	0.200	2.07

Results of the unstayed flat heads are presented in the table below. Two head types are planned for the main vessel, one CF type for low pressures at 350 MOP that has instrumentation ports, and one C-Ring type for high pressure (850 MOP) for vessel pressure testing and to be modified for a future head design (and subsequently proof tested along with a Safety Note Addendum). Stress concentration factors for the circular holes in a plate with internal pressure were used from empirical data in Wiley[2]. Although not a perfectly matching model to Wiley, the concentration factors used are conservative. The stress concentration factor (2.278) reduced the allowable stress to 7,331 psi from 16,700 psi. Hole reinforcement requirements were also calculated using the ASME Codes. These results confirmed the thickness requirements using Wiley stress concentration factors.

Results of two types of Conflat feedthrough heads mounted to the 10.5 inch CF flange are also presented below. All head thickness calculations use the ASME head equation involving bending with the exception of the 2 3/4 inch CF where both bending and no bending cases were used. This flange was bored out to leave a head depth of 0.125" by 1.5" in diameter. The flange thickness around its mounting holes and under its knife edge remains at the nominal flange thickness of 0.5 inches. Thus, calculations were made for both and summarized below. A minimum thickness for the 1/2 inch VCR plug is calculated. The pressure side of a VCR plug is bored out 1/4 inch in diameter to this minimum thickness to be used as a gamma port.

Flange type	MOP	Required head thickness (in)	Actual head thickness (in)	Required hub thickness (in)	Actual hub thickness (in)
Conflat flange, Cu seal AAA99-104240	350	1.261	1.5	0.624	1.250
C-Ring type metal seal. AAA99-104243	850	1.247	1.980	0.661	1.250
Conflat flange, 4 5/8" Ø, x 0.750" thick. Commercial product	850	0.613	0.750	0.423	0.810

1 1/3" CF	350	0.100	0.300	N/A	N/A
2 3/4" CF	350	0.178 / 0.092	0.5 / 0.125	N/A	N/A
VCR plug	850	0.052	0.052	N/A	N/A

Blind holes in the unstayed flat head were analyzed on the basis of area replacement. If the actual cross-sectional area available was greater than the cross-sectional area required, reinforcement was not required. The following table summarizes the results for the 350 MOP flat head. These calculations can also apply to blind mounting holes of the same dimension for mounting and handling the head with the caveat that hole can not be placed near other openings or reinforcements.

Hole type:	Area available (in^2)	Area required (in^2)
8-32 mini conflat holes	0.195	0.051
8-32 mounting bracket holes (internal)	0.205	0.041
1/4-28 medium conflat holes	0.250	0.125

Conflat (CF) flanges are used as connecting members and instrumentation feedthroughs in this pressure vessel design. Five 1 1/3 inch on a 5.5 inch bolt circle pattern and one 2 3/4 inch centrally located CF flanges are used on the 350 MOP head. A 4 5/8 inch CF flange is used on the detector port (850 MOP).

CF flanges were pressure tested in 1992 under the safety note END 92-072. The 1 1/3 inch nominally sized CF flanges with stainless steel bolts started leaking at ~15,000 psi. The 4 5/8 inch CF flange had no leakage with water as the pressure medium up to 1200 psi and minor (10^{-6} Torr-L/s) leaking with helium from 500 psi to 930 psi. All tests were done without catastrophic failure. Leakage occurred around the copper seal. A blank 2 3/4 inch was not proof tested.

For operation, the mating 1 1/3 inch CF flange to the CF port on the 350 MOP head has a high voltage feedthrough that is not rated by the manufacturer (Ceramaseal) because it is a special order. The manufacturer welded the high voltage feedthrough to an opening in the flange. LLNL has proof tested this component to burst (5850 psi). There is a concern for brittle fracture or weld failure due to cracking by mishandling that is addressed in the Fragment Hazard Mitigation paragraph below. The mating 2 3/4 inch CF flange will be proof tested at 604 psig along with the rest of the head. The mating 4 5/8 inch CF flange will be blanked off for pressure testing and initial operational tests. An addendum to this note will follow at a later date to address the attachment method of the detector to the mating flange. It will then be proof tested at 1467 psig.

Fracture Critical Components

This vessel is considered a Category IV risk according to MEDSS. Its failure has the potential for moderate injury and material testing is recommended.

The material used in this vessel is standard ASTM 304L and 316L stainless steel. Material testing was not done for the following reasons:

(1) SA316L and SA304L are standard materials with strict manufacturing requirements.
(2) ASME Boiler and Pressure Vessel Code does not require testing for austenitic stainless steels.
(3) the large critical crack depths (a_{cr}) and lengths calculated using conservative stress intensity factors (K_{Ic}) from literature.
(4) the number of cycles to failure were > 10^5; far larger than the ≤ 10^2 cycles expected using crack growth rates[7] from literature.
(5) The leak-before-break criterion is satisfied by a factor of ~10 or greater (136740 / 13824). Also, the CF type flanges used in the TPC design practically guarantee a leak before failure as demonstrated by earlier proof testing.
(6) 316 and 304 stainless steel both have excellent toughness properties at cryogenic temperatures. Sharpy V-notch impact test data[6,8] on 304 stainless steel indicates a slightly lowered toughness from room temperature to –196°C (150 to 124 ft-lb). For 316, the toughness lowered 13% from 141 to 122 ft-lb.

The table below summarizes the fracture toughness calculations in Appendix C.

	K_{Ic} (psi in^1/2)	K_I (psi in^1/2)	a_{cr} surface flaw (in)	a_{cr} sub-surface flaw (in)	2c length of surface flaw (in)	2c length of sub-surface flaw (in)
Main vessel	136740	10115	77.3	93.6	309.3	374.2
Ellipsoidal head	136740	13824	42.2	51.1	168.9	204.4
Flat head	136740	6528	183.1	221.6	732.5	886.3

The Unbrako bolts recommended above in the Head / Flange Calculations section are rated at their maximum tensile strength at –400°F. The alternative, 304 stainless steel fasteners have the same safe fracture critical properties as the vessel. No fracture critical calculations were performed for fasteners.

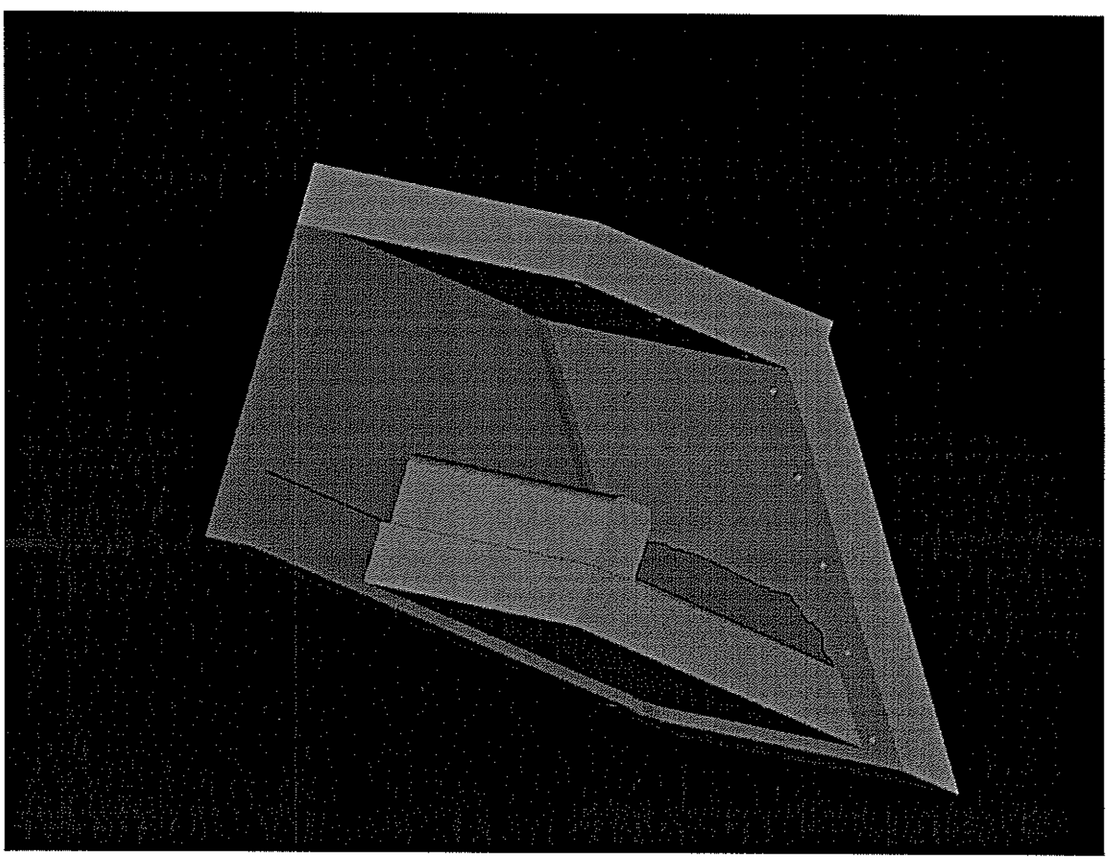
A proof test at 1.5xMAWP and at the working cryogenic temperature is planned for this vessel. Proof testing is the crux of pressure vessel qualification and is best stated from literature[5] as follows:

"The critical flaw size associated with proof test conditions can also be used for life expectance considerations. Specifically, if a pressure vessel survives a given proof test it can be concluded that the largest defect present in the structure is smaller than the critical flaw size at the proof test conditions. Therefore, in the absence of non-destructive inspection, this flaw size can be considered the existing flaw size at the beginning of life at the operating conditions and would, in turn, serve as the basis for further crack growth consideration".

A physical inspection of the TPC for cracks is required between every experiment or experimental cycle. Careful handling of the head, vessel and its related hardware is important so that the welds attaching the various components (high voltage feedthroughs, VCR stubs) are not damaged. If any of these components are bent by mishandling, the suspect welds must be radiographicly inspected and re-proof tested.

Fragment Hazard Mitigation

A fragment deflector/stop was designed to deflect and capture a potential Ceramaseal feedthrough mishap if it were propelled from head of the vessel. It will be placed as close as practical to the TPC head and still allow operation of the vessel. The basic design is based on ballistic gun range technology where the fragment is deflected from a 45° wall into a sand trap (red arrow shows path in Figure below). All walls are made from 2.5" thick lexan that can stop the projectile if it were propelled normal into it. The opening in the stop (11" x 14") is sufficiently oversized to the Ceramaseal bolt circle diameter (5.5") and the sand trap baffle is made from 1/4" lexan to allow fragment passage. This stop will be interlocked during vessel operation. Calculation filename "fragmant.nb" in Appendix C details the shielding calculations obtained from MEDSS.



A Kevlar drape will also be employed to shield the operator from a potential stray fragment reflected back out of the catch.

The system pressure requirements are summarized as follows:

Component	Maximum Operating Pressure (psig)	Maximum Allowable Working Pressure (MAWP) (psig)	Pressure Relief Setting (psig)	Proof Test Pressure (psig)
Main Pressure Vessel (sketch) AAA99-104242 (weld flange)	850	978	978	1467
Flat Head, Metal C-Ring: AAA99-104243	850	978	978	1467
Conlfat flange, 4 5/8" Ø, x 0.750" thick. commercial CF flange (blank)	850	978	978	1467
VCR Plug, 1/2" Ø, modified commercial	850	978	978	1467
Flat Head Conflat Type: AAA99-104240	350	402	402	604
Conlfat flange, 2 3/4" Ø, x 0.500" thick. modified commercial CF flange	350	402	402	604
Ceramaseal: 19543-04-CF; 1 1/3" Ø, x 0.300" thick. modified commercial CF flange	350	402	402	604

E. Testing Requirements

Detailed testing procedures have been developed and are enclosed as Appendices B. The proof test criterion for each system is 150% of MAWP.

F. Labeling Requirements

Upon completion of the testing procedures, the LLNL pressure inspector will certify the inspection of this system by completion of an LLNL Pressure Test/Inspection Record, Form LL3586, and by attaching an LLNL Pressure Tested Label, properly filled out to the individual components identified below. Appropriate additional information will be inserted as required.

LLNL PRESSURE TESTED
FOR MANNED AREA

ASSY. Pressure vessel
SAFETY NOTE MESN99-020-OA
M.A.W.P 978 PSIG.
FLUID He, Xe, Ar, CH4, CO2, P10
TEMP. -320 TO ambient °F
REMARKS Main Pressure Vessel (AAA99-104242, weld flange)
TEST NO. T.R.
EXPIRATION DATE
BY DATE

LLNL PRESSURE TESTED
FOR MANNED AREA

ASSY. AAA98-104240
SAFETY NOTE MESN99-020-OA
M.A.W.P 402 PSIG.
FLUID He, Xe, Ar, CH4, CO2, P10
TEMP. -320 TO ambient °F
REMARKS 350 MOP CF type head
TEST NO. T.R.
EXPIRATION DATE
BY DATE

LLNL PRESSURE TESTED
FOR MANNED AREA

ASSY. AAA99-104243-00
SAFETY NOTE MESN99-020-OA
M.A.W.P 978 PSIG.
FLUID He, Xe, Ar, CH4, CO2, P10
TEMP. -320 TO ambient °F
REMARKS 850 MOP C-Ring Head
TEST NO. T.R.
EXPIRATION DATE
BY DATE

LLNL PRESSURE TESTED
FOR MANNED AREA

ASSY. AAA99-104241-00
SAFETY NOTE MESN99-020-OA
M.A.W.P 402 PSIG.
FLUID He, Xe, Ar, CH4, CO2, P10
TEMP. -320 TO ambient °F
REMARKS 350 MOP CF Type Head
TEST NO. T.R.
EXPIRATION DATE
BY DATE

G. Associated Procedures

The concerns are asphyxiation, cold temperature and radiation exposure of personnel. Responsibility for an OSP resides with the user.

H. References and Notes

1. The defining drawings are as follows:

Drawing Title	LLNL
Pressure Chamber Lid Blank	AAA98-1104241
Pressure Chamber Lid	AAA98-1104240
Pressure Chamber Lid Blank C Ring 850 MOP	AAA98-1104243
Pressure Chamber Weld Flange 850 MOP	AAA98-1104242
Xenon Chamber Model 8" (sketch)	N/A
Xenon Chamber Model 8" associated sketches	N/A

2. 1995 ASME Boiler and Pressure Vessel Code, Section VIII, Division I.

3. Design of Piping Systems, John Wiley & Sons, Inc. 1974.

4. Degraded Piping Program - Phase II, Sixth Program Report, Oct. 1986 – September 1987, USNRC

5. Fracture 1969, Chapman and Hall Ltd. IBN 412094703

6. Handbook of Stainless Steels, D. Peckner, I. Bernstein, McGraw-Hill, 1977

7. Metal Fatigue in Engineering, H. Fuchs, R. Stephens, John Wiley & Sons, Inc. 1980.

8. Austenetic Steels at Low Temperatures, R.P. Reed, T Horiuchi, Plenum Press, 1982.

APPENDIX A: PROOF

TESTING PROCEDURE FOR

THE TPC

A.1 General

This procedure is for proof testing the TPC shown in Figure 1. Initial pressure and leak tests of the system will be conducted in Building 343 because it provides an adequate barricade for conducting the test and keeps personnel exposure to a minimum. Final leak testing of joints made up after installation and retest of the systems in the future will be conducted at the B132 facility.

A.2 Hazards

The Health and Safety Manual Supplement 32.05, Section 2 – "Standard Procedure for Pressure Testing with Gas" applies.

A.3 Pretest Procedure

Use the system indicated in Figure A1 as the test source. Support the chamber horizontally. Cool the chambers' LN2 pipe and surrounding metal with an LN2 filled dewar supplied by the experimenter to simulate the thermal stresses during actual operation. Let the metal 'soak' for 20-30 minutes before proof testing.

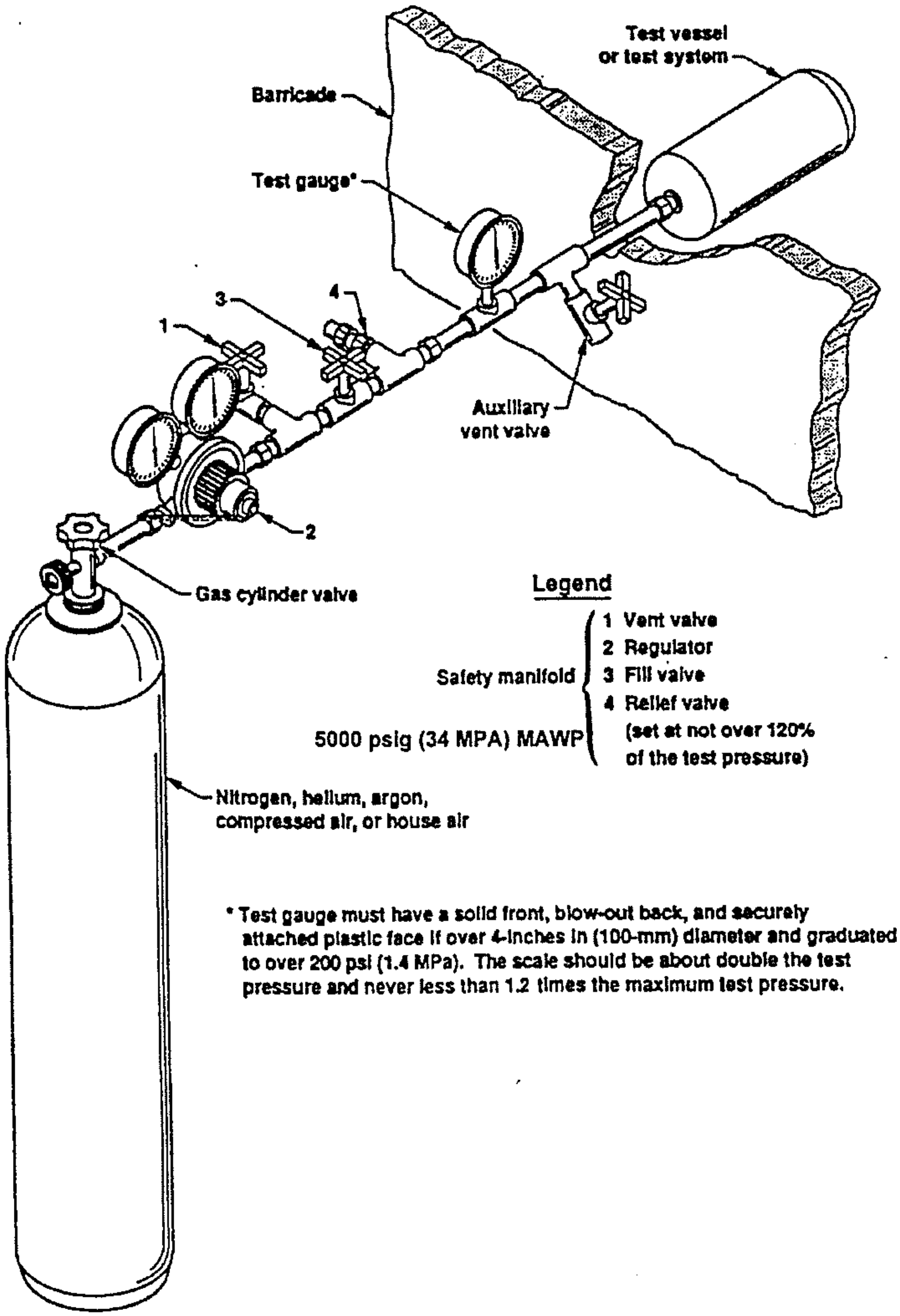


Figure A1 – Gas Test System

A.4 Test Procedure

Refer to Figure 1 and Appendix A for component designations.

A.4.1 High Pressure (1467 psig) Helium System Pressure Test

The two TPC's will first be tested to 1.5 x MAWP, or 1.5 x 978 = 1467 psig using the following components:

Vessel (2 ea., requires 2 separate proof tests)
C-Ring type lid (AAA99-104243)
4 3/4" CF blank for the detector port
Modified VCR plug(s) at the VCR ports

1. Install the hardware described above for the 1467psig proof test.
2. Apply 1467psig test pressure to one of the VCR ports.
3. Hold test pressure at 1467psig for 15 minutes.
4. Vent system down to 150 psig and leak check all joints under pressure with Snoop.
5. Vent helium to atmospheric pressure.

A.4.2 Moderate Pressure (604 psig) Helium System Pressure Test

A single TPC will also be tested to 1.5 x MAWP, or 1.5 x 402 = 604 psig using the following components. Two tests are required to qualify both heads.

Vessel
CF type lid (AAA99-104240, AAA99-104241)
2 3/4" CF modified blank for the x-ray port
1 1/3" CF flanges with high voltage feedthroughs
4 3/4" CF blank for the detector port
Modified VCR plug at the VCR ports

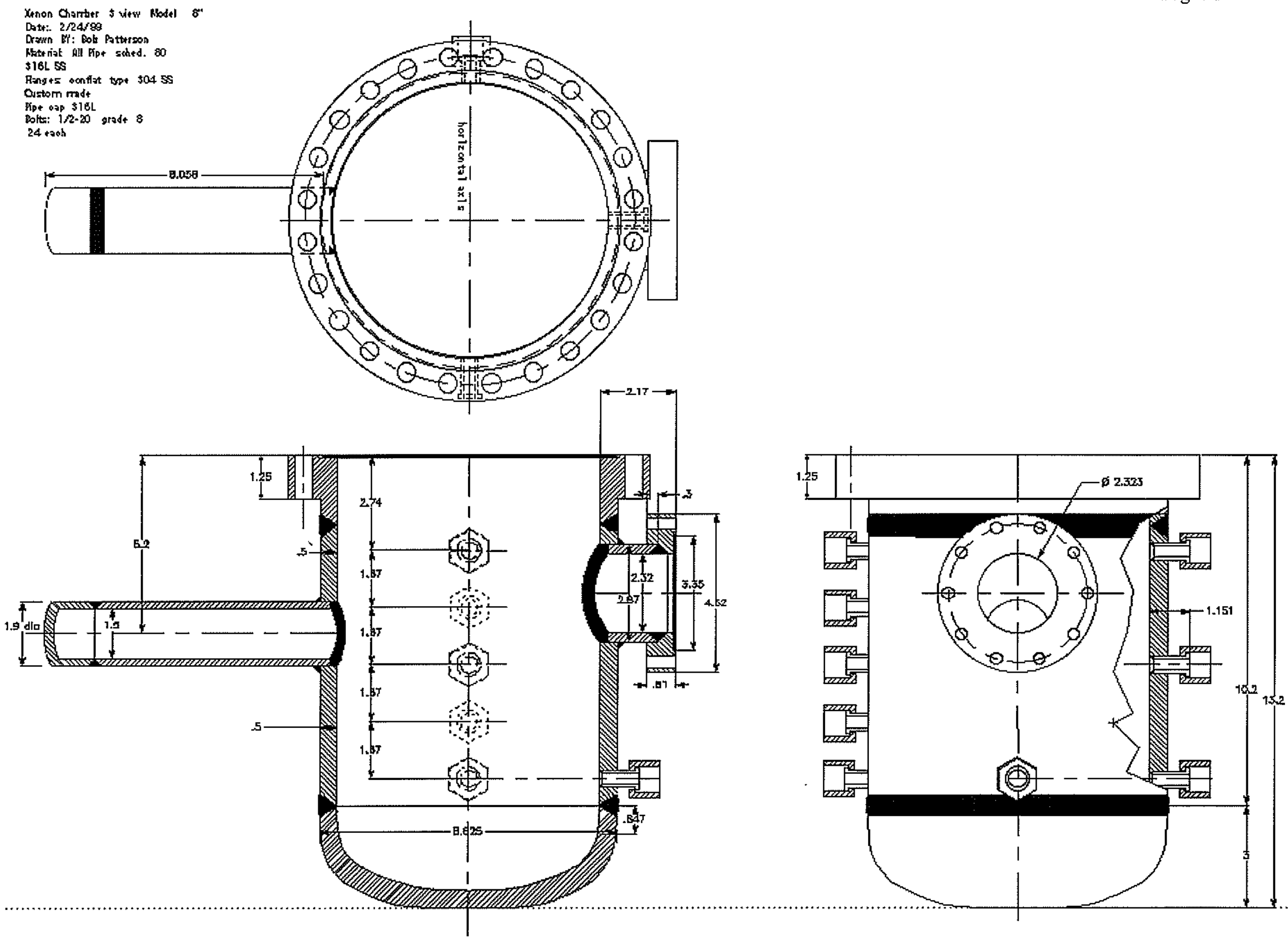
1. Install the hardware described above for the 604 psig proof test.
2. Apply 604 psig test pressure to one of the VCR ports.
3. Hold test pressure at 604 psig for 15 minutes.
4. Vent system down to 150 psig and leak check all joints under pressure with Snoop.
5. Vent helium to atmospheric pressure.

A.4.3 Documentation

Test records shall include an LLNL Pressure test/inspection record for the separate pieces of the vessel. The pressure inspector will send the original copies of the test reports to LLNL Pressure Safety (L-384).

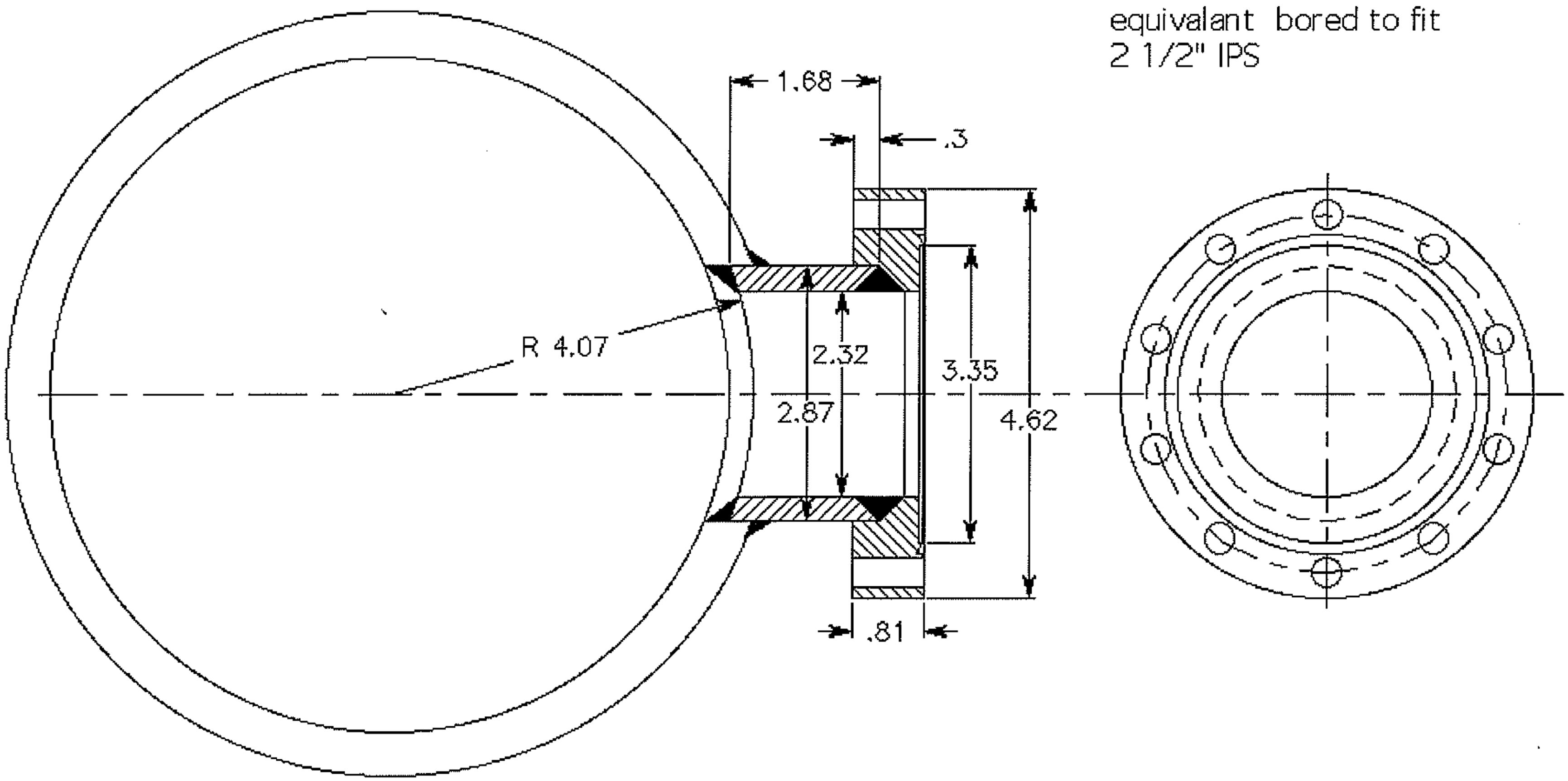
APPENDIX B

DRAWINGS


Xenon Chamber 3 view Model 8"
Date: 2/24/99
Drawn BY: Bob Patterson
Material: All Pipe sched. 80
316L SS
Flanges: conflat type 304 SS
Custom made
Pipe cap 316L
Bolts: 1/2-20 grade 8
24 each
8.058
horizontal axis
2.17
.3
1.25
2.74
5.2
.5
1.37
1.37
1.37
1.37
2.32
2.87
3.35
4.52
.81
1.9 dia
1.5
.5
.847
8.625
1.25
ø 2.323
1.151
10.2
13.2
3

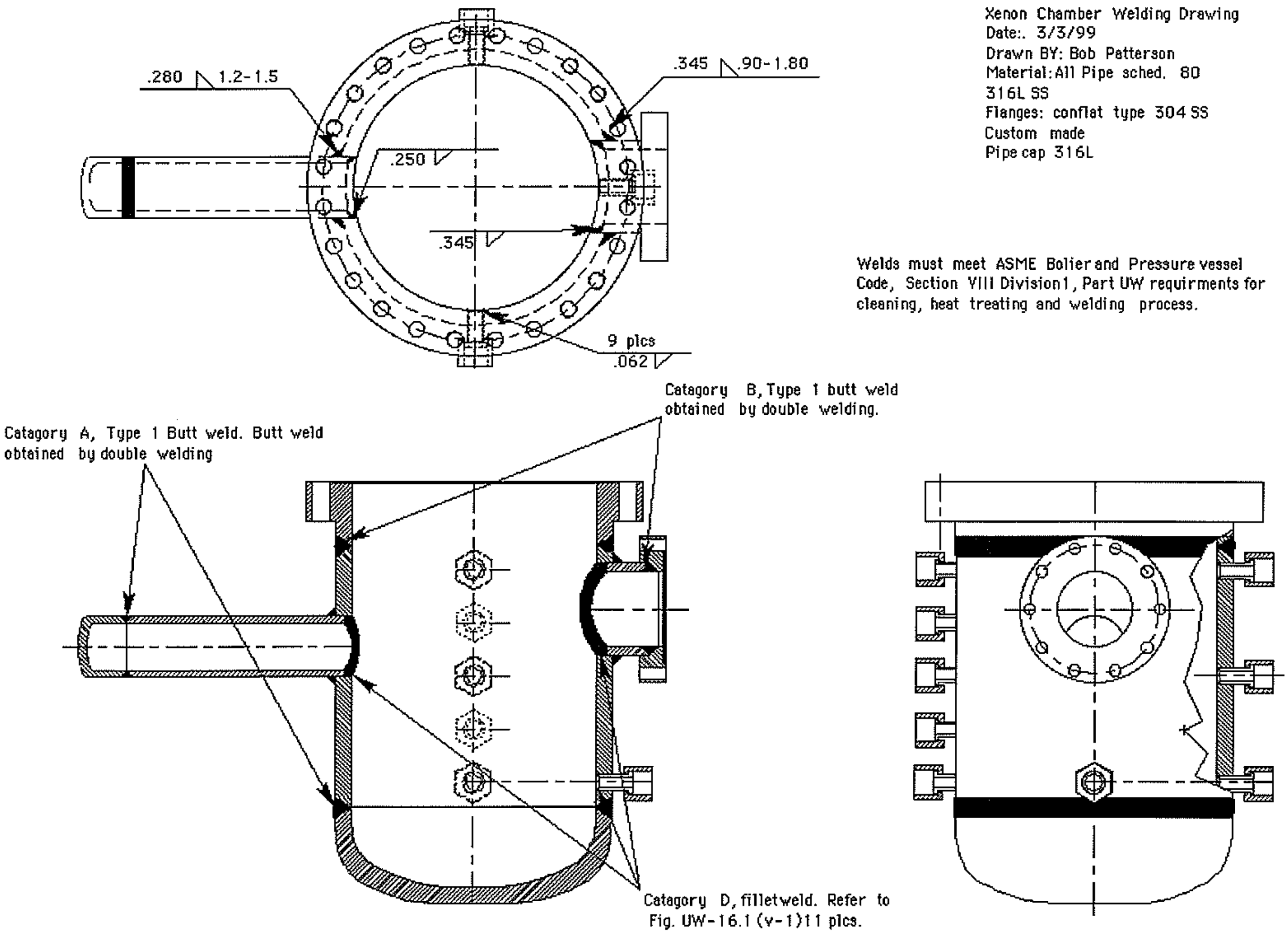
Detector Port
Date:. 2/24/99
Drawn By: Bob Patterson
Material: Pipe 2 1/2 IPS sched. 80
316L SS
2 ea. required

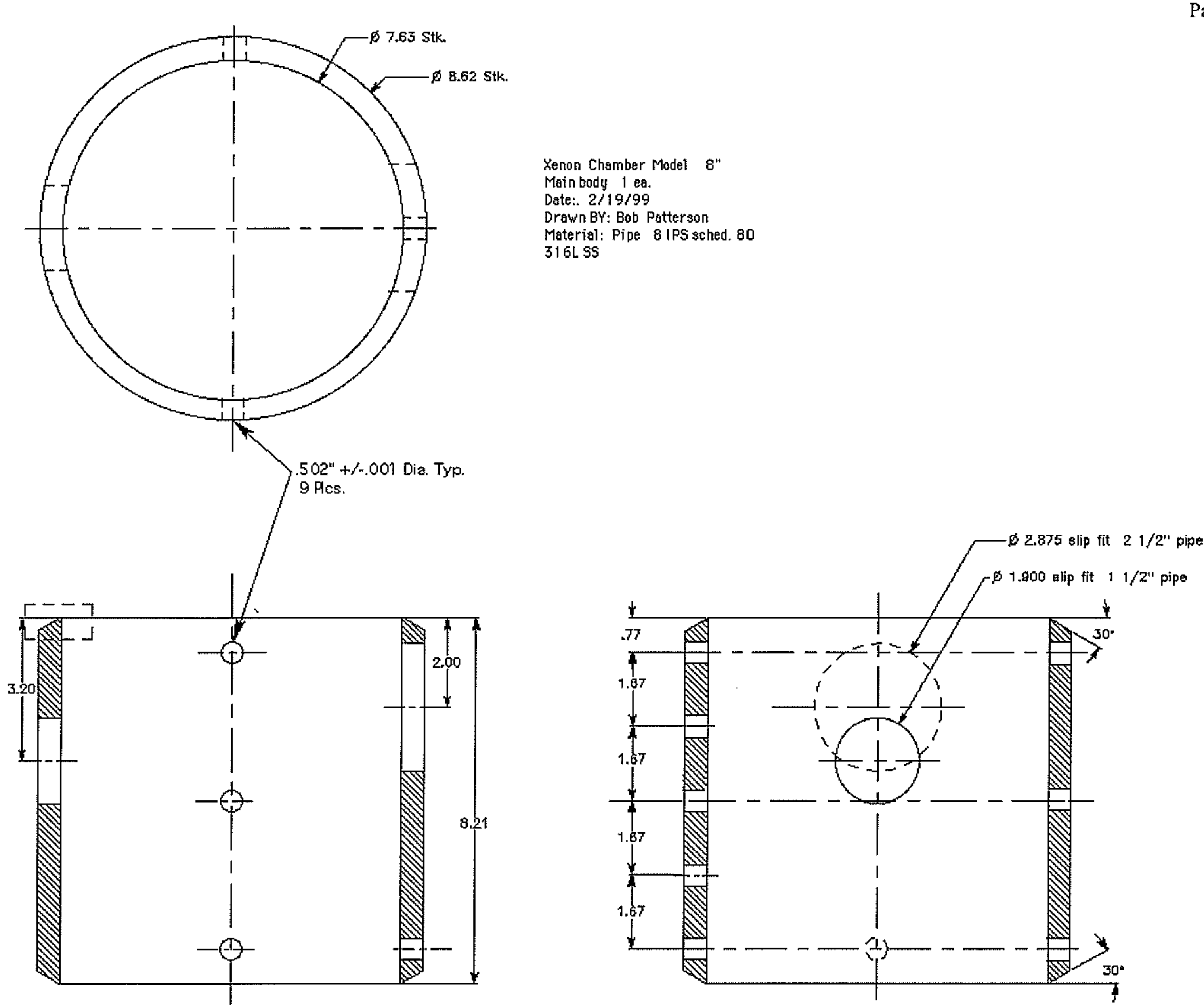
Flange MDC 4 5/8" OD
F458000 conflat blank or
equivalant bored to fit
2 1/2" IPS

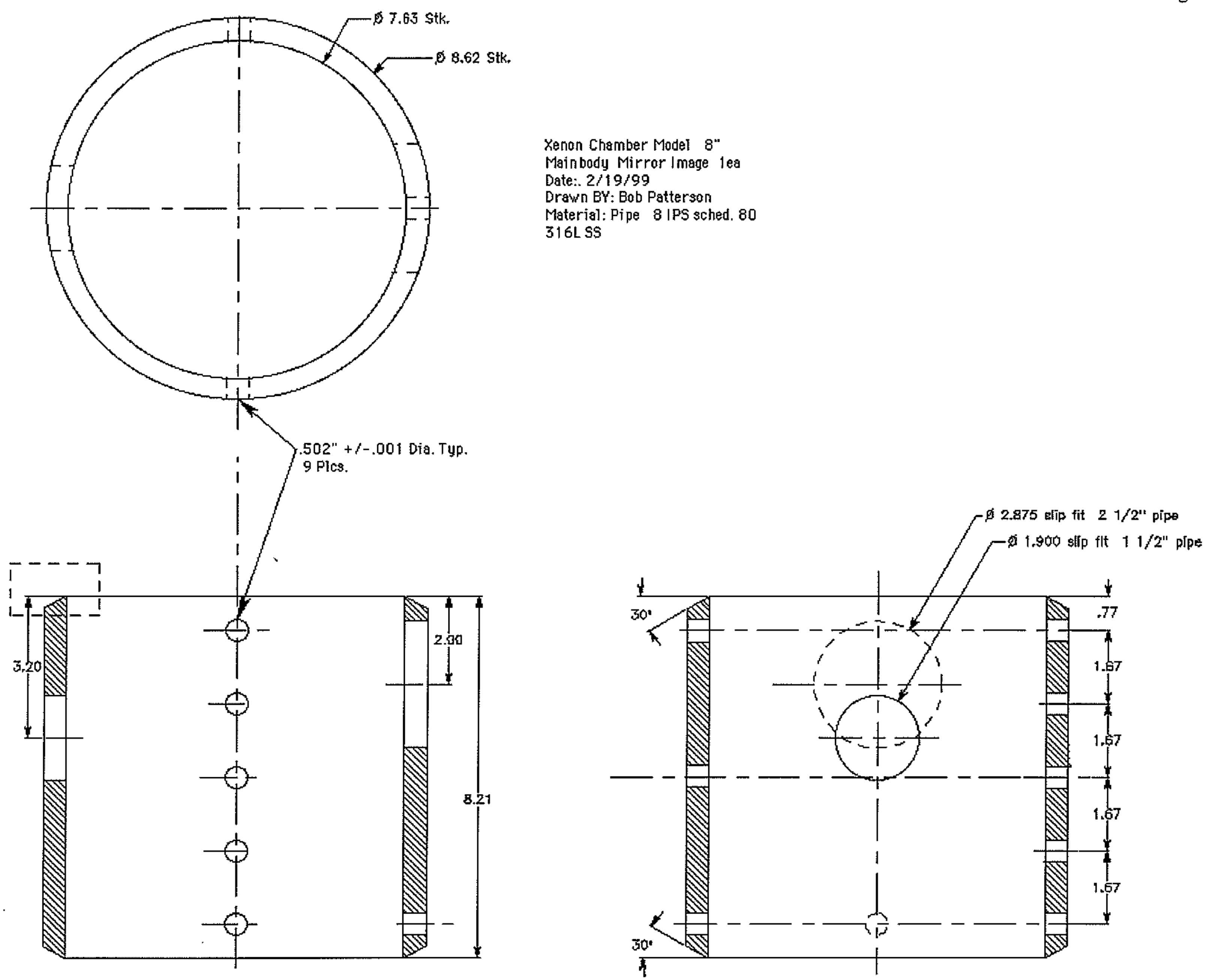


Xenon Chamber Welding Drawing
Date:. 3/3/99
Drawn BY: Bob Patterson
Material: All Pipe sched. 80
316L SS
Flanges: conflat type 304 SS
Custom made
Pipe cap 316L

Welds must meet ASME Bolier and Pressure vessel Code, Section VIII Division1, Part UW requirments for cleaning, heat treating and welding process.

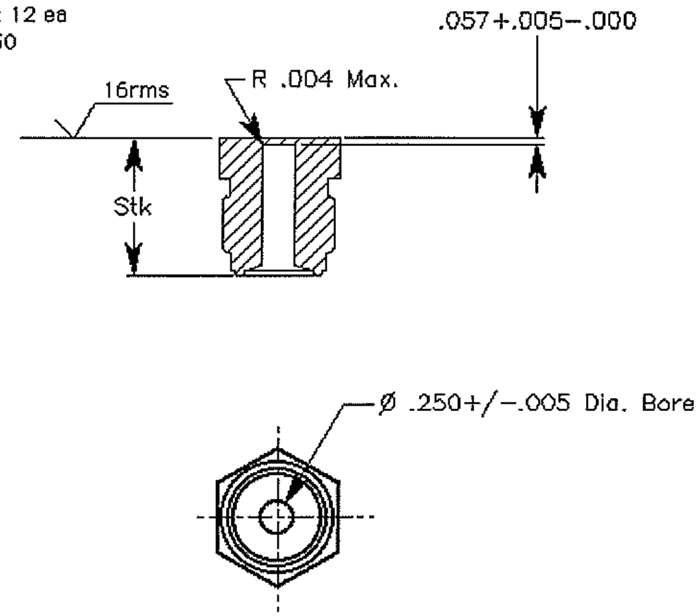



Ø 7.63 Stk.
Ø 8.62 Stk.
Xenon Chamber Model 8"
Main body 1 ea.
Date:. 2/19/99
Drawn BY: Bob Patterson
Material: Pipe 8 IPS sched. 80
316L SS
.502" +/-.001 Dia. Typ.
9 Plcs.
3.20
2.00
8.21
Ø 2.875 slip fit 2 1/2" pipe
Ø 1.900 slip fit 1 1/2" pipe
.77
1.67
1.67
1.67
1.67
30°
30°


Ø 7.63 Stk.
Ø 8.62 Stk.
Xenon Chamber Model 8"
Mainbody Mirror Image 1ea
Date:. 2/19/99
Drawn BY: Bob Patterson
Material: Pipe 8 IPS sched. 80
316L SS
.502" +/-.001 Dia. Typ.
9 Plcs.
Ø 2.875 slip fit 2 1/2" pipe
Ø 1.900 slip fit 1 1/2" pipe
3.20
2.30
8.21
30°
.77
1.67
1.67
1.67
1.67
30°

	Checked by	Date	Drawing #

Gamma window
VCR Plug Modified
P/N SS-8-VCR-P
Quantity: 12 ea
MAWP 850

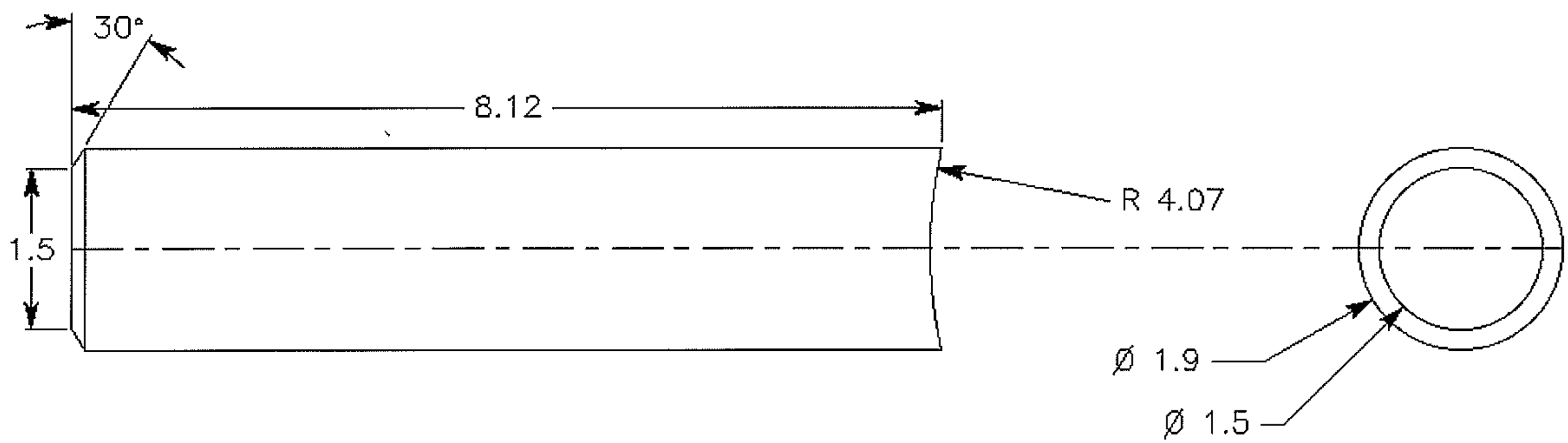


Note: Modify existing VCR Male Plug
Remachine top surface
Parts supplied by requestor

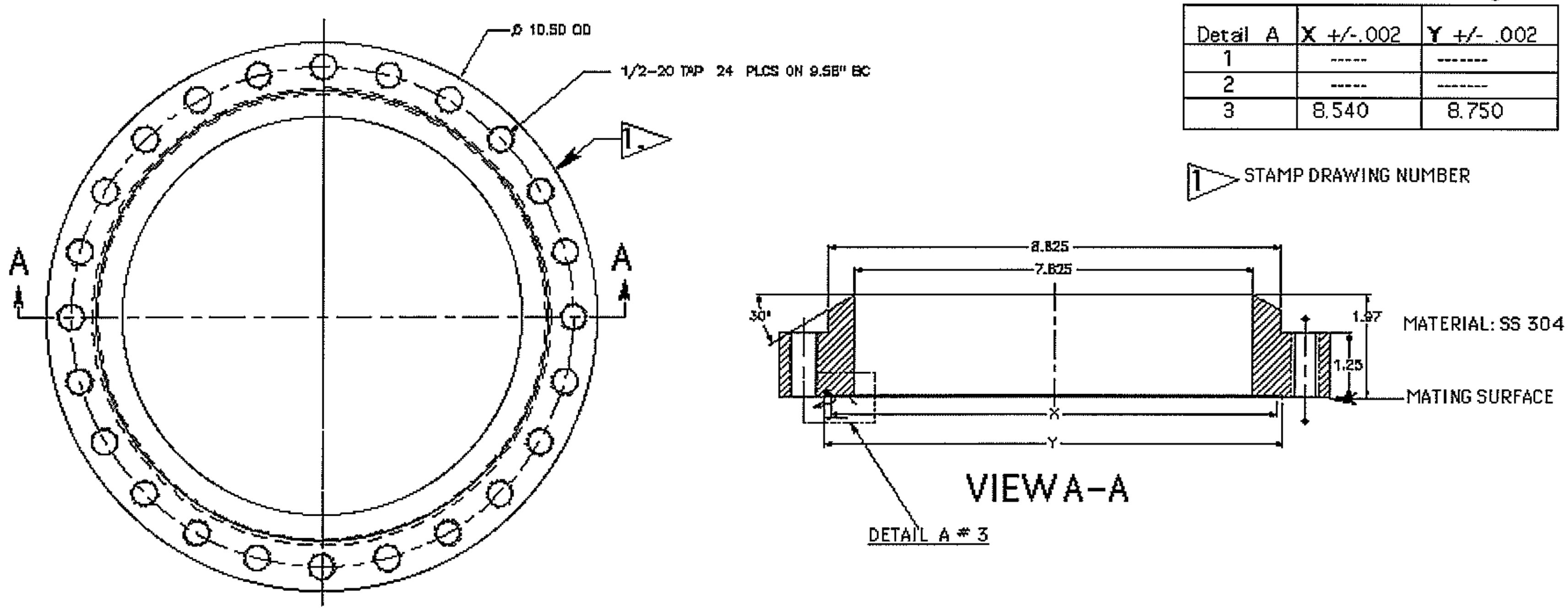
All Dims. in inches

Date Drawn 4/23/99	Lawrence Livermore National Laboratory
Scale 1 to 1	Drawn by Robert Patterson Employee no 686200
Acct. No. 8960-04	Building 132 S Room 1170 L-171 Phone 422-7599

PumpingTubulation
Date:. 2/25/99
Drawn By: Bob Patterson
Material: Pipe1 1/2 IPS sched. 80
316L SS
2 ea. required



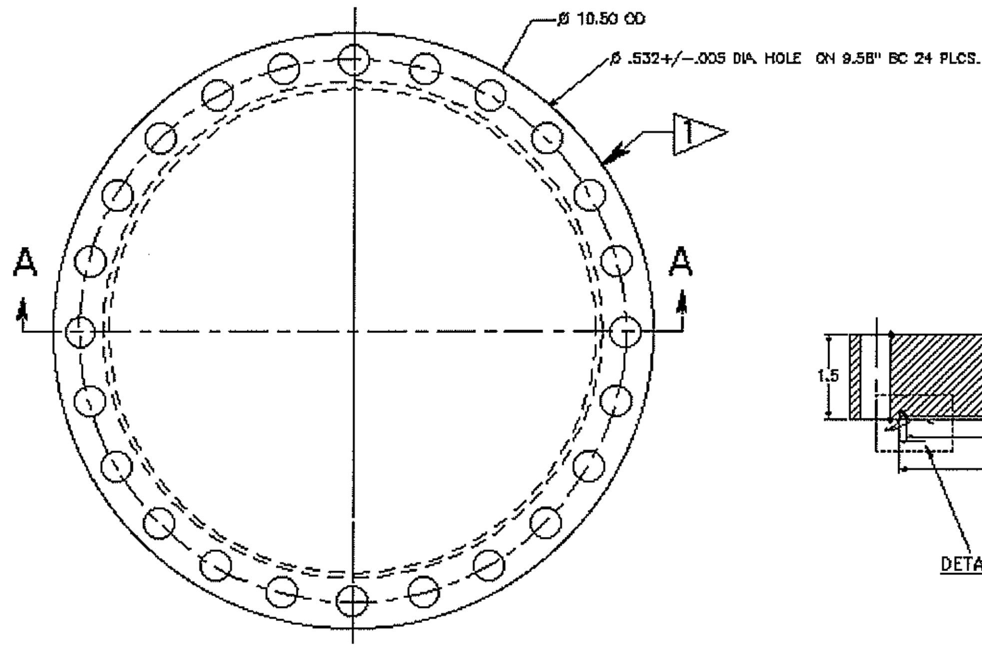
Detail A	X +/-.002	Y +/- .002
1	-----	-------
2	-----	-------
3	8.540	8.750



LEVEL 1 DRAWING

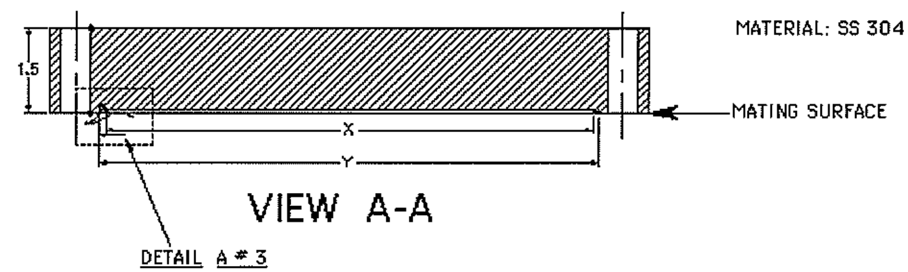
PressureChamber
Weld Flange 850 MOP

AAA99-104242-00



Detail A	X +/-.002	Y +/-.002
1	-----	--------
2	-----	--------
3	8.540	8.750

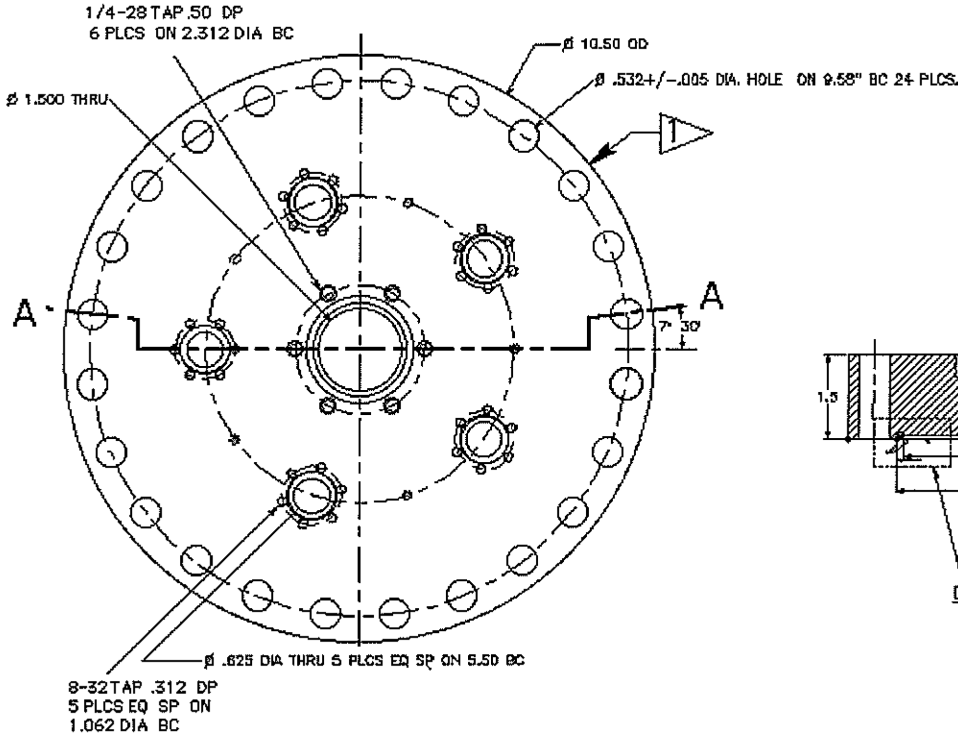
1 STAMP DRAWING NUMBER



LEVEL 1 DRAWING

Pressure Chamber Lid
Blank 350 MOP

AAA99-104241-00



Detail A	X +/-.002	Y +/- .002
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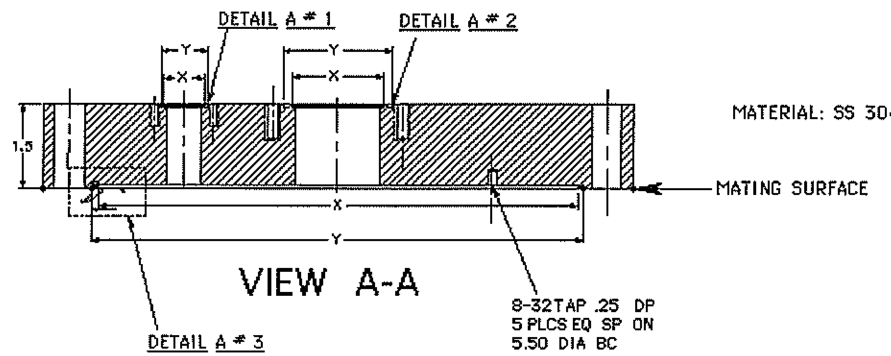
2	1.650	1.902
3	8.540	8.750

1 STAMP DRAWING NUMBER

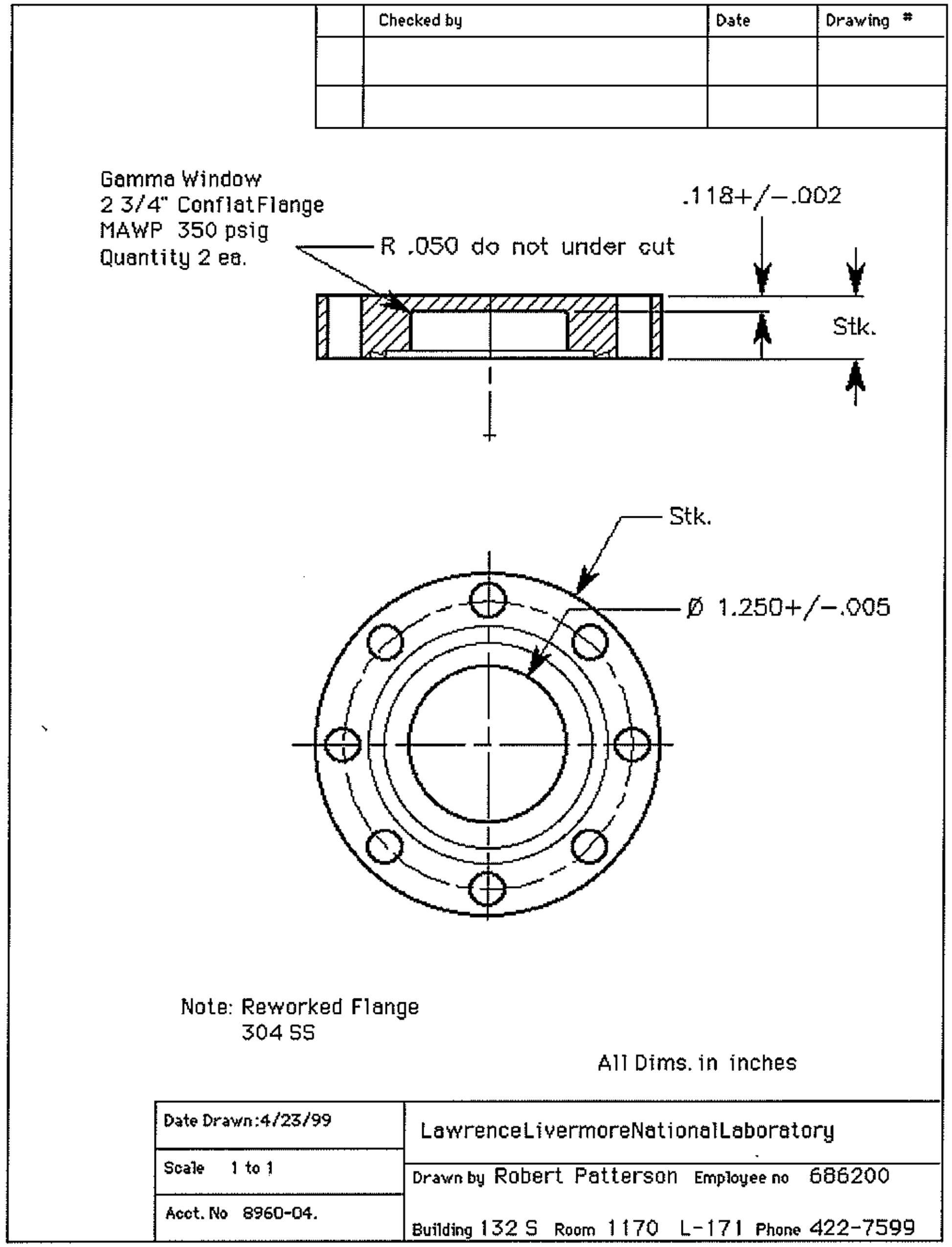


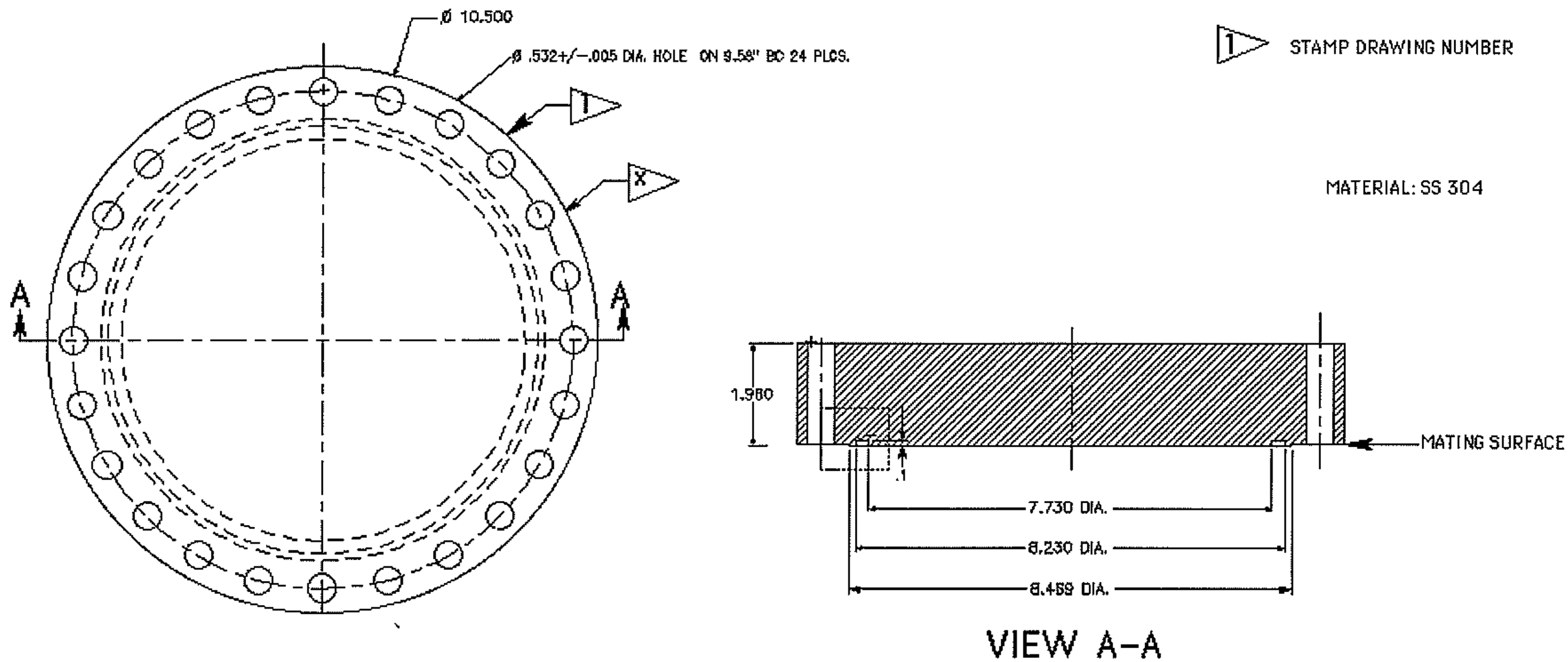
VIEW A-A

LEVEL 1 DRAWING

Pressure Chamber Lid
C F Mini 350 MOP

AAA99-104240-00


Checked by
Date
Drawing #
Gamma Window
2 3/4" ConflatFlange
MAWP 350 psig
Quantity 2 ea.
R .050 do not under cut
.118+/-.002
Stk.
Stk.
Ø 1.250+/-.005
Note: Reworked Flange
304 SS
All Dims. in inches
Date Drawn: 4/23/99
Scale 1 to 1
Acct. No 8960-04.
LawrenceLivermoreNationalLaboratory
Drawn by Robert Patterson Employee no 686200
Building 132 S Room 1170 L-171 Phone 422-7599



LEVEL 1 DRAWING

PressureChamber Lid
Blank C'Ring 850 MOP

AAA99-104243-00

APPENDIX C

CALCULATIONS

File name	Calculations performed
Energy_vessel.nb	energy calculations, peak and static overpressure
Vessel_stress2.nb	main vessel stress calculations, wall thickness, maximum pressure, proof test stress
Ellipsoidal_head_stress.nb	main vessel head thickness, max. pressure, proof test stress
Ellipsoidal_head_stress_LN2..nb	LN2 head thickness, max. pressure, proof test stress
Flange stress_hub_978.nb	main vessel C-Ring head bolt load, moment, stresses
Flange stress_hub_small_978.nb	detector port bolt load, moment, stresses
Flange stress_hub_403.nb	main vessel CF head bolt load, moment, stresses
Xe_vessel_det.nb	detector pipe stress calculations, wall thickness, maximum pressure, proof test stress
Detector_shell.nb	detector pipe weld reinforcement, area required, area available
Weld_load_stress.nb	detector pipe weld load allowable, strength of connecting elements (welds)
Xe_vessel_VCR.nb	VCR pipe stress calculations, wall thickness, maximum pressure, proof test stress
VCR_shell.nb	VCR pipe weld reinforcement, area required, area available
Weld_load_stress_VCR.nb	VCR pipe weld load allowable, strength of connecting elements (welds)
VCR_gamma_port.nb	VCR minimum head thickness calculation
Xe_vessel_LN2.nb	LN2 pipe stress calculations, wall thickness, maximum pressure, proof test stress
LN2_shell.nb	LN2 pipe weld reinforcement, area required, area available
Weld_load_stress_LN2.nb	LN2 pipe weld load allowable, strength of connecting elements (welds)
Head_350_K_openings2.nb	main vessel CF type flat head: stress concentration factor, thickness, distance between hole centers
Head_850_no_openings2.nb	main vessel C-ring type flat head: head thickness, hub thickness
Head_850_4.625_no_openings2.nb	detector port CF type flat head: head thickness, hub thickness
Bolt_load_1.33CF_350.nb	1.33 CF flange bolt load, head thickness
Bolt_load_2.75CF_350.nb	2.75 CF flange bolt load, head thickness
Misc.nb	main vessel: distance between openings, blind mounting hole depth, reinforcement of blind holes on CF flanges mounted on 10.5" Ø CF flange, impact testing
Fracture_critical_mat'l.nb	K_{ic}, K_I, critical crack lengths, Life cycles
Fragmant.nb	shielding calculations

```
(* Energy in Xenon Pressure Vessel *)
```

```
MAWP = 978
```
$P_1 = MAWP$

$P_2 = 14.7$

$K = 1.66$

$R_i = 3.8125$

$D2 = 1.5$

$D3 = 2.32$

$$V_1 = \frac{\pi\,(2\,R_i)^2}{4}\,(12.2 - 0.5)\ (*\ in^3\ *)$$

$$V_2 = \frac{\pi\,(D2)^2}{4}\,(8.058\ -\ 0.2)\ (*\ in^3\ *)$$

$$V_3 = \frac{\pi\,(D3)^2}{4}\,2.17\ (*\ in^3\ *)$$

$$V_T = V_1 + V_2 + V_3\ (*\ in^3\ *)$$

$$Energy = \frac{P_1\,V_T}{12\,(K-1)}\left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{K-1}{K}}\right)\ (*\ ft\text{-}lb\ *)$$

$$Energy_{TNT} = \frac{Energy}{3414.1}\ (*\ g\ TNT\ *)$$

$$Energy_{lb} = Energy_{TNT} * 0.002200\ (*\ lb.\ TNT\ *)$$

```
978

978

14.7

1.66

3.8125

1.5

2.32

534.263

13.8862

9.1733

557.323

55852.

16.3592

0.0359903
```

(* From MEDSS, 30 psi is the threshold for fatalities. 0.2 to 15 psi cause physiological damage (ear, lung, etc.) However, the detailed calculation that follow (and proof tests of Conflat heads) show this vessel will leak before catastrophic failure. *)

(* The following is an analysis of the static overpressure in the confined room *)

```
Psov = 1.15×10^4 Energy_lb/(20×30×10) (* psig *)
```

$$P_{sov} = 1.15\times10^{4}\ \frac{\text{Energy}_{lb}}{20\times30\times10}\ (\text{* psig *})$$

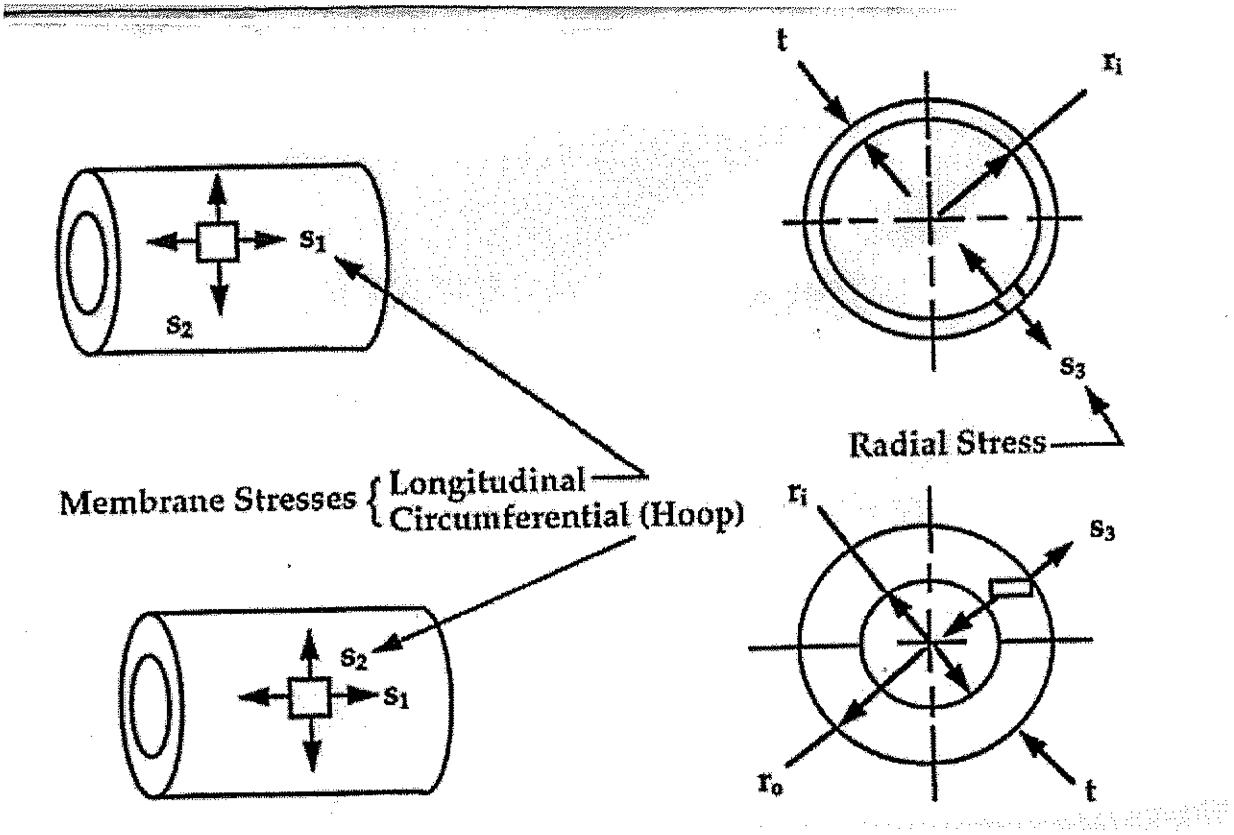
0.0689813

(* The peak overpressure is simply 6 X static *)

$$P_{pov} = 6\times P_{sov}\ (\text{* psig *})$$

0.413888

(*Xenon Pressure Vessel Stress Calculations*)



```
In[12]:=
        MAWP = 978
        σa = 16700 (*allowable stress for 316 L SST*)
        σy = 37000
        Ri = 3.8125
        Ro = 4.3125
        t = Ro - Ri
        Ratio = Ro/Ri
        If [1.1 < Ratio < 1.5, medium wall]
        If [Ratio < 1.1, thin wall]
        If [Ratio > 1.5, thick wall]

Out[12]= 978

Out[13]= 16700

Out[14]= 37000

Out[15]= 3.8125

Out[16]= 4.3125

Out[17]= 0.5

Out[18]= 1.13115

Out[19]= medium wall
```

In[22]:=

```
(*Longitudinal Stress, S1*)
```

$$S_1 = \frac{(\mathrm{MAWP}\ R_i^2)}{(R_o^2 - R_i^2)}$$

```
(*Circumferential Stress, S2*)
```

$$S_2 = \frac{\mathrm{MAWP}\ (R_o^2 + R_i^2)}{(R_o^2 - R_i^2)}$$

```
(*Radial Stress, S3*)
```

$$S_3 = -\mathrm{MAWP}$$

```
(*Von Mises Stress*)
```

$$\sigma_m = \sqrt{0.5\ ((S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2)}$$

Out[22]= 3499.17

Out[23]= 7976.34

Out[24]= -978

Out[25]= 7754.69

```
(*wall thickness, in., max. pressure, psi*)

(*Circumferential / Longitudinal Stress: wall thickness, in., max. pressure, psi*)
```

$E_f = 0.7$
```
(*butt weld efficiency based on no inspection, Table UW-12*)
(*Circumferential butt welds connecting
  ellipsoidal head and hub to cylinder are Catagory A/B, Type 1 welds*)
```

```
p = 1.67 (* in., longitudinal pitch of tube holes *)
d = 0.5 (* in., diamnter of tube hole*)
```

$E_{flig} = \frac{p - d}{p}$ (* UG-53, Ligaments *)

If [E_f < E_{flig}, E_f = E_f, E_f = E_{flig}]

$t_c = \frac{(\text{MAWP } R_i)}{(\sigma_a E_f - 0.6\,\text{MAWP})}$ (*UG27 c 1*)

$P_c = \frac{(\sigma_a E_f t)}{(R_i + 0.6\,t)}$ (*UG27 c 1*)

$SF_{uc} = \frac{P_c}{\text{MAWP}}$ (* P_c uses allowable stress so SF ~5 is also inlcluded*)

$t_l = \frac{(\text{MAWP } R_i)}{(2\,\sigma_a E_f + 0.4\,\text{MAWP})}$ (*UG27 c 2*)

$P_l = \frac{(2\,\sigma_a E_f t)}{(R_i - 0.4\,t)}$ (*UG27 c 2*)

$SF_{ul} = \frac{P_l}{\text{MAWP}}$ (* P_l uses allowable stress so SF ~5 is also inlcluded*)

If[P_c < P_l, "circumferential stress applies", "longitudinal stress applies"]
If[t_c > t_l, "circumferential stress applies", "longitudinal stress applies"]

Out[15]= 0.7

Out[16]= 1.67

Out[17]= 0.5

Out[18]= 0.700599

Out[19]= 0.7

Out[20]= 0.335815

Out[21]= 1421.28

Out[22]= 1.45325

Out[23]= 0.156855

Out[189]= 3235.99

Out[190]= 13.2351

Out[191]= circumferential stress applies

Out[192]= circumferential stress applies

(*Check of Von Mises stress at 1.5 × MAWP for pressure test*)
MAWP = 1.5 × 978

(*Longitudinal Stress, S_1*)

$$S_1 = \frac{(MAWP\ R_i^2)}{(R_o^2 - R_i^2)}$$

(*Circumferential Stress, S_2*)

$$S_2 = \frac{MAWP\ (R_o^2 + R_i^2)}{(R_o^2 - R_i^2)}$$

(*Radial Stress, S_3*)

$$S_3 = -MAWP$$

(*Von Mises Stress*)

$$\sigma_m = \sqrt{0.5\ ((S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2)}$$

$$N_r = \frac{\sigma_y}{\sigma_m}$$

If[N_r > 1, "vessel OK at 1.5 × MAWP during pressure test"]

1467.

5248.76

11964.5

-1467.

11632.

3.18087

vessel OK at 1.5 × MAWP during pressure test

```
(*Xenon Pressure Vessel Stress Calculations*)
(*Ellipsoidal Head*)
```

In[704]:=

```
MAWP = 978
```
σ_a = 16700 (*allowable stress for 304 SST*)

σ_y = 32000

D_i = 7.625

t_w = 0.5

```
(*Circumferential butt welds connecting
  ellipsoidal head and hub to cylinder are Catagory A, Type 1 welds*)
```

E_f = 0.7 (*butt weld efficiency based on no inspection, Table UW-12*)

Out[704]= 978

Out[705]= 16700

Out[706]= 32000

Out[707]= 7.625

Out[708]= 0.5

Out[709]= 0.7

In[710]:= (*Wall thickness, in., max. pressure, psi*)

(*Circumferential Stress: wall thickness, in., max. pressure, psi*)

$$t_h = \frac{(\mathrm{MAWP}\, D_i)}{(2\,\sigma_a\, E_f\, -\, 0.2\,\mathrm{MAWP})} \quad (*\mathrm{UG32}\ \ (d)*)$$

$$P_m = \frac{(2\,\sigma_a\, E_f\ \ t_w)}{(D_i\ +0.2\ t_w)} \quad (*\mathrm{UG32}\ \ (d)*)$$

$$SF_{uc} = \frac{P_m\, 4}{\mathrm{MAWP}}$$

Out[710]= 0.321649

Out[711]= 1513.27

Out[712]= 6.18924

In[713]:=

(*Check of stress at 1.5 × MAWP for pressure test*)

MAWP = 1.5 × 978

$$\mathbf{Solve}\left[\mathbf{MAWP} == \frac{(2\,\sigma\,E_f\;t_w)}{(D_i\;+0.2\;t_w)},\ \sigma\right]$$

$$SF_Y = \frac{\sigma_Y}{\sigma}$$

Out[713]= 1467.

Out[714]= $\{\{\sigma \to 16189.4\}\}$

Out[715]= $\frac{32000}{\sigma}$ $\quad \frac{16{,}700}{16{,}189.4} \geq 1.0$ ✓✓

```
(*Xenon Pressure Vessel Stress Calculations*)
(*Ellipsoidal Head, LN2 Trap*)
```

In[716]:=

$MAWP = 978$

$\sigma_a = 16700$ (*allowable stress for 304 SST*)

$\sigma_y = 32000$

$D_i = 1.5$

$t_w = 0.2$

(*Circumferential butt welds connecting ellipsoidal head and hub to cylinder are Catagory A, Type 1 welds*)

$E_f = 0.7$ (*butt weld efficiency based on no inspection, Table UW-12*)

Out[716]= 978

Out[717]= 16700

Out[718]= 32000

Out[719]= 1.5

Out[720]= 0.2

Out[721]= 0.7

In[722]:= (*Wall thickness, in., max. pressure, psi*)

(*Circumferential Stress: wall thickness, in., max. pressure, psi*)

$$t_h = \frac{(MAWP\ D_i)}{(2\ \sigma_a\ E_f\ -\ 0.2\ MAWP)}$$ (*UG32 (d)*)

$$P_m = \frac{(2\ \sigma_a\ E_f\ t_w)}{(D_i\ +0.2\ t_w)}$$ (*UG32 (d)*)

$$SF_{uc} = \frac{P_m\ 4}{MAWP}$$

Out[722]= 0.0632753

Out[723]= 3036.36

Out[724]= 12.4187

In[725]:=

```
(*Check of stress at 1.5 × MAWP for pressure test*)
MAWP = 1.5 × 978
```

$$\text{Solve}\left[\text{MAWP} == \frac{(2\,\sigma\,E_f\ t_w)}{(D_i + 0.2\ t_w)},\ \sigma\right]$$

$$SF_y = \frac{\sigma_y}{\sigma}$$

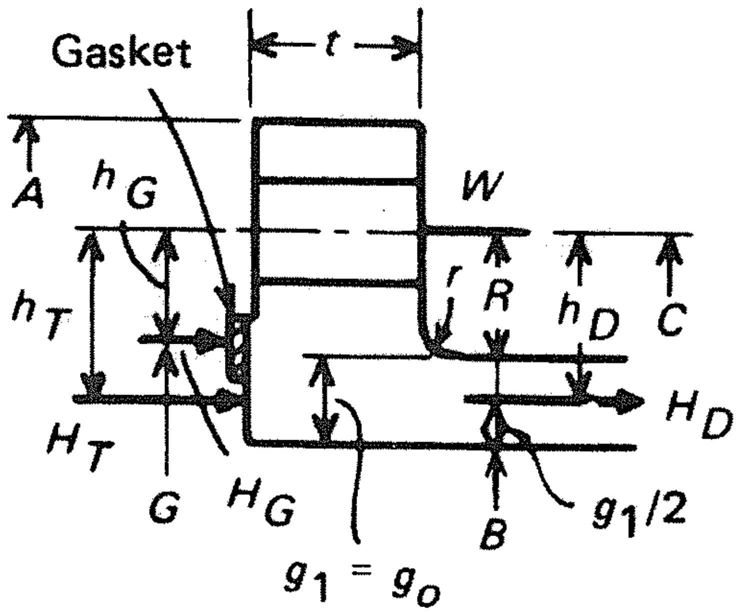
Out[725]= 1467.

Out[726]= $\{\{\sigma \to 8068.5\}\}$

Out[727]= $\frac{32000}{\sigma}$

$\frac{161700}{8068.5} \geq 1$

(2.07)



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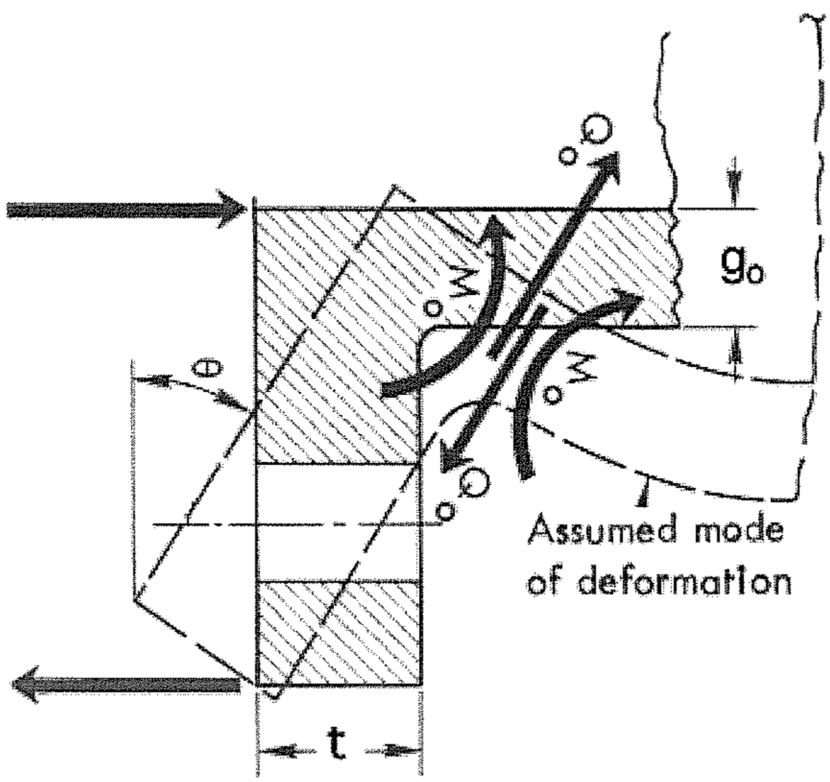



```
(* Bolted Flange Connections with Ring Type Joint *)
(* Integral Flange Type, Appendix 2, Figure 2-4 (5) shown above *)
(* 850 psia MOP, custom flange, ring type joint, metal seal *)
```

```
In[1]:= (* Bolt Load at operating conditions *)

        G = 7.980 (* Diameter, in. at gasket load location *)
        P = 978  (* MAWP, internal design pressure *)
        m = 6.5 (* gasket factor ring joint, Table 2-4.1 *)
        Ng = 0.25 (* width of ring type gasket *)
        bo = Ng/8 (* Table 2-5.2 (6) *)
        If[bo <= 0.25, b = bo, b = .5 Sqrt[bo]]
        y = 26000 (* psi, design seating stress for metal seal, Table 2-5.1 *)
        H = 0.785 G^2 P (* lb., Total hydrostatic end force *)
        Hp = 2 b × π G m P (* lb., Total joint-contact surface compression load *)
        Wm1 = H + Hp (* Minimum required bolt load, for operating *)
        Wm2 = π G b y (* Minimum required bolt load, for gasket seating *)

Out[1]= 7.98

Out[2]= 978

Out[3]= 6.5

Out[4]= 0.25

Out[5]= 0.03125

Out[6]= 0.03125

Out[7]= 26000

Out[8]= 48889.4

Out[9]= 9960.59

Out[10]= 58849.9

Out[11]= 20369.3
```

```
(* Flange Design Bolt Load*)
Ab = 0.1406 × 24 (*cross sectional area of 1/2-13 screw*)
SF = 4 (* MEDSS *)
ST = 81000
(*Unbrako -  KS 1216 1/2-13 SHCS, 160, ksi tensile strength; T = -400 °F to 1200 °F
 OR ASTM-A493-95 Grade S30430; 81 ksi tensile strength *)
Sa = ST ÷ SF
Lb = Sa × Ab  (* lb., Max allowable bolt load *)
Am1 = Wm1 / Sa (* in^2,cross-sectional area of bolts under operating condition *)
Am2 = Wm2 / Sa (*  in^2, cross-sectional area of bolts for gasket seating *)
 If[Am1 > Am2, Am = Am1, Am = Am2]
(* in^2, total required cross-sectional area of bolts *)

Wo = Wm1 (* lb., Flange design bolt load, for operating *)
Wg = (Am + Ab) Sa / 2  (* lb., Flange design bolt load, for gasket seating *)
```

Out[12]= 3.3744

Out[13]= 4

Out[14]= 81000

Out[15]= 20250

Out[16]= 68331.6

Out[17]= 2.90617

Out[18]= 1.00589

Out[19]= 2.90617

Out[20]= 58849.9

Out[21]= 63590.8

In[22]:=

(* Flange Moment *)

(* Table 2-6, integral flange *)

$C_b = 9.58$ (* in., bolt circle diameter *)

$g_1 = 0.5$ (* in., hub flange thickness *)

$B = 7.625$ (*in., inside diameter of flange *)

$test = 20 g_1$

$$R = \frac{(C_b - B)}{2} - g_1$$

$h_D =$

$R + 0.5 g_1$ (* in., radial distance from bolt circle to the circle on which h_D acts *)

$$h_G = \frac{(C_b - G)}{2}$$

$$h_T = \frac{(R + g_1 + h_G)}{2}$$

$H_D = 0.785 B^2 P$ (* lb., total hydrostatic force on area inside of flange *)

$M_D = H_D h_D$

$H_T = H - H_D$

(* lb., difference, total hydrostatic end force less H_D *)

$M_T = H_T h_T$

$H_G = W_o - H$ (* lb., gasket load *)

$M_G = H_G h_G$

$M_o = M_D + M_T + M_G$ (* in-lb., total flange moment due to operating conditions *)

$M_g = W_o \frac{(C_b - G)}{2}$ (* in-lb., total flange moment due to gasket seating *)

If[$M_o > M_g$, "operating conditions control", "gasket seating conditions control"]

If[$M_o > M_g$, $M_o = M_o$, $M_o = M_g$]

Out[22]= 9.58

Out[23]= 0.5

Out[24]= 7.625

Out[25]= 10.

Out[26]= 0.4775

Out[27]= 0.7275

Out[28]= 0.8

Out[29]= 0.88875

Out[30]= 44636.3

Out[31]= 32472.9

Out[32]= 4253.05

Out[33]= 3779.9

```
Out[120]= 9960.59

Out[121]= 7968.47

Out[122]= 44221.3

Out[123]= 47080.

Out[124]= gasket seating conditions control

Out[125]= 47080.

In[126]:= (*Flange Stress *)

          ε = 1 (*hub stress correction factor*)
          t = 1.25 (* in., flange thickness *)
          h = 0.125 (* in., hub length *)
          te = 2 g1
          A = 10.5 (* in., OD of flange *)
          K = A / B
```

$$T = \frac{K^2 (1 + 8.55246 \text{Log}[10, K]) - 1}{(1.04720 + 1.9448 K^2) (K - 1)} \text{ (* factor, Fig. 2-7.1*)}$$

$$U = \frac{K^2 (1 + 8.55246 \text{Log}[10, K]) - 1}{1.36136 (K^2 - 1) (K - 1)} \text{ (* factor, Fig. 2-7.1*)}$$

$$Y = \frac{1}{K - 1} \left(0.66845 + 5.71690 \frac{K^2 \text{Log}[10, K]}{(K^2 - 1)}\right) \text{(* factor, Fig. 2-7.1*)}$$

$$Z = \frac{K^2 + 1}{K^2 - 1} \text{ (* factor, Fig. 2-7.1*)}$$

$$g_o = g_1$$

$$g_1 / g_o$$

$$h_o = \sqrt{B\, g_o}$$

$$h / h_o$$

$$V = 0.550103 \text{ (* Fig. 2-7.3 Integral flange factor *)}$$

$$d_f = \frac{U}{V} h_o\, g_o^2$$

$$L = \frac{t_e + 1}{T} + \frac{t^3}{d_f}$$

$$S_H = \frac{\epsilon M_o}{L\, g_1^2\, B} \text{(* psi, Longitudinal hub stress *)}$$

$$S_R = \frac{(1.33\, t_e + 1) M_o}{L\, t^2\, B} \text{ (* psi, Radial flange stress *)}$$

(* psi, Tangental flange stress *)

$$S_T = \frac{Y M_o}{t^2 B} - Z\, S_R$$

```
Out[126]= 1

Out[127]= 1.25

Out[128]= 0.125

Out[129]= 1.

Out[130]= 10.5
```

Out[131]= 1.37705

Out[132]= 1.7642

Out[133]= 6.84641

Out[134]= 6.23025

Out[135]= 3.23148

Out[136]= 0.5

Out[137]= 1.

Out[138]= 1.95256

Out[139]= 0.0640184

Out[140]= 0.550103

Out[141]= 6.07525

Out[142]= 1.45515

Out[143]= 16972.6

Out[144]= 6327.39

Out[145]= 4172.77

In[146]:=

```
(* Allowable Flange Stress *)

Sf = 16700 (* allowable stress for 316 L -20 to 100 °F, Table 1 A, Section II *)
 If[SH < 1.5 Sf, "hub stress OK", "hub stress too large"]
 If[SR < Sf, "radial stress OK", "radial stress too large"]
 If[ST < Sf, "tangental stress OK", "tangental stress too large"]
 If[(SH + SR)/2 < Sf, "average stress1 OK", "average stress1 too large"]
 If[(SH + ST)/2 < Sf, "average stress2 OK", "average stress2 too large"]
```

Out[146]= 16700

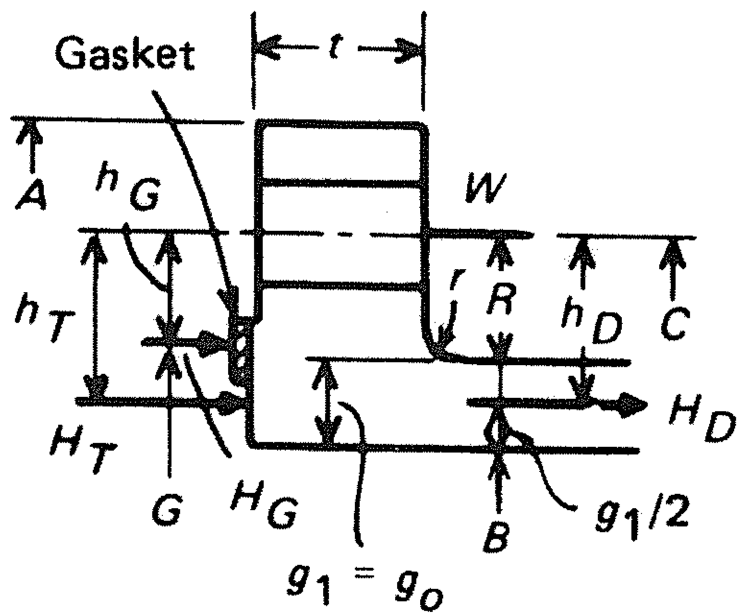
Out[147]= hub stress OK

Out[148]= radial stress OK

Out[149]= tangental stress OK

Out[150]= average stress1 OK

Out[151]= average stress2 OK



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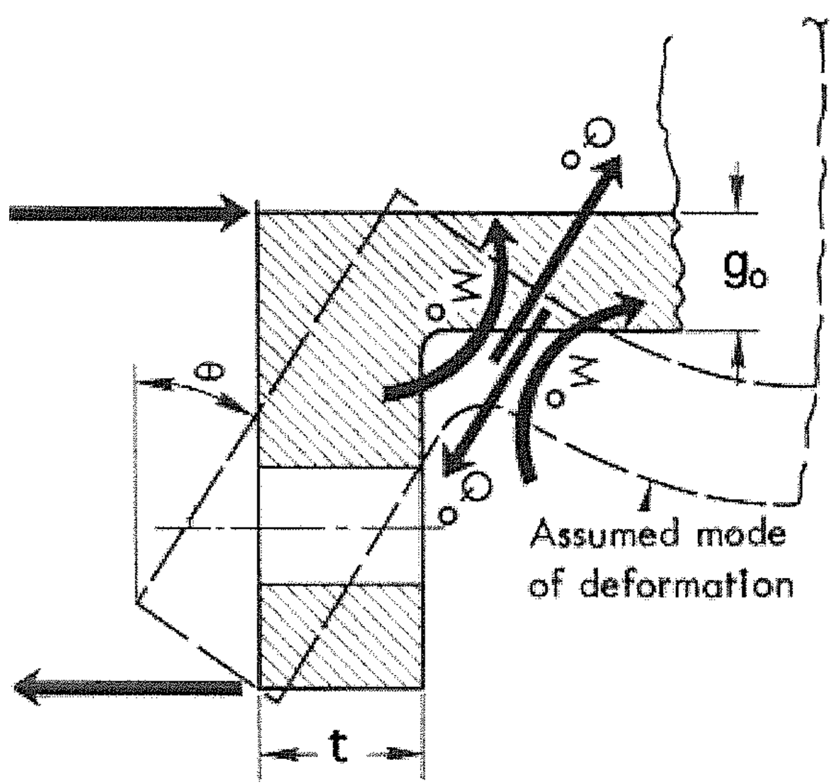



```
(* Bolted Flange Connections
  with flat metal Copper Gasket, Xe chamber Detector Port *)
(* Integral Flange Type, Appendix 2, Figure 2-4 (5) shown above *)
(* 850 psia MOP, conflat type head *)
```

In[249]:= (* Bolt Load at operating conditions *)

$G = 3.35$ (* Diameter, in. at gasket load location *)
$P = 978$ (* MAWP, internal design pressure *)
$m = 4.75$ (* gasket factor flat Cu gasket, Table 2-4.1 *)
$N_g = 0.5$ (* width of Cu gasket *)
$b_o = \frac{N_g}{32}$ (* N/4 for multiple serrations Table 2-5.2 (5),
assume N/32 given a single knife edge serration as used in Conflats *)
$\text{If}\left[b_o <= 0.25,\ b = b_o,\ b = .5\sqrt{b_o}\right]$
$y = 13000$ (* psi, design seating stress for soft copper, Table 2-5.1 *)
$H = 0.785\, G^2\, P$ (* lb., Total hydrostatic end force *)
$H_p = 2\, b \times \pi\, G\, m\, P$ (* lb., Total joint-contact surface compression load *)
$W_{m1} = H + H_p$ (* Minimum required bolt load, for operating *)
$W_{m2} = \pi\, G\, b\, y$ (* Minimum required bolt load, for gasket seating *)

Out[249]= 3.35

Out[250]= 978

Out[251]= 4.75

Out[252]= 0.5

Out[253]= 0.015625

Out[254]= 0.015625

Out[255]= 13000

Out[256]= 8615.85

Out[257]= 1527.84

Out[258]= 10143.7

Out[259]= 2137.76

```
In[12]:= (* Flange Design Bolt Load*)
         Ab = 0.0519 × 10 (*cross sectional area of 5/16-24 screw*)
         SF = 4 (* MEDSS *)
         ST = 81000
         (*Unbrako -  KS 1216 5/16-24 SHCS, 160, ksi tensile strength; T = -400 °F to 1200 °F
          OR ASTM-A493-95 Grade S30430; 81 ksi tensile strength *)
         Sa = ST ÷ SF
         Lb = Sa × Ab  (* lb., Max allowable bolt load *)
         Am1 = Wm1 / Sa (* in^2,cross-sectional area of bolts under operating condition *)
         Am2 = Wm2 / Sa (*  in^2, cross-sectional area of bolts for gasket seating *)
          If[Am1 > Am2, Am = Am1, Am = Am2]
         (* in^2, total required cross-sectional area of bolts *)

         Wo = Wm1 (* lb., Flange design bolt load, for operating *)
         Wg = (Am + Ab) Sa / 2  (* lb., Flange design bolt load, for gasket seating *)

Out[12]= 0.519

Out[13]= 4

Out[14]= 81000

Out[15]= 20250

Out[16]= 10509.7

Out[17]= 0.500923

Out[18]= 0.105568

Out[19]= 0.500923

Out[20]= 10143.7

Out[21]= 10326.7
```

In[332]:=

```
(* Flange Moment *)
(* Table 2-6, integral flange *)
```

$C_b = 4.030$ (* in., bolt circle diameter *)

$g_1 = 0.275$ (* in., hub flange thickness *)

$B = 2.32$ (*in., inside diameter of flange *)

$test = 20\, g_1$ (* Refer to Appx 2, 2-3 notations, for design options *)

$$R = \frac{(C_b - B)}{2} - g_1$$

$h_D =$

$R + 0.5\, g_1$ (* in., radial distance from bolt circle to the circle on which h_D acts *)

$$h_G = \frac{(C_b - G)}{2}$$

$$h_T = \frac{(R + g_1 + h_G)}{2}$$

$H_D = 0.785\, B^2\, P$ (* lb., total hydrostatic force on area inside of flange *)

$M_D = H_D\, h_D$

$H_T = H - H_D$

(* lb., difference, total hydrostatic end force less H_D *)

$M_T = H_T\, h_T$

$H_G = W_o - H$ (* lb., gasket load *)

$M_G = H_G\, h_G$

$M_o = M_D + M_T + M_G$ (* in-lb., total flange moment due to operating conditions *)

$M_g = W_o \frac{(C_b - G)}{2}$ (* in-lb., total flange moment due to gasket seating *)

If[$M_o > M_g$, "operating conditions control", "gasket seating conditions control"]

If[$M_o > M_g$, $M_o = M_o$, $M_o = M_g$]

Out[332]= 4.03

Out[333]= 0.275

Out[334]= 2.32

Out[335]= 5.5

Out[336]= 0.58

Out[337]= 0.7175

Out[338]= 0.34

Out[339]= 0.5975

Out[340]= 4132.23

Out[341]= 2964.87

Out[342]= 4483.62

Out[343]= 2678.96

Out[185]= 1527.84

Out[186]= 519.465

Out[187]= 6163.3

Out[188]= 3448.85

Out[189]= operating conditions control

Out[190]= 6163.3

In[191]:= **(*Flange Stress *)**

ϵ = 1 (*hub stress correction factor*)
t = 0.81 (* in., flange thickness *)
h = 0.0 (* in., hub length *)
$t_e = 2\, g_1$
A = 4.63 (* in., OD of flange *)
K = A / B

$$T = \frac{K^2\ (1 + 8.55246\,\mathrm{Log}[10,\ K]) - 1}{(1.04720 + 1.9448\,K^2)\ (K - 1)}$$ (* factor, Fig. 2-7.1*)

$$U = \frac{K^2\ (1 + 8.55246\,\mathrm{Log}[10,\ K]) - 1}{1.36136\ (K^2 - 1)\ (K - 1)}$$ (* factor, Fig. 2-7.1*)

$$Y = \frac{1}{K - 1}\left(0.66845 + 5.71690\ \frac{K^2\,\mathrm{Log}[10,\ K]}{(K^2 - 1)}\right)$$ (* factor, Fig. 2-7.1*)

$$Z = \frac{K^2 + 1}{K^2 - 1}$$ (* factor, Fig. 2-7.1*)

$g_o = g_1$
g_1 / g_o
$h_o = \sqrt{B\, g_o}$
h / h_o
V = 0.550103 (* Fig. 2-7.3 Integral flange factor *)

$$d_f = \frac{U}{V}\, h_o\, g_o{}^2$$

$$L = \frac{t_e + 1}{T} + \frac{t^3}{d_f}$$

$$S_H = \frac{\epsilon\, M_o}{L\, g_1{}^2\, B}$$ (* psi, Longitudinal hub stress *)

$$S_R = \frac{(1.33\, t_e + 1)\, M_o}{L\, t^2\, B}$$ (* psi, Radial flange stress *)

(* psi, Tangental flange stress *)

$$S_T = \frac{Y\, M_o}{t^2\, B} - Z\, S_R$$

Out[191]= 1

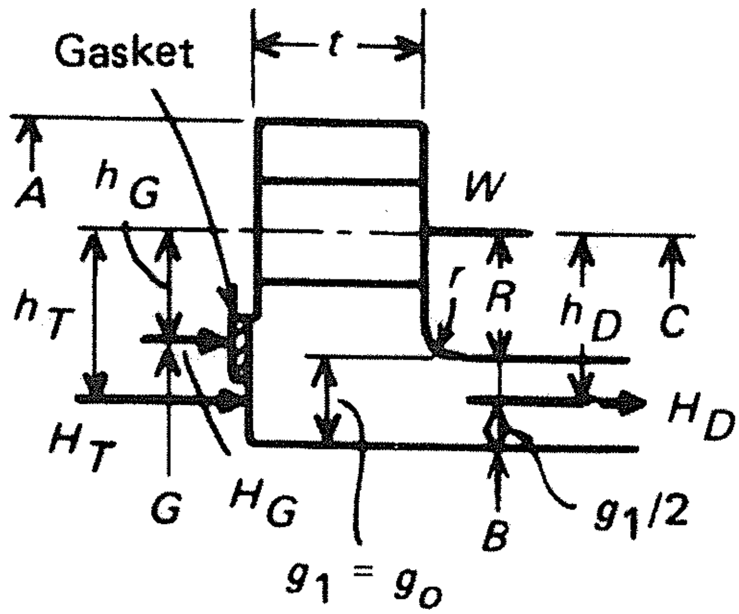
Out[192]= 0.81

Out[193]= 0.

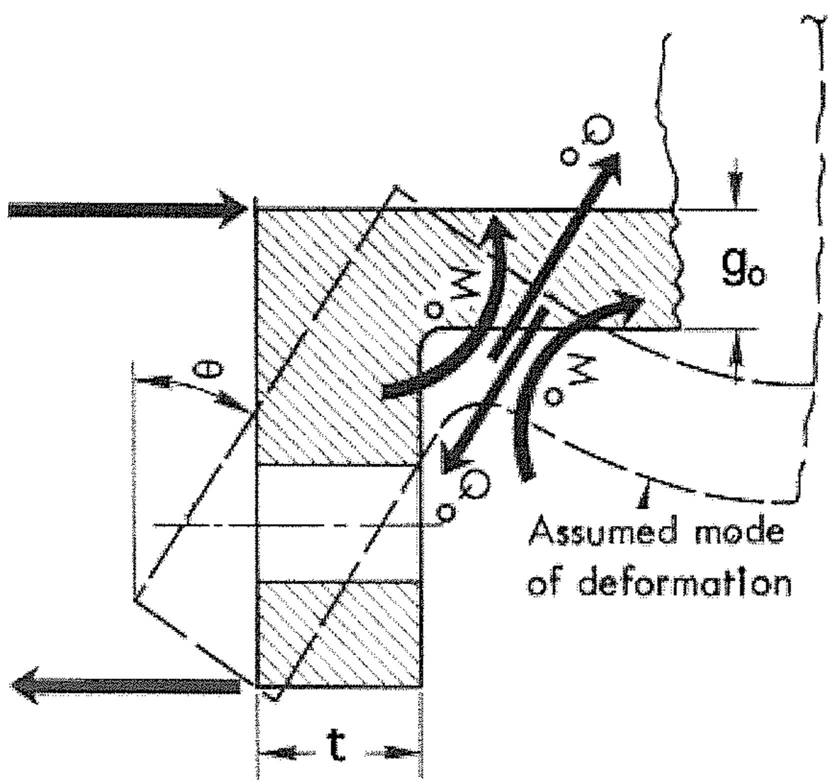
Out[194]= 0.55

Out[195]= 4.63

```
Out[196]= 1.99569

Out[197]= 1.50825

Out[198]= 3.26596

Out[199]= 2.97203

Out[200]= 1.67052

Out[201]= 0.275

Out[202]= 1.

Out[203]= 0.798749

Out[204]= 0.

Out[205]= 0.550103

Out[206]= 0.358627

Out[207]= 2.50956

Out[208]= 13997.9

Out[209]= 2793.7

Out[210]= 7367.04

In[211]:=
    (* Allowable Flange Stress *)

    Sf = 16700 (* allowable stress for 304 L -20 to 100 °F, Table 1 A, Section II *)
     If[SH < 1.5 Sf, "hub stress OK", "hub stress too large"]
     If[SR < Sf, "radial stress OK", "radial stress too large"]
     If[ST < Sf, "tangental stress OK", "tangental stress too large"]
     If[(SH + SR)/2 < Sf, "average stress1 OK", "average stress1 too large"]
     If[(SH + ST)/2 < Sf, "average stress2 OK", "average stress2 too large"]

Out[211]= 16700

Out[212]= hub stress OK

Out[213]= radial stress OK

Out[214]= tangental stress OK

Out[215]= average stress1 OK

Out[216]= average stress2 OK
```



(5)




```
(* Bolted Flange Connections with flat metal Copper Gasket *)
(* Integral Flange Type, Appendix 2, Figure 2-4 (5) shown above *)
(* 350 psia MOP, conflat type head *)
```

```
In[184]:= (* Bolt Load at operating conditions *)

          G = 8.54 (* Diameter, in. at gasket load location *)
          P = 403  (* MAWP, internal design pressure *)
          m = 4.75 (* gasket factor flat Cu gasket, Table 2-4.1 *)
          N_g = 0.5 (* width of Cu gasket *)
          b_o = N_g/32 (* N/4 for multiple serrations Table 2-5.2 (5),
           assume N/32 given a single knife edge serration as used in Conflats *)
          b = b_o
          y = 13000 (* psi, design seating stress for soft copper, Table 2-5.1 *)
          H = 0.785 G^2 P (* lb., Total hydrostatic end force *)
          H_p = 2 b × π G m P (* lb., Total joint-contact surface compression load *)
          W_m1 = H + H_p (* Minimum required bolt load, for operating *)
          W_m2 = π G b y (* Minimum required bolt load, for gasket seating *)

Out[184]= 8.54

Out[185]= 403

Out[186]= 4.75

Out[187]= 0.5

Out[188]= 0.015625

Out[189]= 0.015625

Out[190]= 13000

Out[191]= 23072.3

Out[192]= 1604.93

Out[193]= 24677.2

Out[194]= 5449.68
```

```
In[33]:= (* Flange Design Bolt Load*)
         A_b = 0.1406 × 24 (*cross sectional area of 1/2-13 screw*)
         SF = 4 (* MEDSS *)
         S_T = 81000
         (*Unbrako -  KS 1216 1/2-13 SHCS, 160, ksi tensile strength; T = -400 °F to 1200 °F
          OR ASTM-A493-95 Grade S30430; 81 ksi tensile strength *)
         S_a = S_T ÷ SF
         L_b = S_a × A_b  (* lb., Max allowable bolt load *)
         A_m1 = W_m1 / S_a (* in^2,cross-sectional area of bolts under operating condition *)
         A_m2 = W_m2 / S_a (*  in^2, cross-sectional area of bolts for gasket seating *)
          If[A_m1 > A_m2, A_m = A_m1, A_m = A_m2]
         (* in^2, total required cross-sectional area of bolts *)

         W_o = W_m1 (* lb., Flange design bolt load, for operating *)
         W_g = (A_m + A_b) S_a / 2  (* lb., Flange design bolt load, for gasket seating *)

Out[33]= 3.3744

Out[34]= 4

Out[35]= 81000

Out[36]= 20250

Out[37]= 68331.6

Out[38]= 1.21863

Out[39]= 0.26912

Out[40]= 1.21863

Out[41]= 24677.2

Out[42]= 46504.4
```

```
In[205]:=
        (* Flange Moment *)
        (* Table 2-6, integral flange *)
        Cb = 9.58 (* in., bolt circle diameter *)
        g1 = 0.5 (* in., hub flange thickness *)
        B = 7.625 (*in., inside diameter of flange *)
        test = 20 g1
        R = (Cb - B)/2 - g1
        hD =
         R + 0.5 g1 (* in., radial distance from bolt circle to the circle on which hD acts *)
        hG = (Cb - G)/2
        hT = (R + g1 + hG)/2
        HD = 0.785 B^2 P (* lb., total hydrostatic force on area inside of flange *)
         MD = HD hD

        HT = H - HD
        (* lb., difference, total hydrostatic end force less HD *)
         MT = HT hT

        HG = Wo - H (* lb., gasket load *)
         MG = HG hG

        Mo = MD + MT + MG (* in-lb., total flange moment due to operating conditions *)

        Mg = Wo (Cb - G)/2 (* in-lb., total flange moment due to gasket seating *)
         If[Mo > Mg, "operating conditions control", "gasket seating conditions control"]
         If[Mo > Mg, Mo = Mo, Mo = Mg]

Out[205]= 9.58

Out[206]= 0.5

Out[207]= 7.625

Out[208]= 10.

Out[209]= 0.4775

Out[210]= 0.7275

Out[211]= 0.52

Out[212]= 0.74875

Out[213]= 18393.1

Out[214]= 13381.

Out[215]= 4679.2

Out[216]= 3503.55
```

Out[250]= 1604.93

Out[251]= 834.564

Out[252]= 17719.1

Out[253]= 12832.1

Out[254]= operating conditions control

Out[255]= 17719.1

In[256]:= (*Flange Stress *)

ϵ = 1 (*hub stress correction factor*)
t = 1.25 (* in., flange thickness *)
h = 0.125 (* in., hub length *)
$t_e = 2\, g_1$
A = 10.5 (* in., OD of flange *)
K = A / B

$$T = \frac{K^2\ (1 + 8.55246\ \mathrm{Log}[10,\ K]) - 1}{(1.04720 + 1.9448\, K^2)\ (K - 1)} \text{ (* factor, Fig. 2-7.1*)}$$

$$U = \frac{K^2\ (1 + 8.55246\ \mathrm{Log}[10,\ K]) - 1}{1.36136\ (K^2 - 1)\ (K - 1)} \text{ (* factor, Fig. 2-7.1*)}$$

$$Y = \frac{1}{K - 1}\left(0.66845 + 5.71690\ \frac{K^2\ \mathrm{Log}[10,\ K]}{(K^2 - 1)}\right) \text{ (* factor, Fig. 2-7.1*)}$$

$$Z = \frac{K^2 + 1}{K^2 - 1} \text{ (* factor, Fig. 2-7.1*)}$$

$g_o = g_1$
g_1 / g_o
$h_o = \sqrt{B\, g_o}$
h / h_o
V = 0.550103 (* Fig. 2-7.3 Integral flange factor *)

$$d_f = \frac{U}{V}\, h_o\, g_o^{\,2}$$

$$L = \frac{t_e + 1}{T} + \frac{t^3}{d_f}$$

$$S_H = \frac{\epsilon\, M_o}{L\, g_1^{\,2}\, B} \text{ (* psi, Longitudinal hub stress *)}$$

$$S_R = \frac{(1.33\, t_e + 1)\, M_o}{L\, t^2\, B} \text{ (* psi, Radial flange stress *)}$$

(* psi, Tangental flange stress *)

$$S_T = \frac{Y\, M_o}{t^2\, B} - Z\, S_R$$

Out[256]= 1

Out[257]= 1.25

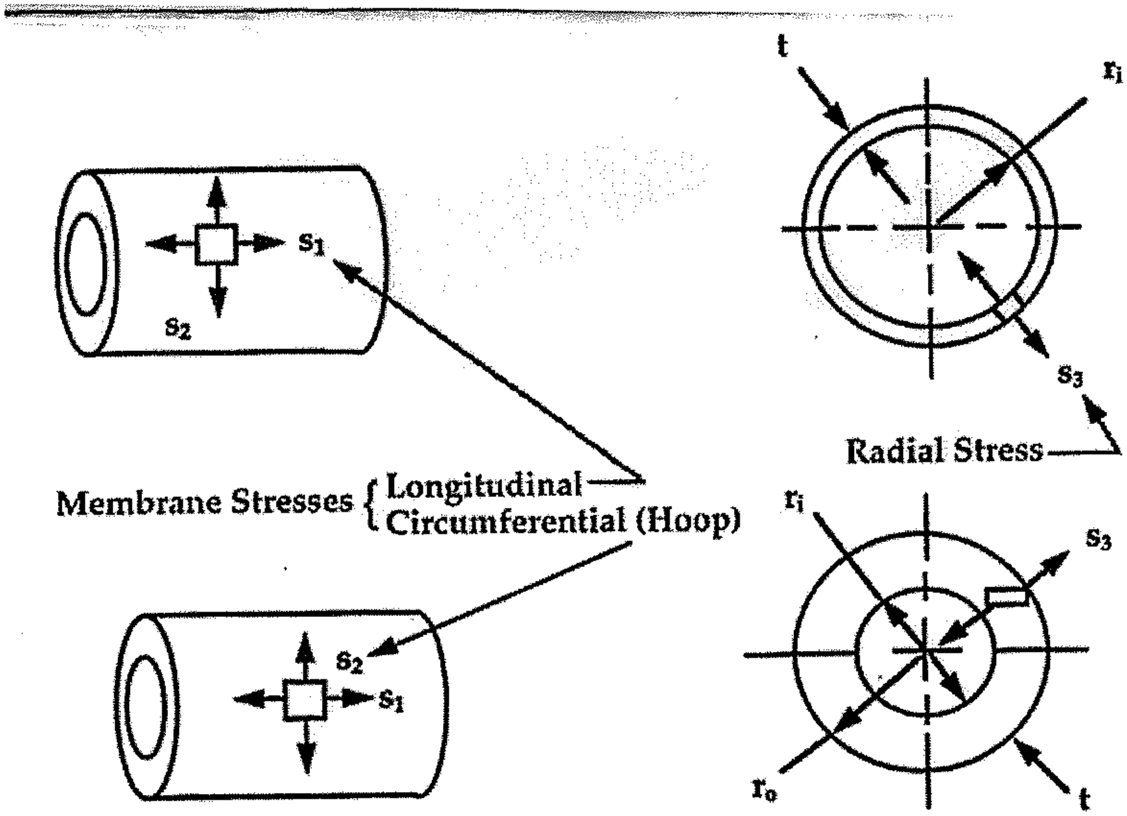
Out[258]= 0.125

Out[259]= 1.

Out[260]= 10.5

```
Out[261]= 1.37705

Out[262]= 1.7642

Out[263]= 6.84641

Out[264]= 6.23025

Out[265]= 3.23148

Out[266]= 0.5

Out[267]= 1.

Out[268]= 1.95256

Out[269]= 0.0640184

Out[270]= 0.550103

Out[271]= 6.07525

Out[272]= 1.45515

Out[273]= 6387.84

Out[274]= 2381.39

Out[275]= 1570.47

In[276]:=
        (* Allowable Flange Stress *)

        Sf = 16700 (* allowable stress for 316 L -20 to 100 °F, Table 1 A, Section II *)
         If[SH < 1.5 Sf, "hub stress OK", "hub stress too large"]
         If[SR < Sf, "radial stress OK", "radial stress too large"]
         If[ST < Sf, "tangental stress OK", "tangental stress too large"]
         If[(SH + SR)/2 < Sf, "average stress1 OK", "average stress1 too large"]
         If[(SH + ST)/2 < Sf, "average stress2 OK", "average stress2 too large"]

Out[276]= 16700

Out[277]= hub stress OK

Out[278]= radial stress OK

Out[279]= tangental stress OK

Out[280]= average stress1 OK

Out[281]= average stress2 OK
```

(*Xenon Pressure Vessel Stress Calculations — Detector Port*)



In[376]:=

```
MAWP = 978
σa = 16700 (*allowable stress for 316 L SST*)
σy = 37000
Ri = 1.1615
Ro = 1.4375
t = Ro - Ri
Ratio = Ro/Ri
If [1.1 < Ratio < 1.5, medium wall]
If [Ratio < 1.1, thin wall]
If [Ratio > 1.5, thick wall]
```

Out[376]= 978

Out[377]= 16700

Out[378]= 37000

Out[379]= 1.1615

Out[380]= 1.4375

Out[381]= 0.276

Out[382]= 1.23762

Out[383]= medium wall

In[386]:=

```
(*Longitudinal Stress, S1*)
```

$$S_1 = \frac{(MAWP\ R_i^2)}{(R_o^2 - R_i^2)}$$

```
(*Circumferential Stress, S2*)
```

$$S_2 = \frac{MAWP\ (R_o^2 + R_i^2)}{(R_o^2 - R_i^2)}$$

```
(*Radial Stress, S3*)
```

$$S_3 = -MAWP$$

```
(*Von Mises Stress*)
```

$$\sigma_m = \sqrt{0.5\ ((S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2)}$$

Out[386]= 1839.34

Out[387]= 4656.68

Out[388]= -978

Out[389]= 4879.78

In[42]:=

```
(*wall thickness, in., max. pressure, psi*)

(*Circumferential Stress: wall thickness, in., max. pressure, psi*)
```

$E_f = 0.7$ (*butt weld efficiency based on no inspection, Table UW-12*)

$t_c = \frac{(MAWP\ R_i)}{(\sigma_a E_f - 0.6\, MAWP)}$ (*UG27 c 1*)

$P_c = \frac{(\sigma_a E_f\ t)}{(R_i + 0.6\, t)}$ (*UG27 c 1*)

$SF_{uc} = \frac{P_c}{MAWP}$ (* P_c uses allowable stress so SF ~5 is also inlcluded*)

(*Longitudinal Stress: wall thickness, in., max. pressure, psi*)
(*Circumferential butt welds connecting
ellipsoidal head and hub to cylinder are Catagory A, Type 1 welds*)

$E_f = 0.7$ (*butt weld efficiency based on no inspection, Table UW-12*)

$t_l = \frac{(MAWP\ R_i)}{(2\,\sigma_a E_f + 0.4\, MAWP)}$ (*UG27 c 2*)

$P_l = \frac{(2\,\sigma_a E_f\ t)}{(R_i - 0.4\, t)}$ (*UG27 c 2*)

$SF_{ul} = \frac{P_l}{MAWP}$ (* P_l uses allowable stress so SF ~5 is also inlcluded*)

If[$P_c < P_l$, "circumferential stress applies", "longitudinal stress applies"]
If[$t_c > t_l$, "circumferential stress applies", "longitudinal stress applies"]

Out[42]= 0.7

Out[43]= 0.102308

Out[44]= 2431.2

Out[45]= 2.48589

Out[46]= 0.7

Out[47]= 0.0477867

Out[48]= 6139.17

Out[49]= 6.27727

Out[50]= circumferential stress applies

Out[51]= circumferential stress applies

In[400]:=

(*Check of Von Mises stress at 1.5 × MAWP for pressure test*)

MAWP = 1.5 × 978

(*Longitudinal Stress, S_1*)

$$S_1 = \frac{(MAWP\ R_i^2)}{(R_o^2 - R_i^2)}$$

(*Circumferential Stress, S_2*)

$$S_2 = \frac{MAWP\ (R_o^2 + R_i^2)}{(R_o^2 - R_i^2)}$$

(*Radial Stress, S_3*)

S_3 = -MAWP

(*Von Mises Stress*)

$$\sigma_m = \sqrt{0.5\ ((S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2)}$$

$$N_r = \frac{\sigma_y}{\sigma_m}$$

If[N_r > 1, "vessel OK at 1.5 × MAWP during pressure test"]

Out[400]= 1467.

Out[401]= 2759.01

Out[402]= 6985.02

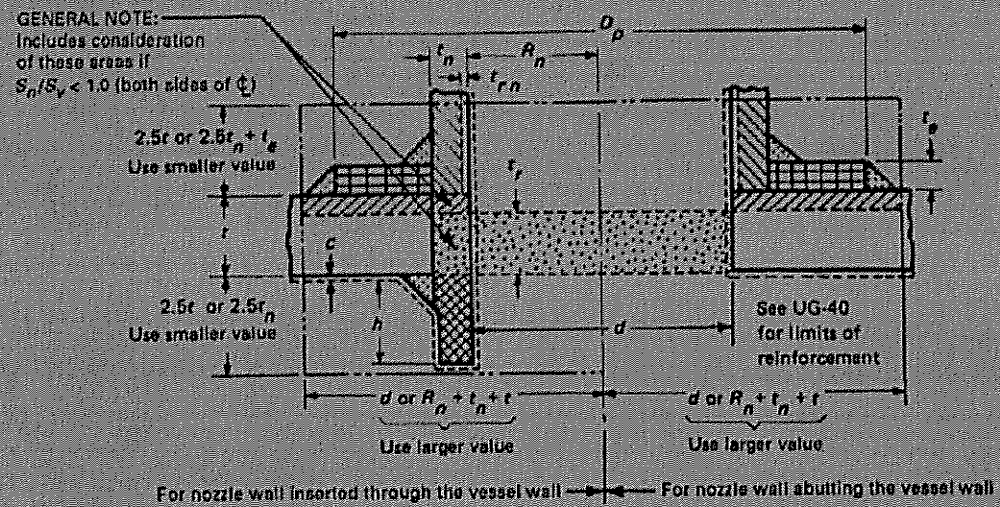
Out[403]= -1467.

Out[404]= 7319.66

Out[405]= 5.05488

Out[406]= vessel OK at 1.5 × MAWP during pressure test

PART UG — GENERAL REQUIREMENTS Fig. UG-37.1



Without Reinforcing Element

$A = d t_r F + 2t_n t_r F(1 - f_{r1})$	Area required
$A_1 = d(E_1 t - F t_r) - 2t_n(E_1 t - F t_r)(1 - f_{r1})$ or $= 2(t + t_n)(E_1 t - F t_r) - 2t_n(E_1 t - F t_r)(1 - f_{r1})$	Area available in shell; use larger value
$A_2 = 5(t_n - t_{rn}) f_{r2} t$ or $= 5(t_n - t_{rn}) f_{r2} t_n$	Area available in nozzle projecting outward; use smaller value
$A_3 = 2(t_n - c) f_{r2} h$	Area available in inward nozzle
A_{41} = outward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in outward weld
A_{43} = inward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in inward weld
If $A_1 + A_2 + A_3 + A_{41} + A_{43} \geq A$	Opening is adequately reinforced
If $A_1 + A_2 + A_3 + A_{41} + A_{43} < A$	Opening is not adequately reinforced so reinforcing elements must be added and/or thicknesses must be increased

With Reinforcing Element Added

A = same as A, above	Area required
A_1 = same as A_1, above	Area available
$A_2 = 5(t_n - t_{rn}) f_{r2} t$ or $= 2(t_n - t_{rn})(2.5t_n + t_e) f_{r2}$	Area available in nozzle projecting outward; use smaller area
A_3 = same as A_3, above	Area available in inward nozzle
A_{41} = outward nozzle weld = $(\text{leg})^2 f_{r3}$	Area available in outward weld
A_{42} = outer element weld = $(\text{leg})^2 f_{r4}$	Area available in outer weld
A_{43} = inward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in inward weld
$A_5 = (D_p - d - 2t_n) t_e f_{r4}$ [Note (1)]	Area available in element
If $A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 \geq A$	Opening is adequately reinforced

NOTE:
(1) This formula is applicable for a rectangular cross-sectional element that falls within the limits of reinforcement.

FIG. UG-37.1 NOMENCLATURE AND FORMULAS FOR REINFORCED OPENINGS
(This Figure Illustrates a Common Nozzle Configuration and Is Not Intended to Prohibit Other Configurations Permitted by the Code.)

```
        (*Opening reinforcement calculations*)
        (* Detector pipe to shell wall*)
        (* A3 = 0, A5 = 0, A42 = 0 *)

        (* Sch 80 Pipe, 3 " Ø *)
        Tx = 2.875 (*OD*)
        d = 2.323 (*ID min*)
        tn = (Tx - d) / 2 (*nozzel wall thickness*)
        te = 1.25 * tn (*weld leg height*)

Out[445]= 2.875

Out[446]= 2.323

Out[447]= 0.276

Out[448]= 0.345

In[473]:= F = 1 (*correction factor*)
          tr = 0.335815 (*minimum shell thickness, Vessel_stress2.nb*)
          fr1 = 1 (*strength reducton factor*)
          t = 0.5 (*shell wall thickness*)
          E1 = 1 (*joint efficiency*)
          A = d tr F + 2 tn tr F (1 - fr1)
           A1a = d (E1 t - F tr) - 2 tn (E1 t - F tr) (1 - fr1)
           A1b = 2 (t + tn) (E1 t - F tr) - 2 tn (E1 t - F tr) (1 - fr1)
             If[A1a > A1b, A1 = A1a, A1 = A1b]

Out[473]= 1

Out[474]= 0.335815

Out[475]= 1

Out[476]= 0.5

Out[477]= 1

Out[478]= 0.780098

Out[479]= 0.381402

Out[480]= 0.254815

Out[481]= 0.381402
```

```
In[482]:= fr2 = 1 (*strength reducton factor*)
          trn = 0.10230807 (*requierd nozzel thickness, Xe_vessel _det.nb*)
          A2a = 5 (tn - trn) fr2 t
          A2b = 5 (tn - trn) fr2 tn
          If[A2a < A2b, A2 = A2a, A2 = A2b]

Out[482]= 1

Out[483]= 0.10230807

Out[484]= 0.43423

Out[485]= 0.239695

Out[486]= 0.239695
```

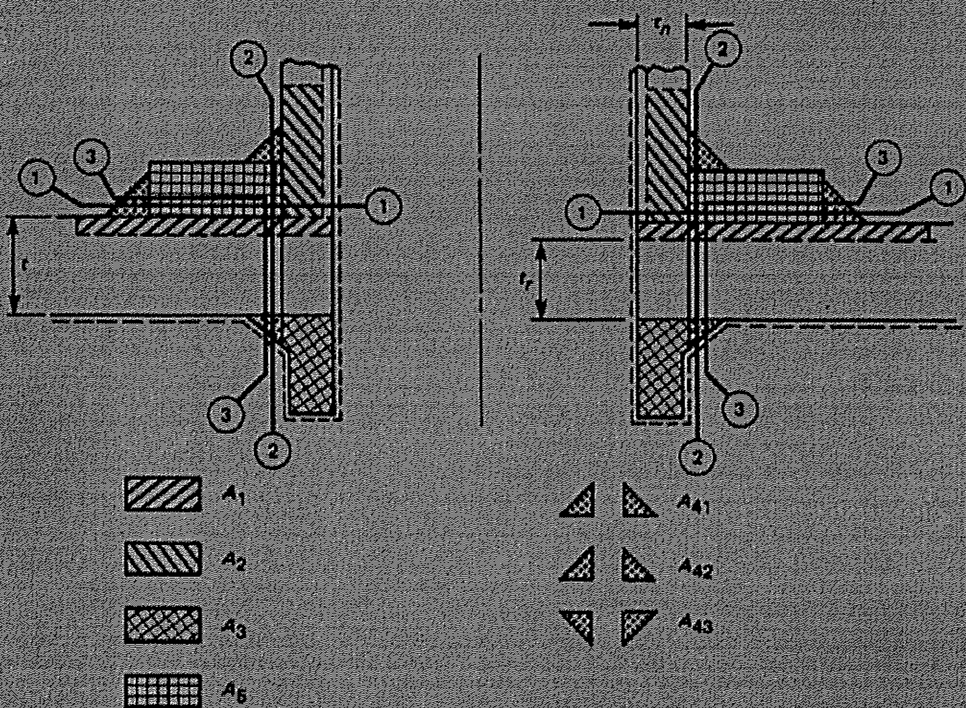
```
fr3 = 1 (*strength reducton factor*)
```

$$A43 = te^2\ fr3$$

$$A41 = \frac{te^2\ fr3}{2}$$ (* 1/2 the area, skip weld on outside*)

```
Out[492]= 1

Out[493]= 0.119025

Out[494]= 0.0595125

In[498]:= (A1 + A2 + A43 + A41)
          A
          (A1 + A2 + A43 + A41) >= A
          (*If actual area > area required, then no additional reinforcement required *)

Out[498]= 0.799634

Out[499]= 0.780098

Out[500]= True
```

Fig. UG-41.1 1995 SECTION VIII — DIVISION 1



W = total weld load [UG-41(b)(2)]
$= [A - A_1 + 2t_n f_{r1}(E_1 t - F t_r)]S_v$
$W_{1\text{-}1}$ = weld load for strength path 1-1 [UG-41(b)(1)]
$= [A_2 + A_5 + A_{41} + A_{42}]S_v$
$W_{2\text{-}2}$ = weld load for strength path 2-2 [UG-41(b)(1)]
$= [A_2 + A_3 + A_{41} + A_{43} + 2t_n t f_{r1}]S_v$
$W_{3\text{-}3}$ = weld load for strength path 3-3 [UG-41(b)(1)]
$= [A_2 + A_3 + A_5 + A_{41} + A_{42} + A_{43} + 2t_n t f_{r1}]S_v$

GENERAL NOTES:
(a) Areas A_1, A_2, A_3, A_5, and A_{41} are modified by f_{rx} factors.
(b) Nomenclature is the same as in UG-37 and Fig. UG-37.1.

(a) Depicts Typical Nozzle Detail With Neck Inserted Through the Vessel Wall

FIG. UG-41.1 NOZZLE ATTACHMENT WELD LOADS AND WELD STRENGTH PATHS TO BE CONSIDERED

```
(*must run "detector_shell.nb" file first to save variables defined below
  into memory*)
```

```
In[523]:= (*Load / Stress  Carried by Welds*)
          A
          A1
          A2
          A3 = 0
          A5 = 0
          A41
          A42 = 0
          A43

Out[523]= 0.780098

Out[524]= 0.381402

Out[525]= 0.239695

Out[526]= 0

Out[527]= 0

Out[528]= 0.0595125

Out[529]= 0

Out[530]= 0.119025

In[533]:= Sv = 16700 (* allowable stress*)
          W = (A - A1 + 2 tn fr1 (E1 t - F tr)) Sv

Out[533]= 16700

Out[534]= 8171.75

In[535]:=
          W1-1 = (A2 + A5 + A41 + A42) Sv

Out[535]= 4996.76

In[536]:= W2-2 = (A2 + A3 + A41 + A43 + 2 tn t fr1) Sv

Out[536]= 11593.7

In[537]:=
          W3-3 = (A2 + A3 + A5 + A41 + A42 + A43 + 2 tn t fr1) Sv

Out[537]= 11593.7

        (* W (total weld load) << W1-1, W2-2, W3-3, (weld load available)*)

In[539]:= (*Allowable Unit Stresses*)
          (*Fillet Weld Shear, UW 15 c*)
          σfw = 0.49 (Sv)

Out[539]= 8183.
```

```
In[540]:= (*Nozzel Wall Shear, UG 45 c*)
          σnw = 0.7 (Sv)

Out[540]= 11690.

In[541]:=
          (*Strength of Connection Elements*)
          (*Fillet Weld Shear*)
```

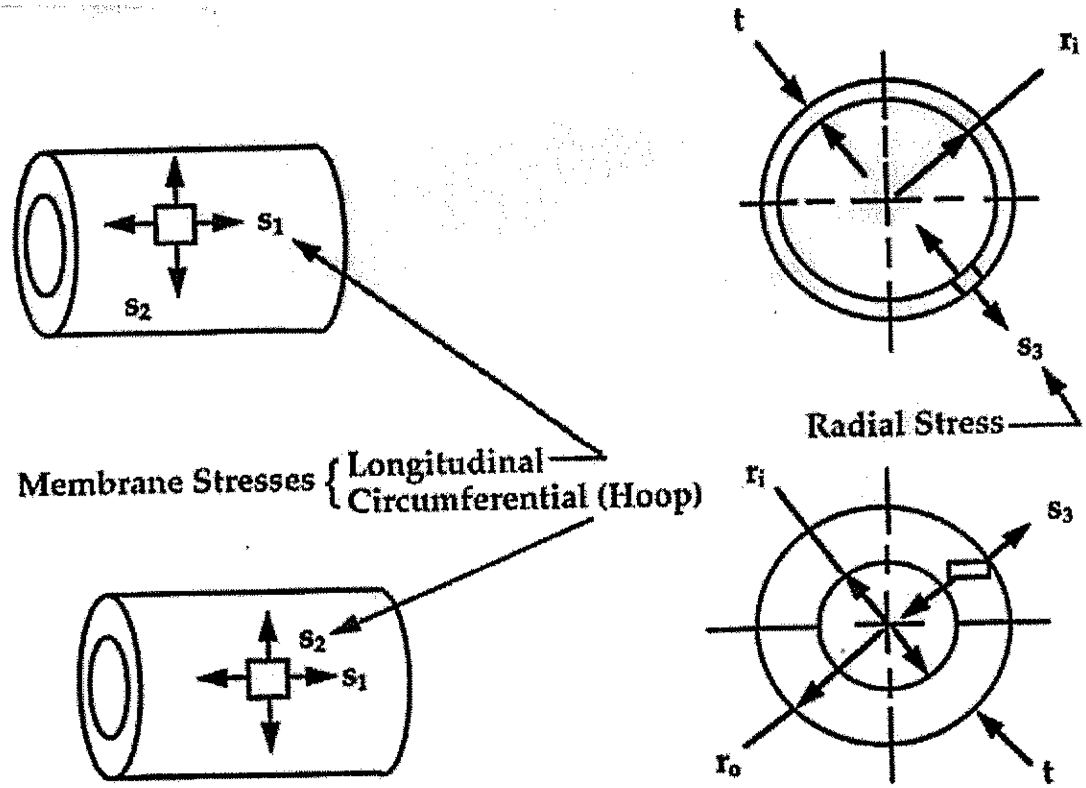
$$W_{fw} = \frac{\pi}{2} \, Tx\, te\, \sigma_{fw}$$

```
Out[541]= 12749.4

In[542]:=
          (*Strength of Connection Elements*)
          (*Nozzel Wall Shear*)
```

$$W_{nw} = \frac{\pi}{2} \, \frac{(Tx + d)}{2} \, tn\, \sigma_{nw}$$

```
Out[542]= 13171.9

In[543]:=
          WS1-1 = Wnw
          WS2-2 = Wfw

Out[543]= 13171.9

Out[544]= 12749.4

      (*All Paths WS1-1, WS2-2, are stronger than the required strength W*)
```

(*Xenon Pressure Vessel Stress Calculations - VCR Port*)



```
In[545]:=
        MAWP = 978
        σa = 16700 (*allowable stress for 316 L SST*)
        σy = 37000
        Ri = 0.40 / 2.
        Ro = 0.5 / 2.
        t = Ro - Ri
        Ratio = Ro / Ri
        If [1.1 < Ratio < 1.5, medium wall]
        If [Ratio < 1.1, thin wall]
        If [Ratio > 1.5, thick wall]

Out[545]= 978

Out[546]= 16700

Out[547]= 37000

Out[548]= 0.2

Out[549]= 0.25

Out[550]= 0.05

Out[551]= 1.25

Out[552]= medium wall
```

In[555]:=

```
(*Longitudinal Stress, S1*)
```

$$S_1 = \frac{(\text{MAWP}\ R_i{}^2)}{(R_o{}^2 - R_i{}^2)}$$

```
(*Circumferential Stress, S2*)
```

$$S_2 = \frac{\text{MAWP}\ (R_o{}^2 + R_i{}^2)}{(R_o{}^2 - R_i{}^2)}$$

```
(*Radial Stress, S3*)
```

$$S_3 = -\text{MAWP}$$

```
(*Von Mises Stress*)
```

$$\sigma_m = \sqrt{0.5\ ((S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2)}$$

Out[555]= 1738.67

Out[556]= 4455.33

Out[557]= -978

Out[558]= 4705.4

In[73]:=

(*wall thickness, in., max. pressure, psi*)

(*Circumferential Stress: wall thickness, in., max. pressure, psi*)

$E_f = 1.0$ (*efficiency*)

$$t_c = \frac{(\text{MAWP}\ R_i)}{(\sigma_a\ E_f - 0.6\ \text{MAWP})}$$ (*UG27 c 1*)

$$P_c = \frac{(\sigma_a\ E_f\ t)}{(R_i + 0.6\ t)}$$ (*UG27 c 1*)

$$SF_{uc} = \frac{P_c}{\text{MAWP}}$$ (* P_c uses allowable stress so SF ~5 is also inlcluded*)

(*Longitudinal Stress: wall thickness, in., max. pressure, psi*)
(*Circumferential butt welds connecting
ellipsoidal head and hub to cylinder are Catagory A, Type 1 welds*)

$E_f = 1.0$ (*efficincy*)

$$t_1 = \frac{(\text{MAWP}\ R_i)}{(2\ \sigma_a\ E_f + 0.4\ \text{MAWP})}$$ (*UG27 c 2*)

$$P_1 = \frac{(2\ \sigma_a\ E_f\ t)}{(R_i - 0.4\ t)}$$ (*UG27 c 2*)

$$SF_{ul} = \frac{P_1}{\text{MAWP}}$$ (* P_1 uses allowable stress so SF ~5 is also inlcluded*)

If[$P_c < P_1$, "circumferential stress applies", "longitudinal stress applies"]
If[$t_c > t_1$, "circumferential stress applies", "longitudinal stress applies"]

Out[73]= 1.

Out[74]= 0.0121391

Out[75]= 3630.43

Out[76]= 3.7121

Out[77]= 1.

Out[78]= 0.00578849

Out[79]= 9277.78

Out[80]= 9.48648

Out[81]= circumferential stress applies

Out[82]= circumferential stress applies

In[569]:=

```
(*Check of Von Mises stress at 1.5 × MAWP for pressure test*)
MAWP = 1.5 × 978

(*Longitudinal Stress, S1*)
```

$$S_1 = \frac{(MAWP\ R_i{}^2)}{(R_o{}^2 - R_i{}^2)}$$

```
(*Circumferential Stress, S2*)
```

$$S_2 = \frac{MAWP\ (R_o{}^2 + R_i{}^2)}{(R_o{}^2 - R_i{}^2)}$$

```
(*Radial Stress, S3*)
```

$$S_3 = -MAWP$$

```
(*Von Mises Stress*)
```

$$\sigma_m = \sqrt{0.5\ ((S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2)}$$

$$N_r = \frac{\sigma_y}{\sigma_m}$$

If[N_r > 1, "vessel OK at 1.5 × MAWP during pressure test"]

Out[569]= 1467.

Out[570]= 2608.

Out[571]= 6683.

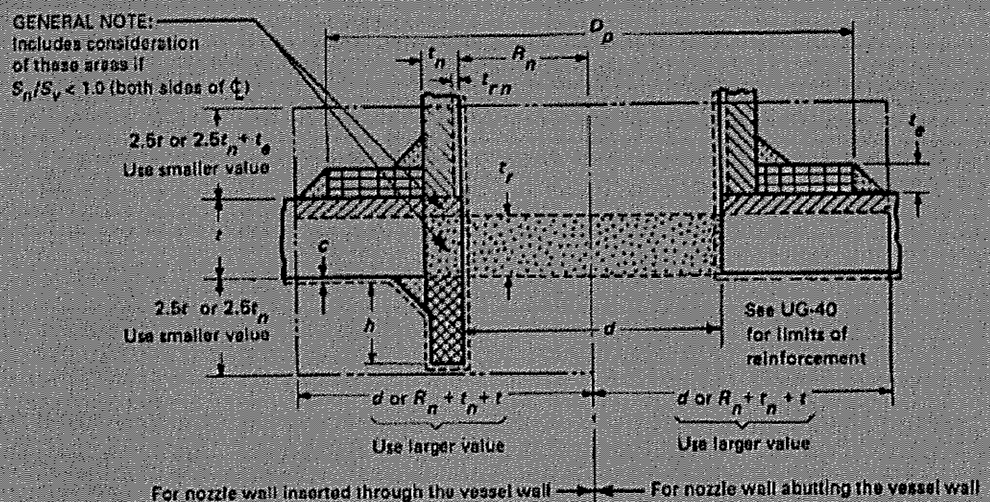
Out[572]= -1467.

Out[573]= 7058.11

Out[574]= 5.2422

Out[575]= vessel OK at 1.5 × MAWP during pressure test

PART UG — GENERAL REQUIREMENTS Fig. UG-37.1



Without Reinforcing Element

$= A = d\,t_r F + 2t_n t_r F(1 - f_{r1})$	Area required
$= A_1 \begin{cases} = d(E_1 t - F t_r) - 2t_n (E_1 t - F t_r)(1 - f_{r1}) \\ = 2(t + t_n)(E_1 t - F t_r) - 2t_n (E_1 t - F t_r)(1 - f_{r1}) \end{cases}$	Area available in shell; use larger value
$= A_2 \begin{cases} = 5(t_n - t_{rn}) f_{r2} t \\ = 5(t_n - t_{rn}) f_{r2} t_n \end{cases}$	Area available in nozzle projecting outward; use smaller value
$= A_3 = 2(t_n - c) f_{r2} h$	Area available in inward nozzle
$= A_{41}$ = outward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in outward weld
$= A_{43}$ = inward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in inward weld
If $A_1 + A_2 + A_3 + A_{41} + A_{43} \geq A$	Opening is adequately reinforced
If $A_1 + A_2 + A_3 + A_{41} + A_{43} < A$	Opening is not adequately reinforced so reinforcing elements must be added and/or thicknesses must be increased

With Reinforcing Element Added

A = same as A, above	Area required
A_1 = same as A_1, above	Area available
$A_2 \begin{cases} = 5(t_n - t_{rn}) f_{r2} t \\ = 2(t_n - t_{rn})(2.5t_n + t_e) f_{r2} \end{cases}$	Area available in nozzle projecting outward; use smaller area
A_3 = same as A_3, above	Area available in inward nozzle
$= A_{41}$ = outward nozzle weld = $(\text{leg})^2 f_{r3}$	Area available in outward weld
$= A_{42}$ = outer element weld = $(\text{leg})^2 f_{r4}$	Area available in outer weld
$= A_{43}$ = inward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in inward weld
$= A_5 = (D_p - d - 2t_n) t_e f_{r4}$ [Note (1)]	Area available in element
If $A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 \geq A$	Opening is adequately reinforced

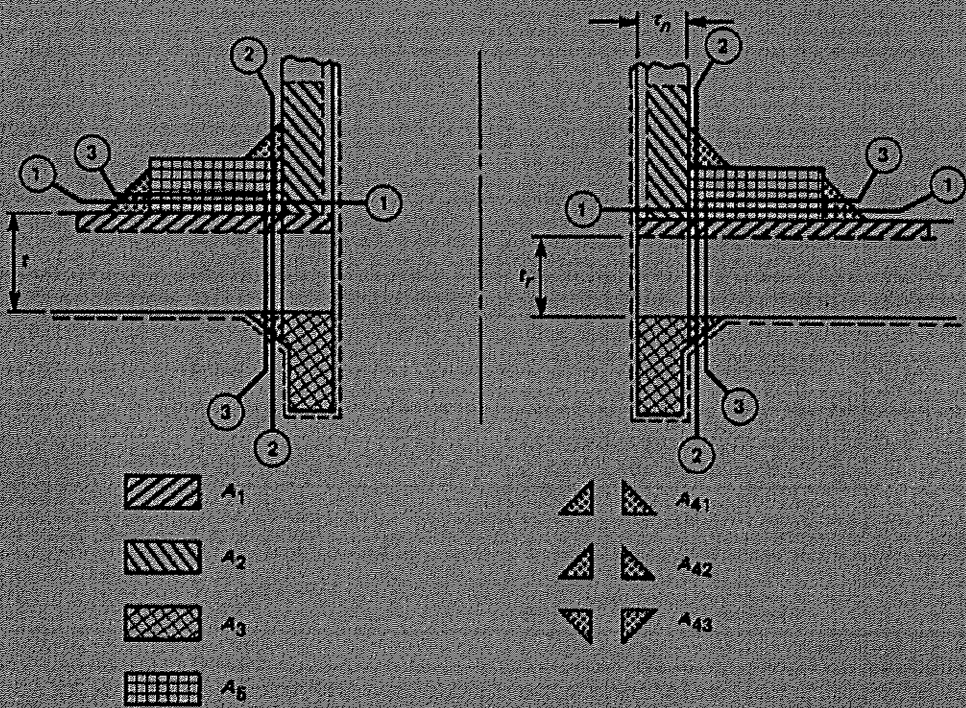
NOTE:
(1) This formula is applicable for a rectangular cross-sectional element that falls within the limits of reinforcement.

FIG. UG-37.1 NOMENCLATURE AND FORMULAS FOR REINFORCED OPENINGS
(This Figure Illustrates a Common Nozzle Configuration and Is Not Intended to Prohibit Other Configurations Permitted by the Code.)

```
In[576]:= (* Opening Reinforcement Calculations*)
          (* VCR Gland to shell wall*)
          (* A3 = 0, A5 = 0, A43 = 0, A42 = 0 *)

          (* VCR, 0.5 " Ø *)
          Tx = .5 (*OD*)
          d = 0.40 (*ID min*)
          tn = (Tx - d) / 2 (*nozzel wall thickness*)
          te = 1.25 * tn (*weld leg height*)

Out[576]= 0.5

Out[577]= 0.4

Out[578]= 0.05

Out[579]= 0.0625

In[580]:= F = 1 (*correction factor*)
          tr = 0.335815 (*minimum shell thickness, Vessel_stress2.nb*)
          fr1 = 1 (*strength reducton factor*)
          t = 0.5 (*shell wall thickness*)
          E1 = 1 (*joint efficiency*)
          A = d tr F + 2 tn tr F (1 - fr1)
           A1a = d (E1 t - F tr) - 2 tn (E1 t - F tr) (1 - fr1)
           A1b = 2 (t + tn) (E1 t - F tr) - 2 tn (E1 t - F tr) (1 - fr1)
             If[A1a > A1b, A1 = A1a, A1 = A1b]

Out[580]= 1

Out[581]= 0.335815

Out[582]= 1

Out[583]= 0.5

Out[584]= 1

Out[585]= 0.134326

Out[586]= 0.065674

Out[587]= 0.180604

Out[588]= 0.180604
```

```
In[589]:= fr2 = 1 (*strength reducton factor*)
          trn = 0.0121391(*required nozzel thickness, Xe_vessel _VCR.nb*)
          A2a = 5 (tn - trn) fr2 t
          A2b = 5 (tn - trn) fr2 tn
          If[A2a < A2b, A2 = A2a, A2 = A2b]

Out[589]= 1

Out[590]= 0.0121391

Out[591]= 0.0946522

Out[592]= 0.00946522

Out[593]= 0.00946522

In[594]:= fr3 = 1 (*strength reducton factor*)
          A43 = te^2 fr3

Out[594]= 1

Out[595]= 0.00390625

In[596]:= (A1 + A2 + A43)
          A
          (A1 + A2 + A43) >= A
          (*If actual area > area required, then no additional reinforcement required *)

Out[596]= 0.193975

Out[597]= 0.134326

Out[598]= True
```

Fig. UG-41.1 1995 SECTION VIII — DIVISION 1



W = total weld load [UG-41(b)(2)]
$= [A - A_1 + 2t_n f_{r1}(E_1 t - F t_r)]S_v$
W_{1-1} = weld load for strength path 1-1 [UG-41(b)(1)]
$= (A_2 + A_5 + A_{41} + A_{42})S_v$
W_{2-2} = weld load for strength path 2-2 [UG-41(b)(1)]
$= (A_2 + A_3 + A_{41} + A_{43} + 2t_n t f_{r1})S_v$
W_{3-3} = weld load for strength path 3-3 [UG-41(b)(1)]
$= (A_2 + A_3 + A_5 + A_{41} + A_{42} + A_{43} + 2t_n t f_{r1})S_v$

GENERAL NOTES:
(a) Areas A_1, A_2, A_3, A_5, and A_{4l} are modified by f_{rx} factors.
(b) Nomenclature is the same as in UG-37 and Fig. UG-37.1.

(a) Depicts Typical Nozzle Detail With Neck Inserted Through the Vessel Wall

FIG. UG-41.1 NOZZLE ATTACHMENT WELD LOADS AND WELD STRENGTH PATHS TO BE CONSIDERED

```
(*must run "VCR_shell.nb" file first to save variables defined below into memory*)
```

```
In[599]:= (*Load / Stress  Carried by Welds*)
          A
          A1
          A2
          A3 = 0
          A5 = 0
          A41 = 0
          A42 = 0
          A43

Out[599]= 0.134326

Out[600]= 0.180604

Out[601]= 0.00946522

Out[602]= 0

Out[603]= 0

Out[604]= 0

Out[605]= 0

Out[606]= 0.00390625

In[607]:= Sv = 16700
          W = (A - A1 + 2 tn fr1 (E1 t - F tr)) Sv

Out[607]= 16700

Out[608]= -498.645

In[610]:=
          W1-1 = (A2 + A5 + A41 + A42) Sv

Out[610]= 158.069

In[611]:= W2-2 = (A2 + A3 + A41 + A43 + 2 tn t fr1) Sv

Out[611]= 1058.3

In[612]:= W3-3 = (A2 + A3 + A5 + A41 + A42 + A43 + 2 tn t fr1) Sv

Out[612]= 1058.3

       (* W (total weld load) << W1-1, W2-2, W3-3, (weld load available)*)

In[613]:= (*Allowable Unit Stresses*)
          (*Fillet Weld Shear, UW 15 c*)
          σfw = 0.49 (Sv)

Out[613]= 8183.
```

```
In[614]:= (*Nozzel Wall Shear, UG 45 c*)
```
$\sigma_{nw} = 0.7\ (Sv)$

```
Out[614]= 11690.

In[615]:=
          (*Strength of Connection Elements*)
          (*Fillet Weld Shear*)
```

$$W_{fw} = \frac{\pi}{2}\, Tx\, te\, \sigma_{fw}$$

```
Out[615]= 401.682

In[616]:=
          (*Strength of Connection Elements*)
          (*Nozzel Wall Shear*)
```

$$W_{nw} = \frac{\pi}{2}\, \frac{(Tx + d)}{2}\, tn\, \sigma_{nw}$$

```
Out[616]= 413.159

In[617]:=
```
$WS_{1-1} = W_{nw}$
$WS_{2-2} = W_{fw}$

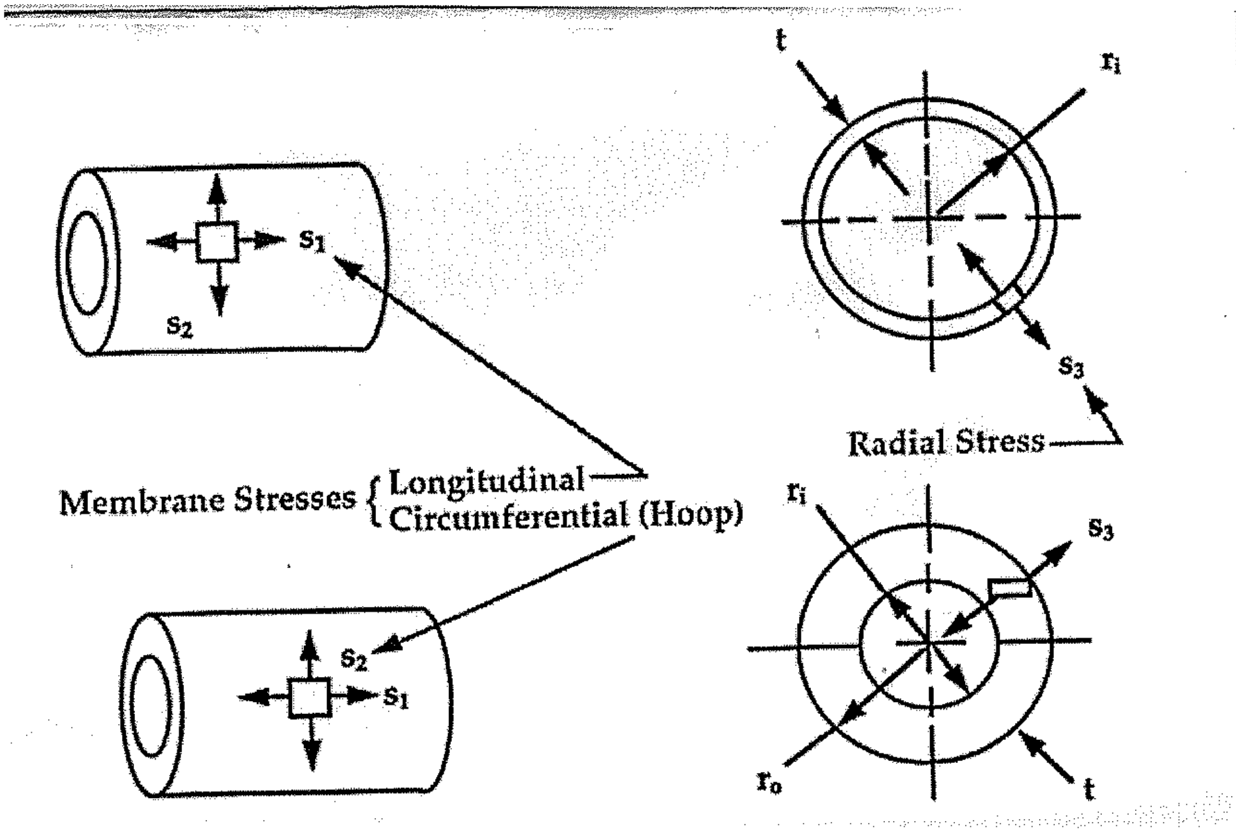
```
Out[617]= 413.159

Out[618]= 401.682
```

(*All Paths WS_{1-1}, WS_{2-2}, are stronger than the required strength W*)

```
(* VCR Port Xe chamber gamma ray feedthru *)
(* 850 psia MOP,
 stainless steel VCR plug with 0.250 " Ø counterbore. Determine head thickness: *)

In[24]:= (* Load at operating conditions *)
         G = 0.250 (* Diameter, in. at gasket load location *)
         P = 850 * 1.15  (* MAWP, internal design pressure *)
         H = 0.785 G^2 P (* lb., Total hydrostatic end force *)

Out[24]= 0.25

Out[25]= 977.5

Out[26]= 47.9586

In[27]:=
         Ca = 0.75   (*Head attachment constant, UG-34 (r)*)
         Ef = 1.0   (*Efficiency Factor *)
         dga = G  (*in. hole cross-sectional diameter*)
         σu = 77000.0 (*psi, 316 L ultimate strength*)
         σa = 16700   (*psi, 316 L allowable strength*)
         th = dga Sqrt[(Ca × P)/(σa × Ef)] (* in., required head thickness, no bending moment *)

Out[27]= 0.75

Out[28]= 1.

Out[29]= 0.25

Out[30]= 77000.

Out[31]= 16700

Out[32]= 0.0523806
```

$$th = d_{ga}\sqrt{\frac{C_a \times P}{\sigma_a \times E_f}}$$

(*Xenon Pressure Vessel Stress Calculations — LN2 Port*)



In[619]:=

```
MAWP = 978
σa = 16700 (*allowable stress for 316 L SST*)
σy = 37000
Ri = 1.5 / 2.
Ro = 1.9 / 2.
t = Ro - Ri
Ratio = Ro / Ri
If [1.1 < Ratio < 1.5, medium wall]
If [Ratio < 1.1, thin wall]
If [Ratio > 1.5, thick wall]
```

Out[619]= 978

Out[620]= 16700

Out[621]= 37000

Out[622]= 0.75

Out[623]= 0.95

Out[624]= 0.2

Out[625]= 1.26667

Out[626]= medium wall

In[629]:=

```
(*Longitudinal Stress, S1*)
```

$$S_1 = \frac{(MAWP\ R_i^2)}{(R_o^2 - R_i^2)}$$

```
(*Circumferential Stress, S2*)
```

$$S_2 = \frac{MAWP\ (R_o^2 + R_i^2)}{(R_o^2 - R_i^2)}$$

```
(*Radial Stress, S3*)
```

$$S_3 = -MAWP$$

```
(*Von Mises Stress*)
```

$$\sigma_m = \sqrt{0.5\ ((S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2)}$$

Out[629]= 1618.01

Out[630]= 4214.03

Out[631]= -978

Out[632]= 4496.43

In[104]:=

```
(*wall thickness, in., max. pressure, psi*)

(*Circumferential Stress: wall thickness, in., max. pressure, psi*)
```

$E_f = 0.7$ (*butt weld efficiency based on no inspection, Table UW-12*)

$t_c = \frac{(\text{MAWP}\ R_i)}{(\sigma_a\ E_f\ -\ 0.6\ \text{MAWP})}$ (*UG27 c 1*)

$P_c = \frac{(\sigma_a\ E_f\ t)}{(R_i\ +0.6\ t)}$ (*UG27 c 1*)

$SF_{uc} = \frac{P_c}{\text{MAWP}}$ (* P_c uses allowable stress so SF ~5 is also inlcluded*)

```
(*Longitudinal Stress: wall thickness, in., max. pressure, psi*)
(*Circumferential butt welds connecting
  ellipsoidal head and hub to cylinder are Catagory A, Type 1 welds*)
```

$E_f = 0.7$ (*butt weld efficiency based on no inspection, Table UW-12*)

$t_l = \frac{(\text{MAWP}\ R_i)}{(2\ \sigma_a\ E_f\ +\ 0.4\ \text{MAWP})}$ (*UG27 c 2*)

$P_l = \frac{(2\ \sigma_a\ E_f\ t)}{(R_i\ -0.4\ t)}$ (*UG27 c 2*)

$SF_{ul} = \frac{P_l}{\text{MAWP}}$ (* P_l uses allowable stress so SF ~5 is also inlcluded*)

If[P_c < P_l, "circumferential stress applies", "longitudinal stress applies"]
If[t_c > t_l, "circumferential stress applies", "longitudinal stress applies"]

Out[104]= 0.7

Out[105]= 0.066062

Out[106]= 2687.36

Out[107]= 2.74781

Out[108]= 0.7

Out[109]= 0.0308567

Out[110]= 6979.1

Out[111]= 7.1361

Out[112]= circumferential stress applies

Out[113]= circumferential stress applies

In[643]:=

(*Check of Von Mises stress at 1.5 × MAWP for pressure test*)
MAWP = 1.5 × 978

(*Longitudinal Stress, S_1*)

$$S_1 = \frac{(MAWP\ R_i^2)}{(R_o^2 - R_i^2)}$$

(*Circumferential Stress, S_2*)

$$S_2 = \frac{MAWP\ (R_o^2 + R_i^2)}{(R_o^2 - R_i^2)}$$

(*Radial Stress, S_3*)

$$S_3 = -MAWP$$

(*Von Mises Stress*)

$$\sigma_m = \sqrt{0.5\ ((S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2)}$$

$$N_r = \frac{\sigma_y}{\sigma_m}$$

If[N_r > 1, "vessel OK at 1.5 × MAWP during pressure test"]

Out[643]= 1467.

Out[644]= 2427.02

Out[645]= 6321.04

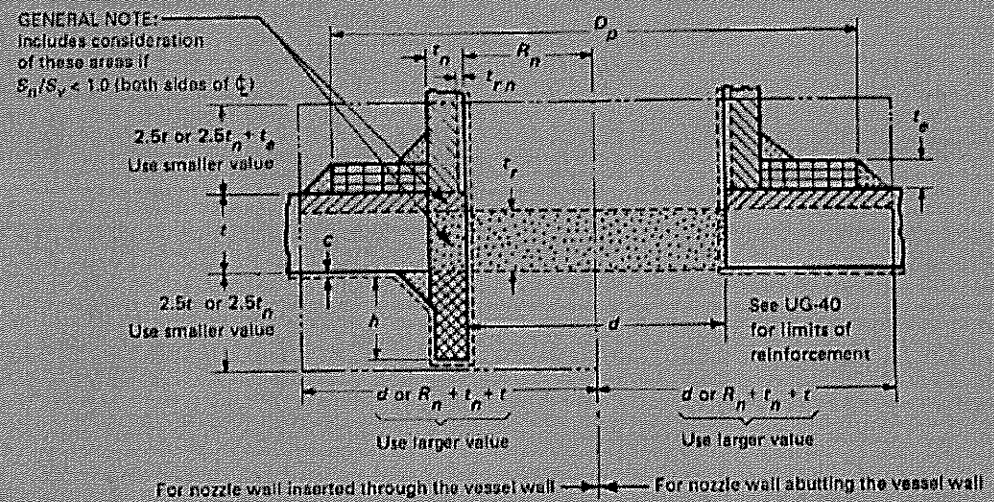
Out[646]= -1467.

Out[647]= 6744.64

Out[648]= 5.48583

Out[649]= vessel OK at 1.5 × MAWP during pressure test

PART UG — GENERAL REQUIREMENTS **Fig. UG-37.1**



Without Reinforcing Element

$= A \quad = d\,t_r F + 2t_n t_r F(1 - f_{r1})$	Area required
$= A_1 \begin{cases} = d(E_1 t - Ft_r) - 2t_n (E_1 t - Ft_r)(1 - f_{r1}) \\ = 2(t + t_n)(E_1 t - Ft_r) - 2t_n(E_1 t - Ft_r)(1 - f_{r1}) \end{cases}$	Area available in shell; use larger value
$= A_2 \begin{cases} = 5(t_n - t_{rn})\, f_{r2} t \\ = 5(t_n - t_{rn}) f_{r2} t_n \end{cases}$	Area available in nozzle projecting outward; use smaller value
$= A_3 \quad = 2(t_n - c) f_{r2} h$	Area available in inward nozzle
$= A_{41}$ = outward nozzle weld = (leg)$^2 f_{r2}$	Area available in outward weld
$= A_{43}$ = inward nozzle weld = (leg)$^2 f_{r2}$	Area available in inward weld
If $A_1 + A_2 + A_3 + A_{41} + A_{43} \geq A$	Opening is adequately reinforced
If $A_1 + A_2 + A_3 + A_{41} + A_{43} < A$	Opening is not adequately reinforced so reinforcing elements must be added and/or thicknesses must be increased

With Reinforcing Element Added

A = same as A, above	Area required
A_1 = same as A_1, above	Area available
$A_2 \begin{cases} = 5(t_n - t_{rn}) f_{r2} t \\ = 2(t_n - t_{rn})(2.5t_n + t_e) f_{r2} \end{cases}$	Area available in nozzle projecting outward; use smaller area
A_3 = same as A_3, above	Area available in inward nozzle
$= A_{41}$ = outward nozzle weld = (leg)$^2 f_{r3}$	Area available in outward weld
$= A_{42}$ = outer element weld = (leg)$^2 f_{r4}$	Area available in outer weld
$= A_{43}$ = inward nozzle weld = (leg)$^2 f_{r2}$	Area available in inward weld
$= A_5 = (D_p - d - 2t_n)\, t_e\, f_{r4}$ [Note (1)]	Area available in element
If $A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 \geq A$	Opening is adequately reinforced

NOTE:
(1) This formula is applicable for a rectangular cross-sectional element that falls within the limits of reinforcement.

FIG. UG-37.1 NOMENCLATURE AND FORMULAS FOR REINFORCED OPENINGS
(This Figure Illustrates a Common Nozzle Configuration and Is Not Intended to Prohibit Other Configurations Permitted by the Code.)

```
In[4]:= (* Opening Reinforcement Calculations*)
        (* LN2 Trap to shell wall*)
        (* A3 = 0, A5 = 0, A42 = 0 *)

        (* Pipe Sch. 80 1.5 " Ø *)
        Tx = 1.9 (*OD*)
        d = 1.5 (*ID min*)
        tn = (Tx - d) / 2 (*nozzel wall thickness*)
        te = 1.25 * tn (*weld leg height*)

Out[4]= 1.9

Out[5]= 1.5

Out[6]= 0.2

Out[7]= 0.25

In[8]:= F = 1 (*correction factor*)
        tr = 0.335815 (*minimum shell thickness, Vessel_stress2.nb*)
        fr1 = 1 (*strength reducton factor*)
        t = 0.5 (*shell wall thickness*)
        E1 = 1 (*joint efficiency*)
        A = d tr F + 2 tn tr F (1 - fr1)
         A1a = d (E1 t - F tr) - 2 tn (E1 t - F tr) (1 - fr1)
         A1b = 2 (t + tn) (E1 t - F tr) - 2 tn (E1 t - F tr) (1 - fr1)
          If[A1a > A1b, A1 = A1a, A1 = A1b]

Out[8]= 1

Out[9]= 0.335815

Out[10]= 1

Out[11]= 0.5

Out[12]= 1

Out[13]= 0.503722

Out[14]= 0.246278

Out[15]= 0.229859

Out[16]= 0.246278
```

```
In[17]:= fr2 = 1 (*strength reducton factor*)
         trn = 0.066062 (*required nozzel thickness, Xe_vessel _LN2.nb*)
         A2a = 5 (tn - trn) fr2 t
         A2b = 5 (tn - trn) fr2 tn
         If [A2a < A2b, A2 = A2a, A2 = A2b]

Out[17]= 1

Out[18]= 0.066062

Out[19]= 0.334845

Out[20]= 0.133938

Out[21]= 0.133938
```

```
In[72]:= fr3 = 1 (*strength reducton factor*)
```

$A43 = te^2\, fr3$

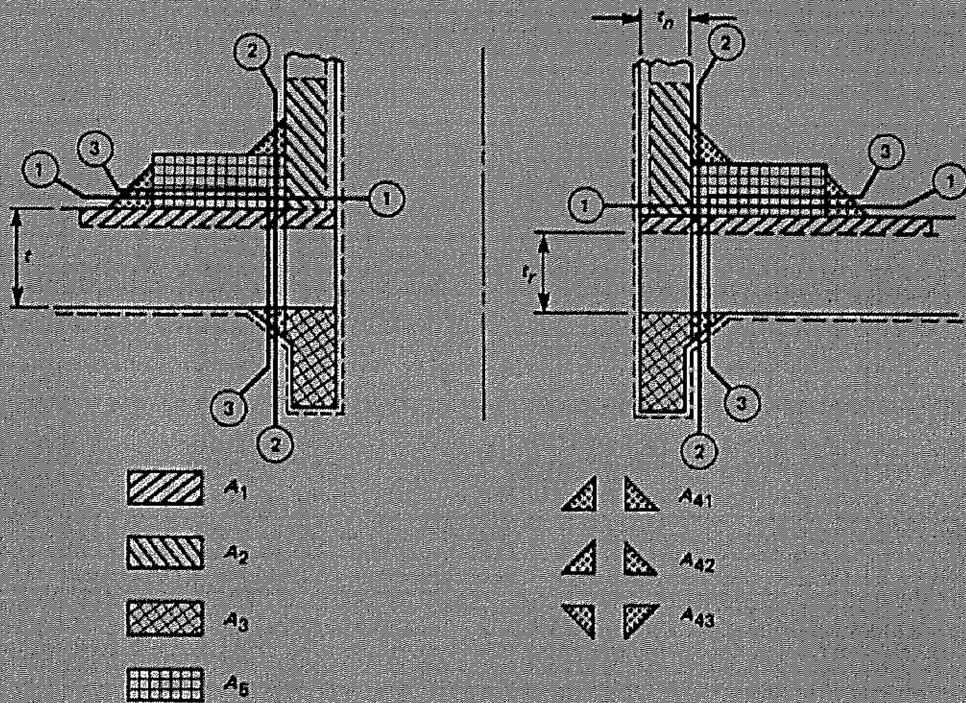
$te_o = 1.4\, tn$

$A41 = \frac{te_o{}^2\, fr3}{1.2}$ (* 80 % the area, skip weld on outside*)

```
Out[72]= 1

Out[73]= 0.0625

Out[74]= 0.28

Out[75]= 0.0653333
```

```
In[76]:= (A1 + A2 + A43 + A41)
         A
         (A1 + A2 + A43 + A41) >= A
         (*If actual area > area required, then no additional reinforcement required *)

Out[76]= 0.508049

Out[77]= 0.503722

Out[78]= True
```

Fig. UG-41.1 1995 SECTION VIII — DIVISION 1



W = total weld load [UG-41(b)(2)]
$= [A - A_1 + 2t_n f_{r1}(E_1 t - F t_r)]S_v$
$W_{1\text{-}1}$ = weld load for strength path 1-1 [UG-41(b)(1)]
$= (A_2 + A_5 + A_{41} + A_{42})S_v$
$W_{2\text{-}2}$ = weld load for strength path 2-2 [UG-41(b)(1)]
$= (A_2 + A_3 + A_{41} + A_{43} + 2t_n t f_{r1})S_v$
$W_{3\text{-}3}$ = weld load for strength path 3-3 [UG-41(b)(1)]
$= (A_2 + A_3 + A_5 + A_{41} + A_{42} + A_{43} + 2t_n t f_{r1})S_v$

GENERAL NOTES:
(a) Areas A_1, A_2, A_3, A_5, and A_{4i} are modified by f_{rx} factors.
(b) Nomenclature is the same as in UG-37 and Fig. UG-37.1.

(a) Depicts Typical Nozzle Detail With Neck Inserted Through the Vessel Wall

FIG. UG-41.1 NOZZLE ATTACHMENT WELD LOADS AND WELD STRENGTH PATHS TO BE CONSIDERED

(*must run "LN2_shell.nb" file first to save variables defined below into memory*)

```
In[79]:= (*Load / Stress  Carried by Welds*)
         A
         A1
         A2
         A3  = 0
         A5 = 0
         A41
         A42 = 0
         A43

Out[79]= 0.503722

Out[80]= 0.246278

Out[81]= 0.133938

Out[82]= 0

Out[83]= 0

Out[84]= 0.0653333

Out[85]= 0

Out[86]= 0.0625

In[87]:= Sv = 16700
         W = (A - A1 + 2 tn fr1 (E1 t - F tr)) Sv

Out[87]= 16700

Out[88]= 5396.09

In[89]:=
```

W_{1-1} = (A2 + A5 + A41 + A42) Sv

```
Out[89]= 3327.83
```

In[90]:= W_{2-2} = (A2 + A3 + A41 + A43 + 2 tn t fr1) Sv

```
Out[90]= 7711.58
```

W_{3-3} = (A2 + A3 + A5 + A41 + A42 + A43 + 2 tn t fr1) Sv

7664.26

(* W (total weld load) << W_{1-1}, W_{2-2}, W_{3-3}, (weld load available)*)

```
In[91]:= (*Allowable Unit Stresses*)
         (*Fillet Weld Shear, UW 15 c*)
```

σ_{fw} = 0.49 (Sv)

```
Out[91]= 8183.
```

In[92]:= (*Nozzel Wall Shear, UG 45 c*)

$\sigma_{nw} = 0.7\ (Sv)$

Out[92]= 11690.

In[93]:=

(*Strength of Connection Elements*)

(*Fillet Weld Shear*)

$$W_{fw} = \frac{\pi}{2}\,Tx\,te\,\sigma_{fw}$$

Out[93]= 6105.57

In[94]:=

(*Strength of Connection Elements*)

(*Nozzel Wall Shear*)

$$W_{nw} = \frac{\pi}{2}\,\frac{(Tx + d)}{2}\,tn\,\sigma_{nw}$$

Out[94]= 6243.29

In[95]:=

$WS_{1-1} = W_{nw}$

$WS_{2-2} = W_{fw}$

Out[95]= 6243.29

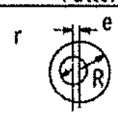
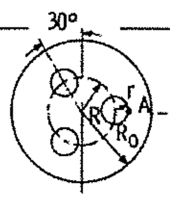
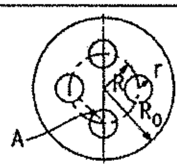
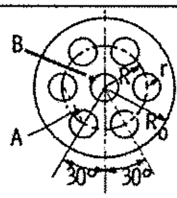
Out[96]= 6105.57

(*All Paths WS_{1-1}, WS_{2-2}, are stronger than the required strength W*)

```
(* Stress Concentration Factor for mini-conflat openings in head. Reference: Wiley *)
```

Stress Concentration Factors

Table 4
Maximum K_t for circular plate with circular holes with internal pressure only

	Pattern	Spacing	Maximum K_t	Location	Reference
1		$r/R = 0.5$	See Fig. 126	See Fig. 126	223, 228, 229
2		$R/R_0 = 0.5$ $r/R_0 = 0.2$	See Fig. 127	See Fig. 127	230
3		$R/R_0 = 0.5$ $r/R_0 = 0.2$	See Fig. 127	See Fig. 127	230
		$R/R_0 = 0.5$ $r/R_0 = 0.25$	2.45	A	223
4		$R/R_0 = 0.6$ $r/R_0 = 0.2$	2.278 Pressure in All Holes 1.521 Pressure in Center Hole Only	A B	223

```
In[1039]:= (*must run "Flange stress_hub_403.nb"
              file first to save variables defined below into memory*)

            (* Assumptions:Chamber head design has 5 mini-conflat holes,not 6,
             Kt will be conservative. Chamber head has 1.5 " diameter inner hole, #4 model has
              equal diameter holes throughout,  Kt will not be conservative. Assume
              these 2 opposites have a cancelling effect and given Kt is valid. Actual BC hole
              radius is 0.3125 ". But, for this analysis assume all holes 0.75" radius *)

            Ro = 5.25 (*in.,Flange outside radius*)
            rh = 0.750  (*in.,BC hole radius*)
            Ri = 0.750 (*in.,inner hole radius*)
            R = 5.5 / 2 (* in., mini conflat bolt circle radius *)
            l = Ri/Ro (*graph constant*)
            m = R/Ro (*graph constant*)
            o = rh/Ro (*graph constant*)

Out[1039]= 5.25

Out[1040]= 0.75

Out[1041]= 0.75

Out[1042]= 2.75

Out[1043]= 0.142857

Out[1044]= 0.52381

Out[1045]= 0.142857

In[1256]:= (* R/Ro and rh/Ro are slightly less than #4 model,
             but Kt will be less (conservative) for chamber head design.*)
             Kt = 2.278

Out[1256]= 2.278
```

```
In[1257]:=
```

$C_a = 0.3$ (*flange attachment constant*)
$p = 350 \times 1.15$ (*psi, MAWP*)
$E_f = 1.0$ (*Efficiency Factor *)
$d_{ga} = 8.54$ (*in. gasket diameter*)
W_{m1}
(* lb., Must run "Flange stress_hub _ 403. nb " to
define this variable. Minimum required bolt load, for operating *)
$h_G = 0.520$ (* in., Must run
"Flange stress_hub_403.nb" to define this variable.Bending moment length *)
$\sigma_u = 77000.0$ (*psi, 316 L ultimate strength*)
$\sigma_a = \frac{16700}{K_t}$ (*psi, 316 L allowable strength*)

```
Out[1257]= 0.3

Out[1258]= 402.5

Out[1259]= 1.

Out[1260]= 8.54

Out[1261]= 24677.2

Out[1262]= 0.52

Out[1263]= 77000.

Out[1264]= 7330.99
```

In[1265]:= $t_h = d_{ga} \sqrt{\frac{(C_a * p)}{(\sigma_a * E_f)} + \frac{(1.9 * W_{m1} * h_G)}{(\sigma_a * E_f * {d_{ga}}^3)}}$ (* in., required flange thickness *)

```
Out[1265]= 1.26123
```

In[1266]:= If[t_h < 1.5, "flange thickness is OK", "flange thickness is NOT OK"]

```
Out[1266]= flange thickness is OK
```

In[1565]:= (* Maximum distance between hole centers for a cluster of holes in head, UG-36 (3) (d)*)
(* No two unreinforced openings shall have their centers closer than: *)
R

$d_{betweenholes1} = \mathrm{Sin}\left[36 \times \frac{\pi}{180}\right] \times R \times 2$

$d_2 = 0.625$
$d_1 = d_2$ (* diameter of holes *)
$C_d = 2.5\ (d_1 + d_2)$

If[$d_{betweenholes1} > C_d$,
"distance between 5.5 BC holes OK", "distance between 5.5 BC holes NOT OK"]

$d_{betweenholes2} = R - .75 - .3125$
$d_2 = 0.625$
$d_1 = 1.500$
$C_d = 2.5\ (d_1 + d_2)$

If[$d_{betweenholes2} > C_d$, "distance between center hole and 5.5 BC holes OK", "distance between center hole and 5.5 BC holes NOT OK,,,,, use alternative UG-39(d)"]

$d_{betweenholes2} < 2\left(\frac{(d_1 + d_2)}{2}\right)$

$d_{betweenholes2} > 1.25\left(\frac{(d_1 + d_2)}{2}\right)$

(* in., required flange thickness,
UG-39 (e) (1) (2), using alternative to Area reinforcment of UG39 (b) (1) *)

$\sigma_a = 16700$

$ef = \frac{R - \left(\frac{(d_1 + d_2)}{2}\right)}{R}$ (* UG39 (e) (2) *)

$fs = \sqrt{0.5 / ef}$

$t_h = d_{ga} \sqrt{fs \times 2\left(\frac{(C_e * p)}{(\sigma_a * E_f)} + \frac{(1.9 * W_{m1} * h_G)}{(\sigma_a * E_f * d_{ga}{}^3)}\right)}$

Out[1565]= 2.75

Out[1566]= 3.23282

Out[1567]= 0.625

Out[1568]= 0.625

Out[1569]= 3.125

Out[1570]= distance between 5.5 BC holes OK

Out[1571]= 1.6875

```
Out[1572]= 0.625

Out[1573]= 1.5

Out[1574]= 5.3125

Out[1575]= distance between center hole and 5.5 BC holes NOT OK,,,,, use alternative UG-39(d)

Out[1576]= True

Out[1577]= True

Out[1578]= 16700

Out[1579]= 0.613636

Out[1580]= 0.902671

Out[1581]= 1.12279
```

```
Out[1589]= 0.625

Out[1590]= 1.5

Out[1591]= 5.3125

Out[1592]= distance between center hole and 5.5 BC holes NOT OK,,,,, use alternative UG-39(d)

Out[1593]= True

Out[1594]= True

Out[1595]= 16700

Out[1596]= 0.613636

Out[1597]= 0.902671

Out[1598]= 1.24385
```

```
(*must run "Flange stress_hub_978.nb"
  file first to save variables defined below into memory*)

(* Assumptions: Chamber head design has has no holes *)

Ro = 5.25 (*in.,Flange outside radius*)
```

$C_a = 0.3$ (*flange attachment constant*)
$p = 850 \times 1.15$ (*psi, MAWP*)
$E_f = 1.0$ (*Efficiency Factor *)
$d_{ga} = G$ (*in. gasket diameter*)
W_{m1}
(* lb., Must run "Flange stress_hub _ 978.nb
  " to define this variable. Minimum required bolt load, for operating *)
$h_G = 0.520$ (* in.,
 Must run "Flange stress_hub_978.nb" to define this variable.Bending moment length *)
$\sigma_u = 77000.0$ (*psi, 316 L ultimate strength*)
$\sigma_a = 16700$ (*psi, 316 L allowable strength*)

0.3

977.5

1.

7.98

58849.9

0.52

77000.

16700

$$t_h = d_{ga}\sqrt{\frac{(C_a * p)}{(\sigma_a * E_f)} + \frac{(1.9 * W_{m1} * h_G)}{(\sigma_a * E_f * {d_{ga}}^3)}}$$ (* in., required flange thickness *)

1.2468

If[t_h < 1.980, "flange thickness is OK", "flange thickness is NOT OK"]

flange thickness is OK

(* Minimum thickness of plate under gasket (hub) *)

$tg = d_{ga}\sqrt{(1.9 * W_{m1} * h_G) / (\sigma_a * {d_{ga}}^3)}$ (* UG34, sketch (k) *)

If[tg < 1.980, "flange hub thickness is OK", "flange hub thickness is NOT OK"]

0.660529

flange hub thickness is OK

```
(*must run "Flange stress_hub_small_978.nb"
  file first to save variables defined below into memory*)

(* Assumptions: Head design has has no holes *)

Ro = 4.63 / 2 (*in.,Flange outside radius*)
```

$C_e = 0.3$ (*flange attachment constant*)
$p = 850 \times 1.15$ (*psi, MAWP*)
$E_f = 1.0$ (*Efficiency Factor *)
$d_{ga} = G$ (*in. gasket diameter*)
W_{m1}
(* lb., Must run "Flange stress_hub _small _ 978. nb
" to define this variable. Minimum required bolt load, for operating *)
$h_G = 0.520$ (* in., Must run
"Flange stress_hub_small_978.nb" to define this variable.Bending moment length *)

$\sigma_a = 16700$ (*psi, 304 L allowable strength*)

```
0.3

977.5

1.

3.35

10143.7

0.52

16700
```

$$t_h = d_{ga} \sqrt{\frac{(C_e * p)}{(\sigma_a * E_f)} + \frac{(1.9 * W_{m1} * h_G)}{(\sigma_a * E_f * d_{ga}{}^3)}}$$ (* in., required flange thickness *)

0.613356

If[t_h < 0.750, "flange thickness is OK", "flange thickness is NOT OK"]

flange thickness is OK

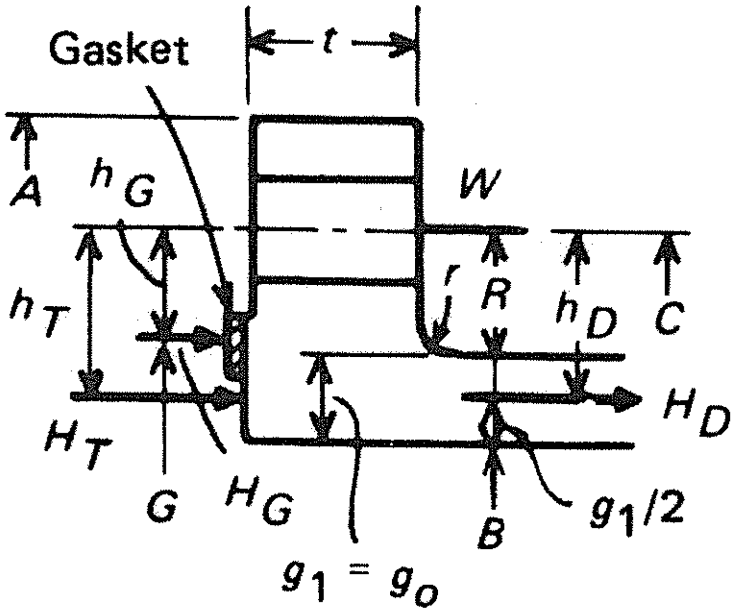
(* Minimum thickness of plate under gasket (hub) *)

$tg = d_{ga} \sqrt{(1.9 * W_{m1} * h_G) / (\sigma_a * d_{ga}{}^3)}$ (* UG34, sketch (k) *)

If[tg < 0.81, "flange hub thickness is OK", "flange hub thickness is NOT OK"]

0.423249

flange hub thickness is OK



(5)

```
(* Bolted Flange Connections with flat metal Copper Gasket,
 Xe chamber high voltage feedthru *)
(* 350 psia MOP, conflat type head *)
```

```
(* Bolt Load at operating conditions *)
G = 0.72 (* Diameter, in. at gasket load location *)
P = 350*1.15  (* MAWP, internal design pressure *)
m = 4.75 (* gasket factor flat Cu gasket, Table 2-4.1 *)
Ng = 0.25 (* width of Cu gasket *)
bo = Ng/32 (* N/4 for multiple serrations Table 2-5.2 (5),
 assume N/32 given a single knife edge serration as used in Conflats *)
If[bo <= 0.25, b = bo, b = .5 Sqrt[bo]]
y = 13000 (* psi, design seating stress for soft copper, Table 2-5.1 *)
H = 0.785 G^2 P (* lb., Total hydrostatic end force *)
Hp = 2 b × π G m P (* lb., Total joint-contact surface compression load *)
Wm1 = H + Hp (* Minimum required bolt load, for operating *)
Wm2 = π G b y (* Minimum required bolt load, for gasket seating *)
```

$$N_g = 0.25,\quad b_o = \frac{N_g}{32},\quad b = .5\sqrt{b_o},\quad H = 0.785\,G^2 P,\quad H_p = 2b \times \pi G m P,\quad W_{m1} = H + H_p,\quad W_{m2} = \pi G b y$$

0.72

402.5

4.75

0.25

0.0078125

0.0078125

13000

163.795

67.5712

231.366

229.729

In[54]:= (* Design Bolt Load*)

$A_b = \frac{\pi \times 0.1248^2}{4} \times 6$ (*cross sectional area of #8-32 screw*)

SF = 4 (* MEDSS *)

$S_T = 81000$

(*Unbrako - KS 1216 8-32 SHCS, 160, ksi tensile strength; T = -400 °F to 1200 °F OR ASTM-A493-95 Grade S30430; 81 ksi tensile strength *)

$S_a = S_T \div SF$

$L_b = S_a \times A_b$ (* lb., Max allowable bolt load *)

$A_{m1} = W_{m1} / S_a$ (* in^2,cross-sectional area of bolts under operating condition *)

$A_{m2} = W_{m2} / S_a$ (* in^2, cross-sectional area of bolts for gasket seating *)

If[$A_{m1} > A_{m2}$, $A_m = A_{m1}$, $A_m = A_{m2}$]

(* in^2, total required cross-sectional area of bolts *)

$W_o = W_{m1}$ (* lb., Flange design bolt load, for operating *)

$W_g = \frac{(A_m + A_b)\ S_a}{2}$ (* lb., Flange design bolt load, for gasket seating *)

$SF_u = L_b\ SF / W_g$

Out[54]= 0.0733956

Out[55]= 4

Out[56]= 81000

Out[57]= 20250

Out[58]= 1486.26

Out[59]= 0.0114255

Out[60]= 0.0113446

Out[61]= 0.0114255

Out[62]= 231.366

Out[63]= 858.814

Out[64]= 6.92239

$C_e = 0.3$ (*flange attachment constant*)
$p = 350 \times 1.15$ (*psi, MAWP*)
$E_f = 1.0$ (*Efficiency Factor *)
$d_{ga} = G$ (*in. gasket diameter*)
W_{m1}
(* lb., Minimum required bolt load, for operating *)
$h_G = 0.171$ (* in., Bending moment length *)
$\sigma_u = 77000.0$ (*psi, 304 L ultimate strength*)
$\sigma_a = 16700$ (*psi, 304 L allowable strength*)

$$t_h = d_{ga} \sqrt{\frac{(C_e * p)}{(\sigma_a * E_f)} + \frac{(1.9 * W_{m1} * h_G)}{(\sigma_a * E_f * d_{ga}{}^3)}} \quad \text{(* in., required flange thickness *)}$$

0.3

402.5

1.

0.72

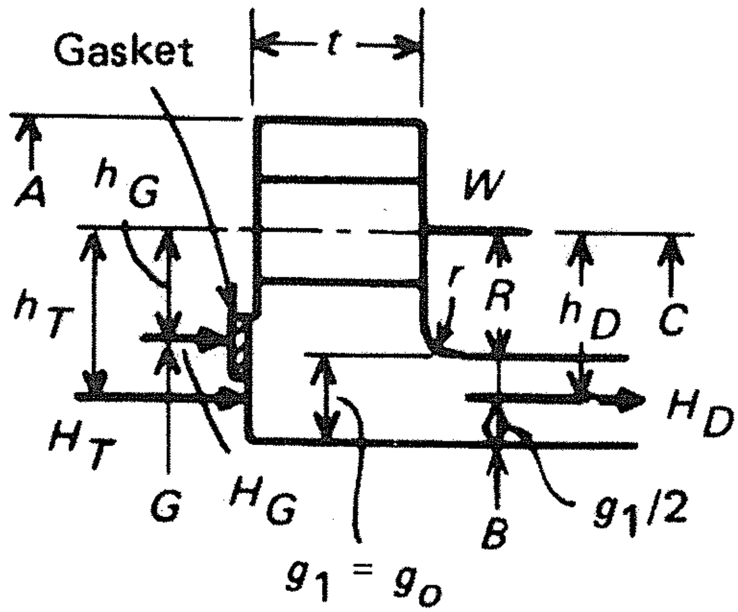
231.366

0.171

77000.

16700

0.1



(5)

```
(* Bolted Flange Connections with flat metal Copper Gasket,
 Xe chamber gamma ray feedthru *)
(* 350 psia MOP, conflat type head *)
```

```
(* Bolt Load at operating conditions *)
G = 1.650 (* Diameter, in. at gasket load location *)
P = 350 * 1.15  (* MAWP, internal design pressure *)
m = 4.75 (* gasket factor flat Cu gasket, Table 2-4.1 *)
N_g = 0.25 (* width of Cu gasket *)
b_o = N_g/32 (* N/4 for multiple serrations Table 2-5.2 (5),
 assume N/32 given a single knife edge serration as used in Conflats *)
If[b_o <= 0.25, b = b_o, b = .5 Sqrt[b_o]]
y = 13000 (* psi, design seating stress for soft copper, Table 2-5.1 *)
H = 0.785 G^2 P (* lb., Total hydrostatic end force *)
H_p = 2 b × π G m P (* lb., Total joint-contact surface compression load *)
W_m1 = H + H_p (* Minimum required bolt load, for operating *)
W_m2 = π G b y (* Minimum required bolt load, for gasket seating *)
```

1.65

402.5

4.75

0.25

0.0078125

0.0078125

13000

860.208

154.851

1015.06

526.462

```
In[76]:= (* Design Bolt Load*)
```

$A_b = \frac{\pi \times 0.2052^2}{4} \times 6$ `(*cross sectional area of 1/4-28 screw*)`

$SF = 4$ `(* MEDSS *)`

$S_T = 81000$

`(*Unbrako - KS 1216 1/4-28 SHCS, 160, ksi tensile strength; T = -400 °F to 1200 °F OR ASTM-A493-95 Grade S30430; 81 ksi tensile strength *)`

$S_a = S_T \div SF$

$L_b = S_a \times A_b$ `(* lb., Max allowable bolt load *)`

$A_{m1} = W_{m1} / S_a$ (* in^2,cross-sectional area of bolts under operating condition *)

$A_{m2} = W_{m2} / S_a$ (* in^2, cross-sectional area of bolts for gasket seating *)

If$[A_{m1} > A_{m2}, A_m = A_{m1}, A_m = A_{m2}]$

(* in^2, total required cross-sectional area of bolts *)

$W_o = W_{m1}$ `(* lb., Flange design bolt load, for operating *)`

$W_g = \frac{(A_m + A_b)\ S_a}{2}$ `(* lb., Flange design bolt load, for gasket seating *)`

$SF_u = L_b\ SF / W_g$

```
Out[76]= 0.198425

Out[77]= 4

Out[78]= 81000

Out[79]= 20250

Out[80]= 4018.1

Out[81]= 0.0501264

Out[82]= 0.0259981

Out[83]= 0.0501264

Out[84]= 1015.06

Out[85]= 2516.58

Out[86]= 6.38661
```

```
Ca = 0.13    (*flange attachment constant*)
p = 350 × 1.15  (*psi, MAWP*)
Ef = 1.0    (*Efficiency Factor *)
dga = G  (*in. gasket diameter*)
Wm1
(* lb., Minimum required bolt load, for operating *)
hG = 0.331 (* in., Bending moment length *)
σu = 77000.0 (*psi, 304 L ultimate strength*)
σa = 16700   (*psi, 304 L allowable strength*)
```

$$th = d_{ga}\sqrt{\frac{C_a \times p}{\sigma_a \times E_f}}$$ (* in., required flange thickness, no bending moment *)

$$t_{ho} = d_{ga}\sqrt{\frac{(C_a * p)}{(\sigma_a * E_f)} + \frac{(1.9 * W_{m1} * h_G)}{(\sigma_a * E_f * d_{ga}{}^3)}}$$ (* in., required flange thickness *)

```
p = 0  (*psi, MAWP*)
Wm2
(* lb., Minimum required bolt load, for gasket seating *)
```

$$t_{hg} = d_{ga}\sqrt{\frac{(C_a * p)}{(\sigma_a * E_f)} + \frac{(1.9 * W_{m2} * h_G)}{(\sigma_a * E_f * d_{ga}{}^3)}}$$ (* in., required flange thickness *)

```
0.13

402.5

1.

1.65

1015.06

0.331

77000.

16700

0.0923592

0.178038

0

526.462

0.109616
```

```
(* Miscellaneous Calculations *)

(* UG-36 Openings in Pressure vessels*)
(* UG-36 (c) (3) (d),  No two unreinforced openings shall have their centers
  closer than the sum of their diameters: *)

(* This applies to all holes in the shell of the vessel. Actual holes have
  reinforcement built into the design so this is concervative. *)
```

$d_1 = 2.87$

$d_2 = 0.5$

$l_s = d_1 + d_2$

$l = \frac{\pi \times 7.625}{4}$ (* distance between holes*)

$l >= l_s$

2.87

0.5

3.37

5.98866

True

```
(* Drilled holes not penetrating shell *)
(* holes must be less than 2 " dia. & not less than 0.25 "*)
```

$D_i = 7.625$

$t = 0.5$

D_i / t

$D_i / t >= 10$

7.625

0.5

15.25

True

$d_h = 0.375$ (* UNF 3/8-16, major dia. *)

d_h / D_i

0.375

0.0491803

```
Rt = 0.375 (* from graph *)
If [ dh / Di < 0.03, Rt = .25, Rt = Rt ]

tmn = t (Rt) (* Appx. 30 Fig. 30-1, remaining wall thickness*)

0.375

0.375

0.1875        0.5 - .1875 = 0.313 IN.


dh = 0.250 (* UNF 1/4-20, major dia. *)
dh / Di

0.25

0.0327869

Rt = 0.256 (* from graph *)
If [ dh / Di < 0.03, Rt = .25, Rt = Rt ]

tmn = t (Rt) (* Appx. 30 Fig. 30-1, remaining wall thickness*)


0.256

0.256

0.128        0.5 - 0.128 = 0.372 IN.

(* Drilled / tapped holes in unstayed flat head *)
(* reinforcement required, replacement of area *)
(* 8-32 for mini conflats *)
tr = 1.26123 (* in., minimum required flange thickness *)
ta = 1.5 (* in., actual flange thickness *)
dh = 0.164 (* in., hole diameter *)
dd = 0.312 (* in., depth of hole *)

Ar = dh × dd (* in.^2, area required *)
Aa = dh (ta - dd) (* in.^2, area available *)
If [ Aa > Ar, "Reinforcement OK for mini-conflat blind holes",
 "Reinforcement NOT OK for mini-conflat blind holes" ]

1.26123

1.5

0.164

0.312

0.051168

0.194832

Reinforcement OK for mini-conflat blind holes
```

```
(* Drilled / tapped holes in unstayed flat head *)
(* reinforcement required, replacement of area *)
(* 8-32 for mounting brackets inside vessel *)
tr = 1.26123 (* in., minimum required flange thickness *)
ta = 1.5 (* in., actual flange thickness *)
dh = 0.164 (* in., hole diameter *)
dd = 0.25 (* in., depth of hole *)

Ar = dh × dd (* in.^2, area required *)
Aa = dh (ta - dd) (* in.^2, area available *)
If [ Aa > Ar, "Reinforcement OK for mounting bracket blind holes",
 "Reinforcement NOT OK for mounting bracket blind holes" ]
```

```
1.26123

1.5

0.164

0.25

0.041

0.205

Reinforcement OK for mounting bracket blind holes
```

```
(* Drilled / tapped holes in unstayed flat head *)
(* reinforcement required, replacement of area *)
(* 1/4-28 for medium conflat, center hole *)
tr = 1.26123 (* in., minimum required flange thickness *)
ta = 1.5 (* in., actual flange thickness *)
dh = 0.250 (* in., hole diameter *)
dd = 0.5 (* in., depth of hole *)

Ar = dh × dd (* in.^2, area required *)
Aa = dh (ta - dd) (* in.^2, area available *)
If [ Aa > Ar, "Reinforcement OK for medium conflat blind holes",
 "Reinforcement NOT OK for medium conflat blind holes" ]
```

```
1.26123

1.5

0.25

0.5

0.125

0.25

Reinforcement OK for medium conflat blind holes
```

```
(* Weld impact testing exemption calculation *)
(* UHA-51 (g) *)
 Sa = 16700
 S1 = 3499.17
 S_vm = 7754.6879
 If [S1 / Sa < 0.4, "Impact testing NOT required for weld",
 "Impact testing REQUIRED required for weld"]
```

```
16700

3499.17

7754.6879

Impact testing NOT required for weld
```

```
(* Base material Impact testing exemption *)
(* UHA-51 (d) (1) (a) austenitic chromium-nickel stainless steels: 304, 304 L,
 316, 316 L.
```

```
(* Fracture Critical Components *)

In[91]:= (* The applied stress is: *)
```

$R_i = 3.8125$
$R_o = 4.3125$
MAWP = 978
$\sigma_a = 16700$ (*allowable stress for 316 L SST*)
$\sigma_y = 37000$
$R_i = 3.8125$
$R_o = 4.3125$
$t = R_o - R_i$

$$\text{Ratio} = \frac{R_o}{R_i}$$

```
If [1.1 < Ratio < 1.5, medium wall]
If [Ratio < 1.1, thin wall]
If [Ratio > 1.5, thick wall]
```

(*Circumferential Stress, S_2*)

$$S_2 = \frac{\text{MAWP}\,(R_o{}^2 + R_i{}^2)}{(R_o{}^2 - R_i{}^2)}$$

```
Out[91]= 3.8125

Out[92]= 4.3125

Out[93]= 978

Out[94]= 16700

Out[95]= 37000

Out[96]= 3.8125

Out[97]= 4.3125

Out[98]= 0.5

Out[99]= 1.13115

Out[100]= medium wall

Out[103]= 7976.34
```

(* First consider actual stress intensity factors from literature (testing). Then apply this $K_{Ic} = K_I$ value to the Xe vessel at its MAWP/stress *)

```
(* Degraded Piping Program Phase II, 4/99 *)
(* Material 304 and 316 stainless steel, range: 561 to 13,400 in-lb/in^2,
Ji used is the lowest measurable value in all tests, parent or welded material *)
```

In[14]:=

```
Ji = 561   (* in-lb/in^2 *)

v = 0.33 (* Poisson' s ratio *)
```

$$E_y = \frac{29.7\times10^6}{(1-v^2)} \text{ (* psi *)}$$

$$K_{Ic} = \sqrt{Ji\times E_y} \text{ (* psi } \sqrt{in} \text{ *)}$$

$$a_{crs} = \frac{1}{1.21\,\pi}\left(\frac{K_{Ic}}{S_2}\right)^2 \text{ (* in., crack critical length, surface flaw *)}$$

$$a_{cri} = \frac{1}{\pi}\left(\frac{K_{Ic}}{S_2}\right)^2 \text{ (* in., crack critical length, imbedded flaw *)}$$

$$lengthc_{crs} = 4\times a_{crs}$$

$$lengthc_{cri} = 4\times a_{cri}$$

```
(* considering leak before break  criteria,
 leak occurs occurs before catastrophic failure in a pressure vessel when *)
```

$$\sqrt{\frac{\pi\, t\, S_2{}^2}{1-\frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}} \text{ (* Fracture and Fatigue Control in Structures,}$$

```
 Rolfe and Barsom, Prentice-Hall, 1977, pg. 394*)
```

$$K_{Ic} >= \sqrt{\frac{\pi\, t\, S_2{}^2}{1-\frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}} \text{ (* Hoop stress applies, } S_2 \text{ *)}$$

$$\text{If}\left[K_{Ic} >= \sqrt{\frac{\pi\, t\, S_2{}^2}{1-\frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}}\right.,$$

```
 "Leaking should occur before failure", "failure may occur before leaking"]

(* LIFE Expectancy Cycles *)
```

$$a_o = 0.125 \text{ (* in., initial flaw size *)}$$

$$A = 3.0\times10^{-10} \text{ (* Metal Fatigue in Engineering, 1980, John Wiley \& Sons, pg 86 *)}$$

$$n = 3.25 \text{ (* Metal Fatigue in Engineering, 1980, John Wiley \& Sons, pg 86 *)}$$

$$N_f = \frac{2}{2-n}\left(\frac{1}{A\left(1.12\,\frac{S_2}{1000}\sqrt{\pi}\right)^n}\right)\left(a_{crs}{}^{\left(\frac{2-n}{2}\right)} - a_o{}^{\left(\frac{2-n}{2}\right)}\right)$$

```
(* Damage Tolerant Design Handbook" V.2, 1983 *)
```

$$N_f = \frac{2}{2-n}\left(\frac{1}{A\left(1.12\,\frac{S_2}{1000}\sqrt{\pi}\right)^n}\right)\left(a_{cri}{}^{\left(\frac{2-n}{2}\right)} - a_o{}^{\left(\frac{2-n}{2}\right)}\right)$$

```
(* Damage Tolerant Design Handbook" V.2, 1983 *)
```

Out[14]= 561

Out[15]= 0.33

```
Out[16]= 3.33296×10^7
Out[17]= 136740.
Out[18]= 77.3126
Out[19]= 93.5482
Out[20]= 309.25
Out[21]= 374.193
Out[22]= 10115.1
Out[23]= True
Out[24]= Leaking should occur before failure
Out[25]= 0.125
Out[26]= 3.×10^-10
Out[27]= 3.25
Out[28]= 2.42577×10^6
Out[29]= 2.43076×10^6
```

In[60]:= (* For the ellipsoidal head *)

$D_i = 7.625$

$t_w = 0.5$

$E_f = 0.7$ (*butt weld efficiency based on no inspection, Table UW-12*)

$$\sigma = \frac{MAWP\,(D_i + 0.2\,t_w)}{(2\,E_f\,t_w)}$$

$S_2 = \sigma$

$$a_{crs} = \frac{1}{1.21\,\pi}\left(\frac{K_{Ic}}{S_2}\right)^2$$ (* in., crack critical length, surface flaw *)

$$a_{cri} = \frac{1}{\pi}\left(\frac{K_{Ic}}{S_2}\right)^2$$ (* in., crack critical length, imbedded flaw *)

$lengthc_{crs} = 4 \times a_{crs}$

$lengthc_{cri} = 4 \times a_{cri}$

(* considering leak before break criteria,
leak occurs occurs before catastrophic failure in a pressure vessel when *)

$$\sqrt{\frac{\pi\,t\,S_2{}^2}{1 - \frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}}$$ (* Fracture and Fatigue Control in Structures,

Rolfe and Barsom, Prentice-Hall, 1977, pg. 394*)

$$K_{Ic} >= \sqrt{\frac{\pi\,t\,S_2{}^2}{1 - \frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}}$$ (* Hoop stress applies, S_2 *)

$$\text{If}\left[K_{Ic} >= \sqrt{\frac{\pi\,t\,S_2{}^2}{1 - \frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}},\right.$$

"Leaking should occur before failure", "failure may occur before leaking"]

(* LIFE Expectancy Cycles *)

$a_o = 0.125$ (* in., initial flaw size *)

$A = 3.0 \times 10^{-10}$ (* Metal Fatigue in Engineering, 1980, John Wiley & Sons, pg 86 *)

$n = 3.25$ (* Metal Fatigue in Engineering, 1980, John Wiley & Sons, pg 86 *)

$$N_f = \frac{2}{2-n}\left(\frac{1}{A\left(1.12\,\frac{S_2}{1000}\,\sqrt{\pi}\right)^n}\right)\left(a_{crs}{}^{\left(\frac{2-n}{2}\right)} - a_o{}^{\left(\frac{2-n}{2}\right)}\right)$$

(* Damage Tolerant Design Handbook" V.2, 1983 *)

$$N_f = \frac{2}{2-n}\left(\frac{1}{A\left(1.12\,\frac{S_2}{1000}\,\sqrt{\pi}\right)^n}\right)\left(a_{cri}{}^{\left(\frac{2-n}{2}\right)} - a_o{}^{\left(\frac{2-n}{2}\right)}\right)$$

(* Damage Tolerant Design Handbook" V.2, 1983 *)

Out[60]= 7.625

Out[61]= 0.5

Out[62]= 0.7

```
Out[63]= 10792.9

Out[64]= 10792.9

Out[65]= 42.2259

Out[66]= 51.0934

Out[67]= 168.904

Out[68]= 204.373

Out[69]= 13824.2

Out[70]= True

Out[71]= Leaking should occur before failure

Out[72]= 0.125
```

Out[73]= $3.\times10^{-10}$

```
Out[74]= 3.25

Out[75]= 900192.

Out[76]= 902921.
```

In[104]:= (* For the flat head *)

$\sigma = 5182.85$ (* from head_ 350 _K _openings2.nb*)
$S_2 = \sigma$

$a_{crs} = \frac{1}{1.21\,\pi}\left(\frac{K_{Ic}}{S_2}\right)^2$ (* in., crack critical length, surface flaw *)

$a_{cri} = \frac{1}{\pi}\left(\frac{K_{Ic}}{S_2}\right)^2$ (* in., crack critical length, imbedded flaw *)

$\text{lengthc}_{crs} = 4 \times a_{crs}$
$\text{lengthc}_{cri} = 4 \times a_{cri}$

(* considering leak before break criteria,
leak occurs occurs before catastrophic failure in a pressure vessel when *)

$\sqrt{\frac{\pi\, t\, S_2{}^2}{1 - \frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}}$ (* Fracture and Fatigue Control in Structures,

Rolfe and Barsom, Prentice-Hall, 1977, pg. 394*)

$K_{Ic} >= \sqrt{\frac{\pi\, t\, S_2{}^2}{1 - \frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}}$ (* Hoop stress applies, S_2 *)

$\text{If}\left[K_{Ic} >= \sqrt{\frac{\pi\, t\, S_2{}^2}{1 - \frac{1}{2}\left(\frac{S_2}{\sigma_y}\right)^2}},\right.$

"Leaking should occur before failure", "failure may occur before leaking"]

(* LIFE Expectancy Cycles *)

$a_o = 0.125$ (* in., initial flaw size *)
$A = 3.0 \times 10^{-10}$ (* Metal Fatigue in Engineering, 1980, John Wiley & Sons, pg 86 *)
$n = 3.25$ (* Metal Fatigue in Engineering, 1980, John Wiley & Sons, pg 86 *)

$N_f = \frac{2}{2-n}\left(\frac{1}{A\left(1.12\,\frac{S_2}{1000}\sqrt{\pi}\right)^n}\right)\left(a_{crs}{}^{\left(\frac{2-n}{2}\right)} - a_o{}^{\left(\frac{2-n}{2}\right)}\right)$

(* Damage Tolerant Design Handbook" V.2, 1983 *)

$N_f = \frac{2}{2-n}\left(\frac{1}{A\left(1.12\,\frac{S_2}{1000}\sqrt{\pi}\right)^n}\right)\left(a_{cri}{}^{\left(\frac{2-n}{2}\right)} - a_o{}^{\left(\frac{2-n}{2}\right)}\right)$

(* Damage Tolerant Design Handbook" V.2, 1983 *)

Out[104]= 5182.85

Out[105]= 5182.85

Out[106]= 183.113

Out[107]= 221.567

Out[108]= 732.454

Out[109]= 886.269

```
Out[110]= 6527.84

Out[111]= True

Out[112]= Leaking should occur before failure

Out[113]= 0.125
```

Out[114]= $3. \times 10^{-10}$

Out[115]= 3.25

Out[116]= 9.92359×10^{6}

Out[117]= 9.93542×10^{6}

```
(* Fragment Evaluation *)

(* It is assumed that the most vulnerable point in this vessel is the Ceramaseal high
   voltage feedthroughs mounted to the mini-Conflats which are mounted to
   the 350 MOP head. These could easily be bumped or damaged by mishandling
   resulting in a fragment / projectile. The following will estimate the
   shielding thickness required for personnel protection near the vessel
   head. Also assume all of the energy is transferred to a single fragment. *)
```

In[22]:= $m_{fg} = 37.7$ (* g; actual measurment *)

$m_{fs} = m_{fg} * 6.852\times10^{-5}$ (* lb.s^2/ft; slugs *)

$v_f = \sqrt{\frac{2\,\text{Energy}}{m_{fs}}}$ (* ft/s *)

Out[22]= 37.7

Out[23]= 0.0025832

Out[24]= 6575.9

In[21]:= (* Check zero mass velocity. Fragemnt can travel no faster than the shock wave that is driving it. *)

$P_{ratio} = \frac{P_1}{P_2}$ (*   *)

$g = 32.2$ (* ft/s^2 *)
$T = 528$ (* °R *)
$k = 1.4$
$R = 53.3$ (* ft-lb/lb-°R *)
$a = \sqrt{k\,g\,R\,T}$
$v_{f1} = a \times 2.55$ (* ft/s Figure 12 Zero Mass velocity *)
$v_{f12} = v_{f1}\,\mathrm{Cos}\left[\frac{0.785398}{2}\right]$ (* MEDSS eqn. 38 *)
$v_{f1m} = v_{f1} \times 0.3048$ (* m/s *)
$v_{f1m2} = v_{rf} \times 0.3048$ (* m/s *)
$m_1 = \frac{2\,\mathrm{Energy}}{v_{f1}^2}\,32.2$ (* lb_m; largest fragment that can achieve this velocity *)

Out[94]= 66.5306

Out[95]= 32.2

Out[96]= 528

Out[97]= 1.4

Out[98]= 53.3

Out[99]= 1126.35

Out[100]= 2872.19

Out[101]= 2653.55

Out[102]= 875.443

Out[103]= 808.804

Out[104]= 0.436013

(* The Ceramaseal feedthrough mass is <
m_1 so it can only achive this maximum velocity.

Fragment shielding evaluation...... *)

In[105]:= $T_m = 6\times10^{-5}\left(\frac{m_{fg}}{1000}\right)^{0.33} v_{f1m}$ (* UK formula *)

$T_m = T_m \times 12 \times 3.28084$ (* in *)

(* Thor formula: Lexan *)

$\alpha = 1.814$

$\beta = -1.652$

$c_1 = 7.329$

$$A_f = \frac{\pi\, 0.5^2}{4}$$

$$T_{in} = \frac{1}{A_f}\left(\frac{v_{f1}}{10^{c_1}\left(7000\,\frac{m_1}{32.2}\right)^{\beta}}\right)^{\frac{1}{\alpha}}$$

Out[105]= 0.0178062

Out[106]= 0.701032

Out[107]= 1.814

Out[108]= -1.652

Out[109]= 7.329

Out[110]= 0.19635

Out[111]= 2.36231

(* After ricochet the shielding thickness needs to be:*)

```
(* Thor formula: Lexan *)
```

$$T_m = 6\times10^{-5}\left(\frac{m_{fg}}{1000}\right)^{0.33} v_{flm2} \text{ (* m; UK formula *)}$$

$$T_m = T_m \times 12\times 3.28084 \text{ (* in *)}$$

(* Thor formula: Lexan *)

$\alpha = 1.814$

$\beta = -1.652$

$c_1 = 7.329$

$$A_f = \frac{\pi\, 0.5^2}{4}$$

$$T_{in} = \frac{1}{A_f}\left(\frac{v_{f12}}{10^{c_1}\ (7000\ \frac{m_1}{32.2})^{\beta}}\right)^{\frac{1}{\alpha}}$$

Out[112]= 0.0164508

Out[113]= 0.647669

Out[114]= 1.814

Out[115]= -1.652

Out[116]= 7.329

Out[117]= 0.19635

Out[118]= 2.26142