Next100 Pressure Vessel - User's Design Specification (DRAFT 2)

Derek Shuman[1], **Sara Cárcel**[2] **A. Martínez**[2]

NEXT Collaboration

[1]Lawrence Berkeley National Laboratory (LBNL), Berkeley CA, USA
[2]Instituto de Física Corpuscular (IFIC), CSIC, Univ. de Valencia, Valencia, Spain,

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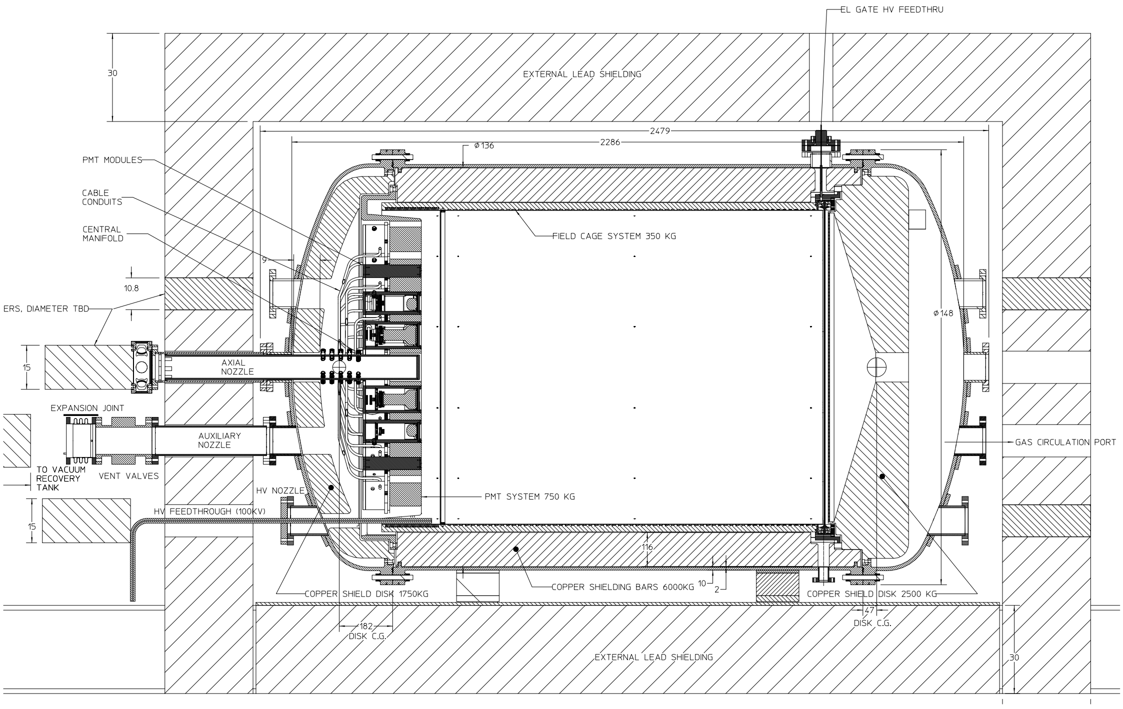

Figure 1: Detector Cross section

1 Introduction

This document is a preliminary User's Design Specification for a pressure vessel to be used in a neutrino physics experiment called NEXT-100. The final specification is expected to be released for RFQ approx. 2 months after this release. Drawings presented here are not to be considered final. Prospective Manufacturers are encouraged to provide feedback on the design, and details of fabrication, as well as preliminary cost and schedule estimates.

2 Purpose

The NEXT Collaboration is a group of physicists and engineers affiliated with Institute of Particle Physics/ University of Valencia, (IFIC) (principal institution), LBNL, and many others. The NEXT-100 experiment proposal is funded by this collaboration to build a detector to look for a phenomenon called neutrinoless double beta decay. The experiment requires a pressure vessel, to be used for gas containment, and additionally as the housing and support for a neutrino detector installed inside. Figure 1, below, shows a cross section of the detector inside the pressure vessel. This pressure vessel is the subject of this Specification.

3 Introductory Requirements Description

The pressure vessel has the following general requirements:

1. Size, Shape, Orientation: 1.360m inner diameter x 2.286m inside length, cylindrical main vessel section with detachable torispheric heads on each end, vessel axis horizontal, with two saddle supports welded to main vessel shell. Welded-in nozzles on both the main vessel and the heads extend the overall size to 2.8m overall length x 1.5m high.
2. Assembly Configuration: 3 parts, a main cylindrical vessel with bolted flange connections to torispheric heads on each end. Flanges are flat faced with double O-rings or, possibly a Helicoflex C-ring/O-ring combination, to provide pressure and vacuum seal on both main flanges and on nozzle flanges.
3. Material: Main Vessel and Heads, shells, nozzles and flanges: Stainless Steel, 316Ti plate (UNS S31635, EN 1.4571) per ASME specification SA-240 or (EU) equivalent; main flange bolting: Inconel 718 ASME specification grade SB-637 (UNS N07718) or (EU) equivalent.
4. Fluid: gaseous xenon (primary), argon, neon, nitrogen, dry air, with small amounts of CF4, CH4, H2, at room temperature to 50C (negligible corrosive, flammable or toxic hazard).
5. Design Pressure Range : -1.5 atm to 15.4 barg (16.4 bara)
6. Leak Tightness: (1X10^{-6} torr*L/sec
7. Design Standard: ASME Pressure Vessel Code section VIII, division 1, using full weld efficiency (fully radiographed double and full penetration welds required). Other design standards allowable in Canfranc, Spain may be used. Note that although the requirement for a User's Design Specification is only required for Pressure Vessels designed under ASME section VIII division 2 rules, we regard this as an essential controlling document for the vessel.
8. Low residual background radioactivity; additional material and process screening, over and above that required by ASME Pressure Vessel Code, or equivalent, will be performed by the Collaboration; full cooperation of Manufacturer is required. Nominal design may be impacted by (now pending) test results.
9. Internal detector components will be supported on internal flanges on the vessel (on both main cylindrical and on torispheric heads), and nozzle flanges. Total weight of detector inside vessel does not exceed 13000 kg (13 metric tons).

These requirements and others are fully detailed in the Requirements section below. This includes requirements outlined in ASME PV code sec VIII, division 2, part 2.2.2 "User's Design Specification". We continue with the general description:

There are two unique and noteworthy aspects of this vessel; the first is a radiopurity requirement and, the second to a lesser extent, the need to mount internal components. The detector inside the vessel is highly sensitive to radiation from trace amounts of uranium (U) and thorium (Th). Austenitic stainless steel alloys contain uranium and thorium in trace amounts, from several parts per trillion (ppt) to hundreds of parts per million (ppm), only the lower levels are acceptable to us. 316Ti (1.4571) has been well characterized by others and found to typically show acceptably low levels; this is the reason we are specifying its use, even though other alloys may also be acceptable.

To assure sufficient radiopurity of materials, the Collaboration will require samples (several kg. each) from all raw material lots (bar, plate, tubing, forging ends, etc.) in order to perform background radiation counts prior to the material be accepted for fabrication, These counts take 1 month each to perform and we can only perform 2 simultaneous sample measurements at one time, so adequate material procurement scheduling is required.

Regarding manufacturing, thoriated TIG welding electrodes, and guns that have been used with such electrodes must not be used. Ceriated, lanthanated, yttriated, or plain tungsten electrodes are acceptable. Special cleaning procedures for material preparation are required, and may be subject to testing by the Collaboration and may be modified

The pressure vessel also serves to support the detector inside. The detector contains a large amount of radiation shielding, in the form of precision machined copper bars and plates, approx. 12000 kg of copper in all. The vessel to head flanges incorporate internal flanges for mounting of both this copper shielding and the detector components. As such all final machining on the must be performed only after a full stress relief anneal is performed after welding operations. Head to vessel flanges are nominally bolted; a flat faced flange design is used having 2 O-rings for seals, with a vacuum sense port in between them to detect leakage. The inner groove will be compatible, if feasible, with a Helicoflex gasket, loaded to its Y1 unit force. Manufacture must only demonstrate proper sealing performance using elastomer O-ring seals in both grooves. Manufacturer is invited offer some details as to preferred fabrication details before final specification is issued.

4 Parties to the Contract

Henceforth in this Document, the parties to the contract are listed and defined as follows:

4.1 Collaboration

The Collaboration is headed by Dr. J.J. Gomez, IFIC. The lead mechanical engineer for the pressure vessel is Derek Shuman, LBNL, with assistance from mechanical engineers: Sara Cárcel and Alberto Martínez (IFIC). Sara will be the prime contact person overseeing fabrication.

4.2 Manufacturer

This is the primary firm contracted with the Collaboration to perform or coordinated the design, fabrication, and testing. Subcontractors are not included, however, the rights of inspection negotiated between the Collaboration and the Manufacturer must be extended to apply to all subcontractors.

4.3 Certification Authority

This is an independent Certification Authority contracted to certify this document for completeness and correctness prior to the commencement of fabrication, and also to certify the Manufacturer's Design Report prior to the acceptance of the vessel by the Collaboration.

4.4 Inspector

This is a qualified person provided by the Certification Authority to perform inspections of all aspects of the design, fabrications and testing, in order to verify that the vessel has been designed, fabricated and tested in full compliance with the appropriate pressure vessel code.

5 Scope of Contract

Manufacturer is to supply, at a minimum, the complete vessel, in a clean condition compatible with high vacuum testing, complete with all flange bolts, nuts, washers, and all blank-off plates used for hydrostatic testing. Optionally, the Manufacturer may additionally supply the nozzle extensions, and other internal parts of the detector. Excess unused plate material shall be returned to Collaboration, if feasible. The Collaboration will supply the pressure relief devices. Here is a detailed list:

1. (1) main vessel, per LBNL drawing 26K591A
2. (2) torispheric heads per LBNL drawing 26K591A
3. (300) studs per LBNL drawing 26K593A
4. (300) heavy hex nuts, M16-1.0, Inconel 718 per ASME SB-637, silver plated for item 3 above
5. (600) heavy narrow washers, 28 mm OD Inconel 718 per ASME SB-637, for 3
6. (4) O-rings, nitrile, 5mm x 1320mm ID
7. (4) O-rings, nitrile, 3mm x 1460mm ID
8. (16) O-rings, nitrile, 3mm x 90mm ID
9. (16) O-rings, nitrile, 3mm x 110mm ID
10. (4) O-rings, nitrile, 3mm x 65mm ID
11. (4) O-rings, nitrile, 3mm x 70mm ID
12. (12) O-rings, nitrile, 3mm x 35mm ID
13. (12) O-rings, nitrile, 2.5mm x 50mm ID
14. (2) heads per LBNL drawing 26K591A
15. (9) cover plates, DN100 per LBNL drawing 26K594A
16. (3) cover plates, DN75 per LBNL drawing 26K595A
17. (7) cover plates, DN40 per LBNL drawing 26K596A
18. hardware (316 SS, silver plated) to attach the above 3 items to vessel flanges
19. Manufacturers Design Report, in both Spanish and English

6 Responsibilities

6.1 Manufacturer

6.1.1 Material Use Planning

Manufacturer is to use materials provided by the Collaboration to fabricate the vessel, unless other arrangement is made. Manufacturer is required to approve any materials provided by the Collaboration with regards to fitness of use. Manufacturer must request any certifications, samples needed for testing. Manufacturer is to specify the range of raw material sizes, and the amounts of each needed to fabricate the vessel All materials and equipment used, that are supplied by the Manufacturer, are subject to approval by the Collaboration, both raw materials that will be part of the vessel, and all other materials and equipment used in the manufacturing process.

6.1.2 Final Design

Manufacturer is responsible for the pressure integrity of the vessel and is required to perform all necessary calculations and analyses, as Manufacturer sees fit. Detailed preliminary calculations are provided by the Collaboration, in the Appendix, as a convenience, and to justify the dimensions of the vessel presented here in this Specification, however manufacturer is ultimately responsible for pressure integrity and sufficiency of design. Manufacturer may propose changes to the design, however these must be approved by the Collaboration. The design presented here is performed according to the rules of ASME section VIII, division 1, with full weld efficiency, which is required in the final design. This requires full penetration double welds on the major welds plus a full radiographic inspection. Other codes that are acceptable in the Jurisdiction of Canfranc, Spain are acceptable in part, or in full, as allowed by the codes themselves.

6.1.3 Fabrication Plan

Manufacturer is to submit a detailed fabrication plan to Collaboration for approval prior to commencement of fabrication, describing the sequence of operations to be used in fabricating and testing the vessel. These shall include (but not be limited to) the following:

1. Main Cylindrical Vessel: shell forming and welding sequence, all dimensions of shell sections, rolling methods, edge preparations and cleaning, welding procedures and equipment, intermediate heat treatments, and inspections.
2. Torispheric Head: shell forming, rolling methods, edge preparations, intermediate heat treatments, and inspections.
3. Head to Vessel Flange fabrication sequence, all dimensions of flange sections, edge preparations, welding procedures, intermediate heat treatments and inspections.
4. Flange to Shell Weld joint design

6.2 Collaboration

Collaboration is responsible for finding and securing the required material in timely manner prior to scheduling construction. The Collaboration is responsible for timely radiopurity testing of material samples from lots. Each of these measurements can take up to one month to complete. A schedule of radiopurity measurements will be drafted once the manufacturing process is fully known.

7 ASME User Design Specification

2.2.2.1 ASME required specifications

a) Installation Site

1) Location - Installed location - Canfranc Spain, inside Canfranc Under Ground Laboratory in LSC Hall 1. Vessel may be staged temporarily at some other location, perhaps for pressure testing, and/or for trial assembly of detector. This location will be either at IFIC in Valencia, or perhaps at University of Zaragoza.

2) Jurisdictional Authority All that are required for the locations listed above

3) Environmental conditions

i) Wind loads - None

ii) **Seismic Design Loads -** 1m/s^2 (0.1g) maximum vertical (over static gravity); 2 m/s^2 maximum horizontal acceleration. Vessel will be mounted on a shock isolating platform, and will be elevated above the hall floor by 1.2m

iii) **Snow Loads -** None

iv) **Lowest one day mean temperature-** 10C . Note - remote possibility exists of cryogen spill underneath pressure vessel, with temperature unknown. Cryogenic liquid is not expected to contact vessel, as the vessel will be mounted on a platform at least 1.5 m above the main hall floor, and a total cryogenic liquid spill will result in at most a few cm of liquid height on floor. Nevertheless, vessel will be immediately vented to 0 barg upon receiving a fault signal indicating a cryogen spill in the LSC hall.

b) **Vessel Identification**

1) **Vessel Number -** "NEXT100-PV1"

2) **Fluids -** gaseous xenon (primary), argon, neon, nitrogen, dry air, with small amounts of CF4, CH4, H2 (<5%), all held at room temperature to 50C (negligible corrosive, flammable or toxic hazard). No liquids will be introduced into the vessel, other than cleaning, in the disassembled condition or perhaps in an assembled condition, unpressurized. Although not presently planned, the vessel may eventually be immersed in a fluid bath, of either ultrapure water, or scintillator fluid (as yet unknown), with the vessel in either the pressurized or vacuum condition. Maximum fluid pressure of this bath will be, at the lowest point of the vessel no higher that 0.35 barg, from hydrostatic head only, the water or scintillating fluid being at atmospheric pressure. There should be no corrosion allowance made in the design for this possible future use; adjustment will be made to operating pressure if needed.

c) **Vessel Configuration and Controlling Dimensions -** The vessel will be oriented with its axis of revolution in the horizontal direction. LBNL Drawing number 26K589A shows the assembled vessel with controlling dimensions, some of which are listed below:

Inside Diameter, Vessel and Head Shells	1360 mm
Inside Length, on Centerline Axis, including Heads	2286 mm
Length, Main Cylindrical Vessel	1600 mm
Torispheric Head Inner Crown Radius (R_c=1.0D, Kloepper)	1360 mm
Torispheric Head Inner Knuckle Radius (R_k=0.1D, Kloepper)	136 mm
Center axis height above floor(including support pads)	800mm

Table 1: Required Geometric Values

d) **Design Features**

1) **Supports -** The vessel shall be designed with saddle supports welded to the main cylindrical vessel. These supports shall be sufficient to support the weight of the pressure vessel, with all internal components and fluids, for static gravity plus the maximum seismic acceleration, described below. The supports shall be designed to accommodate expansion and contraction of the vessel, from both pressure/vacuum and from temperature excursions. The vessel must return to the same position upon returning to normal operating temperature. The proposed design utilizes low friction polymer bearing pads in a 2D kinematic support arrangement; Manufacturer may propose alternate methods

or materials. The vessel may be lifted with slings while empty. It is not foreseen that the vessel will be lifted with the internal copper shielding inside, however jacking screws on the support feet are provide for leveling with the copper inside. The Appendix contains a set of illustrations detailing the proposed design.

The torispheric heads are not required to have lifting lugs welded to them, though Manufacturer may elect to add these, with prior approval. If lugs are added, they must be designed for the entire mass of the head plus its internal copper disk (2500 kg). The collaboration will be using specially designed lift fixtures to handle the heads; these attach to the head using various combinations of the flange bolt (clearance) holes. Some of these clearance holes will be threaded to accept certain lift fixture mounting bolts. See section on Loads below for further description.

e) Design Conditions

Internal Pressure, MPa (g)	External Pressure, MPa (g)
1.54	0.15

Table 2: Design Pressures,(gauge)

f) Operating Conditions

1) Maximum Operating Pressure (MOP) - 14.0 barg

2) Maximum Allowable Working Pressure (MAWP) - 15.4 barg

3) Operating Temperature - 15C-30C. Temperature may rise to 150C under a vacuum (-1.0 barg) condition, but not under pressure condition.

4) Fluid Transients and Flow - A typical operating cycle, following any condition requiring the vessel to be opened, is as follows:

1. Vessel will be pulled to vacuum condition and held for several days. Vessel may be heated to 50C during this operation.
2. Xenon gas will then be introduced at a slow fill rate, no less than 10 min. to fully pressurize.
3. Xenon gas will then be circulated at 200 SLPM through the vessel/ purifier circuit. Detector will be operated during this time, continuously, without interruption of flow or pressure, for as long as possible.

Should the detector need a repair which requires removing at least one of the heads or nozzle attachments, the following sequence of operations is performed:

1. Vessel will be vented by opening a valve connecting to a cryogenic recovery cylinder, depressurization is expected to take at least 30 minutes. Pressure will be reduced to less than 1 torr.
2. Vessel will be filled with clean dry air to 0.01-0.1 barg, then vented to atmosphere.
3. The head assembly fixture will be assembled to the floor.
4. Head to flange studs occupying the threaded holes in the head are be removed
5. The head assembly fixture is then aligned to closely mate with the head flange and is then bolted to the head
6. The remainder of the head to vessel flange studs are removed to allow head to be moved away from the vessel, on the fixture rails, by a distance of 1m

7. A second lift fixture is then bolted to the head using a number of holes at the top; this lift fixture provides a single lift point for attachment to a crane hook, with the lift point over the center of gravity of the head/internal copper shield disk assembly. See section on Loads below for further description.

Under a possible emergency condition defined as an abnormally high pressure drop rate occurring during normal operation, the vessel will be vented in approximately 10 sec. to the large vacuum tank, in order to minimize loss of the xenon into the LSC Hall. This will be accomplished by using at least one remote operated active vent valve that will open fully upon receiving a controller signal. This valve is in parallel with the main pressure relief valve. The active vent valve will be a straight-through design, so that reaction force does not exert a force transverse to the nozzle axis. However, the possibility exists that a right angle valve might mistakenly be substituted, and so the auxiliary nozzles on each head must be designed to withstand a moment caused by the reaction force from this valve. The maximum flow rate will be 25 kg/sec (Xe). The reaction force associated with this flow (xenon) is 3500 N. The vent valve will be located on the end of a nozzle extension that is 58cm long; thus a maximum moment of 2300 N*m may be applied to the nozzle to head weld; this is the design requirement for weld and nozzle sufficiency. The relief valve for fire condition or failed regulator is much lower flow and may be a right angle valve.

g) **Design Fatigue Life -** The vessel is estimated to undergo not more than 200 full pressure cycles, at most (including head removals) during its lifetime. Each vessel head is not expected to be removed more than 50 times. Pressure will remain static for each pressure cycle; i.e. pressure is not varied during vessel operation, only during filling and venting. Some pressure cycles will be less than full MOP. Pressure change rate under typical operating conditions will be low, less than 0.001 bar/sec. There will be a few rapid depressurization events, of maximum 2 bar/sec using a remote actuated vent valve; not more than 10 of these cycles are estimated to occur, primarily from testing.

h) **Materials of Construction**

1) **Vessel -** Stainless Steel plate, 316Ti (UNS S31635, EN 1.4571) per ASME SA-240 specification or equivalent: all vessel shells, nozzles and head-to vessel flanges shall be made from plate unless other forms are allowable, under acceptable pressure vessel codes other than ASME sec VIII, div. 1. Material will be provided by the Collaboration.

2) **Bolting -** Inconel 718 (UNS N77180) to ASME SB-637 standard or equivalent, studs, nuts and washers. Material will be provided by the Collaboration.

3) **Flange Seals**

i) **Main Head to Vessel Flanges-** O-rings: butyl, nitrile, possibly PCTFE or PTFE ((nozzle flanges only), or special low force Helicoflex (type HN200), aluminum jacket. Main head to vessel flanges will be double O-ring sealed, the larger cross section O-ring for pressure, the outer, smaller cross section for vacuum. The annulus between them will incorporate a sense port for leak checking the O-rings.

ii) **Nozzle flanges** All nozzle flanges are sized to ASME standards (section VIII, div 1, Appendix Y) using standard CF knife edge flange bolt patterns. This is to allow use of pressure rated or tested CF components via an adapter flange, if needed. The flange sealing faces shall be plain flat faced, to be used only with special interface flange plates having double O-ring, or O-ring/Helicoflex seals. Design gasket force shall be for Helicoflex Y2 values, if feasible, but at least 3x Y1 value.

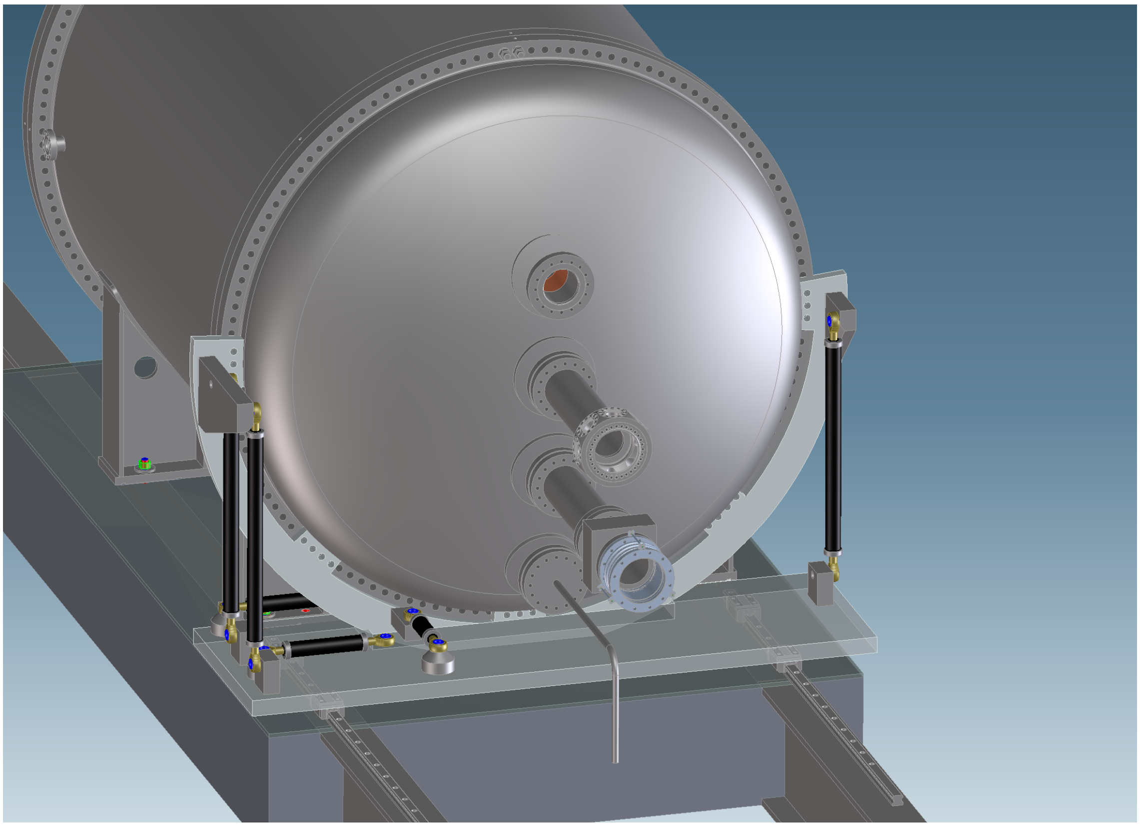

Figure 2: Head Assembly Fixture
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i) Loads and Load Cases - The vessel supports an internal detector of 12000 kg. The weight is almost entirely in the form of copper bars supported by the main cylindrical vessel and copper disks supported by the heads.

A circular array of copper bars, each the full length of the main cylindrical vessel will be mounted to the internal flanges of the main cylindrical vessel; these comprise a total of 6000 kg. An additional 1000 kg of detector components are attached to each end of the bar array, for a total mass of 8000 kg supported by the two internal flanges of the main cylindrical vessel, 4000 kg on each flange. These bars attach to the vessel only by the internal flanges, no contact is made with the main vessel shell.

One head will contain a copper disk of 1500 kg, the other a disk of 2500 kg. , each mounted to the internal flange of the head. The heads must support this weight in any orientation when detached from the main cylindrical vessel. Provision is made for differential thermal expansion between head and disk, to eliminate unwanted stress. The heads are handled separately by attaching lift fixtures to the flange bolt holes, using through bolts and nuts. To perform the initial separation of the head from the vessel, some of these flange holes are threaded for a several mm size larger bolt, in order to attach the head retraction fixture which is used to bring the head up to/away the vessel (on precision rails). The resulting effective hole diameter still yields acceptable stress levels under operating pressure. These threaded holes must be sleeved when the head is bolted to the main vessel, so as to prevent interference between them and the bolt threads. Fig . **??** shows the fixture in use:

j) Overpressure Protection - There are no conditions, short of a fire in the LSC hall that can lead to an overpressure condition. There are no flammable gas mixtures, nor any oxidizing gases inside the vessel at any time when it is closed (dry air may be circulated through the vessel when heads are removed to allow people to work inside). There are only metals, ceramics and common plastic materials such as polyethylene, PEEK, PTFE, PMMA, epoxy, etc. inside the vessel, There are electrical components inside generating no more than 1 kW of heat dissipation, these will be actively cooled with water cooling circuits, either inside the vessel, or outside, using either the xenon gas or the vessel as a heat transfer surface (10C maximum allowable temperature rise above 20C ambient; 30C actual temp). Fast vent capability is incorporated solely for the purpose of minimizing gas loss in the case of an unexpected leak, as the xenon gas is very expensive and the LSC Hall is an enclosed space. Fast venting, in an emergency, will be done by actuating a remote operated vent valve leading directly to a large evacuated recovery cylinder of 20-30 m^3 (thus reducing pressure to <1 bara). The high cost of the xenon, and the enclosed underground cavern combined with the potentially dangerous anesthetic properties of xenon gas preclude venting directly to atmosphere. There will be two relief devices, one passive and one active:

1) Passive: Pilot operated reclosable relief valve, back pressure insensitive - set to 100%MAWP), valve sized for fire or malfunctioning regulator.

2) Pilot operated servo vent valve, 65 mm vent dia. - Set to actuate only upon emergency signal (indicating a substantial leak), at any pressure, for fast vent to vacuum tank. maximum discharge rate is, for xenon at 15 bara 25 kg/sec. This valve will be a straight through design, to eliminate torque on the nozzle from reaction (it may be located at the end of a 60 cm long nozzle extension).

2.2.2.2 Additional Specifications

a) Material Supplied by Collaboration -

As part of the radiopurity requirement below, the Collaboration will find, measure samples and purchase all consumable materials used for fabrication, including plate and welding wire. These shall be purchased and secured prior to commencement of fabrication. Manufacturer shall submit a list of materials to purchase which must include all necessary allowances for trimming and finish machining.

b) Minimum Thickness Design -

This requirement is driven by the radiopurity requirements below; the proposed design assumes a weld efficiency of 1.0 for any division 1 calculations and thus will require a full radiographic inspection of all pressure bearing welds. Welds in category A and B will be required to be full penetration double welds.

The use of a large number of flange bolts and the specification for using fine thread Inconel 718 bolting material follows, this minimizes flange outer diameter; bolt holes will need higher than usual dimensional accuracy and must not be rough drilled prior to final solution annealing.

c) Use of Plate Stock for Flanges -

The flat faced flange design used is designed to div. 1 rules, Appendix Y. Div. 1 allows flanges to be made from plate if no hub is present (which is the case in this design). Plate stock is not ideal for machining into flanges, and there is some possibility that leakage paths from laminar flaws inside the plate may compromise the vacuum tightness of the vessel. 316Ti plate is the only permissible form in Div. 1. and the Collaboration is supplying 50mm stock which precludes rolling and welding into a ring. Manufacturer is cautioned to perform any necessary tests such as X-ray, ultrasound, or helium leak checking of samples, above that which are required under ASME code, so as to assure vacuum tightness as specified in the Specification.

d) Radiopurity Assurance -

In order to assure that all materials used to fabricate the vessel are of high radiopurity and do not become contaminated in the fabrication process, there will be additional material checks of both raw material, and of samples from each fabrication process along the way. Every effort will be made to determine the scope of these checks. Therefore Manufacturer has a responsibility to disclose any and all fabrication processes to be used, both prior to start of fabrication, and through fabrication, inspection and testing. Each material sample test takes 3 weeks to perform, so Manufacturer must be forthcoming in disclosures. Tests may be performed on the following items:

1) Possible Tests on Materials

- **i)** ends from all raw plate used to fabricate vessel shells, flanges, attachments and supports
- **ii)** ends from all finished rollings and spinnings after welding, cylinder and both heads
- **iii)** ends from all bar, pipe and tube stock used for nozzles and nozzle flanges
- **iv)** ends or samples from all bar stock used to fabricate flange bolts and nuts, if applicable

2) Procedures -

- **i)** All welding shall be of the gas tungsten arc (GTAW) in accordance with ASME procedures.
- **ii)** The welding will be done by ASME or equivalent qualified welders, as per ASME requirements.
- **iii)** Filler for welding: 316Ti commercial use filler. Filler material samples shall be submitted to the Collaboration for radiopurity measurement; if unacceptably contaminated, filler metal may be made from from the supplied plate stock.
- **iv)** All parts shall be thoroughly cleaned prior to assembly and welding for welding per the following process:
  - **a.** All supplies and tools to be used are subject to approval by the Collaboration. Manufacturer shall submit a list of list of tools and machinery used for cleaning, to the Collaboration prior to use.
  - **b.** The bending rolls shall be cleaned before the bending operation with an appropriate surface with clean rags.
  - **c.** The welding should be done in a clean enclosed space specific to stainless steel, to prevent inclusion of iron or other contaminants.
  - **d.** Filler must be cleaned and dried prior to use, per ASME or equivalent standards.
  - **e.** The assembly/welding area should be isolated and clean, without contamination of other work.
  - **f.** Thoriated electrodes, as well as guns and shields previously used with thoriated electrodes must *NOT* be used. Plain tungsten (WP EN 26 848, 99.8% minimum tungsten, green), ceriated, yttriated or lanthanated electrodes are acceptable. Shielding gas is argon with a minimum purity of 99.99%, group I, IN 439)

e) Precision Tolerances - To assure that vessel is fabricated on time, and to avoid unnecessary rework, it is imperative to follow a well thought out sequence of fabrications. Manufacturer is to submit a fabrication plan to Collaboration for approval, as mentioned elsewhere. Fabrication requirements:

1) All welding to be performed with flanges in rough machined condition. No flange bolt holes must be present in head to vessel flanges before full solution anneal, below. Nozzle flanges may be prewelded to nozzles, finish machined, then welded to main vessel and heads after main vessel solution anneal below.

2) Torispheric head shells are recommended be in fully solution annealed prior to welding to head flanges.

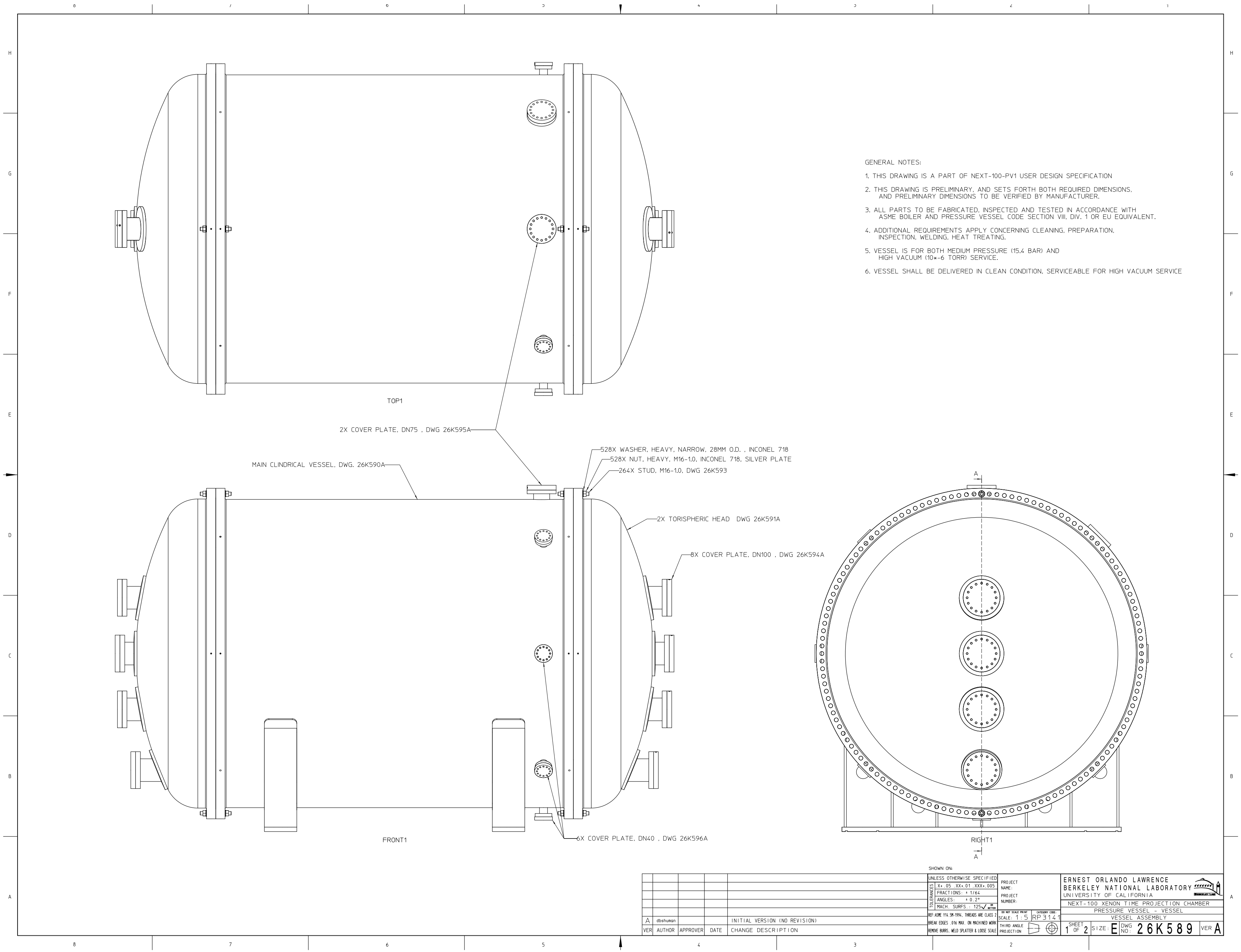
3) Main cylindrical vessel shell is recommended to be fully solution annealed prior to welding to vessel flanges.

4) Vessel and heads shall be full solution annealed after welding (1050C-1120C) for 1 hr minimum, followed by a slow cooldown period of not less than 8 hrs (4 hrs/25mm of section); vessel and heads shall be placed with axes vertical on flat surfaces, in a free unstressed, and unconstrained condition for the duration of this this operation. If annealing is performed as a bright anneal process, argon or full vacuum must be used; hydrogen shall not be used.

5) Saddle supports are to be fabricated separately, and solution annealed as above prior to final machining and welding to main cylindrical vessel.

6) Nozzles and saddle supports may be welded to vessel and heads after final machining of main flanges, if Manufacturer is confident final tolerances on drawing can be met.

8 Drawings


TOP1
FRONT1
RIGHT1
A
A
2X COVER PLATE, DN75 , DWG 26K595A
MAIN CLINDRICAL VESSEL, DWG, 26K590A
528X WASHER, HEAVY, NARROW, 28MM O.D. , INCONEL 718
528X NUT, HEAVY, M16-1.0, INCONEL 718, SILVER PLATE
264X STUD, M16-1.0, DWG 26K593
2X TORISPHERIC HEAD DWG 26K591A
8X COVER PLATE, DN100 , DWG 26K594A
6X COVER PLATE, DN40 , DWG 26K596A
GENERAL NOTES:
1. THIS DRAWING IS A PART OF NEXT-100-PV1 USER DESIGN SPECIFICATION
2. THIS DRAWING IS PRELIMINARY, AND SETS FORTH BOTH REQUIRED DIMENSIONS, AND PRELIMINARY DIMENSIONS TO BE VERIFIED BY MANUFACTURER.
3. ALL PARTS TO BE FABRICATED, INSPECTED AND TESTED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII, DIV. 1 OR EU EQUIVALENT.
4. ADDITIONAL REQUIREMENTS APPLY CONCERNING CLEANING, PREPARATION, INSPECTION, WELDING, HEAT TREATING.
5. VESSEL IS FOR BOTH MEDIUM PRESSURE (15.4 BAR) AND HIGH VACUUM (10**-6 TORR) SERVICE.
6. VESSEL SHALL BE DELIVERED IN CLEAN CONDITION, SERVICEABLE FOR HIGH VACUUM SERVICE
A dbshuman INITIAL VERSION (NO REVISION)
VER AUTHOR APPROVER DATE CHANGE DESCRIPTION
SHOWN ON:
UNLESS OTHERWISE SPECIFIED
TOLERANCES
X± .05 .XX±.01 .XXX±.005
FRACTIONS: ± 1/64
ANGLES: ± 0.2°
MACH. SURFS.: 125
REF ASME Y14.5M-1994, THREADS ARE CLASS 2
BREAK EDGES .016 MAX. ON MACHINED WORK
REMOVE BURRS, WELD SPLATTER & LOOSE SCALE
PROJECT NAME:
PROJECT NUMBER:
DO NOT SCALE PRINT
SCALE: 1:5
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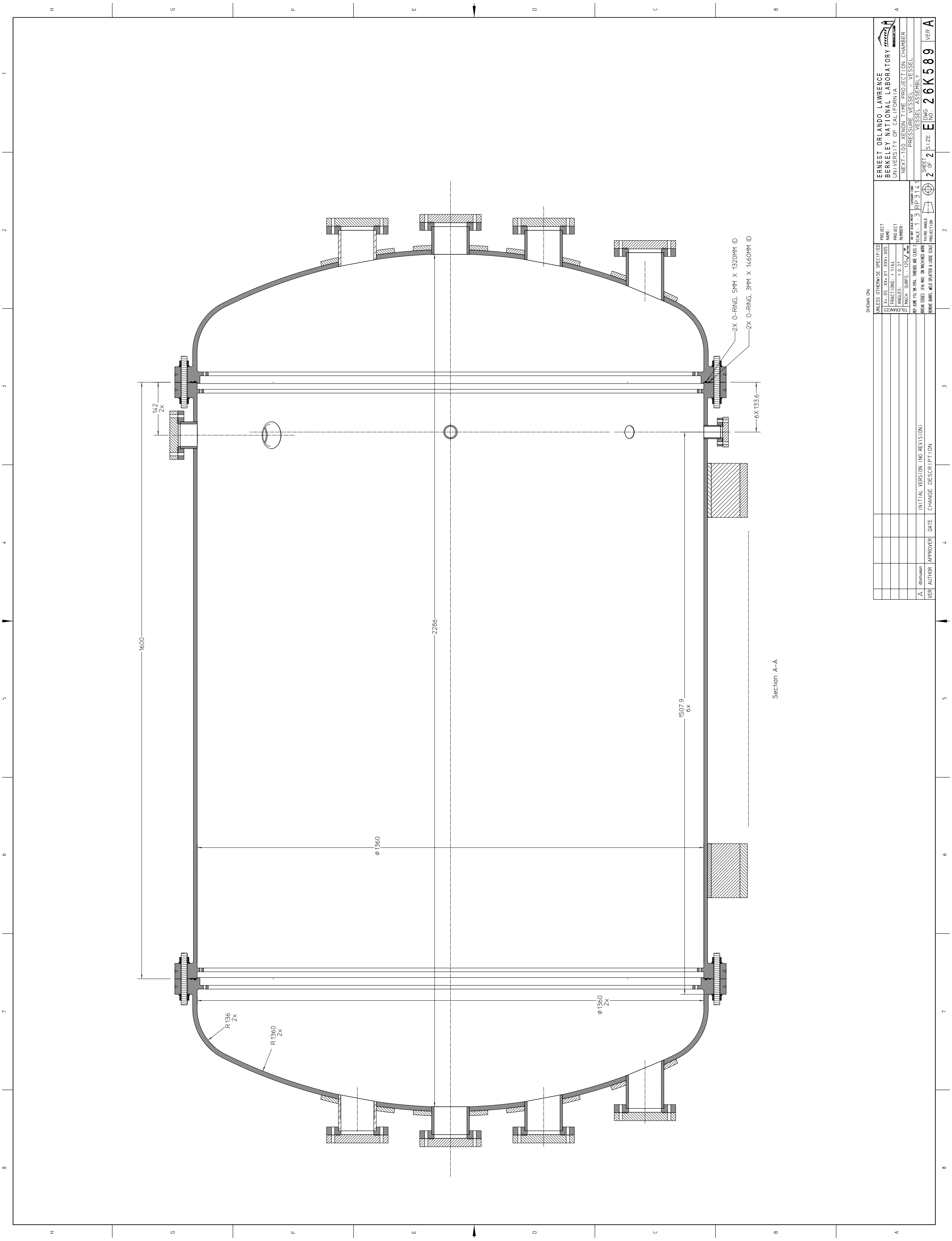
RP3141
THIRD ANGLE PROJECTION
ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
VESSEL ASSEMBLY
SHEET 1 OF 2
SIZE: E
DWG NO: 26K589
VER A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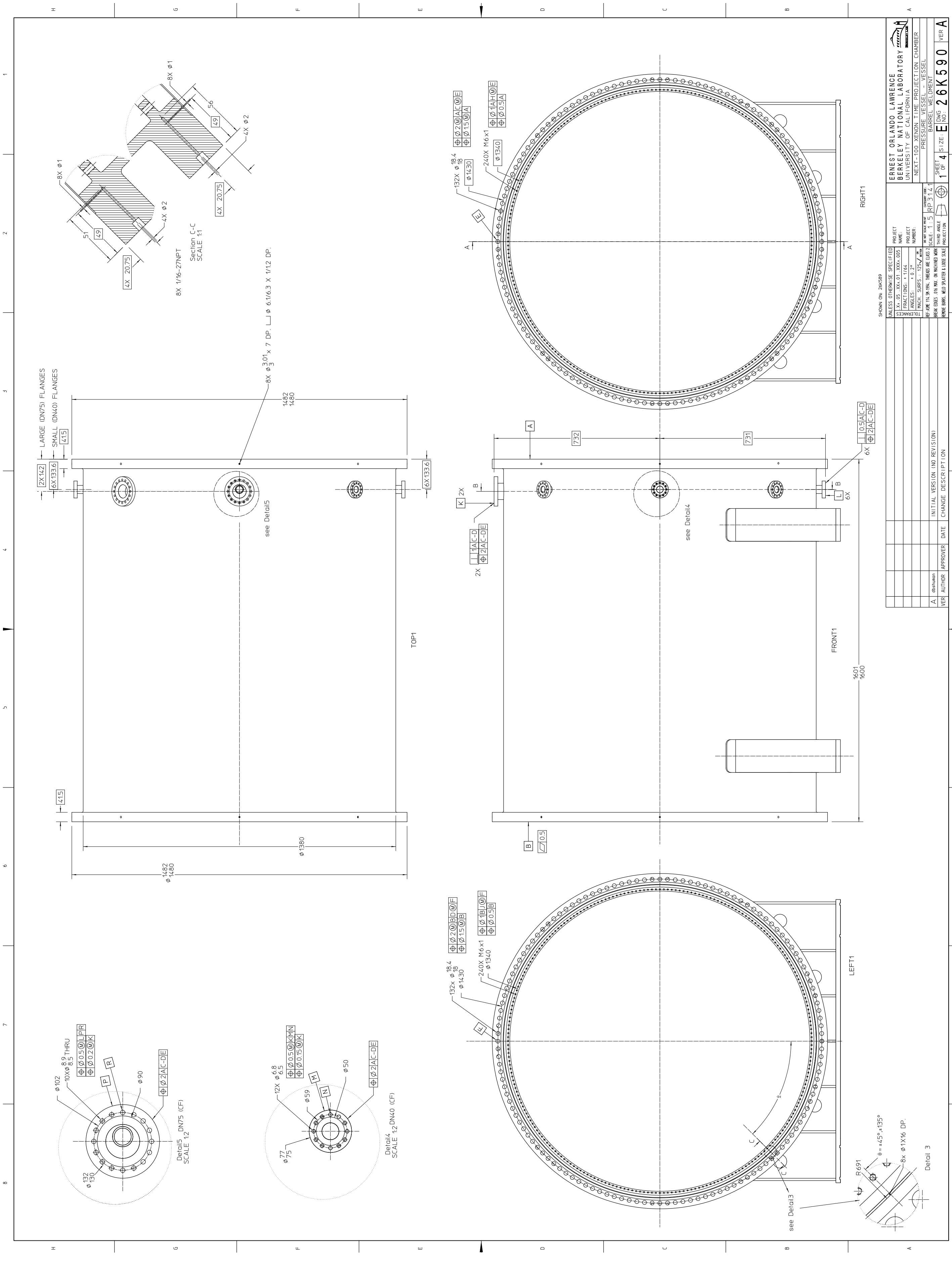
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
VESSEL ASSEMBLY
26K589
INITIAL VERSION (NO REVISION)
2X O-RING, 5MM X 1320MM ID
2X O-RING, 3MM X 1460MM ID
Section A-A
2286
1600
ø1360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BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
BARREL WELDMENT
26K590
TOP1
FRONT1
LEFT1
RIGHT1
Section C-C
SCALE 1:1
Detail5 DN75 (CF)
SCALE 1:2
Detail4 DN40 (CF)
SCALE 1:2
Detail 3
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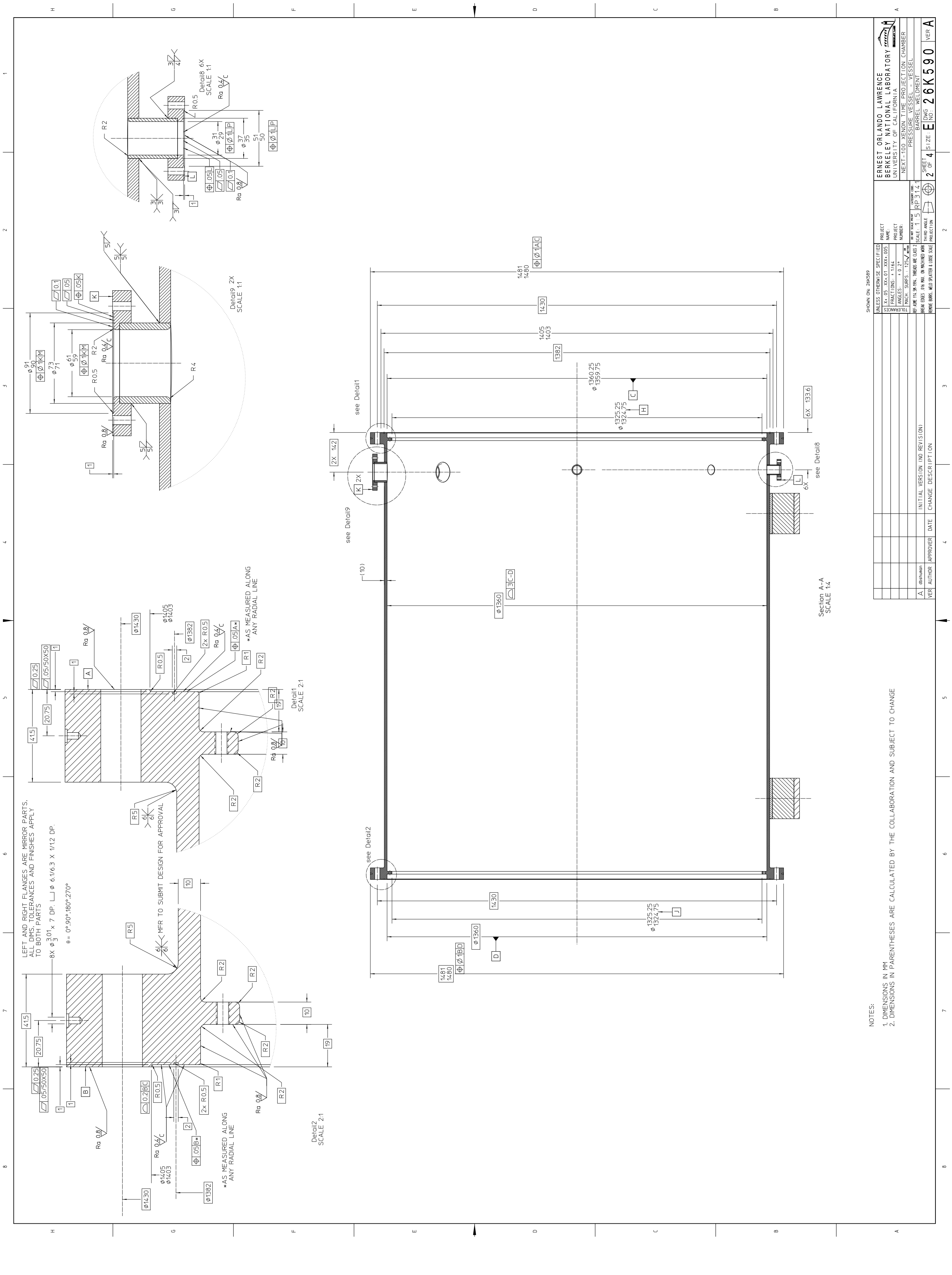
CHANGE DESCRIPTION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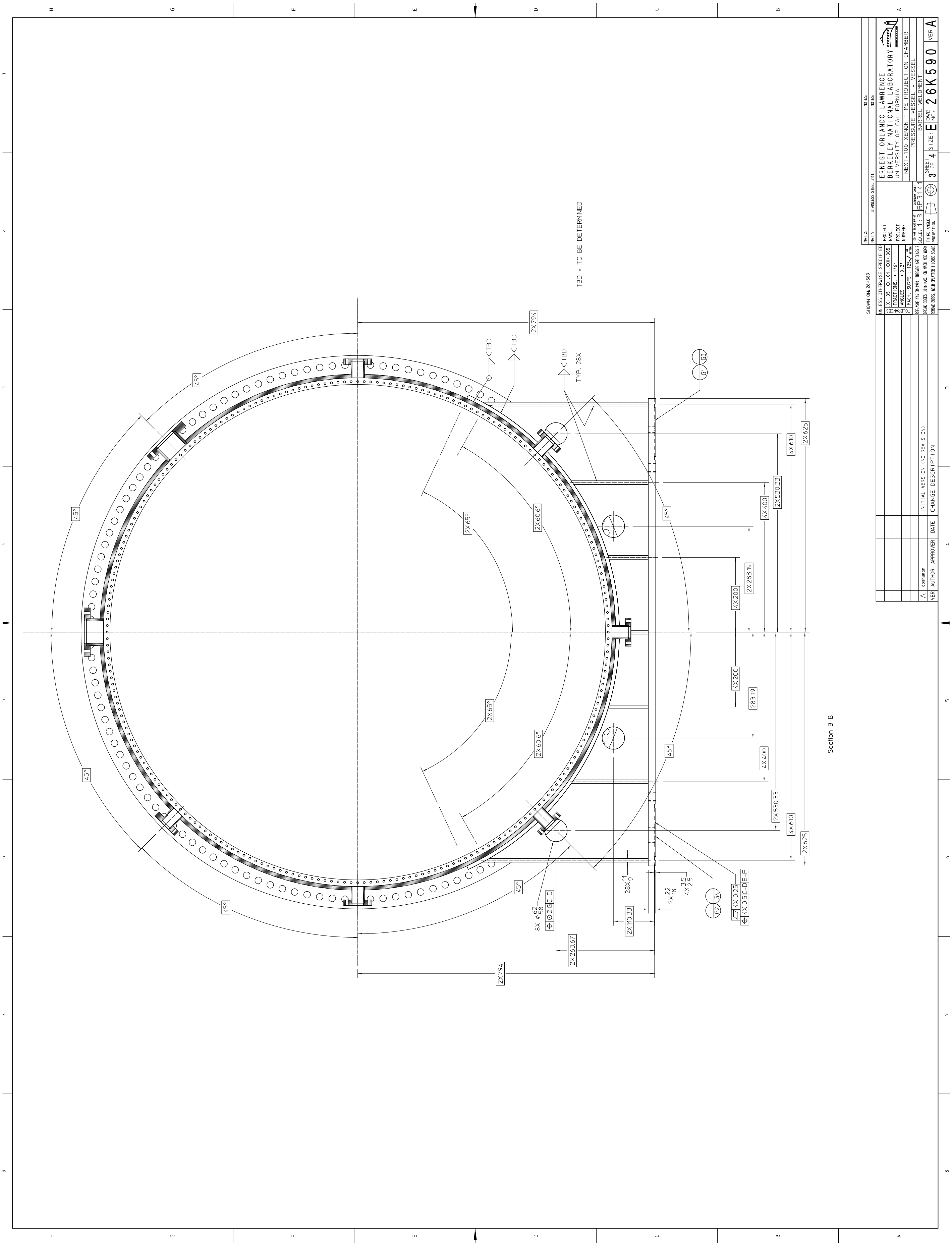
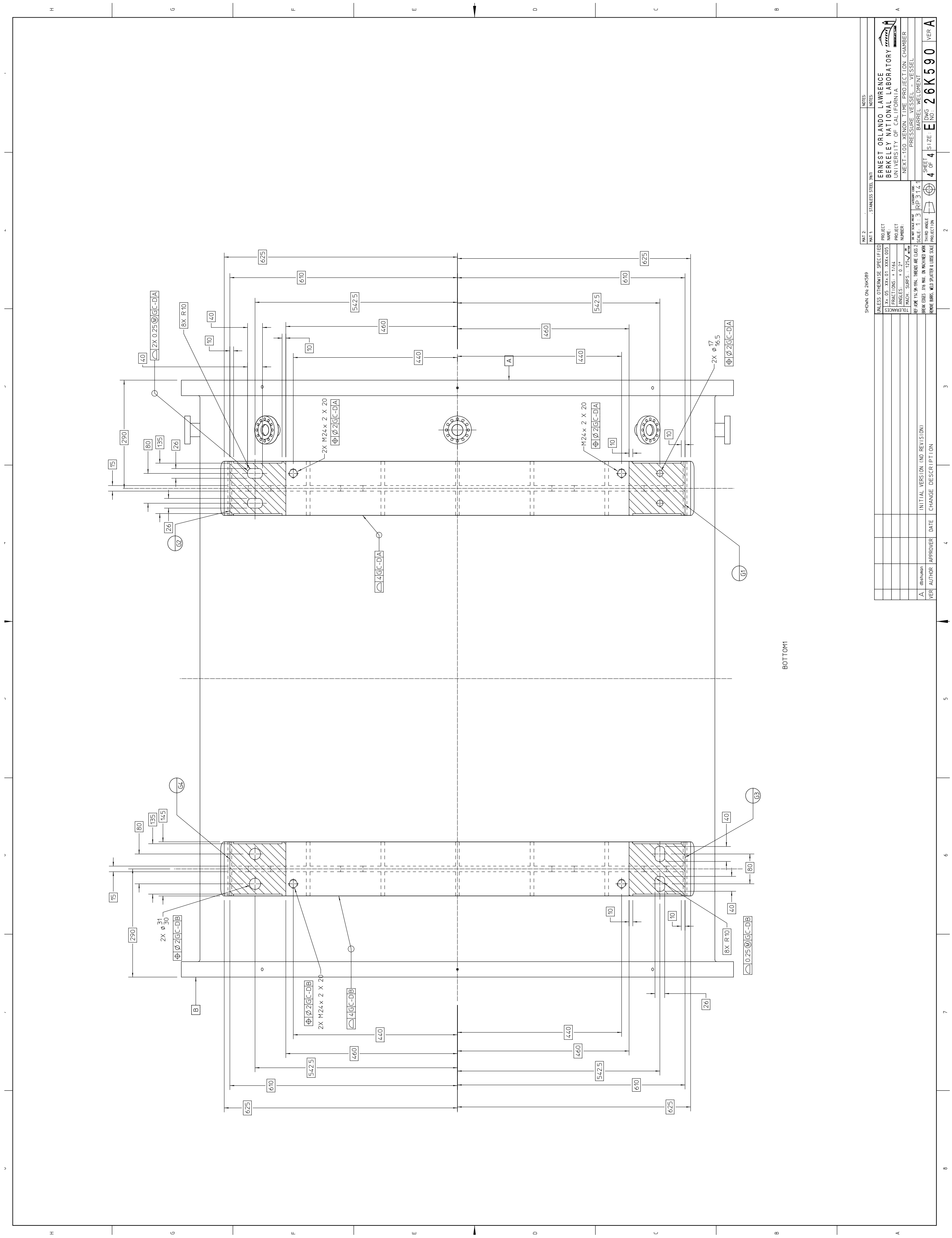
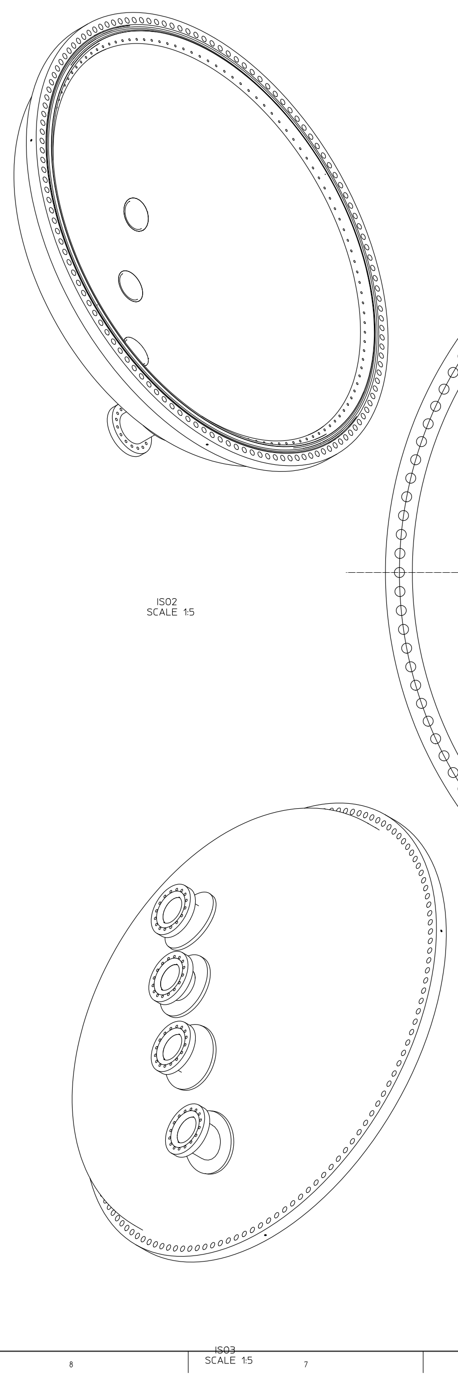
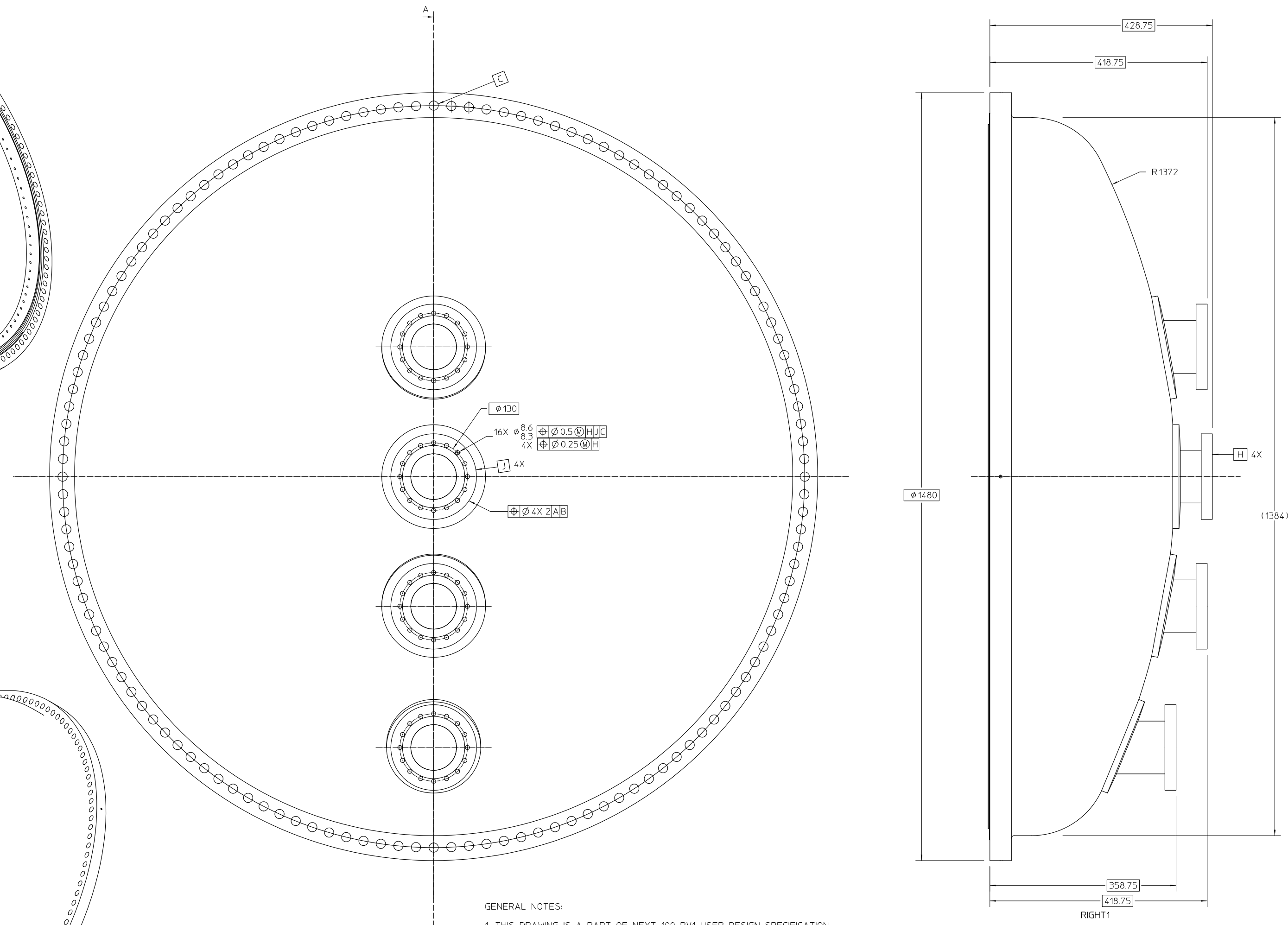
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
BARREL WELDMENT
26K590
NOTES:
1. DIMENSIONS IN MM
2. DIMENSIONS IN PARENTHESES ARE CALCULATED BY THE COLLABORATION AND SUBJECT TO CHANGE
Section A-A
SCALE 1:4
LEFT AND RIGHT FLANGES ARE MIRROR PARTS,
ALL DIMS, TOLERANCES AND FINISHES APPLY
TO BOTH PARTS


TBD = TO BE DETERMINED
Section B-B
TYP. 28X
SHOWN ON: 26K589
INITIAL VERSION (NO REVISION)
CHANGE DESCRIPTION
ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
BARREL WELDMENT
26K590


ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
BARREL WELDMENT
26K590
INITIAL VERSION (NO REVISION)
BOTTOM1





GENERAL NOTES:

1. THIS DRAWING IS A PART OF NEXT-100-PV1 USER DESIGN SPECIFICATION
2. THIS DRAWING IS PRELIMINARY, AND SETS FORTH BOTH REQUIRED DIMENSIONS, AND PRELIMINARY DIMENSIONS TO BE VERIFIED BY MANUFACTURER.
3. ALL PARTS TO BE FABRICATED, INSPECTED AND TESTED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII, DIV. 1 OR EU EQUIVALENT.
4. ADDITIONAL REQUIREMENTS APPLY CONCERNING CLEANING, PREPARATION, INSPECTION, WELDING, HEAT TREATING.
5. VESSEL IS FOR BOTH MEDIUM PRESSURE (15.4 BAR) AND HIGH VACUUM (10**-6 TORR) SERVICE.
6. PART SHALL BE DELIVERED IN CLEAN CONDITION, SERVICEABLE FOR HIGH VACUUM SERVICE

MATERIAL: 316Ti (EN 1.4571, UNSS31635) PER ASME SPEC 240 OR EQUIV

SHOWN ON: 26K589

VER	AUTHOR	APPROVER	DATE	CHANGE DESCRIPTION
A	dbshuman			INITIAL VERSION (NO REVISION)

UNLESS OTHERWISE SPECIFIED
TOLERANCES: X± .05 XX± .01 XXX± .005; FRACTIONS: ± 1/64; ANGLES: ± 0.2°; MACH. SURFS.: 125√
REF ASME Y14.5M-1994. THREADS ARE CLASS 2
BREAK EDGES .016 MAX. ON MACHINED WORK
REMOVE BURRS, WELD SPLATTER & LOOSE SCALE

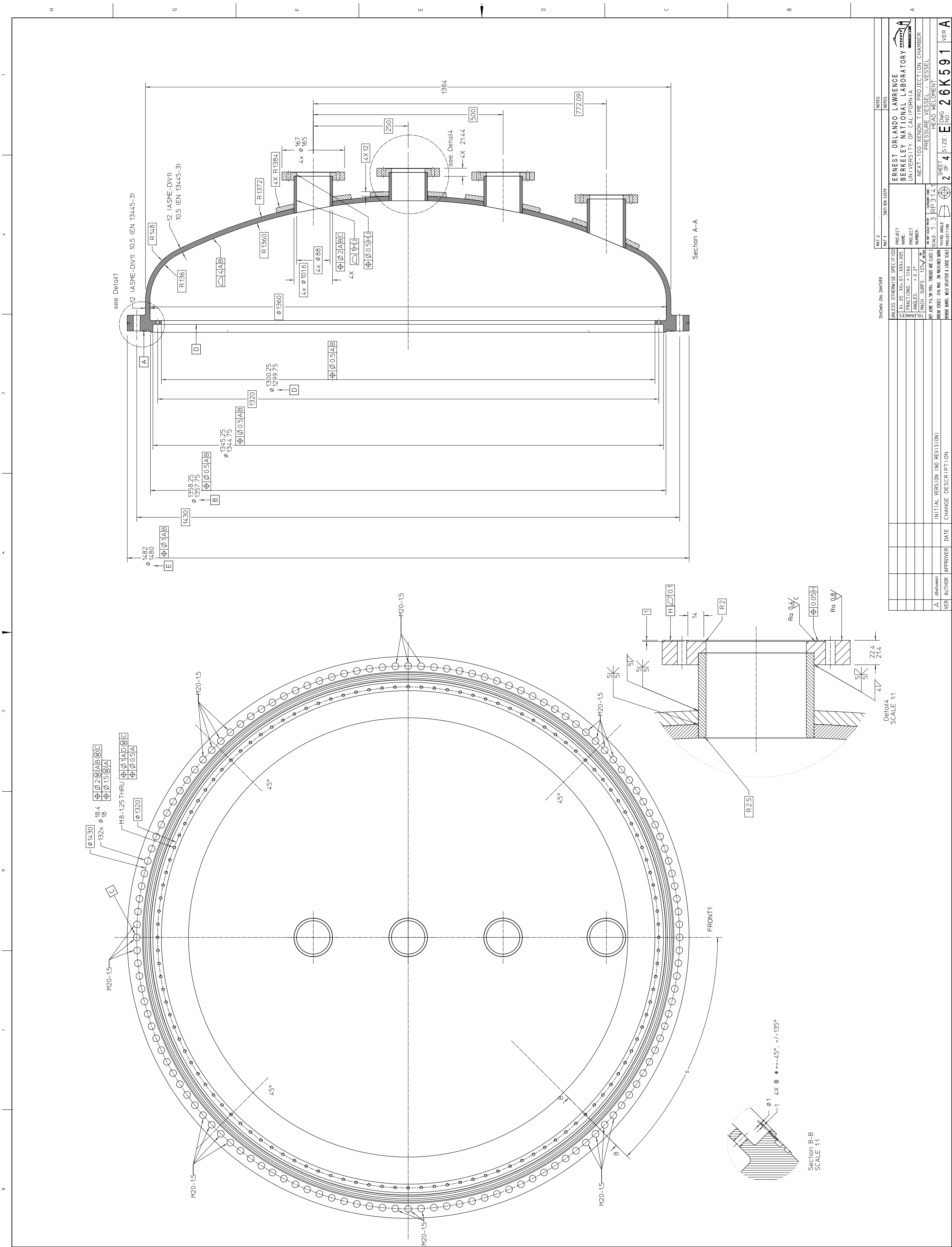
PROJECT NAME:
PROJECT NUMBER:

DO NOT SCALE PRINT — SCALE: 1 : 3
CATEGORY CODE: RP3141
THIRD ANGLE PROJECTION

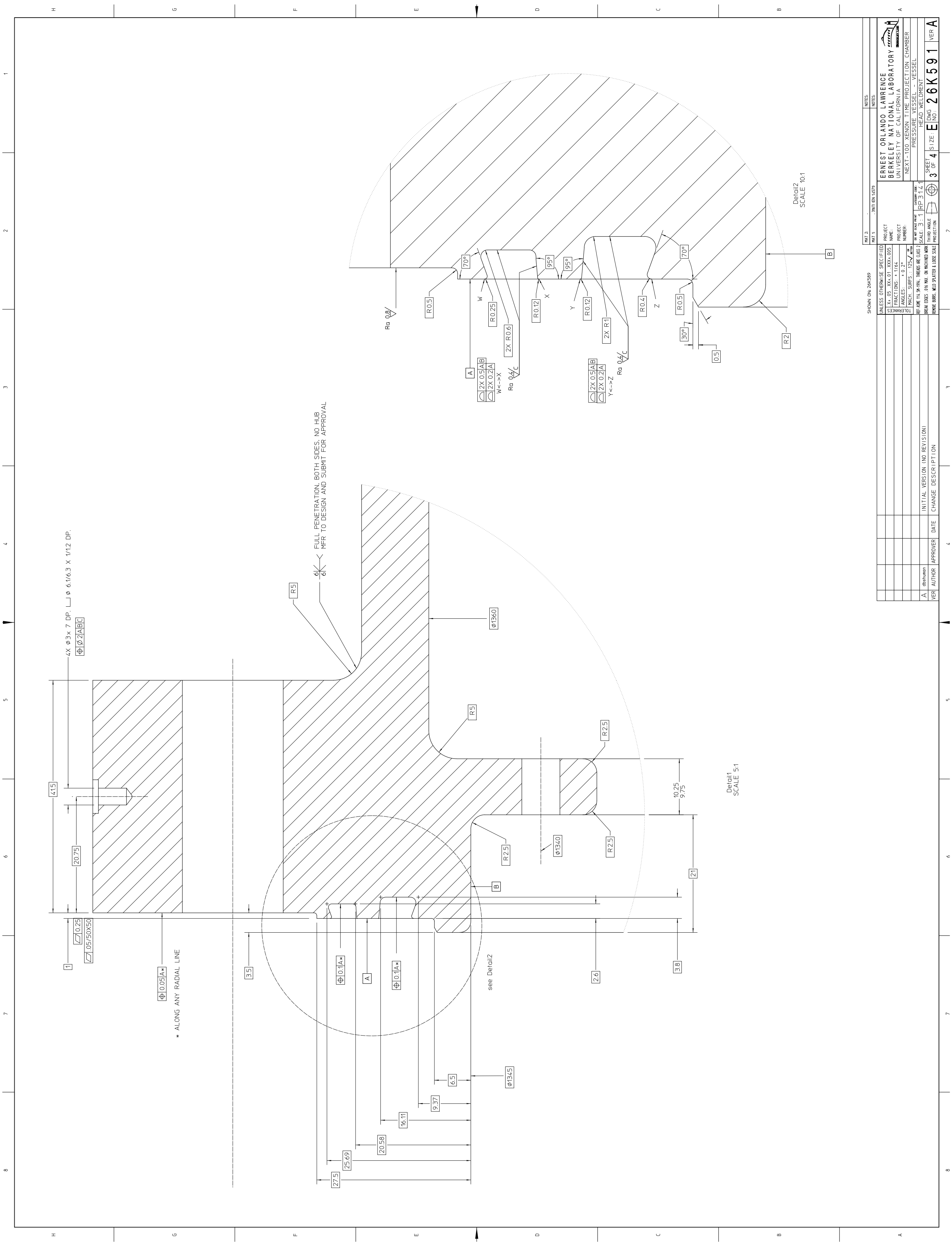
ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA

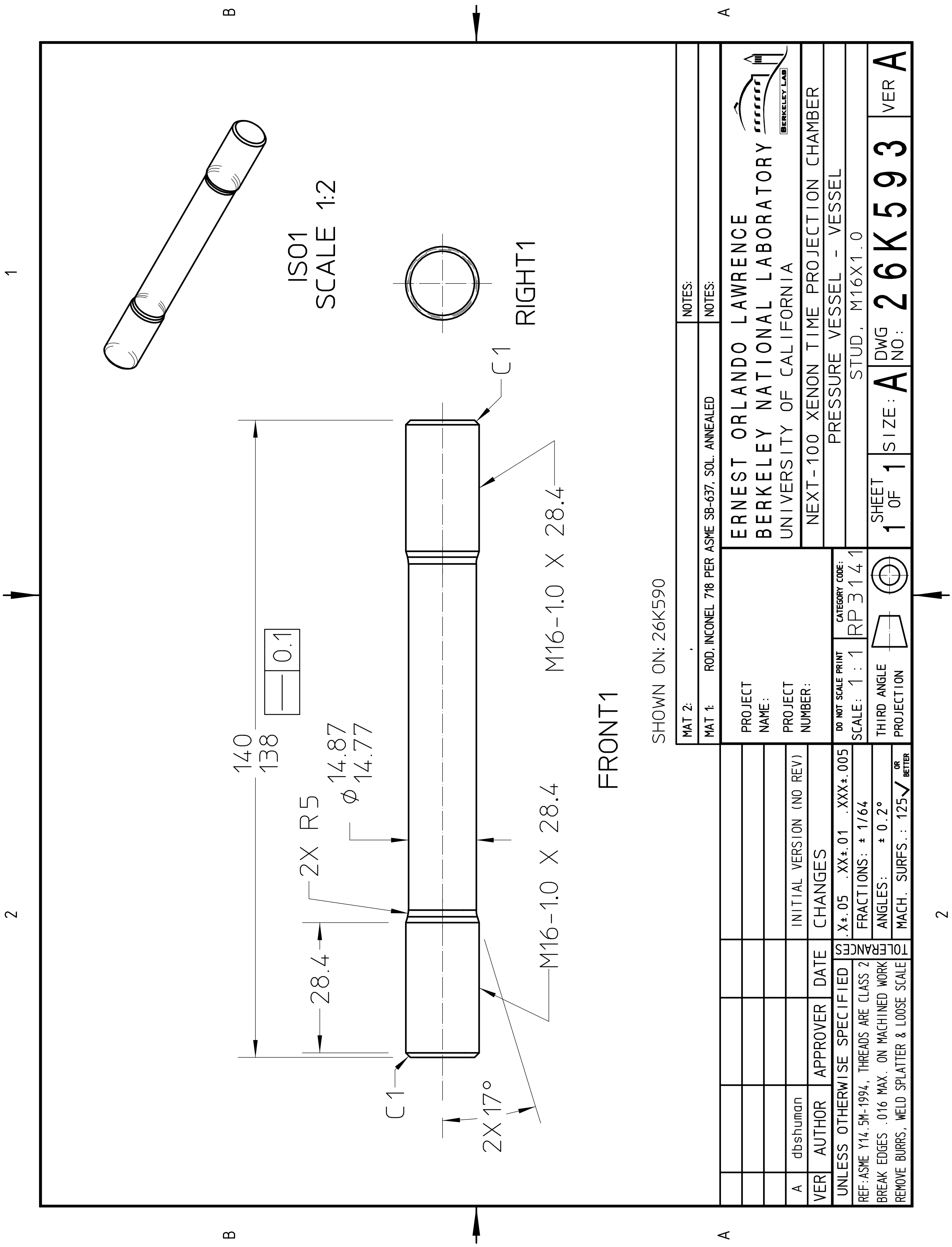
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
HEAD WELDMENT

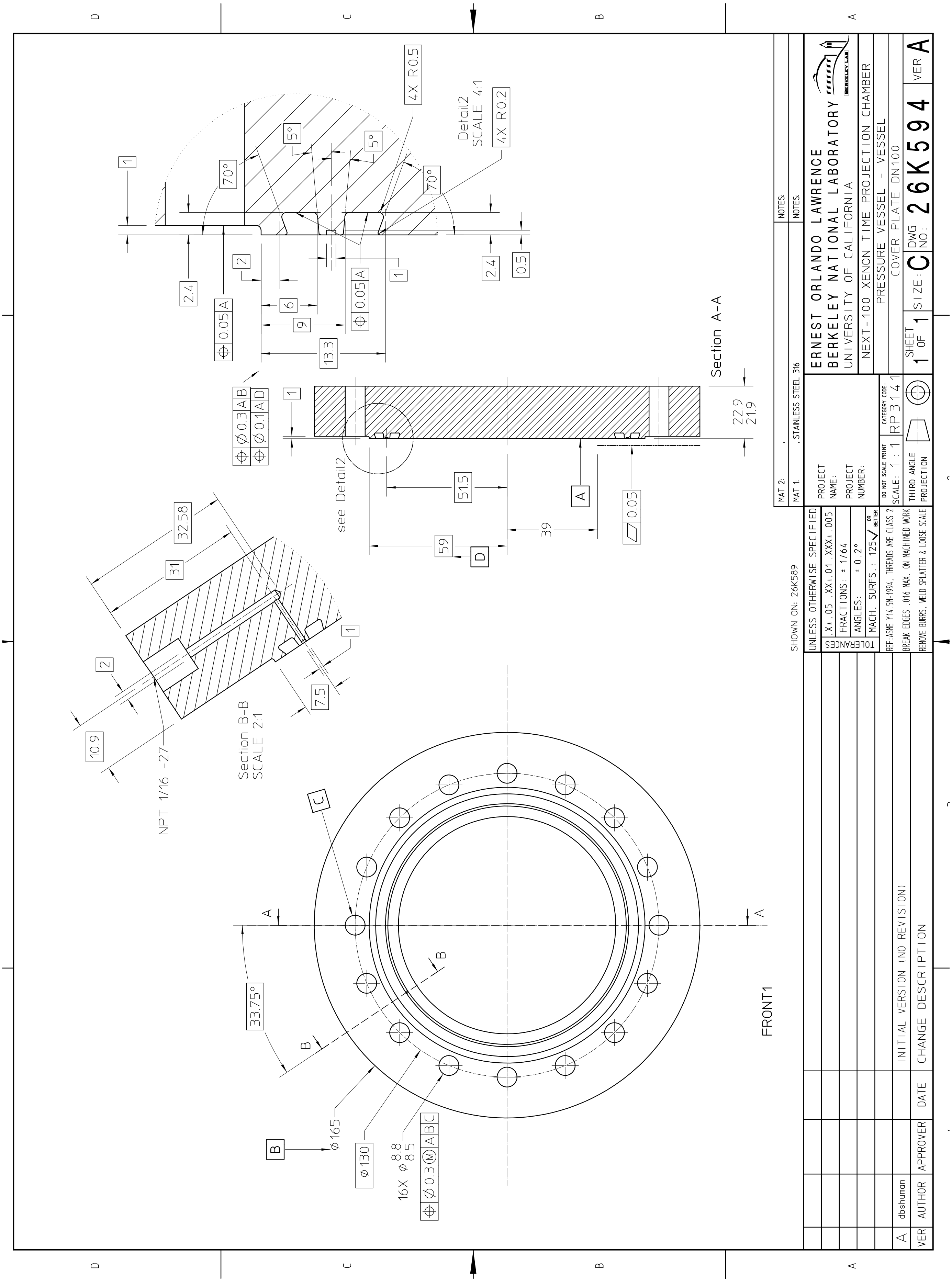
SHEET 1 OF 4 | SIZE E | DWG NO: 26K591 | VER A

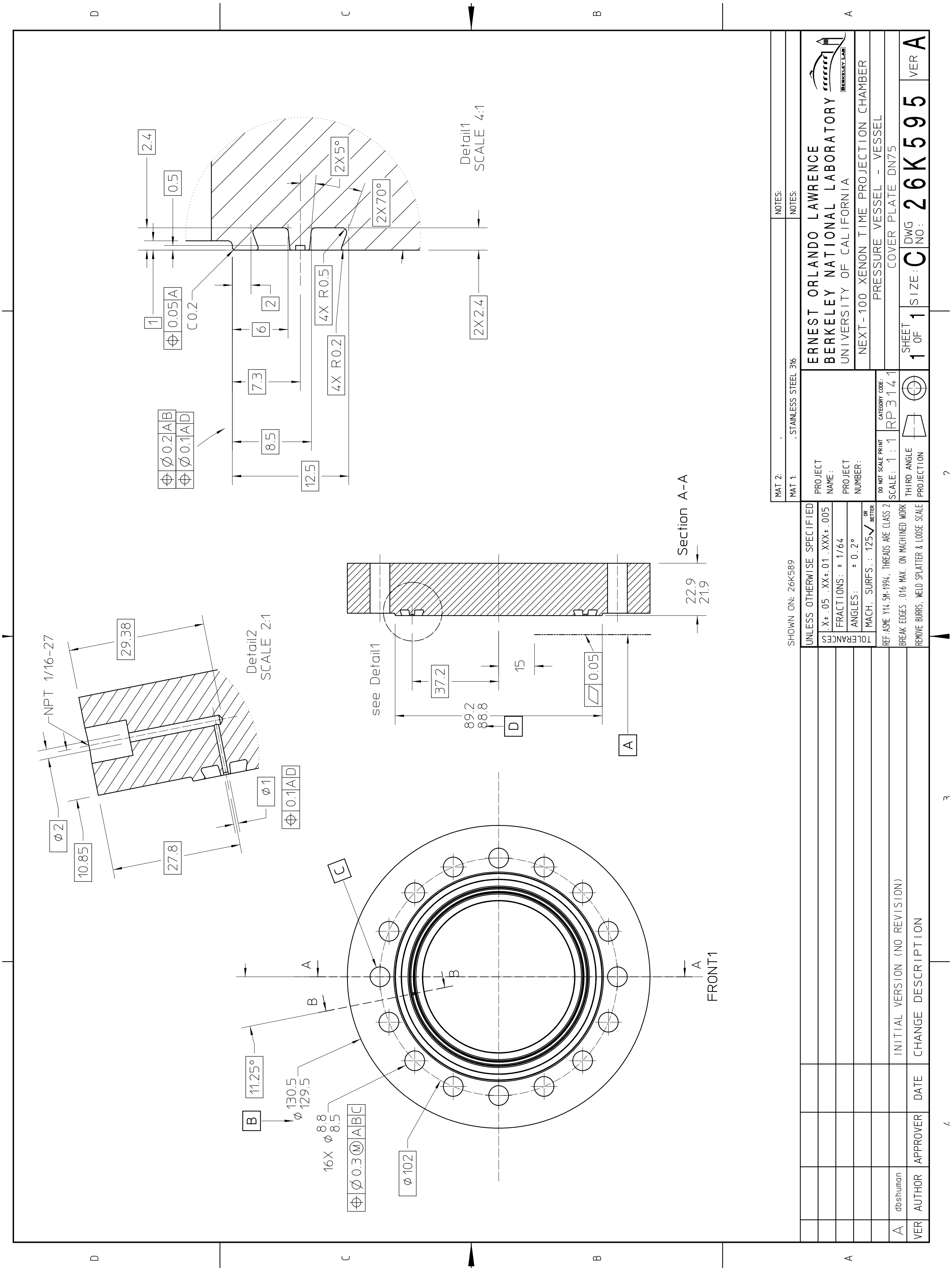


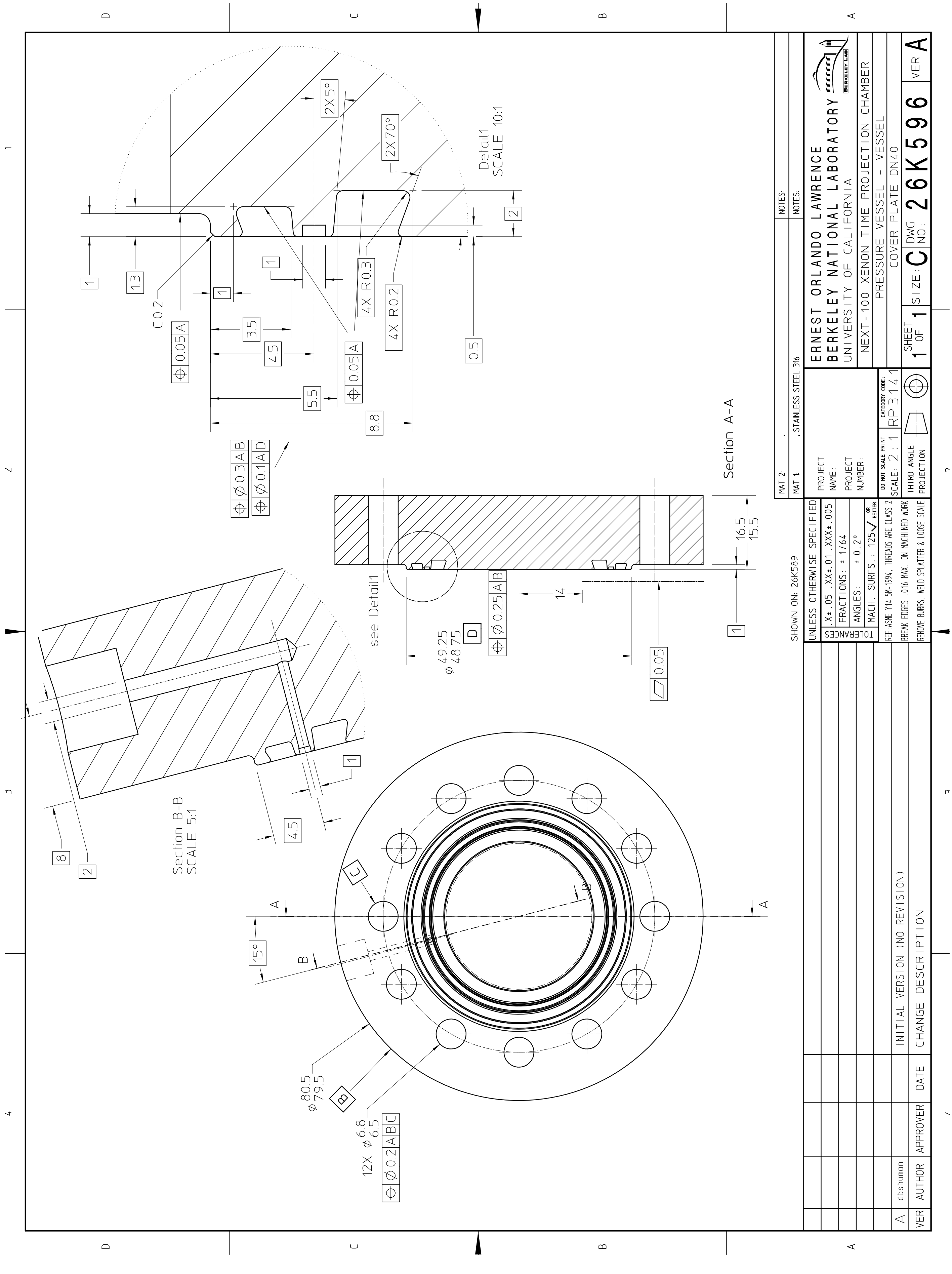
ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
HEAD WELDMENT
26K591
Section A-A
Detail4
SCALE 1:1
Section B-B
SCALE 1:1
INITIAL VERSION (NO REVISION)


ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
HEAD WELDMENT
26K591
Detail1
SCALE 5:1
Detail2
SCALE 10:1
FULL PENETRATION, BOTH SIDES, NO HUB
MFR TO DESIGN AND SUBMIT FOR APPROVAL
* ALONG ANY RADIAL LINE
see Detail2
INITIAL VERSION (NO REVISION)
CHANGE DESCRIPTION


ISO1
SCALE 1:2
RIGHT1
FRONT1
SHOWN ON: 26K590
C1
140
138
0.1
2X R5
⌀ 14.87
14.77
28.4
M16-1.0 X 28.4
M16-1.0 X 28.4
2X 17°
NOTES:
NOTES:
MAT 2:
MAT 1: ROD, INCONEL 718 PER ASME SB-637, SOL. ANNEALED
ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
BERKELEY LAB
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
STUD, M16X1.0
PROJECT NAME:
PROJECT NUMBER:
A dbshuman INITIAL VERSION (NO REV)
VER AUTHOR APPROVER DATE CHANGES
UNLESS OTHERWISE SPECIFIED
REF: ASME Y14.5M-1994, THREADS ARE CLASS 2
BREAK EDGES .016 MAX. ON MACHINED WORK
REMOVE BURRS, WELD SPLATTER & LOOSE SCALE
TOLERANCES
X± .05 .XX± .01 .XXX± .005
FRACTIONS: ± 1/64
ANGLES: ± 0.2°
MACH. SURFS.: 125 OR BETTER
DO NOT SCALE PRINT
SCALE: 1 : 1
CATEGORY CODE:
RP 3141
THIRD ANGLE PROJECTION
SHEET 1 OF 1
SIZE: A
DWG NO: 26K593
VER A


ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
COVER PLATE DN100
SIZE: C
DWG NO: 26K594
VER A
SHEET 1 OF 1
SHOWN ON: 26K589
MAT 1: STAINLESS STEEL 316
MAT 2:
NOTES:
PROJECT NAME:
PROJECT NUMBER:
UNLESS OTHERWISE SPECIFIED
.X± .05 .XX± .01 .XXX± .005
FRACTIONS: ± 1/64
ANGLES: ± 0.2°
MACH. SURFS.: 125 OR BETTER
TOLERANCES
REF: ASME Y14.5M-1994, THREADS ARE CLASS 2
BREAK EDGES .016 MAX. ON MACHINED WORK
REMOVE BURRS, WELD SPLATTER & LOOSE SCALE
DO NOT SCALE PRINT
SCALE: 1 : 1
CATEGORY CODE: RP 314 1
THIRD ANGLE PROJECTION
VER AUTHOR APPROVER DATE CHANGE DESCRIPTION
A dbshuman INITIAL VERSION (NO REVISION)
FRONT1
Section A-A
Section B-B
SCALE 2:1
Detail2
SCALE 4:1
see Detail2
NPT 1/16 -27
4X R0.5
4X R0.2
16X ⌀8.8 8.5
⌀0.3 Ⓜ A B C
⌀165
⌀130
33.75°
70°
5°
2.4
0.5
6
9
13.3
0.05 A
51.5
59
39
22.9
21.9
0.05
⌀0.3 A B
⌀0.1 A D
32.58
31
10.9
7.5
A
B
C
D


ERNEST ORLANDO LAWRENCE
BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
COVER PLATE DN75
SIZE: C
DWG NO: 26K595
VER A
SHEET 1 OF 1
SCALE: 1:1
THIRD ANGLE PROJECTION
CATEGORY CODE: RP3141
MAT 1: , STAINLESS STEEL 316
SHOWN ON: 26K589
UNLESS OTHERWISE SPECIFIED
.X± .05 .XX± .01 .XXX± .005
FRACTIONS: ± 1/64
ANGLES: ± 0.2°
MACH. SURFS.: 125
TOLERANCES
REF: ASME Y14.5M-1994, THREADS ARE CLASS 2
BREAK EDGES .016 MAX. ON MACHINED WORK
REMOVE BURRS, WELD SPLATTER & LOOSE SCALE
INITIAL VERSION (NO REVISION)
VER AUTHOR APPROVER DATE CHANGE DESCRIPTION
A dbshuman
Section A-A
Detail1 SCALE 4:1
Detail2 SCALE 2:1
see Detail1
NPT 1/16-27
FRONT1



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BERKELEY NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA
NEXT-100 XENON TIME PROJECTION CHAMBER
PRESSURE VESSEL - VESSEL
COVER PLATE DN40
SIZE: C
DWG NO: 26K596
VER A
SHEET 1 OF 1
SCALE: 2 : 1
CATEGORY CODE: RP3141
THIRD ANGLE PROJECTION
DO NOT SCALE PRINT
MAT 1: STAINLESS STEEL 316
MAT 2:
NOTES:
PROJECT NAME:
PROJECT NUMBER:
SHOWN ON: 26K589
UNLESS OTHERWISE SPECIFIED
TOLERANCES
.X± .05 .XX± .01 .XXX± .005
FRACTIONS: ± 1/64
ANGLES: ± 0.2°
MACH. SURFS.: 125 OR BETTER
REF: ASME Y14.5M-1994, THREADS ARE CLASS 2
BREAK EDGES .016 MAX. ON MACHINED WORK
REMOVE BURRS, WELD SPLATTER & LOOSE SCALE
VER AUTHOR APPROVER DATE CHANGE DESCRIPTION
A dbshuman INITIAL VERSION (NO REVISION)
Section A-A
Section B-B
SCALE 5:1
Detail1
SCALE 10:1
see Detail1
2X5°
2X70°
4X R0.3
4X R0.2
C0.2
⌀80.5
⌀79.5
12X ⌀6.8
⌀6.5
⌀49.25
⌀48.75
15°

Pressure Vessel Design Calculations

All calculations following by D. Shuman, except where noted.
Mar 27, 2012

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Active volume dimensions, from earlier analyses:

$r_{Xe} = 0.53\,m \qquad l_{Xe} = 1.3\,m$

We consider using a field cage solid insulator/light tube of 3 cm total thk., and a copper liner of 12 cm thickness, including all tolerances and necessary gaps.

$t_{fc} := 3cm \qquad t_{Cu} := 12cm$

Pressure Vessel Inner Radius, Diameter:

$R_{i\_pv} := r_{Xe} + t_{fc} + t_{Cu} \qquad R_{i\_pv} = 0.68\,m \qquad D_{i\_pv} := 2R_{i\_pv} \qquad D_{i\_pv} = 1.36\,m$

Pressure vessel length:

main cyl. vessel overall, inside

$L_v := 1.6m \qquad L_o := 2.2m$

Temperatures:

For pressure operation, the temperature range will be 10C-30C. For vacuum operation, the temperature range will be 10C to 150C (bakeout).

Maximum Operating Pressure (MOP), gauge:

$MOP_{pv} := \left(P_{MOPa} - 1bar\right) \qquad MOP_{pv} = 14\,bar$

Minimum Pressure, gauge:

$P_{min} = -1.5\,bar$ the extra 0.5 atm maintains an upgrade path to a water or scintillator tank

Maximum allowable pressure, gauge (from LBNL Pressure Safety Manual, PUB3000)
From LBNL PUB3000, recommended minimum is, 10% over max operating pressure; this is design pressure at LBNL. This is for spring operated relief valves, to avoid leakage. Use of pilot operated relief valves can reduce this to as little as 2%, as they seal tighter when approaching relief pressure:

$MAWP_{pv} := 1.1MOP_{pv} \qquad MAWP_{pv} = 15.4\,bar$

$P := MAWP_{pv}$

Mass supported internally by pressure vessel

Internal copper shield (ICS)

$M_{ICS\_cyl} := 6000kg \qquad M_{ICS\_eh} := 1500kg \qquad M_{ICS\_tp} := 2500kg$

Detector subsystems, est.

$M_{ep} := 750kg \qquad M_{tp} := 200kg \qquad M_{fc} := 350kg$

Length inside vessel of copper, total

$L_{Cu} := 2.0m$

Mass total of internal copper shielding:

$M_{ICS} := M_{ICS\_cyl} + M_{ICS\_eh} + M_{ICS\_tp} \qquad M_{ICS} = 10000\,kg$

Maximum mass supported on internal flange of each head:

$M_{fl\_h} := M_{ICS\_tp}$

Maximum mass supported on each internal flange of the main cylindrical vessel:

$M_{fl\_v} := 0.5\left(M_{ICS\_cyl} + M_{fc}\right) + M_{ep} \qquad M_{fl\_v} = 3925\,kg$ this mass will be present when heads are not mounted

Estimated approximate total vessel mass carried on supports (numbers from calcs below):

vessel heads flanges

$$M_v := \rho_{SS} \cdot \left(2\pi R_{i\_pv} \cdot 10\text{mm} \cdot L_o + \pi R_{i\_pv}^{\;2} \cdot 12\text{mm} + 4 \cdot 2\pi R_{i\_pv} \cdot 4.2\text{cm} \cdot 5\text{cm}\right) \quad M_v = 1179\,\text{kg} \qquad \rho_{SS} := 8\,\frac{\text{gm}}{\text{cm}^3}$$

Total detector mass:

$$M_{det} := M_{ICS} + M_{ep} + M_{fc} + M_{tp} + M_v \qquad M_{det} = 1.248 \times 10^4\,\text{kg}$$

Vessel wall thicknesses

Material:

We use 316Ti for vessel shells and flanges due to its known good radiopurity and strength.

Design Rules:

ASME Boiler and Pressure Vessel code section VIII, Rules for construction of Pressure vessels division 1 (2010)

316Ti is not an allowed material under section VIII, division 2, so we must use **division 1** rules. The saddle supports are however, deisgned using the methodology given in div. 2, as div. 1 does not provide design formulas (nonmandatory Appendix G)

Maximum allowable material stress, for sec. VIII, division 1 rules from ASME 2009 Pressure Vessel code, sec. II part D, table 1A:

$S_{max\_316Ti\_div1} := 20000psi$ -20F - 100F

Youngs modulus

$E_{SS\_aus} := 193GPa$

color scheme for this document

input check result (all conditions should be true (=1)

$xx := 1$ $xx > 0 = 1$

Choose material, then maximum allowable strength is:

$S := S_{max\_316Ti\_div1}$

Vessel wall thickness, for internal pressure is then (div. 1), Assume all welds are type (1) as defined in UW-12 , are double welds, fully radiographed, so weld efficiency:

$E := 1$

Minimum wall thickness is then:

$$t_{pv\_d1\_min\_ip} := \frac{P \cdot R_{i\_pv}}{S \cdot E - 0.6 \cdot P}$$

$$t_{pv\_d1\_min\_ip} = 7.75\,mm$$

We set wall thickness to be:

$t_{pv} := 10mm$ $t_{pv} > t_{pv\_d1\_min\_ip} = 1$

Maximum Allowable External Pressure

ASME PV code Sec. VIII, Div. 1- UG-28 Thickness of Shells under External Pressure

Maximum length between flanges $L_{ff} := 1.6m$

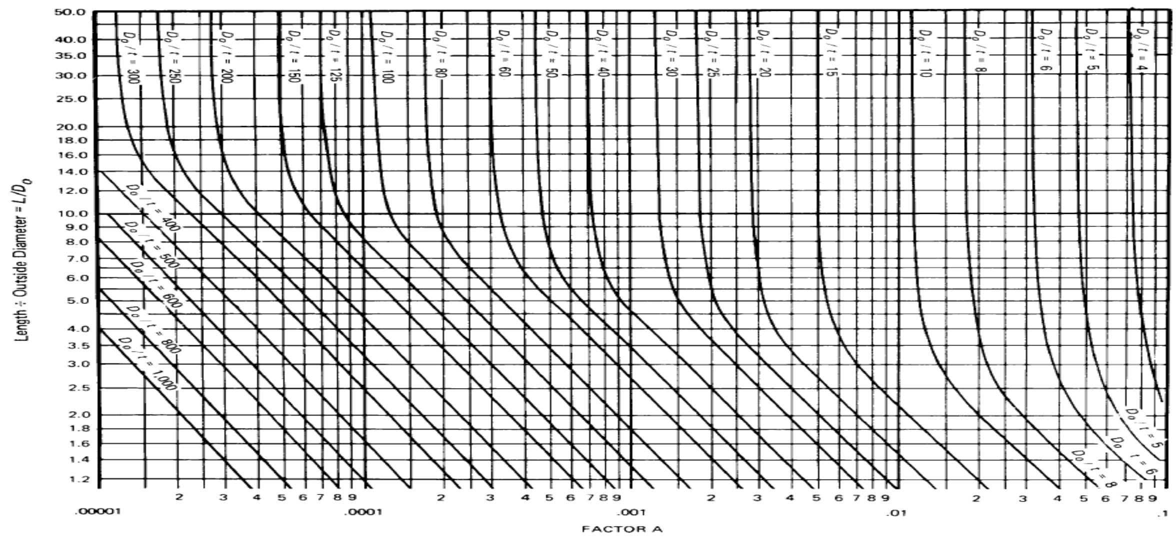
The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{L_{ff}}{2R_{i\_pv}} = 1.2 \qquad \frac{2R_{i\_pv}}{t_{pv}} = 136$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor A:

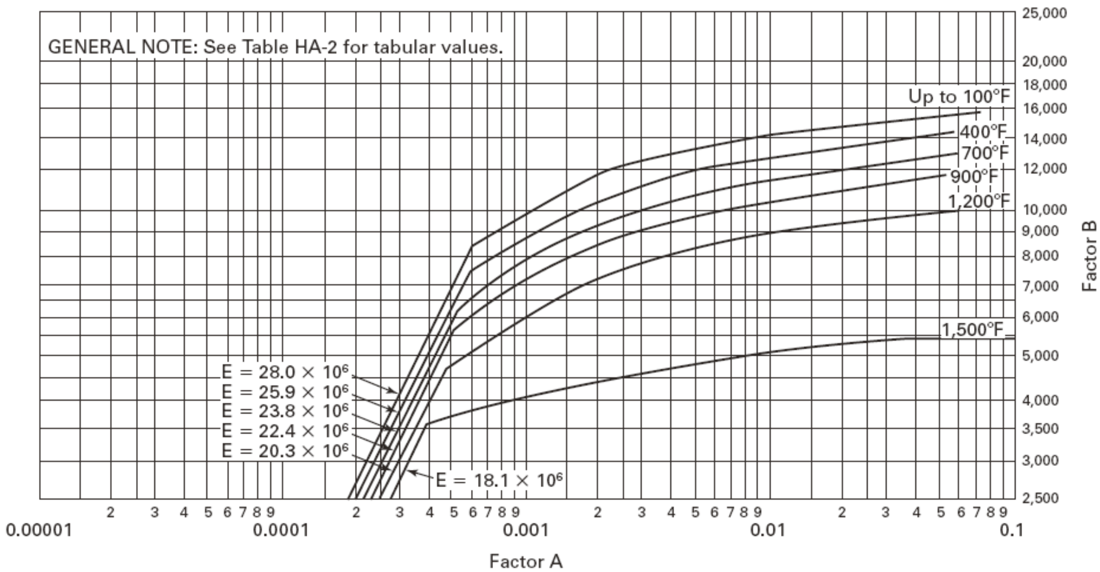
FIG. G GEOMETRIC CHART FOR COMPONENTS UNDER EXTERNAL OR COMPRESSIVE LOADINGS (FOR ALL MATERIALS) [NOTE (14)]



$A_{.} := 0.0005$

Using the factor A in chart (HA-2) in Subpart 3 of Section II, Part D, we find the factor B (@ 400F, since we may bake while pulling vacuum):

FIG. HA-2 CHART FOR DETERMINING SHELL THICKNESS OF COMPONENTS UNDER EXTERNAL PRESSURE DEVELOPED FOR AUSTENITIC STEEL 16Cr–12Ni–2Mo, TYPE 316



$B_{.} := 6200\text{psi}$ @ 400F

The maximum allowable working external pressure is then given by :

$$P_a := \frac{4B_{.}}{3\left(\frac{2R_{i\_pv}}{t_{pv}}\right)} \qquad P_a = 4.1\,\text{bar} \qquad -P_{min} = 1.5\,\text{bar}$$

$P_a > -P_{min} = 1$

Flange thickness, head to vessel main flanges:

inner radius max. allowable pressure

$R_{i\_pv} = 0.68\,m$ $P = 15.4\,bar$ (gauge pressure)

The flange design for O-ring sealing (or other self energizing gasket such as helicoflex) is "flat-faced", with "metal to metal contact outside the bolt circle". This design avoids the high flange bending stresses found in a raised face flange (of Appendix 2) and will result in less flange thickness. The rules for this design are found only in sec VIII division 1 under Appendix Y, and must be used with the allowable stresses of division 1. Flanges and shells will be fabricated from 316Ti (ASME spec SA-240) stainless steel plate. Plate samples will be helium leak checked before fabrication, as well as ultrasound inspected for flat laminar flaws which may create leak paths. The flange bolts and nuts will be inconel 718, (UNS N77180) as this is the highest strength non-corrosive material allowed for bolting.

We will design with enough flange strength to accomodate using a Helicoflex 5mm gasket (smallest size possible) specially designed with a maximum sealing force of 70 N/mm.

Maximum allowable material stresses, for sec VIII, division 1 rules from ASME 2010 Pressure Vessel code, sec. II part D, table 2A (division 1 only):

Maximum allowable design stress for flange

$S_f := S_{max\_316Ti\_div1}$ $S_f = 137.9\,MPa$ $S_f = 2 \times 10^4\,psi$

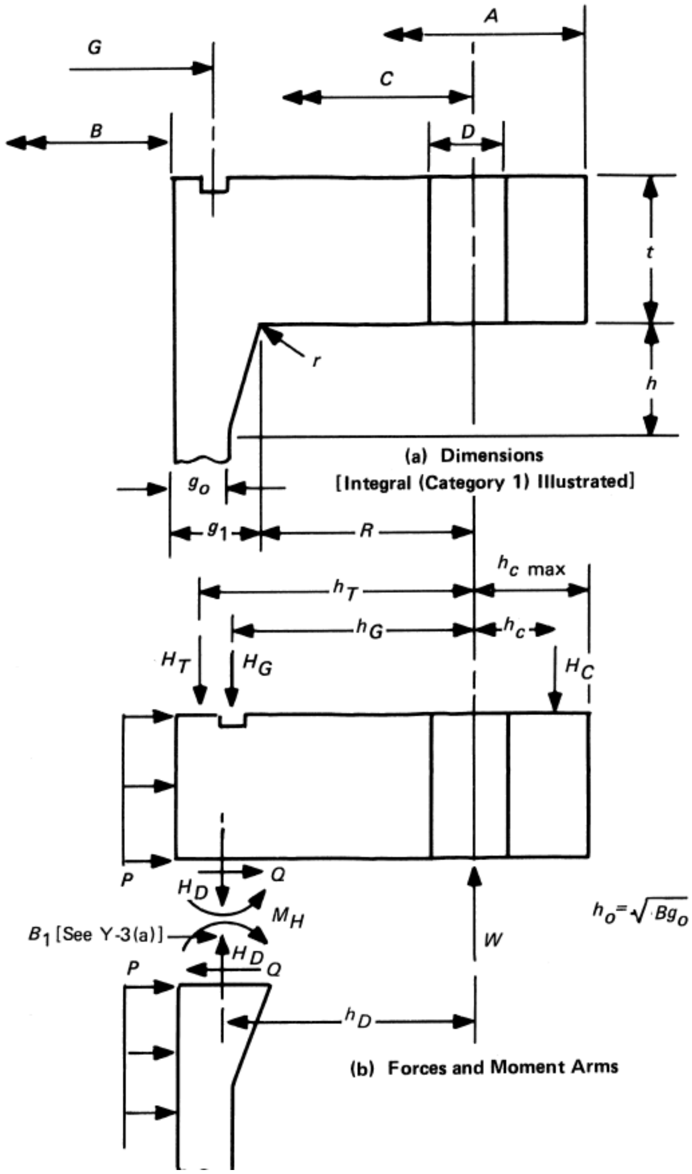
Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

Inconel 718 (UNS N07718) $S_{max\_N07718} := 37000psi$

$S_b := S_{max\_N07718}$ $S_b = 255.1\,MPa$

From sec. VIII div 1, non-mandatory appendix Y for bolted joints having metal-to-metal contact outside of bolt circle. First define, per Y-3:

FIG. Y-3.2 FLANGE DIMENSIONS AND FORCES



hub thickness at flange (no hub)

$g_0 := t_{pv}$ $\quad g_1 := t_{pv}$ $\quad g_0 = 10\,\text{mm}$ $\quad g_1 = 10\,\text{mm}$

corner radius:

$r_1 := \max(.25g_1, 5\text{mm})$ $\quad r_1 = 5\,\text{mm}$

Flange OD

$A := 1.48\text{m}$

Flange ID

$B := 2R_{i\_pv}$ $\quad B = 1.36\,\text{m}$

define:

$B_1 := B + g_1$ $\quad B_1 = 1.37\,\text{m}$

Bolt circle (B.C.) dia, C:

$C := 1.43\cdot\text{m}$

Gasket dia

$G := 2\left(R_{i\_pv} + .65\text{cm}\right)$ $\quad G = 1.373\,\text{m}$ O-ring mean radius as measured in CAD model: $68.65\cdot 2 = 137.3$

Note: this diameter will be correct for Helicoflex gasket, but slightly higher for O-ring, which is fluid and "transmits pressure" out to its OD, howqever the lower gasket unit force of O-ring more than compensates, as per below:

Force of Pressure on head

$H := .785G^2 \cdot MAWP_{pv}$ $\quad H = 2.31 \times 10^6 \, N$

Sealing force, per unit length of circumference:

for O-ring, 0.275" dia., shore A 70 F= ~5 lbs/in for 20% compression, (Parker O-ring handbook); add 50% for smaller second O-ring. (Helicoflex gasket requires high compression, may damage soft Ti surfaces, may move under pressure unless tightly backed, not recommended)

Helicoflex has equivalent formulas using Y as the unit force term and gives several possible values.

for 5mm HN200 with aluminum jacket:

$Y_1 := 70 \frac{N}{mm}$ min value for our pressure and required leak rate (He) $\quad Y_2 := 220 \frac{N}{mm}$ recommended value for large diameter seals, regardless of pressure or leak rate

for gasket diameter $\quad D_j := G \quad D_j = 1.373 \, m$

Force is then either of:

$F_m := \pi D_j \cdot Y_1$ $\quad$ or $\quad F_j := \pi \cdot D_j \cdot Y_2$

$F_m = 3.019 \times 10^5 \, N$ $\quad F_j = 9.489 \times 10^5 \, N$

Helicoflex recommends using Y2 (220 N/mm) for large diameter seals, even though for small diameter one can use the greater of Y1 or Ym=(Y2*(P/Pu)). For 15 bar Y1 is greater than Ym but far smaller than Y2. Sealing is less assured, but will be used in elastic range and so may be reusable. Flange thickness and bolt load increase quite substantially when using Y2 as design basis, which is a large penalty. We plan to recover any Xe leakage, as we have a second O-ring outside the first and a sniff port in between, so we thus design for Y1 (use F_m) and "cross our fingers" : if it doesn't seal we use an O-ring instead and recover permeated Xe with a cold trap. Note: in the cold trap one will get water and N2, O2, that permeates through the outer O-ring as well.

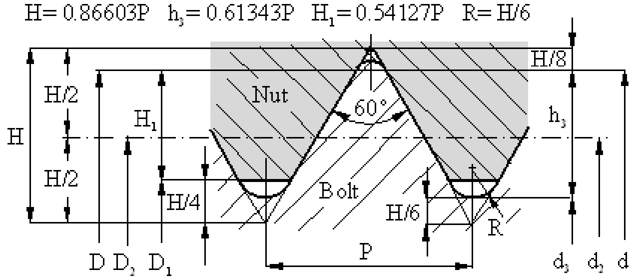
Start by making trial assumption for number of bolts, nominal bolt dia., pitch, and bolt hole dia D,

$n := 132$ $\quad d_b := 16mm$ $\quad$ maximum number of bolts possible, using narrow washers: $\quad n_{max} := \mathrm{trunc}\left(\frac{\pi C}{2.0 d_b}\right)$ $\quad n_{max} = 140$

Choosing ISO fine thread, to maximize root dia.; thread depth is:

$p_t := 1.0mm$ $\quad h_3 := .6134 \cdot p_t$

using nomenclature and formulas from this chart at http://www.tribology-abc.com/calculators/metric-iso.htm



metric screw threads ISO 724 (DIN 13 T1)								
Nominal diameter	Pitch	root radius	pitch diameter	minor diameter		thread height		drill diameter
d = D	P	r	d2=D2	d3	D1	h3	H1	mm
M 1.00	0.25	0.036	0.838	0.693	0.729	0.153	0.135	0.75
M 1.10	0.25	0.036	0.938	0.793	0.829	0.153	0.135	0.85
M 1.20	0.25	0.036	1.038	0.893	0.929	0.153	0.135	0.95
M 1.40	0.30	0.043	1.205	1.032	1.075	0.184	0.162	1.10
M 1.60	0.35	0.051	1.373	1.171	1.221	0.215	0.189	1.25
M 1.80	0.35	0.051	1.573	1.371	1.421	0.215	0.189	1.45
M 2.00	0.40	0.058	1.740	1.509	1.567	0.245	0.217	1.60
M 2.20	0.45	0.065	1.908	1.648	1.713	0.276	0.244	1.75
M 2.50	0.45	0.065	2.208	1.948	2.013	0.276	0.244	2.05
M 3.00	0.50	0.072	2.675	2.387	2.459	0.307	0.271	2.50
M 3.50	0.60	0.087	3.110	2.764	2.850	0.368	0.325	2.90
M 4.00	0.70	0.101	3.545	3.141	3.242	0.429	0.379	3.30
M 4.50	0.75	0.108	4.013	3.580	3.688	0.460	0.406	3.80
M 5.00	0.80	0.115	4.480	4.019	4.134	0.491	0.433	4.20
M 6.00	1.00	0.144	5.350	4.773	4.917	$\boxed{0.613}$	0.541	5.00
M 7.00	1.00	0.144	6.350	5.773	5.917	0.613	0.541	6.00
M 8.00	1.25	0.180	7.188	6.466	6.647	0.767	0.677	6.80
M 9.00	1.25	0.180	8.188	7.466	7.647	0.767	0.677	7.80
M 10.00	1.50	0.217	9.026	8.160	8.376	0.920	$\boxed{0.812}$	8.50
M 11.00	1.50	0.217	10.026	9.160	9.376	0.920	0.812	9.50
M 12.00	1.75	0.253	10.863	9.853	10.106	1.074	0.947	10.20
M 14.00	2.00	0.289	12.701	11.546	11.835	1.227	1.083	12.00
M 16.00	2.00	0.289	14.701	13.546	13.835	1.227	1.083	14.00
M 18.00	2.50	0.361	16.376	14.933	15.394	1.534	1.353	15.50
M 20.00	2.50	0.361	18.376	16.933	17.294	1.534	1.353	17.50

<---use h3 for 1.0 mm pitch

<--- use H1 for 1.5mm pitch

Bolt root dia. is then:

$d_3 := d_b - 2h_3$ $\qquad$ $d_3 = 14.7732\,\text{mm}$

Total bolt cross sectional area:

$A_b := n \cdot \frac{\pi}{4} d_3^{\,2}$ $\qquad$ $A_b = 226.263\,\text{cm}^2$

Check bolt to bolt clearance, here we use narrow thick washers (28mm OD) under the 24mm wide (flat to flat) nuts (28mm is also corner to corner distance on nut), we adopt a minimum bolt spacing of 2x the nominal bolt diameter (to give room for a 24mm socket) :

$\pi C - 2.0 n \cdot d_b \geq 0 = 1$ $\qquad$ actual bolt to bolt distance: $\frac{\pi C}{n} = 34.034\,\text{mm}$

Check nut, washer, socket clearance: $OD_w := 2d_b$ $\qquad$ this is for standard narrow washers, and for wrench sockets which more than cover the nut width across corners

$0.5C - \left(0.5B + g_1 + r_1\right) \geq 0.5OD_w = 1$

Flange hole diameter, minimum for clearance :

$D_{tmin} := d_b + 2\text{mm}$ $\qquad$ $D_{tmin} = 18\,\text{mm}$

We will thread some of these clearance holes for M20-1.5 bolts to allow the head retraction fixture to be bolted up the the flange. The effective diameter of these holes will be the average of nominal and minimum diameters. To avoid thread interference with flange bolts, the studs will be machined to root diameter per **UG-12(b)**.in between threaded ends of 1.5x diameter in length. The actual clearance holes will be 18mm, depending on achievable tolerances, so as to allow threading where needed.

$H_1 := .812\text{mm}$ from chart above

$d_{min\_20\_1.5} := 20\text{mm} - 2 \cdot H_1$ $d_{min\_20\_1.5} = 1.838\,\text{cm}$ this will be max bolt hole size or least material condition (LMC)

$d_{min\_20\_1.5} \geq D_{tmin} = 1$

$D_e := 0.5\left(20 \cdot \text{mm} + d_{min\_20\_1.5}\right)$ $D_e = 1.919\,\text{cm}$

Set:

$D_t := D_e$ $D_t > D_{tmin} = 1$

Compute Forces on flange:

We use a unit gasket seating force of Y1 above

$H_G := F_m$ $H_G = 3.019 \times 10^5\,\text{N}$

$h_G := 0.5(C - G)$ $h_G = 2.85\,\text{cm}$ from Table 2-6 Appendix 2, Integral flanges

$H_D := .785 \cdot B^2 \cdot P$ $H_D = 2.266 \times 10^6\,\text{N}$

$R_. := 0.5(C - B) - g_1$ $R_. = 2.5\,\text{cm}$ radial distance, B.C. to hub-flange intersection, int fl..

$h_D := R_. + 0.5 g_1$ $h_D = 3\,\text{cm}$ from Table 2-6 Appendix 2, Int. fl.

$H_T := H - H_D$ $H_T = 4.353 \times 10^4\,\text{N}$

$h_T := 0.5 \cdot \left(R_. + g_1 + h_G\right)$ $h_T = 31.75\,\text{mm}$ from Table 2-6 Appendix 2, int. fl.

Total Moment on Flange

$M_P := H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G$ $M_P = 7.797 \times 10^4\,\text{J}$

Appendix Y Calculation

$P = 15.4\,\text{bar}$

Choose values for plate thickness and bolt hole dia:

$t := 4.15\text{cm}$ $D := D_t$ $D = 1.919\,\text{cm}$

Going back to main analysis, compute the following quantities:

$\beta := \dfrac{C + B_1}{2B_1}$ $\beta = 1.022$ $h_C := 0.5(A - C)$ $h_C = 2.5\,\text{cm}$

$a := \dfrac{A + C}{2B_1}$ $a = 1.062$ $AR := \dfrac{n \cdot D}{\pi \cdot C}$ $AR = 0.564$ $h_0 := \sqrt{B \cdot g_0}$ $h_0 = 11.662\,\text{cm}$

$$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}}\,\text{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right) \qquad r_B = 7.462 \times 10^{-3}$$

We need factors F and V, most easily found in figs 2-7.2 and 7.3 (Appendix 2) since $\frac{g_1}{g_0} = 1$ these values converge to $F := 0.90892$ $V := 0.550103$

Y-5 Classification and Categorization

We have identical (class 1 assembly) integral (category 1) flanges, so from table Y-6.1, our applicable equations are (5a), (7) - (13), (14a), (15a), (16a)

$J_S := \frac{1}{B_1}\left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B$ $J_S = 0.083$ $J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi \cdot r_B$ $J_P = 0.062$

(5a) $F' := \frac{g_0^{\,2}\left(h_0 + F \cdot t\right)}{V}$ $F' = 2.806 \times 10^{-5}\,m^3$ $M_P = 7.797 \times 10^{4}\,N \cdot m$

$A = 1.48\,m$ $B = 1.36\,m$

$K := \frac{A}{B}$ $K = 1.088$ $Z := \frac{K^2 + 1}{K^2 - 1}$ $Z = 11.854$

$f := 1$ hub stress correction factor for integral flanges, use f =1 for g1/g0=1 (fig 2-7.6)

$t_s := 0mm$ no spacer between flanges

$l := 2t + t_s + 0.5d_b$ $l = 9.1\,cm$ strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

http://www.hightempmetals.com/techdata/hitempInconel718data.php

Elastic constants:

$E := E_{SS\_aus}$ $E = 193\,GPa$ $E_{Inconel\_718} := 208GPa$ $E_{bolt} := E_{Inconel\_718}$

Flange Moment due to Flange-hub interaction

$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'}$ $M_S = -1.8 \times 10^{3}\,N \cdot m$ (7)

Slope of Flange at I.D.

$\theta_B := \frac{5.46}{E \cdot \pi t^3}\left(J_S \cdot M_S + J_P \cdot M_P\right)$ $\theta_B = 5.903 \times 10^{-4}$ (8) opening half gap = $\theta_B \cdot 3cm = 0.018\,mm$

$E \cdot \theta_B = 113.924\,MPa$

Contact Force between flanges, at h_C:

$H_C := \frac{M_P + M_S}{h_C}$ $H_C = 3.045 \times 10^{6}\,N$ (9)

Bolt Load at operating condition:

$W_{m1} := H + H_G + H_C$ $W_{m1} = 5.657 \times 10^{6}\,N$ (10)

Operating Bolt Stress

$\sigma_b := \frac{W_{m1}}{A_b}$ $\sigma_b = 250\,MPa$ $S_b = 255.1\,MPa$ (11)

$r_E := \frac{E}{E_{bolt}}$ $r_E = 0.928$ elasticity factor

Design Prestress in bolts

$$S_i := \sigma_b - \frac{1.159 \cdot h_C^{\,2} \cdot \left(M_P + M_S\right)}{a \cdot t^3 \cdot l \cdot r_E \cdot B_1} \qquad S_i = 243.7\,\text{MPa} \qquad (12)$$

Radial Flange stress at bolt circle

$$S_{R\_BC} := \frac{6\left(M_P + M_S\right)}{t^2\left(\pi \cdot C - n \cdot D\right)} \qquad S_{R\_BC} = 135.4\,\text{MPa} \qquad (13)$$

Radial Flange stress at inside diameter

$$S_{R\_ID} := -\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_{R\_ID} = 1.61\,\text{MPa} \qquad (14a)$$

Tangential Flange stress at inside diameter

$$S_T := \frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_T = 2.46\,\text{MPa} \qquad (15a)$$

Longitudinal hub stress

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91\left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V} \qquad S_H = 19.372\,\text{MPa} \qquad (16a)$$

Y-7 Bolt and Flange stress allowables: $S_b = 255.1\,\text{MPa}$ $\quad S_f = 137.9\,\text{MPa}$

(a) $\sigma_b < S_b = 1$

(b)

(1) $S_H < 1.5S_f = 1$ $\qquad S_n$ not applicable

(2) not applicable

(c) $S_{R\_BC} < S_f = 1$

$S_{R\_ID} < S_f = 1$

(d) $S_T < S_f = 1$

(e) $\frac{S_H + S_{R\_BC}}{2} < S_f = 1$

$\frac{S_H + S_{R\_ID}}{2} < S_f = 1$

(f) not applicable

Bolt force

$$F_{bolt} := \sigma_b \cdot .785 \cdot d_b^{\,2} \qquad F_{bolt} = 5.024 \times 10^4\,\text{N}$$

Bolt torque required, minimum:

$T_{bolt\_min} := 0.2F_{bolt} \cdot d_b$ $\quad T_{bolt\_min} = 160.8\,N \cdot m \quad T_{bolt\_min} = 118.6\,lbf \cdot ft$ for pressure test use 1.5x this value

This is the minimum amount of bolt preload needed to assure joint does not open under pressure. An additional amount of bolt preload is needed to maintain a minimum frictional shear resistance to assure head does not slide downward from weight; we do not want to depend on lip to carry this. Non-mandatory Appendix S of div. 1 makes permissible higher bolt stresses than indicated above when needed to assure full gasket sealing and other conditions. This is consistent with proper preloaded joint practice, for properly designed joints where connection stiffness is much greater than bolt stiffness, and we are a long way from the yield stress of the bolts

$M_{head} := 2500kg \qquad \mu_{SS\_SS} := .7$ typ. coefficient of friction, stainless steel (both) clean and dry

$V_{head} := M_{head} \cdot g \qquad V_{head} = 2.452 \times 10^4\,N$

$F_n := \dfrac{V_{head}}{\mu_{SS\_SS}} \qquad F_n = 3.502 \times 10^4\,N$ this is total required force, force required per bolt is:

$F_{n\_bolt} := \dfrac{F_n}{n} \qquad F_{n\_bolt} = 265.331\,N$ this is insignificant compared to that required for pressure.

Let bolt torque for normal operation be then 25% greater than minimum:

$T_{bolt} := 1.25T_{bolt\_min} \qquad T_{bolt} = 201\,N \cdot m \qquad T_{bolt} = 148\,ft \cdot lbf$

It is recommended that a pneumatic torque wrench be used for tightening of bolts. Anti-seize lubricant (checked for radiopurity) should be used on threads and washers. Bolts should be tightened in 1/3 full torque increments, but there is no specific tightening pattern to be used, as gasket compression is not determined by bolt tightness. The head lift fixture may be retracted once all bolts not occupied by lift fixture have been tightened to the first 1/3 torque increment; there will be adequate frictional shear resistance to eliminate head slippage while detaching lift fixture. Bolts should be run up uniformly to fully close gap before proceeding with tightening. Do not forget to install sleeves in all threaded holes after removing lift fixture.

Additional Calculations for Shielding Weight:

Shear stress in inner flange lip from shield (could happen only if flange bolts come loose, are left loose, or if joint opens under pressure, otherwise friction of faces will support shield, given additional tension, as permissible under non-mandatory Appendix S above)

Masses of Copper shielding in cyl and heads (maybe extra in tracking head)

$t_{Cu} = 0.12\,m \qquad t_{Cu\_h} := 20cm \qquad L_{ff} = 1.6\,m \qquad \rho_{Cu} = 9 \times 10^3 \dfrac{kg}{m^3}$

$M_{sh\_head} := \rho_{Cu} \cdot \pi R_{i\_pv}^{\ 2} \cdot t_{Cu\_h} \qquad M_{sh\_head} = 2.615 \times 10^3\,kg$

$M_{sh\_cyl} := \rho_{Cu} \cdot 2\pi \cdot R_{i\_pv} \cdot t_{Cu} \cdot L_{ff} \qquad M_{sh\_cyl} = 7.383 \times 10^3\,kg$

$M_{sh} := M_{sh\_cyl} + 2M_{sh\_head} \qquad M_{sh} = 1.261 \times 10^4\,kg$ slightly less than this, due to gaps

$t_{lip} := 3mm$

Shear stress in lip (projected force):

$\tau_{lip} := \dfrac{M_{sh\_head} \cdot g}{R_{i\_pv} \cdot t_{lip}} \qquad \tau_{lip} = 12.57\,MPa$

Shear stress on O-ring land (section between inner and outer O-ring), from pressurized O-ring. This is assumed to be the primary stress. There is some edge moment but the "beam" is a very short one. This shear stress is not in the same direction as the nominal tangential (hoop) stress of the flange.

$t_{land\_radial} := .36cm \qquad w_{land\_axial} := .41cm$

$F_{O\_ring\_land} := 2\pi R_{i\_pv} \cdot w_{land\_axial} \cdot P$

$A_{O\_ring\_land} := 2\pi R_{i\_pv} \cdot t_{land\_radial}$

$$\tau_{land} := \frac{F_{O\_ring\_land}}{A_{O\_ring\_land}} \qquad \tau_{land} = 1.778\,MPa$$

Bolt loads from Cu bars

The internal copper shield bars are attached to the inside flanges with M6-1 bolts. The worst case for attachment ar the bars with collimation holes; these are narrow where they attach. For a flange hole pattern of 240 bolts, there are 5 attachment holes at each end.

$d_{root\_M6} := 4.77mm$

On the tracking side, the bars will be pulled up tight to the inside flange. On the energy side they must float axially, this is done using a special shoulder bolt which provides a loose double shear connection. Worst case would be single shear, where the tracking side bolts are left loose.

$M_{cubar\_vfan} := 225kg$

$$\tau_{bolt\_cubar} := \frac{0.5M_{cubar\_vfan} \cdot g}{5 \cdot \frac{\pi}{4} d_{root\_M6}^{2}} \qquad \tau_{bolt\_cubar} = 12.347\,MPa$$

This stress is inconsequential, as bolts will be ASME SB-98 silicon copper UNS C65500 - HO2 (half hard cond); this material should be radiopure and has > 20% elongation in the hard condition. Shear strength in yield is 50% Sy.

$S_{y\_65500\_H2} := 38000psi \qquad S_{y\_65500\_H2} = 262.001\,MPa$

$S_{sy\_65500\_H2} := 0.5S_{y\_65500\_H2} \qquad S_{sy\_65500\_H2} = 131\,MPa$

O-Ring groove dimensions

the Recommmended range of compression for static face seals is 21-30% in the Parker O-ring handbook; Trelleborg recommend 15-30%. For each nominal size, there are several cross sections, metric, JIS and A-568. Ity ios recommended by this author to design a groove which can accomodate all these cross sections with squezze in the acceptable range, so as to give the most flexibility.

For large diameter O-rings, Parker recommends using one size smaller to avoid sag. This is feasible for the inner O-ring, as the undercut lip is on the ID of the groove, but will not work on the outer vacuum O-ring as the undecut must be on the OD (otherwise the undercut may reduce seal effectiveness). Using an O-ring 1 or 2 sizes larger on the outer O-ring may develop enough compressive stress to retain O-ring in groove, but this should be tested. Stiffer compounds may help here if there is a problem Regardless, the groove dimensions should account for the stretch or compression of the O-ring which changes its effective cross section diameter. There are several close sizes that Trelleborg makes unspliced O-rings from (these are strongly preferred) and a stiffer than normal compound could be used for the vacuum O-ring, if needed

Inner (pressure bearing) O-ring:

Groove wall radii (average), depth, inner corner radii:

$R_{Ogpo} := 688.7mm$ $R_{Ogpi} := 682.25mm$ $d_{Opg} := 3.8mm$ $r_{ip} := 1mm$

O-ring inner radius, cross section diameter, unstretched

$R_{Opi} := 660mm$ $d_{Op} := \begin{pmatrix} 5 \\ 5.34 \end{pmatrix} mm$ metric size / AS - 568 size

O-ring elongation (tangential direction, normal to cross section)

$\varepsilon_{Opt} := 1 - \frac{R_{Opi}}{R_{Ogpi}}$ $\varepsilon_{Opt} = 3.261\,\%$ recommeded less than 3% (Trelleborg); 3% is our min. target

Bulk Modulus of most rubber polymers is very high, material is essentially incompressible (Poisson's ratio = -0.5)

Strain, O-ring cross section, in axial direction

$\varepsilon_{Opa} := -0.5\varepsilon_{Opt}$ $\varepsilon_{Opa} = -0.016$

O-ring dia., stretched:

$d_{Ops} := d_{Op} \cdot (1 + \varepsilon_{Opa})$ $d_{Ops} = \begin{pmatrix} 4.918 \\ 5.253 \end{pmatrix} mm$

Resulting squeeze (using the vectorize operator to continue parallel calculations)

$sq_p := \overrightarrow{\frac{d_{Ops} - d_{Opg}}{d_{Ops}}}$ $sq_p = \begin{pmatrix} 22.74 \\ 27.659 \end{pmatrix} \%$ $15\% < sq_p < 30\% = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

O-ring groove cross sectional area,

$A_{Opg} := \overrightarrow{\left[d_{Opg} \cdot (R_{Ogpo} - R_{Ogpi}) - \left(\frac{1}{2} - \frac{\pi}{2} \right) \cdot r_{ip}^{2} \right]}$ $A_{Opg} = 2.558 \times 10^{-5} m^2$

Trelleborg recommends no more than 85% fill ratio

$R_{fp} := \frac{\overrightarrow{\frac{\pi}{4} d_{Ops}^{2}}}{A_{Opg}}$ $R_{fp} = \begin{pmatrix} 74.274 \\ 84.718 \end{pmatrix} \%$ $R_{fp} < 85\% = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

Outer (vacuum) O-ring:

Groove wall radii (average), depth, inner corner radii:

$R_{Ogvo} := 697.66mm$ $R_{Ogvi} := 692.93mm$ $d_{Ovg} := 2.6mm$ $r_{iv} := 0.6mm$

O-ring inner radius, cross section diameter, unstretched

$R_{Ovi} := 730mm$ $d_{Ov} := \begin{pmatrix} 3 \\ 3.55 \end{pmatrix} mm$ metric size
metric/JIS size note: there are several intermediate sizes

O-ring elongation (tangential direction, normal to cross section)

$\varepsilon_{Ovt} := 1 - \frac{R_{Ovi}}{R_{Ogvi}}$ $\varepsilon_{Ovt} = -5.35\,\%$ recommended less than 3% (Trelleborg); we go for ~5% here as compression should not compromise integrity

Bulk Modulus of most rubber polymers is very high, material is essentially incompressible (Poisson's ratio = -0.5)

Strain, O-ring cross section, in axial direction

$\varepsilon_{Ova} := -0.5\varepsilon_{Ovt}$ $\varepsilon_{Ova} = 0.027$

O-ring dia., stretched:

$d_{Ovs} := d_{Ov} \cdot (1 + \varepsilon_{Ova})$ $d_{Ovs} = \begin{pmatrix} 3.08 \\ 3.645 \end{pmatrix} mm$

Resulting squeeze

$sq_v := \overrightarrow{\frac{d_{Ovs} - d_{Ovg}}{d_{Ovs}}}$ $sq_v = \begin{pmatrix} 15.591 \\ 28.669 \end{pmatrix} \%$ $15\% < sq_v < 30\% = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

O-ring groove cross sectional area,

$A_{Ovg} := \overrightarrow{\left[d_{Ovg} \cdot (R_{Ogvo} - R_{Ogvi}) - \left(\frac{1}{2} - \frac{\pi}{2} \right) \cdot r_{iv}^{2} \right]}$ $A_{Ovg} = 1.268 \times 10^{-5}\, m^2$

Fill ratio; Trelleborg recommends no more than 85%:

$R_{fv} := \frac{\overrightarrow{\frac{\pi}{4} d_{Ovs}^{2}}}{A_{Ovg}}$ $R_{fv} = \begin{pmatrix} 58.752 \\ 82.269 \end{pmatrix} \%$ $R_{fv} < 85\% = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$ We should have a comfortable margin here

Support Design using rules of div 2, part 4.15:

From the diagram below the rules are only applicable to flange attached heads if there is a flat cover or tubesheet inside, effectively maintaining the flanges circular. Since the PMT carrier plate and shielding is firmly bolted in, it serves this purpose and we may proceed. We must also compute the case with the heads attached, as there will be additional load

a) Design Method- although not specifically stated, the formulas for bending moments at the center and at the supports are likely based on a uniform loading of the vessel wall from the vessel contents. In this design, the internal weight (primarily of the copper shield) is applied at the flanges; there is no contact with the vessel shell. We calculate both ways and take the worst case.

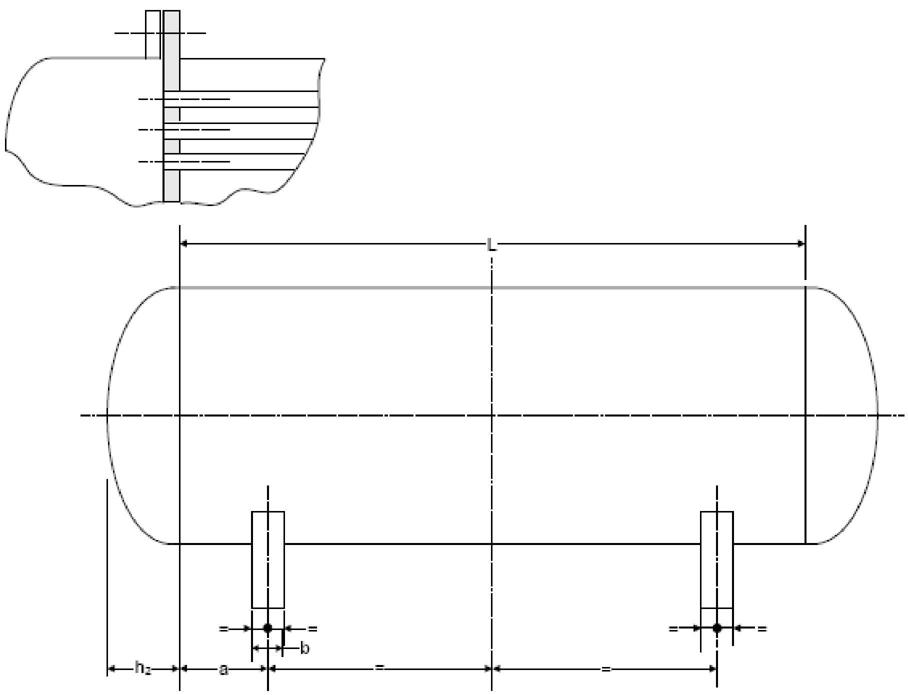


Figure 4.15.1 – Horizontal Vessel on Saddle Supports

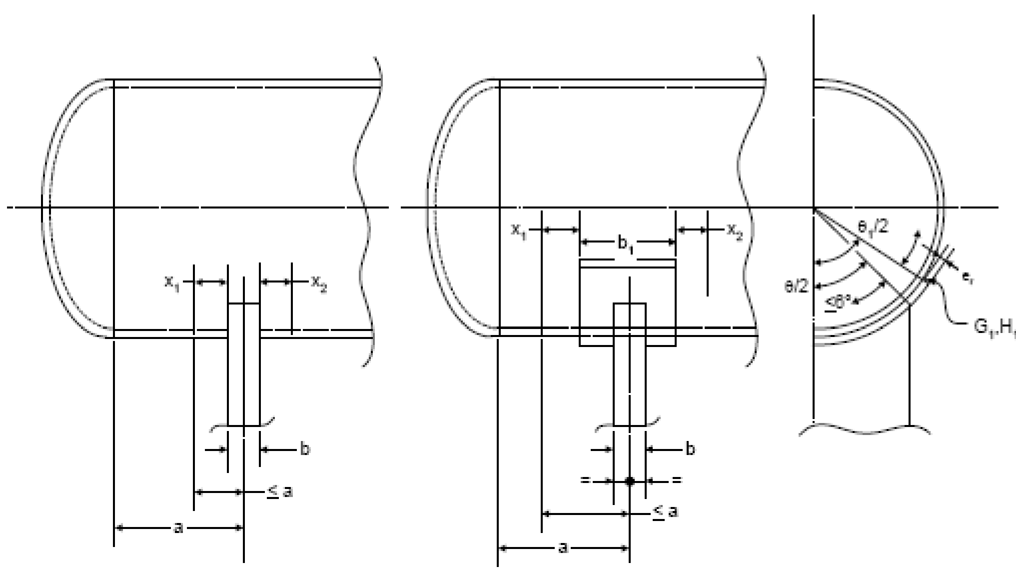


Figure 4.15.2 – Cylindrical Shell Without Stiffening Rings

$L := L_{ff}$ $M_{tot} := 12000kg$ $L = 1.6\,m$

$b := 1.5cm$ $a_{min} := .18L_{ff}$ $a_{min} = 28.8\,cm$ $a := 29cm$ $\theta := 120deg$ $R_m := R_{i\_pv} + 0.5t_{pv}$

$b_1 := \min\left[\left(b + 1.56\cdot\sqrt{R_m\cdot t_{pv}}\right), 2\cdot a\right]$ $b_1 = 14.411\,cm$ $h_2 := 20cm$ $k := 0.1$

$\theta_1 := \theta + \frac{\theta}{12}$ $\theta_1 = 130\,deg$

maximum reaction load at each support:

$Q := 0.5M_{tot}\cdot g$ $Q = 5.884\times 10^4\,N$

$$M_1 := -Q\cdot a\cdot\left(1 - \frac{1 - \frac{a}{L} + \frac{R_m^{\ 2} - h_2^{\ 2}}{2\cdot a\cdot L}}{1 + \frac{4h_2}{3L}}\right)$$

$M_1 = 1.676\times 10^3\,N\cdot m$ $Q\cdot a = 1.706\times 10^4\,J$

$$M_2 := \frac{Q\cdot L}{4}\cdot\left[\frac{1 + \frac{2\cdot\left(R_m^{\ 2} - h_2^{\ 2}\right)}{L^2}}{1 + \frac{4\cdot h_2}{3L}} - \frac{4a}{L}\right]$$

$M_2 = 9.875\times 10^3\,N\cdot m$

$M_{1'} := Q\cdot a$ $M_{1'} = 1.706\times 10^4\,N\cdot m$

$M_{2'} := M_{1'}$ $M_{2'} = 1.706\times 10^4\,N\cdot m$

$$T := \frac{Q\cdot(L - 2a)}{L + \frac{4h_2}{3}}$$

$T = 3.215\times 10^4\,N$

4.15.3.3 - longitudinal stresses

distributed load (ASME assumption) end load (actual)

$\sigma_1 := \frac{P\cdot R_m}{2t_{pv}} - \frac{M_2}{\pi R_m^{\ 2} t_{pv}}$ $\sigma_1 = 52.789\,MPa$ $\sigma_{1'} := \frac{P\cdot R_m}{2t_{pv}} - \frac{M_{2'}}{\pi R_m^{\ 2} t_{pv}}$ $\sigma_{1'} = 52.301\,MPa$

$\sigma_2 := \frac{P\cdot R_m}{2t_{pv}} + \frac{M_2}{\pi R_m^{\ 2} t_{pv}}$ $\sigma_2 = 54.128\,MPa$ $\sigma_{2'} := \frac{P\cdot R_m}{2t_{pv}} + \frac{M_{2'}}{\pi R_m^{\ 2} t_{pv}}$ $\sigma_{2'} = 54.616\,MPa$

same stress at supports, since these are stiffened, as a<0.5Rm and close to a torispheric head $a < 0.5R_m = 1$

$\sigma_3 := \frac{P\cdot R_m}{2t_{pv}} - \frac{M_1}{\pi R_m^{\ 2} t_{pv}}$ $\sigma_3 = 53.345\,MPa$ $\sigma_{3'} := \frac{P\cdot R_m}{2t_{pv}} - \frac{M_{1'}}{\pi R_m^{\ 2} t_{pv}}$ $\sigma_{3'} = 52.301\,MPa$

$$\sigma_4 := \frac{P \cdot R_m}{2t_{pv}} + \frac{M_1}{\pi R_m^{\ 2} t_{pv}} \qquad \sigma_4 = 53.572\,\text{MPa} \qquad\qquad \sigma_{4'} := \frac{P \cdot R_m}{2t_{pv}} + \frac{M_{1'}}{\pi R_m^{\ 2} t_{pv}} \qquad \sigma_{4'} = 54.616\,\text{MPa}$$

4.15.3.4 - Shear stresses

$$\Delta := \frac{\pi}{6} + \frac{5\theta}{12} \qquad \Delta = 1.396$$

$$\alpha := 0.95\left(\pi - \frac{\theta}{2}\right) \qquad \alpha = 1.99$$

$$K_2 := \frac{\sin(\alpha)}{\pi - \alpha + \sin(\alpha)\cos(\alpha)} \qquad K_2 = 1.171$$

here we use c), formula for cyl. shell with no stiffening rings and which is not stiffened by a formed head, flat cover or tubesheet. This is worst case, as we havea flange, which can be considered as one half of a stiffening ring pair for each support.

c) $$\tau_1 := \frac{K_2 \cdot T}{\pi R_m \cdot t_{pv}} \qquad \tau_1 = 1.749\,\text{MPa} \qquad (4.15.14)$$

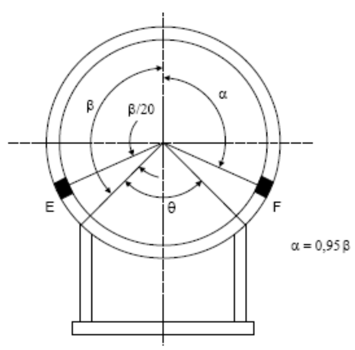
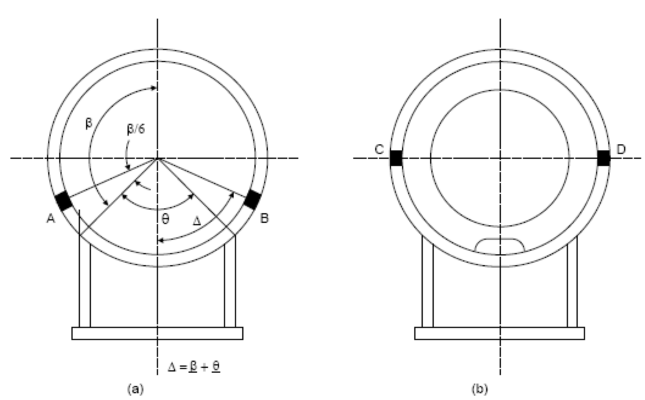


Figure 4.15.5 – Locations of Maximum Longitudinal Normal Stress and Shear Stress in the Cylinder



4.15.3.5 Circumferential Stress

$$K_5 := \frac{1 + \cos(\alpha)}{\pi - \alpha + \sin(\alpha) \cdot \cos(\alpha)} \qquad K_5 = 0.76$$

$$\beta := \pi - \frac{\theta}{2} \qquad \beta = 2.094$$

$$K_6 := \frac{\frac{3\cdot\cos(\beta)}{4}\cdot\left(\frac{\sin(\beta)}{\beta}\right)^2 - \frac{5\cdot\sin(\beta)\cdot\cos(\beta)}{4\cdot\beta} + \frac{\cos(\beta)^3}{2} - \frac{\sin(\beta)}{4\cdot\beta} + \frac{\cos(\beta)}{4} - \beta\cdot\sin(\beta)\cdot\left[\left(\frac{\sin(\beta)}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin(2\cdot\beta)}{4\cdot\beta}\right]}{2\cdot\pi\cdot\left[\left(\frac{\sin(\beta)}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin(2\cdot\beta)}{4\cdot\beta}\right]}$$

$$K_6 = -0.221$$

$$\frac{a}{R_m} < 0.5 = 1$$

$$K_7 := \frac{K_6}{4} \qquad K_7 = -0.055$$

a) Max circ bending moment

1) Cyl shell without a stiffening ring

$$M_\beta := K_7\cdot Q\cdot R_m \qquad M_\beta = -2.223\times 10^3\ \mathrm{N\cdot m}$$

c) Circ. stress in shell, without stiffening rings

$$x_1 := 0.78\sqrt{R_m\cdot t_{pv}} \qquad x_1 = 6.456\,\mathrm{cm} \qquad x_2 := x_1 \qquad k = 0.1$$

$$\sigma_6 := \frac{-K_5\cdot Q\cdot k}{t_{pv}\cdot(b + x_1 + x_2)} \qquad \sigma_6 = -3.104\,\mathrm{MPa}$$

$$L < 8R_m = 1 \qquad L = 1.6\,\mathrm{m} \qquad b_1 = 14.411\,\mathrm{cm}$$

$$\sigma_7 := \frac{-Q}{4t_{pv}\cdot(b + x_1 + x_2)} - \frac{12K_7\cdot Q\cdot R_m}{L\cdot t_{pv}^2} \qquad \sigma_7 = 156.484\,\mathrm{MPa} \qquad (4.15.25)$$

too high; we need a reinforcement plate of thickness;

$t_r := t_{pv}$ strength ratio: $\eta := 1$ (4.15.29)

$$\sigma_{7r} := \frac{-Q}{4(t_{pv} + \eta\cdot t_r)\cdot b_1} - \frac{12K_7\cdot Q\cdot R_m}{L\cdot(t_{pv} + \eta\cdot t_r)^2} \qquad \sigma_{7r} = 36.569\,\mathrm{MPa} \qquad (4.15.28)$$

3) f) Acceptance Criteria

$$S = 1.379\times 10^8\ \mathrm{Pa} \qquad S = 2\times 10^4\ \mathrm{psi}$$

$$|\sigma_{7r}| < 1.25S = 1$$

4) this section not applicable as $t_r > 2t_{pv} = 0$

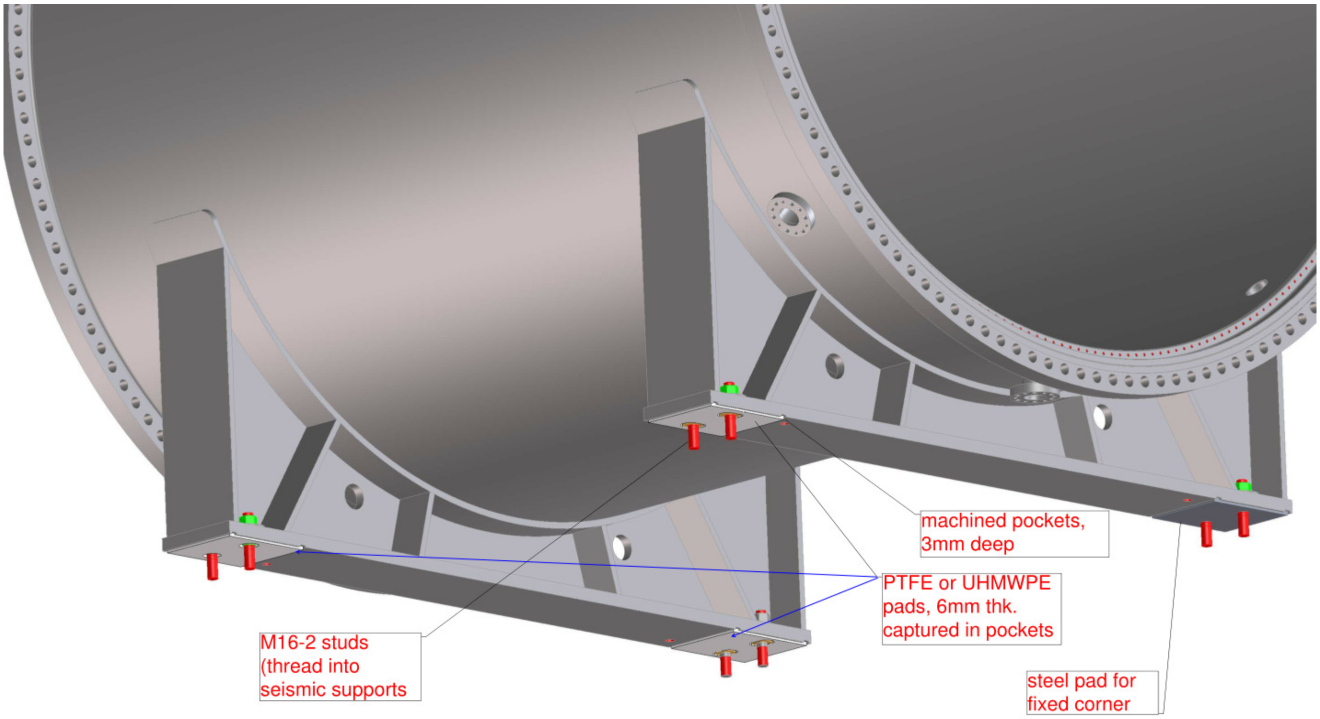
4.15.3.6 - Saddle support, horizontal force given below must be resisted by low point of saddle (where height = h_s)

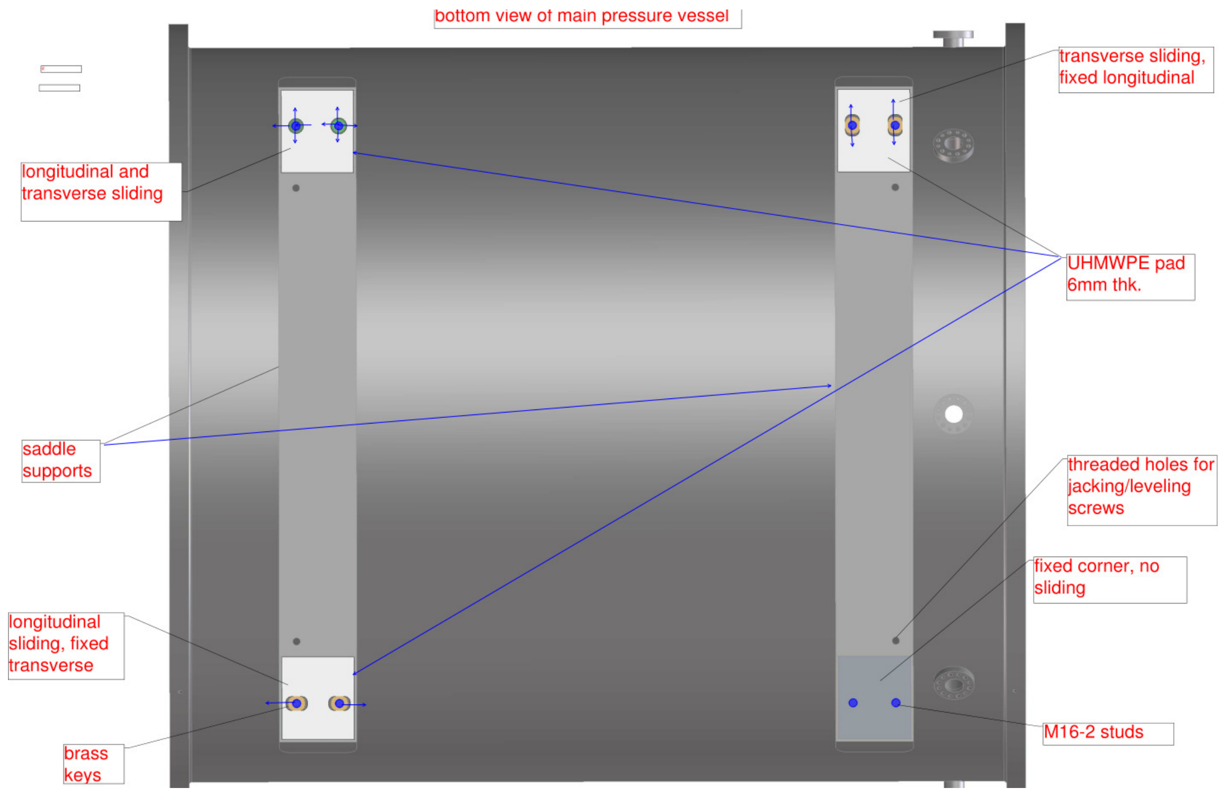
$$F_h := Q \cdot \left(\frac{1 + \cos(\beta) - 0.5 \cdot \sin(\beta)^2}{\pi - \beta + \beta \cdot \sin(\beta) \cos(\beta)} \right) \qquad F_h = 5.242 \times 10^4 \, N \qquad h_s := 9cm$$

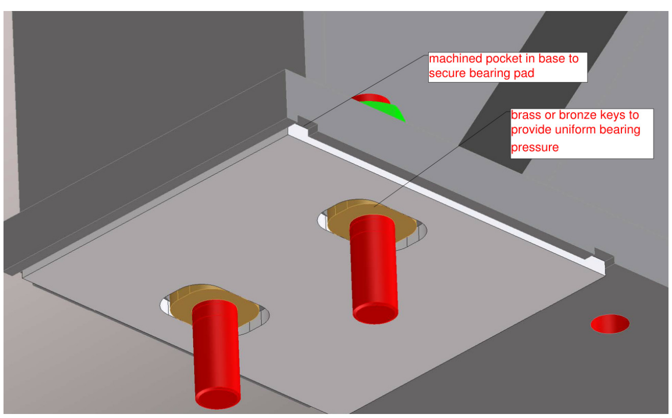
$$\sigma_h := \frac{F_h}{b \cdot h_s} \qquad \sigma_h = 38.833 \, MPa$$

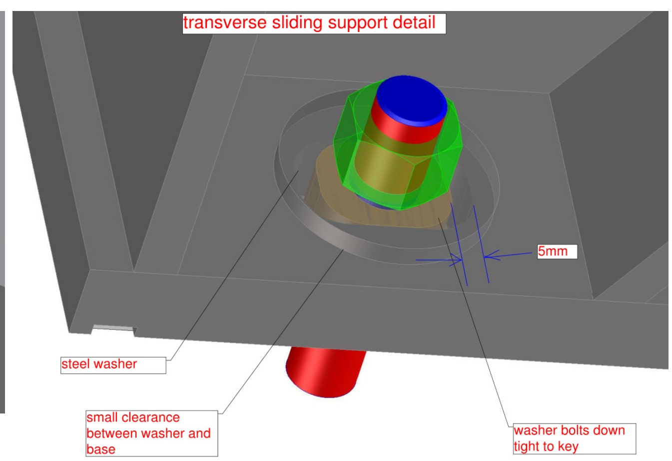
Support on, and Attachment to floor

The vessel has four points of connection ("corners") to the seismic platform, two on each saddle support; each connection point consisting of two M16mm studs. Other possibilities exist. Pressurization or thermal excursion (bakeout, cryogen spill from ArDM) will result in dimensional changes of the vessel, so it is required to use low friction pads under the supports to constrain the vessel in a 2D kinematic fashion. One corner is fixed, two others are slotted to allow sliding in one direction (orthogonal to each other), and a full clearance hole pattern at the fourth corner allows sliding in both directions.









Vessel length and width change under pressurization and heating:

length between saddle supports:

$L_s := L_{ff} - 2a \quad L_s = 1.02\,m$

$R_{i\_pv} = 0.68\,m \quad t_{pv} = 10\,mm$

Pressure load, longitudinal

$H_D = 2.266\times 10^6\,N$

saddle support width, transverse

$w_s := 1.2m$

stresses in vessel shell, longitudinal and tangential (hoop):

$$\sigma_{long} := \frac{H_D}{2\pi R_{i\_pv}\cdot t_{pv}} \qquad \sigma_{long} = 53.041\,MPa \qquad \sigma_{hoop} := \frac{P\cdot R_{i\_pv}}{t_{pv}} \qquad \sigma_{hoop} = 106.137\,MPa$$

length width changes from pressure:

$E_{SS\_aus} = 193\,GPa$

$$\delta L_s := \frac{\sigma_{long} \cdot L_s}{E_{SS\_aus}} \qquad \delta L_s = 0.28\,mm$$

$$\delta w_s := \frac{\sigma_{hoop} \cdot w_s}{E_{SS\_aus}} \qquad \delta w_s = 0.66\,mm$$

in reality, the support itself will restrain a significant portion of this deflection, since the saddle is welded to the vessel shell

thermal growth, 150C bakeout

$\alpha_{SS} := 16 \cdot 10^{-6} K^{-1}$ up to 100C

$\Delta T_v := 150K - 20K$

$\varepsilon_{th\_SS} := \alpha_{SS} \cdot \Delta T_v$

$\delta_{v\_t} := \varepsilon_{th\_SS} \cdot w_s \qquad \delta_{v\_t} = 2.496\,mm$

$\delta_{v\_l} := \varepsilon_{th\_SS} \cdot L_s \qquad \delta_{v\_l} = 2.122\,mm$

bakeout will only be performed under vacuum condition.

These deflections (from either pressure or thermal excursion) are substantial enough to warrant the use of low friction pads under three of the four supports, which will allow the vessel to slide both lengthwise and widthwise when pressurizing/depressurizing or baking. In addition there is a remote possibility of cryogen spillage, perhaps from ArDM which may chill the vessel, so a capacity for contraction equal to the above expansion should be designed in. Bolt holes should be slotted, with sliding keys to give uniform bearing pressure on slots under transverse loads, as described above. In addition, each corner should have one large tapped hole for a leveling/jacking screw that will allow bearing pad replacement, in situ.

Bolt shear stress from seismic acceleration

The maximum horizontal acceleration from a seismic event is expected to be much less than 1 m/s²; we use a design value here of:

$$a_{horiz} := 2\frac{m}{s^2}$$

$F_{horiz} := M_{tot} \cdot a_{horiz} \qquad F_{horiz} = 2.4 \times 10^4\,N$

Bolt area required:

We calculate for all horizontal load taken on two corners only, since we will have sliding supports. We calculate for austenitic stainless steel bolts:

$S_{sup\_bolt} := S_f \qquad S_{sup\_bolt} = 137.895\,MPa$

maximum shear stress:

$S_{s\_sup\_bolt} := 0.5 S_{sup\_bolt}$

bolt area required, per corner

$$A_{sup\_bolts} := \frac{0.5 F_{horiz}}{S_{s\_sup\_bolt}} \qquad A_{sup\_bolts} = 1.74\,cm^2$$

assume 2 bolts per corner, for redundancy and symmetry about support web. with 2 bolts, the only critical dimension to match between the holes in the support and the holes in the seismic frame are the distance between the hole pairs (hole pattern rotation need not be matched). The sliding keys can be custom machined if needed to compensate for mismatch.

$$d_{sup\_bolt} := \sqrt{\frac{4}{\pi} \cdot 0.5 A_{sup\_bolts}} \qquad d_{sup\_bolt} = 10.526\,\text{mm}$$ this is required minimum root diameter

Support uses (2) M16-2.0 bolts on each corner, root diameter is 12mm

Bearing design

Assume a full square contact patch under each corner;accounting for bolts and keys:

$$A_{bearing} := b_1^{\,2} - 4A_{sup\_bolts} \qquad A_{bearing} = 200.724\,\text{cm}^2$$

Bearing pressure is then (assuming a non-leveled condition where full weight is supported on two diagonal corners):

$$P_{bearing} := 0.5\frac{M_{tot} \cdot g}{A_{bearing}} \qquad P_{bearing} = 425.162\,\text{psi}$$

Maximum allowable bearing pressures and temperatures (we may bake vessel at 150C with copper shielding inside)

frrom Slideways bearing catalogue (similar to table 10-4 in J. Shigley, Mech. Engin. 3rd ed.)

Physical Properties of Various Materials

Physical Properties	UHMW	OF/UHMW	Wood	MD-Nylon	Nylon	PTFE	Acetal
PV Capacity (psi-fpm)	2,000	6,000	15,000	3,500	2,700	1,000	3,000
Max Pressure (psi)	1,200	600	1,000	2,000	2,000	500	1,000
Max Velocity (fpm)	100	500	500	150	100	400	100
Max Continuous Temp (˚F)	180	160	160	220	180	500	200
Dynamic Coefficient of Friction vs. Steel (dry)	.15-.20	.13-.16	0.09	.15-.35	.16-.43	.04-.10	.15-.35

Material for Bearing Pad

We choose only unfilled plastics, as most fillers are not radiopure (possible exception: bronze filled PTFE). PTFE (unfilled), @500 psi , has little margin for stability, but any creep flow will act to equalize pressure over all 4 supports, resulting in a lower, stable pressure. Furthermore it is the only material that can withstand 150C, although the temperature at the supports will be substantially less than 150C, due to the poor thermal conductivity of SS. Cooling of supports should be performed in case of bakeout, regardless. Bronze-filled PTFE, UHMWPE (non-oil-filled), nylon, or acetal may also be used; cooling of support pads during bakeout would be mandatory.

Jacking screw diameter

Each jacking screw must be able to lift half the entire weight of the detector. We look for a low grade bolt that can support this force

$$F_{js} := 0.5M_{tot} \cdot g \qquad F_{js} = 5.884 \times 10^4\,\text{N} \qquad F_{js} = 1.323 \times 10^4\,\text{lbf}$$

Use 90% yield strength as allowable stress (non critical)

$$S_{y\_316Ti} := 30000\text{psi}$$

$$A_{js} := \frac{F_{js}}{0.9 \cdot S_{y\_316Ti}} \qquad A_{js} = 3.161\,\text{cm}^2$$

$$d_{js\_root} := \sqrt{\frac{4}{\pi} A_{js}} \qquad d_{js\_root} = 20.061\,\text{mm}$$

Use an M24-2 bolt at each corner. Lubricate or PTFE coat (preferred)

Saddle support bending stress

Cross section of saddle support is an I-beam, with a central "web" connecting two "flanges" We check bending stress in support at bottom of vessel, where cross section height is a minimum.

Vessel axis height (axis above floor) $\quad R_{o\_pv} := R_{i\_pv} + t_{pv} \qquad R_{o\_pv} = 69\,cm$

$h_v := 80cm$

flange and web thicknesses, widths:

$t_{fll} := 2cm \quad w_{fll} := b_1 \quad w_{fll} = 14.411\,cm \quad t_w := 1.5cm \quad t_r = 1\,cm$

I-beam web height, not including flanges:

$h_w := h_v - \left(R_{o\_pv} + t_r + t_{fll}\right) \qquad h_w = 8\,cm$

I-beam Area Moment of Inertia:

Parallel axis theorem

sum moments of flanges and web about axis thru top surface , then divide by total area to find neutral axis

$A_r := b_1 \cdot t_r \qquad A_w := t_w \cdot h_w \qquad A_{fll} := w_{fll} \cdot t_{fll}$

$$c_1 := \frac{A_r \cdot \left(0.5 \cdot t_r\right) + A_w \cdot \left(t_r + 0.5 \cdot h_w\right) + A_{fll} \cdot \left(t_r + h_w + 0.5 \cdot t_{fll}\right)}{A_r + A_w + A_{fll}} \qquad c_1 = 6.435\,cm \quad \text{down from top surface}$$

$$I_r := \frac{b_1 \cdot t_r^{\,3}}{12} \qquad I_w := \frac{t_w \cdot h_w^{\,3}}{12} \qquad I_{fll} := \frac{w_{fll} \cdot t_{fll}^{\,3}}{12}$$

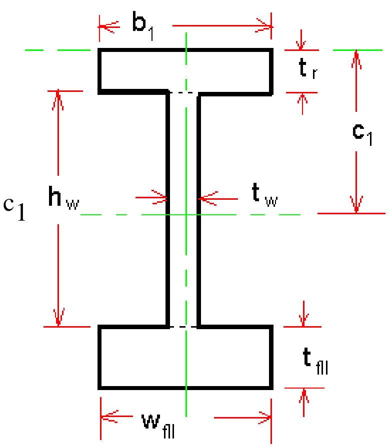
$I_r = 1.201\,cm^4 \qquad I_w = 64\,cm^4 \qquad I_{fll} = 9.608\,cm^4$

$d_r := 0.5t_r - c_1 \qquad d_w := \left(t_r + 0.5 \cdot h_w\right) - c_1 \qquad d_{fll} := \left(t_r + h_w + 0.5 \cdot t_{fll}\right) - c_1$

$d_r = -5.935\,cm \qquad d_w = -1.435\,cm \qquad d_{fll} = 3.565\,cm$

$$I_{s\_min} := \left(I_r + A_r \cdot d_r^{\,2}\right) + \left(I_w + A_w \cdot d_w^{\,2}\right) + \left(I_{fll} + A_{fll} \cdot d_{fll}^{\,2}\right)$$

$I_{s\_min} = 973.459\,cm^4$



Consider as a uniformly loaded beam, simply supported on each end

load per unit width (along the long dimension; transverse to vessel axis)

$$\omega := \frac{0.55M_{tot} \cdot g}{w_s} \qquad \omega = 539.366\,\frac{N}{cm} \qquad M_{tot} = 1.2 \times 10^4\,kg$$

Moment at center:

$$M_{sup\_max} := \frac{\omega \cdot w_s^{\,2}}{8} \qquad M_{sup\_max} = 9.709 \times 10^3\,N \cdot m$$

Maximum stress, tensile in flange under vessel

$$\sigma_{sup\_max} := \frac{M_{sup\_max} \cdot 0.5h_w}{I_{s\_min}} \qquad \sigma_{sup\_max} = 39.893\,MPa$$

This is low enough to allow support only at corners; we do not need to support under the full width of the support feet.

ANGEL Torispheric Head Design, using (2010 ASME PV Code Section VIII, div. 1, UG-32 Formed heads and sections, Pressure on Concave Side, Appendix 1-4 rules eq 3

$P = 1.561\times 10^{6}\,\mathrm{Pa}$ $\quad E = 1$ $\quad S = 2\times 10^{4}\,\mathrm{psi}$

I.D.

$D_i := 2R_{i\_pv}$

O.D.

$D_o := D_i + 2t$ $\qquad D_o = 1.443\,\mathrm{m}$

Crown radius: Knuckle radius:

$L_{cr} := 1D_i$ $\quad L_{cr} = 1.36\,\mathrm{m}$ $\qquad r_{kn} := 0.1D_i$ $\qquad r_{kn} = 0.136\,\mathrm{m}$

$E = 1$ $\quad S_{div1} := 20000\mathrm{psi}$ $\qquad S_{div1} = 1.379\times 10^{8}\,\mathrm{Pa}$ $\qquad S_{y\_316Ti} = 206.843\,\mathrm{MPa}$

Appendix 1-4 mandatory Supplemental Design Formulas

UG-32 does not give equations for a range of crown and knuckle radii; these are found in **App 1-4**

$$\frac{L_{cr}}{r_{kn}} = 10$$

$$M := \frac{1}{4}\left(3 + \sqrt{\frac{L_{cr}}{r_{kn}}}\right) \qquad M = 1.541$$

Minimum shell thickness:

$$t_{min} := \frac{P\cdot L_{cr}\cdot M}{2S\cdot E - 0.2P} \qquad t_{min} = 11.871\,\mathrm{mm} \qquad (3)$$

note: we will need full weld efficiency for the above thickness to be permissible, as per UG-32(b)

this formula is only valid if the following equation is true (1-4(a))

$$\frac{t_{min}}{L_{cr}} \ge 0.002 = 1 \qquad \frac{t_{min}}{L_{cr}} = 8.729\times 10^{-3}$$

Set head thickness:

$t_h := 12\mathrm{mm}$

Note: under EN_13455-3 rules for 316Ti, a thinner thickness of 10.25 mm is possible, due to a higher maximum allowable strength at the knuckle. Below is an analysis from Sara Carcel

Torispherical heads, VIII, Div 2			**DIN 28011**			**EN 13445-3 (316Ti, para D**	
	KORBBOGEN		r=0,1L			f	166.666667
D	1360		1360	Diámetro interior		X	0.1
t	10		10			t	10
De	1380		1380			Y	0.00735294
L	1104		1360	Diámetro interior corona		Z	2.13353891
ri	212.52		136			N	0.84954918
L/D	0.81176471	Ok	1	Entre 0,7 y 1, ver 4-49		β0,1	0.86799204
ri/D	0.15626471	Ok	0.1	Mayor de 0,06		β0,2	0.51421113
Li/t	110.4	Ok	136	Entre 20 y 2000		β	0.86799204
β	1.01880199		1.11024234			P	1.52
φ	0.49440713		0.85749293			**eb**	8.66383993
R	752.567792		697.850818	Si φ<β		**ey**	10.2275849
C1	0.71313518	r/D>0,08	0.6742			**es**	6.21577196
C2	1.05371176		1.2			**Thickness**	**10.2275849**
Peth	64.2498476	E=117000	44.2387943				
C3	206	Sy=206MPa	206				
Py	3.37117586		1.57121205				
G	19.0585868		28.1558395				
Pck	6.73919067		3.14731728	G>1			

Nozzle wall thickness required

Internal radius of finished opening

$R_n := 4.4\text{cm}$

Thickness required for internal pressure:

$t_{rn} := \dfrac{P \cdot R_n}{S \cdot E - 0.6 \cdot P}$ $\quad t_{rn} = 0.501\,\text{mm}$

We set nozzle thickness

$t_n := 7\text{mm}$ we are limited by need to maintain CF bolt pattern which has typically a 4.0 inch OD pipe with room for outside fillet weld

$D_{on} := 2(R_n + t_n)$ $\quad D_{on} = 4.016\,\text{in}$

Thickness required for external load

Nozzles on head may be subject to several possible non-pressure loads, simultaneously:

1. Reaction force from pressure relief, (fire) or fast depressure (auxiliary nozzle only)
2. Weight of attached components, including valves, expansion joints, copper or lead shielding plugs, high voltage feedthrough.

The nozzles may all have nozzle extensions rigidly attached which create to possibility of high moments being applied to the nozzles, not just shear loads. We consider the direction and location of center of gravity for these loads:

$L_{ne} := 58\text{cm}$ $\quad \rho_{Pb} := 11.3 \dfrac{\text{gm}}{\text{cm}^3}$

Forces and centers of gravity (l):

$F_{shp} := \pi R_n^{\,2} \cdot L_{ne} \cdot \rho_{Pb} \cdot g$ $\quad F_{shp} = 391\,\text{N}$ $\quad l_{shp} := 0.5 L_{ne}$

$W_{ne} := 2 \cdot \left(2\pi R_n \cdot L_{ne} \cdot t_n \cdot \rho_{SS}\right) \cdot g$ $\quad W_{ne} = 176\,\text{N}$ $\quad l_{Wne} := l_{shp}$ (factor of 2 to account for flange weights)

Fast vent reaction force, as calculated below

$F_{fv} := 3700\text{N}$ worst case is venting upward, at right angles to nozzle axis (we plan to use a straight through valve, regardless, for which reaction force will not produce a bending moment and will simply reduce longitudinal stress from pressure)

Moments:

$M_{shp} := F_{shp} \cdot l_{shp}$ $\quad M_{shp} = 113\,\text{N}\cdot\text{m}$

$M_{Wne} := W_{ne} \cdot l_{Wne}$ $\quad M_{Wne} = 51.1\,\text{N}\cdot\text{m}$

$M_{fv} := F_{fv} \cdot L_{ne}$ $\quad M_{fv} = 2146\,\text{N}\cdot\text{m}$

Total moment:

$M_n := M_{fv} + M_{shp} + M_{Wne}$ $\quad M_n = 2310\,\text{N}\cdot\text{m}$

Moment of Inertia, bending

$I_n := \pi \cdot \left(R_n + 0.5 t_n\right)^3 \cdot t_n$ $\quad I_n = 235.7\,\text{cm}^4$

Stress, bending (longitudinal)

$\sigma_{n\_1} := \dfrac{M_n \cdot \left(R_n + t_n\right)}{I_n}$ $\quad \sigma_{n\_1} = 50\,\text{MPa}$

Stress, circumferential (hoop)

$$\sigma_{n\_c} := \frac{P \cdot R_n}{t_n} \qquad \sigma_{n\_c} = 9.811\,\text{MPa}$$

Criterion for acceptable stress - use maximum shear stress theory:

Maximum shear stress (min. stress is in third direction, = zero on outside of nozzle):

$$\tau_n := \sqrt{\left(\frac{\sigma_{n\_1} - 0\text{MPa}}{2}\right)^2} \qquad \tau_n = 25\,\text{MPa}$$ OK (J. Shigley, Mech.Eng. 3rd ed., eq. (2-9)

Compare with maximum shear stress from minimum thickness nozzle (pressure only, no applied moments)

$$\sigma_{rn} := \frac{P \cdot R_n}{t_{rn}} \qquad \sigma_{rn} = 137\,\text{MPa}$$

$$\tau_{rn} := \sqrt{\left(\frac{\sigma_{rn} - 0\text{MPa}}{2}\right)^2} \qquad \tau_{rn} = 68.5\,\text{MPa}$$

Additional Factor of Safety, over ASME factor of safety:

$$FS_n := \frac{\tau_{rn}}{\tau_n} \qquad FS_n = 2.7$$ OK

External pressure:

Nozzles on head are very short; no analysis needed. Nozzle extensions are longer:

$$L_{ne} = 58\,\text{cm} \qquad t_{ne} := 7\text{mm}$$

$$\frac{L_{ne}}{2R_n} = 6.591 \qquad 2\frac{R_n}{t_{ne}} = 12.571$$

From charts HA-1 and HA-2 above:

$$A_{ne} := .02 \qquad B_{ne} := 13000\text{psi}$$

$$P_{a\_ne} := \frac{4B_{ne}}{3\left(\frac{2R_n}{t_{ne}}\right)} \qquad P_{a\_ne} = 93.795\,\text{bar}$$ OK

UG-37 Reinforcement Required for Openings in Shells and heads

Reinforcement is not required for the DN40 and DN75 flanged nozzles welded to the main cylindrical vessel as per **UG-36** below:

UG-36 (c) (3) Strength and Design of finished Openings:

(3) Openings in vessels not subject to rapid fluctuations in pressure do not require reinforcement other than that inherent in the construction under the following conditions:

<--no rapid fluctuations, condition met

(a) welded, brazed, and flued connections meeting the applicable rules and with a finished opening not larger than:

$3\frac{1}{2}$ in. (89 mm) diameter — in vessel shells or heads with a required minimum thickness of $\frac{3}{8}$ in. (10 mm) or less;

<-- applicable to cyl. vessel, condition met for DN40; DN75 nozzles

$2\frac{3}{8}$ in. (60 mm) diameter — in vessel shells or heads over a required minimum thickness of $\frac{3}{8}$ in. (10 mm);

<-- not applicable to cyl. vessel, but is applicable to heads; condition not met for DN100 nozzles, reinforcement needed

(b) threaded, studded, or expanded connections in which the hole cut in the shell or head is not greater than $2\frac{3}{8}$ in. (60 mm) diameter;

<--not applicable

(c) no two isolated unreinforced openings, in accordance with (a) or (b) above, shall have their centers closer to each other than the sum of their diameters;

<-- condition met

(d) no two unreinforced openings, in a cluster of three or more unreinforced openings in accordance with (a) or (b) above, shall have their centers closer to each other than the following: for cylindrical or conical shells,

<-- condition met

$$(1 + 1.5 \cos \theta)(d_1 + d_2);$$

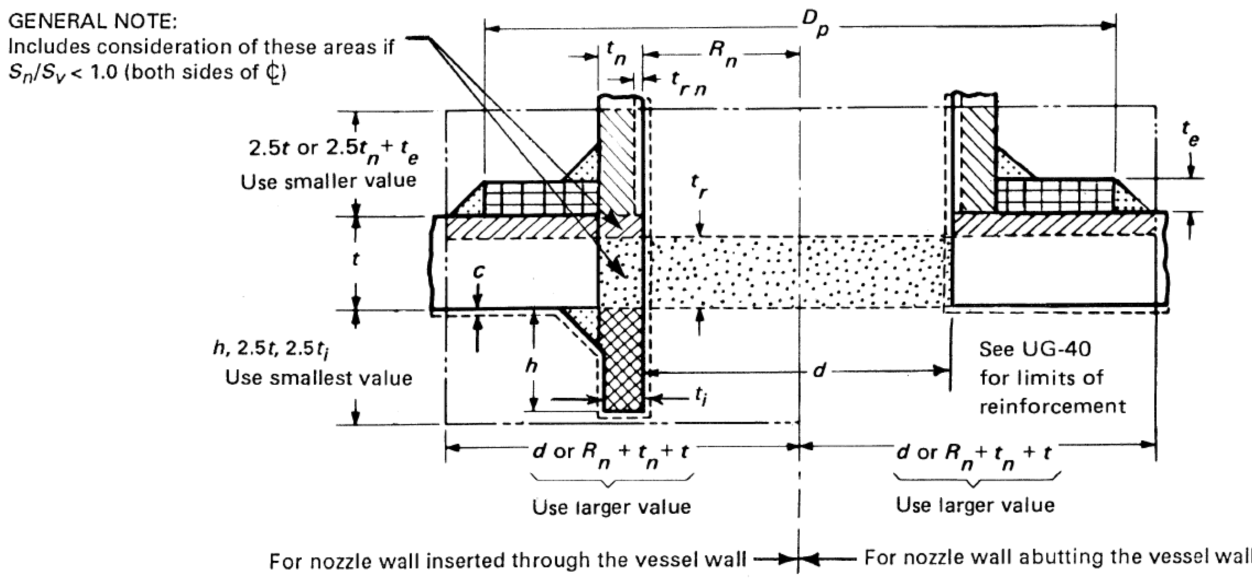
for doubly curved shells and formed or flat heads,

$$2.5(d_1 + d_2)$$

In addition, there are no significant external loads on the radial nozzles of the vessel, only the weight of an HV feedthrough at 45 deg angle; this is insignicant compared to the maximum loads and moments possible on the head nozzles (which are similar in size and thickness). Proceeeding with the head nozzle reinforcement:

Reinforcement for the DN100 flanged nozzles welded to the torispheric heads is required and calculated according to **UG-37** :

FIG. UG-37.1 NOMENCLATURE AND FORMULAS FOR REINFORCED OPENINGS



nozzle and reinforcement are also 316Ti

$t_{rn} = 0.501\,\text{mm}$ $\quad d := 2R_n$ $\quad d = 8.8\,\text{cm}$ $\quad F := 1.0$ $\quad f_r := 1.0$ $\quad f_{r1} := 1.0$ $\quad E_1 := 1$ $\quad f_{r2} := 1$

$t_n := t_h$ $\quad t_r = 12\,\text{mm}$ $\quad t_i := 0\text{mm}$ $\quad h := 0\text{mm}$ $\quad leg_i := 0\text{mm}$ $\quad t := 12\text{mm}$ $\quad leg_n := 1.4t_n$ $\quad leg_n = 0.98\,\text{cm}$

We will need a reinforcing pad, as the head is already minimum thickness and the nozzle is much thinner. Note: this is for ASME; sec VIII, div. 1. European Codes allow thinner thickness for 316Ti.

$t_e := 12\text{mm}$ $\quad D_p := 1.8d$ $\quad D_p = 0.158\,\text{m}$ (from UG-40 Limits of Reinforcement) $\quad f_{r4} := 1$ $\quad leg_e := .71\cdot t_e$

Area or reinforcement required:

$$A_{req} := d\cdot t_r\cdot F + 2t_n\cdot t_r\cdot F\cdot(1 - f_{r1}) \qquad A_{req} = 1.056\times 10^3\,\text{mm}^2$$

Area available in shell:

$$A_{1a} := d\cdot(E_1\cdot t - F\cdot t_r) - 2\cdot t_n\cdot(E_1\cdot t - F\cdot t_r)\cdot(1 - f_{r1})$$
$$A_{1b} := 2\cdot(t + t_n)\cdot(E_1\cdot t - F\cdot t_r) - 2\cdot t_n\cdot(E_1\cdot t - F\cdot t_r)\cdot(1 - f_{r1})$$

$$A_1 := \max(A_{1a}, A_{1b}) \qquad A_1 = 0\,\text{mm}^2$$

Area available in nozzle projecting outwards

$$A_{2a} := 5(t_n - t_{rn})\cdot f_{r2}\cdot t \qquad A_{2a} = 389.914\,\text{mm}^2$$

$$A_{2b} := 2\cdot(t_n - t_{rn})\cdot(2.5t_n + t_e)\cdot f_{r2} \qquad A_{2b} = 383.415\,\text{mm}^2$$

$$A_2 := \min(A_{2a}, A_{2b}) \qquad A_2 = 383.415\,\text{mm}^2$$

Area available in nozzle projecting inwards

$A_{3a} := 5t \cdot t_i \cdot f_{r2}$ $\qquad A_{3a} = 0\,mm^2$

$A_{3b} := 5t_i \cdot t_i \cdot f_{r2}$ $\qquad A_{3b} = 0\,mm^2$

$A_{3c} := 2 \cdot h \cdot t_i \cdot f_{r2}$ $\qquad A_{3c} = 0\,mm^2$

$A_3 := \min(A_{3a}, A_{3b}, A_{3c})$ $\quad A_3 = 0\,mm^2$

Area available in weld, outward

$A_{41} := leg_n^{\,2} \cdot f_{r2}$ $\qquad A_{41} = 96.04\,mm^2$

Area available in outer element weld

$A_{42} := leg_e^{\,2} \cdot f_{r4}$ $\qquad A_{42} = 72.59\,mm^2$

Area available in weld, inward

$A_{43} := leg_i^{\,2} \cdot f_{r2}$ $\qquad A_{43} = 0\,mm^2$

Area available in reinforcement

$A_5 := \left(D_p - d - 2t_n\right) \cdot t_e \cdot f_{r4}$ $\qquad A_5 = 676.8\,mm^2$

Total Area available

$A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 = 1229\,mm^2$

Area required:

$A_{req} = 1056\,mm^2$

$A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 \geq A_{req} = 1$

Torispheric Head, per DIN

A thinner head thickness of 10.58mm is calculated by S. Carcel to DIN formula; this is acceptable. It is not yet clear whether or not reinforcement pads are needed.

Pressure Relief Capacity requirements

There are two possible conditions 1. regulator failure and 2 external fire

$L_{pv} := 2.1m$ length of vessel , inside average $\qquad R_{o\_pv} = 0.69\,m$ outer radius

Pressure vessel outer area:

$A_{pv} := 2\pi R_{o\_pv}^{\;2} + 2\pi R_{o\_pv} \cdot L_{pv}$ $\qquad A_{pv} = 12.096\,m^2$

From Anderson Greenwood Technical Seminar Manual, fire sizing is:

$A_{orif} := \frac{F' \cdot A'}{\sqrt{P_1}} \cdot in^2$ $\qquad F' := .045$ $\qquad A' := \frac{A_{pv}}{ft^2}$ $\qquad P_1 := \frac{MAWP_{pv}}{psi}$ $\qquad A' = 130.198$

$P_1 = 226.38$

$A_{orif} = 0.389\,in^2$ $\qquad k := 1.667$ $\qquad K_D := 1$

$d_{orif} := \frac{4}{\pi} \cdot \sqrt{A_{orif}}$ $\qquad d_{orif} = 0.795\,in$

However we will want to use the higher value which gives a fast vent , so as to safe Xe in case of leak

$A_{vent} := \pi\cdot(30mm)^2 \qquad A_{vent} = 4.383\,in^2$

Mass flow: $\qquad P' := \frac{P}{psi} \qquad T := 535$ R, ambient

$M_a := M_{a\_Xe}\cdot\frac{mol}{gm}$

$$C_g := 520\cdot\sqrt{k\cdot\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} \qquad C_g = 377.641$$

$A_o := \frac{A_{vent}}{in^2} \qquad Z_c := .95$

$$W := \frac{A_o\cdot C_g\cdot K_D\cdot P'\cdot\sqrt{M_a}}{\sqrt{T\cdot Z_c}}\cdot\frac{lb}{hr} \qquad W = 24.419\,\frac{kg}{s} \qquad W = 1.938\times10^5\,\frac{lb}{hr}$$

Reactive force, from same ref. (pg. 49)

$W' := W\cdot\frac{hr}{lb} \qquad P_2 := 0$

$$F_T := \frac{W'\cdot\sqrt{\frac{k\cdot T}{(k+1)\cdot M_a}}}{366}\cdot lbf \qquad F_T = 3.694\times10^3\,N$$

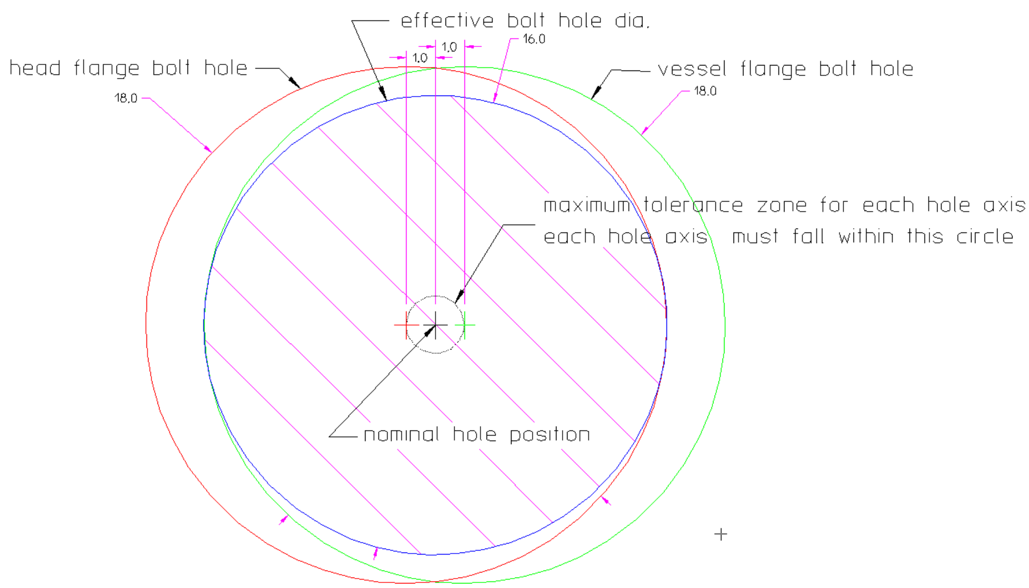
Tolerance analysis

Flange bolt holes:

First consideration is to realize that flanges absolutely must mate and bolt up without interference. This means that tolerances must not be considered to add up in any statistical manner, all features must be considered as being both at their limits of positional tolerance (oppositional) and in their maximum material condition (MMC). That is, all holes and female features are as small as the tolerances allow, and all bolts and male features are as large as the tolerances allow. Materials are all similar, so temperature ranges need not be considered, but part deformations under load must be factored in.

Heads will be assembled to the vessel by first mounting them to an adjustable cradle support which allows precise motions in all 6 degrees of freedom (translations in x,y, and z, plus pitch, yaw and roll about the center axes). This lift fixture is mounted on roller slides that move along the central vessel axis. The head is not assembled to the vessel by hanging it loosely from a crane hook, though this, and other methods are acceptable during construction.

A desirable, but not mandatory, design goal here is to assure that once the shear lip is assembled to the vessel ID, the bolts will all insert without further translational alignment (rotation may still be needed). Thus, if the vessel ID and mating shear lip are at MMC, and there is no remaining clearance between them, then the total bolt hole tolerance is equal to the hole to bolt diametral clearance, at MMC. This must be shared between the head and flange holes so the tolerance for each will be half the diametral tolerance. That is: for a 2 mm diametral clearance, each bolt hole axis may be as much as 1.0 mm off its nominal position; the total will be no more than 1 mm which produces an effective aperture 2 mm smaller than the hole diameter, = 16 mm and bolts will still assemble. In other words, the hole axis must be within a 1 mm radius (2 mm diameter) circle, thus the true position tolerance for the bolt holes is (a cylinder of) 2 mm dia. this is illustrated below:



$d_{cl\_mmc} := 18\text{mm}$

$d_b = 16\,\text{mm}$ note root diameter is less, but threaded portion is 25 mm long and must pass through both holes simultaneously

We need to account for vessel flange deflection under load which will distort the hole pattern. Maximum deflection, in the vertical direction is:

$\delta_{fl} := 0.1\text{mm}$

This is from an ANSYS workbench model with a 60000N load applied to each vessel internal flange, no head present. Assume head flange is undistorted

$t_{bh\_max} := d_{cl\_mmc} - \left(d_b + \delta_{fl}\right)$

$t_{bh\_max} = 1.9\,mm$

This is total maximum positional tolerance diameter for each flange hole, assuming the nominal hole positions of head and vessel flanges are in alignment.

There are two ways to specify bolt hole positional tolerance, either with respect to themselves as a pattern (the pattern otherwise unconstrained) or each hole individually, with respect to the specified datums. The former is a precisional tolerance, the latter an accuracy tolerance, (which is more difficult to achieve).

For the case where there is no remaining clearance between the shear lip and the vessel ID, when both are at MMC, the requirements for accuracy and precision are the same, and t_{bh_max} is the maximum allowable positional tolerance with respect to the datums A/B,C/D, and E/F; that is we have only an absolute accuracy requirement for bolt hole positional tolerance.

Any radial clearance between the shear lip and the vessel ID, with both at MMC, allows the two hole patterns to shift, as an ensemble, with respect to each other. This allows a larger maximum allowable positional tolerance of the holes with respect to the datums A/B,C/D, and E/F (accuracy requirement), but still requires that the hole pattern still be toleranced to t_{bh_max} or smaller, with respect to itself (a local precision or repeatability requirement). This is accomplished with two tolerance blocks on the drawing. The drawback is that the shear lip and vessel id may assemble, but in a shifted condition, such that bolts will not assemble. Additional alignment will then be needed. The cure for this is to tighten the tolerance (from t_{bh_max}) on the hole pattern in reference to itself, by the maximum shift that can occur when the shear lip and vessel ID are in LMC condition. Given that these are large features, their tolerance will necessarily be large.

Set:

$t_{bh} := 1.5mm$ $\qquad$ $t_{bh} < t_{bh\_max} = 1$

$R_{i\_pv} = 680\,mm$ $\qquad$ $\Delta R_{i\_pv} := 0.25mm$ $\quad$ (+/-)

$R_{sl} := 679mm$ $\qquad$ $\Delta R_{sl} := 0.25mm$ $\quad$ (+/-)

Nominal radial clearance between shear lip and vessel, both at MMC:

$r_{cl} := \left(R_{i\_pv} - \Delta R_{i\_pv}\right) - \left(R_{sl} + \Delta R_{sl}\right)$ $\qquad$ $r_{cl} = 0.5\,mm$

Check: $r_{cl} > 2\delta_{fl} = 1$ $\quad$ Head will assemble to flange with full detector mass loading (safety factor>2)

The radial clearance between shear lip and vessel ID (both at MMC) is represents an additional tolerance that we can add to the accuracy tolerance, because we can use it to shift the patterns to match. Since tolerances are specified on a diameter basis, we add 2x the radial offset (minus 2x the deflection):

$t_{bh\_acc\_max} := t_{bh} + 2\left(r_{cl} - \delta_{fl}\right)$ $\qquad$ $t_{bh\_acc\_max} = 2.3\,mm$

Using this value might require a very high alignment precision to find the proper bolt alignment so we use a slightly smaller value

$t_{bh\_acc} := 2mm$

Will head and bolts "auto-assemble" (assemble without further translational alignment) for shear lip and vessel ID at MMC?

Check: $t_{bh\_acc} \le t_{bh\_max} = 0$

If false, additional shift of head relative to vessel may be necessary, even though shear lip assembles to vessel ID. If true, we can proceed to check for the case of shear lip and vessel ID at LMC, below:

Maximum offset of shear lip and vessel ID axes (both at LMC)

$\Delta r_{cl} := \left(R_{i\_pv} + \Delta R_{i\_pv}\right) - \left(R_{sl} - \Delta R_{sl}\right)$ $\qquad$ $\Delta r_{cl} = 1.5\,mm$

Maximum offset of bolt holes for flanges at LMC, bolts and holes at MMC

$$\Delta r_{cl} + t_{bh} = 3\,mm$$

check if bolts will assemble with vessel ID and shear lip assembled at LMC (fully misaligned), without further translation alignment:

$$\Delta r_{cl} + t_{bh} \le t_{bh\_max} = 0$$

We conclude that we may need to further translate and rotate the head relative to the vessel in order to align the bolt holes, even though the shear lip assembles. Since no "autoassembly is possible, we can loosen the accuracy requirement conditionally by specifying the true position tolerance for circular datum C or D at MMC condition; this allows the final bolt hole accuracy tolerance to increase by the amount datum C or D are from MMC.

The head must be retracted for this operation as the actual clearance between the shear lip and the vessel ID will not be known. Furthermore, the adjustment of the struts is not performed simultaneously, and large intermediate translations or rotations of the headmay take place prior to achieving the final small alignment. This would cause an interference of the shear lip with the ID, with possible damage. Internal components may require further retraction of head prior to alignment. LBNL Engineering Note 10182B, D. Shuman, provides a general method and MathCAD worksheet for determining the needed strut adjustments to align a component. The 6 strut head alignment and assembly fixture designed for the heads has Cartesian motions which are largely uncoupled and should be simple enough to adjust intuitively without needing this methodology.

Equivalent maximum radial, angular misalignments of bolt holes (given here as +/- values) for all parts at MMC. These describe square tolerance zones inscribed within the circular tolerance zones

With respect to each other (precisional tolerance)

$$\Delta r_{bh} := \frac{.71}{2} t_{bh} \qquad \Delta r_{bh} = 0.532\,mm \quad (+/-)$$

$$\Delta\theta_{bh} := \frac{.71}{2} \frac{t_{bh}}{R_{i\_pv}} \qquad \Delta\theta_{bh} = 0.045\,deg \quad (+/-)$$

With respect to datums A/B,C/D,E/F   (accuracy tolerance)

$$\Delta r_{bh\_acc} := \frac{.71}{2} t_{bh\_acc} \qquad \Delta r_{bh\_acc} = 0.71\,mm \quad (+/-)$$

$$\Delta\theta_{bh\_acc} := \frac{.71}{2} \frac{t_{bh\_acc}}{R_{i\_pv}} \qquad \Delta\theta_{bh\_acc} = 0.06\,deg \quad (+/-)$$

Head Nozzle and Flange Calculation, DN100 (CF) size Nozzle

Internal radius, nozzle

$R_n := 44\text{mm}$

Nozzle wall thickness

Required nozzle wall thickness, for internal pressure is:

$E_w := 1$

$$t_{rn} := \frac{MAWP \cdot (R_n)}{S_{max\_304L\_div1} \cdot E_w - 0.6 \cdot MAWP} \qquad t_{rn} = 0.601\,\text{mm}$$

We set wall thickness to be:

$t_n := 7\text{mm}$ $\qquad$ $t_n > t_{rn} = 1$

Flange thickness:

Note: we design, if possible for standard CF bolt pattern so as to allow possibility of using CF flanges prebolted to adapter plates, on extra long screws. This allows CF flange/adapter plate to be preassembled and tested for both pressure and leak tightness prior to installing as an assembly onto pressure vessel flange. This will require utmost care to tighten nuts without loosening the CF joint. It is recommended that a torque wrench be used on the bolt head to maintain full tightness while tightening nut on opposite side.

The flange design for helicoflex or O-ring sealing is "flat-faced", with "metal to metal contact outside the bolt circle". This design avoids the high flange bending stresses found in a raised face flange (of Appendix 2) and will result in less flange thickness, even though the rules for this design are found only in sec VIII division 1 under Appendix Y, and must be used with the lower allowable stresses of division 1.

Flanges and shells will be fabricated from 316Ti (UNSS31635, EN 1.4571, ASME spec SA-240) stainless steel plate. The flange bolts and nuts will be either 316 SS, or Inconel 718, (UNS N77180), if required, as this is the highest strength non-corrosive material allowed for bolting.

We will design to use one Helicoflex 3mm thick gasket with aluminum facing (softest).

Maximum allowable material stresses, for sec VIII, division 1 rules from ASME 2010 Pressure Vessel code, sec. II part D, table 2B:

Maximum allowable design stress for flange

$S_f := S_{max\_316Ti}$ $\qquad$ $S_f = 137.9\,\text{MPa}$

Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

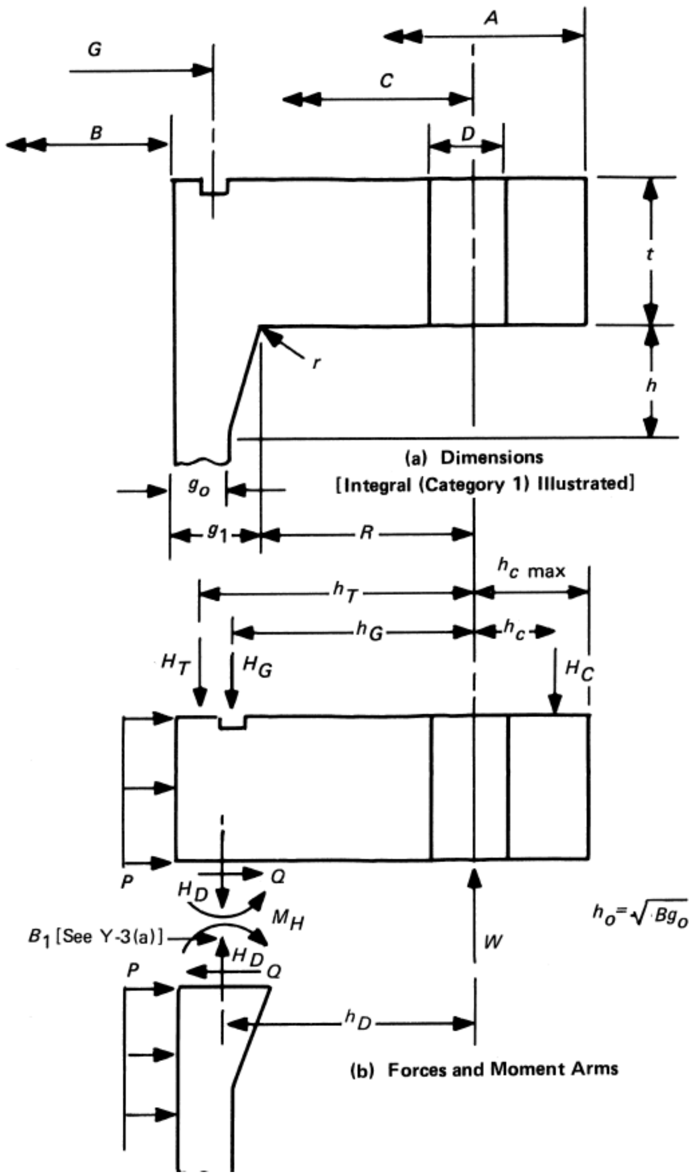
Inconel 718 (UNS N07718) $\qquad$ 316 condition/temper 2 (SA-193, SA-320)

$S_{max\_N07718} := 37000\text{psi}$ $\qquad$ $S_{max\_316\_bolt} := 22000\text{psi}$

$S_b := S_{max\_N07718}$ $\qquad$ $S_b = 255.1\,\text{MPa}$

From sec. VIII div 1, non-mandatory appendix Y for bolted joints having metal-to-metal contact outside of bolt circle. First define, per Y-3:

FIG. Y-3.2 FLANGE DIMENSIONS AND FORCES



hub thickness at flange (no hub)

$g_0 := t_n \quad g_1 := t_n \quad g_0 = 7\,\text{mm} \quad g_1 = 7\,\text{mm}$

corner radius:

$r_1 := \min(.5g_1, 5\text{mm}) \quad r_1 = 3.5\,\text{mm}$

Flange OD

$A := 16.5\text{cm}$

Flange ID

$B := 2R_n \qquad B = 8.8\,\text{cm}$

define:

$B_1 := B + 2g_1 \quad B_1 = 10.2\,\text{cm}$

Bolt circle (B.C.) dia, C:

$C := 13.0\text{cm}$

Gasket dia

$G := 2(R_n + .5\text{cm}) \qquad G = 0.098\,\text{m}$

Force of Pressure on head

$H := .785G^2 \cdot MAWP \qquad H = 1.177 \times 10^4 N$

Sealing force, per unit length of circumference:

for O-ring, 0.275" dia., shore A 70 F= ~5 lbs/in for 20% compression, (Parker o-ring handbook); add 50% for smaller second O-ring. (Helicoflex gasket requires high compression, may damage soft Ti surfaces, may move under pressure unless tightly backed, not recommended)

Helicoflex has equiv. values of Y for the ASME force term F and gives several possible values for 3mm HN200 with aluminum jacket:

$Y_1 := 25\frac{N}{mm}$ min value for our pressure and required leak rate (He)

$Y_2 := 185\frac{N}{mm}$ recommended value for large diameter seals, regardless of pressure or leak rate

for gasket diameter $D_j := G \qquad D_j = 0.098\,m$

Force is then either of:

$F_m := \pi D_j \cdot Y_1$

$F_m = 7.697 \times 10^3 N$

or

$F_j := \pi \cdot D_j \cdot Y_2$

$F_j = 5.696 \times 10^4 N$

Helicoflex recommends using Y2 for large diameter seals, even though for small diameter one can use the greater of Y1 or Ym=(Y2*(P/Pu)). For 15 bar Y1 is greater than Ym but far smaller than Y2. Sealing is less assured, but will be used in elastic range and so may be reusable. Flange thickness and bolt load increase quite substantially when using Y2 as design basis, which is a large penalty. We plan to recover any Xe leakage, as we have a second O-ring outside the first and a sniff port in between, so we thus design for Y1 (use F_m) and "cross our fingers" : if it doesn't seal we use an O-ring instead and recover permeated Xe with a cold trap. Note: in the cold trap one will get water and N2, O2, that permeates in through the outer O-ring as well.

Number of bolts, root dia., pitch, bolt hole dia D, (these are from DN75 CF standard dimensions (VACOM catalog)

$n := 16 \qquad d_b := 8mm \qquad p_t := 1mm \qquad h_3 := .614p_t$

root dia.

$d_3 := d_b - 2h_3 \qquad d_3 = 6.772 \times 10^{-3} m$

$A_b := n \cdot \frac{\pi}{4} \cdot d_3^{\,2} \qquad A_b = 5.763\,cm^2$

Check bolt to bolt clearance, for box wrench b2b spacing is ~ 1.2 in for 1/2in bolt twice bolt dia ($2.4xd_b$):

$\pi C - 2.4n \cdot d_b \geq 0 = 1 \qquad \pi\frac{C}{n \cdot d_b} = 3.191$

Check nut, washer clearance: $OD_w := 2d_b$ this covers the nut width across corners

$0.5C - (0.5B + g_1 + r_1) \geq 0.5OD_w = 1$

Flange hole diameter, minimum for clearance :

$D_{tmin} := d_b + 0.5mm \qquad D_{tmin} = 8.5\,mm$

Set:

$D_t := 9mm \qquad D_t \geq D_{tmin} = 1$

Compute Forces on flange:

$$H_G := \begin{pmatrix} F_m \\ F_j \end{pmatrix} \qquad H_G = \begin{pmatrix} 7.697\times 10^3 \\ 5.696\times 10^4 \end{pmatrix} N$$

from Table 2-6 Appendix 2, Integral flanges

$$h_G := 0.5(C - G) \qquad h_G = 1.6\,cm$$

$$H_D := .785 \cdot B^2 \cdot MAWP \qquad H_D = 9.488\times 10^3\,N$$

Here we add in the force on the nozzle from externally applied moment of:

$M_n := 2310N\cdot m$ from shielding, venting, etc as calculated elsewhere)

This force acts through the nozzle and can be thought of as an an additional force to be added with H_D. To calculate, we compare longitudinal stresses in nozzle from pressure to that from moment:

ID OD Moment of Inertia, nozzle

$$d_{ni} := 2R_n \qquad d_n := 2(R_n + t_n) \qquad I_n := \frac{\pi}{64}\left(d_n^{\,4} - d_{ni}^{\,4}\right) \qquad I_n = 236.963\,cm^4$$

max. bending stress:

$$\sigma_M := \frac{M_n \cdot (R_n + t_n)}{I_n} \qquad \sigma_M = 50\,MPa$$

this is somewhat high, we compare with allowable longitudinal stress from pressure with minimum thickness nozzle (as calculated above)

$$\sigma_{D\_min} := \frac{H_D}{2\pi(R_n + 0.5t_n)\cdot t_{rn}} \qquad \sigma_{D\_min} = 53\,MPa \qquad \sigma_M < \sigma_{D\_min} = 1$$

OK

max long stress from pressure:

$$\sigma_D := \frac{H_D}{2\pi(R_n + 0.5t_n)\cdot t_n} \qquad \sigma_D = 4.54\,MPa$$

then let the equivalent force be:

$$F_n := \frac{\sigma_M}{\sigma_D}\cdot H_D \qquad F_n = 1.039\times 10^5\,N$$

$$R := 0.5(C - B) - g_1 \qquad R = 1.4\,cm$$

radial distance, B.C. to hub-flange intersection, int fl.

$$h_D := R + 0.5g_1 \qquad h_D = 1.75\,cm$$

from Table 2-6 Appendix 2, Int. fl.

$$H_T := H - H_D \qquad H_T = 2.279\times 10^3\,N$$

$$h_T := 0.5(R + g_1 + h_G) \qquad h_T = 18.5\,mm$$

from Table 2-6 Appendix 2, int. fl.

Total Moment on Flange (maximum value)

$$M_P := (H_D + F_n)\cdot h_D + H_T\cdot h_T + H_G\cdot h_G \qquad M_P = \begin{pmatrix} 2.1\times 10^3 \\ 2.9\times 10^3 \end{pmatrix} N\cdot m$$

Appendix Y Calc

$$P := MAWP \qquad P = 15.4\,bar$$

Choose values for plate thickness and bolt hole dia:

$$t := 2.1cm \qquad D := D_t \qquad D = 0.9\,cm$$

Going back to main analysis, compute the following quantities:

$$\beta := \frac{C + B_1}{2B_1} \qquad \beta = 1.137 \quad h_C := 0.5(A - C) \qquad h_C = 0.018\,m$$

$$a := \frac{A + C}{2B_1} \qquad a = 1.446 \qquad AR := \frac{n \cdot D}{\pi \cdot C} \qquad AR = 0.353 \qquad h_0 := \sqrt{B \cdot g_0}$$

$$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}} \operatorname{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right) \qquad r_B = 0.018 \qquad h_0 = 0.025\,m$$

We need factors F and V, most easily found in figs 2-7.2 and 7.3 (Appendix 2)

since $\frac{g_1}{g_0} = 1$ these values converge to $F := 0.90892 \; V := 0.550103$

Y-5 Classification and Categorization

We have identical (class 1 assembly) integral (category 1) flanges, so from table Y-6.1, our applicable equations are (5a), (7)-(13),(14a),(15a),16a)

$$J_S := \frac{1}{B_1}\left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B \qquad J_S = 0.475 \qquad J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi \cdot r_B \qquad J_P = 0.325$$

(5a) $$F' := \frac{g_0^{\,2}(h_0 + F \cdot t)}{V} \qquad F' = 3.911 \times 10^{-6}\,m^3 \qquad M_P = \begin{pmatrix} 2.149 \times 10^3 \\ 2.937 \times 10^3 \end{pmatrix} N \cdot m$$

$A = 16.5\,cm \qquad B = 8.8\,cm$

$$K := \frac{A}{B} \qquad K = 1.875 \quad Z := \frac{K^2 + 1}{K^2 - 1} \qquad Z = 1.795$$

$f := 1$ hub stress correction factor for integral flanges, use f =1 for g1/g0=1 (fig 2-7.6)hu

$t_s := 0mm$ no spacer

$l := 2t + t_s + 0.5d_b \qquad l = 4.6\,cm$ strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

Elastic constants http://www.hightempmetals.com/techdata/hitempInconel718data.php

$E := E_{SS\_aus} \qquad E = 193\,GPa \qquad E_{Inconel\_718} := 208GPa \qquad E_{bolt} := E_{Inconel\_718}$

Flange Moment due to Flange-hub interaction

$$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'} \qquad M_S = \begin{pmatrix} -245.3 \\ -335.3 \end{pmatrix} J \qquad (7)$$

Slope of Flange at I.D.

$$\theta_B := \frac{5.46}{E \cdot \pi t^3}(J_S \cdot M_S + J_P \cdot M_P) \qquad \theta_B = \begin{pmatrix} 5.649 \times 10^{-4} \\ 7.72 \times 10^{-4} \end{pmatrix} \qquad E \cdot \theta_B = \begin{pmatrix} 109.018 \\ 149.001 \end{pmatrix} MPa \quad (7)$$

Contact Force between flanges, at h_C:

$$H_C := \frac{M_P + M_S}{h_C} \qquad H_C = \begin{pmatrix} 1.088 \times 10^5 \\ 1.487 \times 10^5 \end{pmatrix} N \qquad (8)$$

$$W_{m1} := H + H_G + H_C \qquad W_{m1} = \begin{pmatrix} 1.282\times 10^5 \\ 2.174\times 10^5 \end{pmatrix} N$$ (9)

Operating Bolt Stress

$$\sigma_b := \frac{W_{m1}}{A_b} \qquad \sigma_b = \begin{pmatrix} 222.5 \\ 377.2 \end{pmatrix} MPa \qquad S_b = 255.1\,MPa$$ (10)

$$r_E := \frac{E}{E_{bolt}} \qquad r_E = 0.928$$ elasticity factor

Design Prestress in bolts

$$S_i := \sigma_b - \frac{1.159\cdot h_C^{\,2}\cdot\left(M_P + M_S\right)}{a\cdot t^3\cdot l\cdot r_E\cdot B_1} \qquad S_i = \begin{pmatrix} 210.9 \\ 361.4 \end{pmatrix} MPa$$ (11)

Radial Flange stress at bolt circle

$$S_{R\_BC} := \frac{6\left(M_P + M_S\right)}{t^2\left(\pi\cdot C - n\cdot D\right)} \qquad S_{R\_BC} = \begin{pmatrix} 98 \\ 133.9 \end{pmatrix} MPa$$ (12)

Radial Flange stress at inside diameter

$$S_{R\_ID} := -\left(\frac{2F\cdot t}{h_0 + F\cdot t} + 6\right)\cdot\frac{M_S}{\pi B_1\cdot t^2} \qquad S_{R\_ID} = \begin{pmatrix} 11.925 \\ 16.299 \end{pmatrix} MPa$$ (13)

Tangential Flange stress at inside diameter

$$S_T := \frac{t\cdot E\cdot\theta_B}{B_1} + \left(\frac{2F\cdot t\cdot Z}{h_0 + F\cdot t} - 1.8\right)\cdot\frac{M_S}{\pi B_1\cdot t^2} \qquad S_T = \begin{pmatrix} 22.86 \\ 31.24 \end{pmatrix} MPa$$ (14a)

Longitudinal hub stress

$$S_H := \frac{h_0\cdot E\cdot\theta_B\cdot f}{0.91\left(\frac{g_1}{g_0}\right)^2 B_1\cdot V} \qquad S_H = \begin{pmatrix} 52.991 \\ 72.426 \end{pmatrix} MPa$$

Y-7 Flange stress allowables: $S_b = 255.106\,MPa \quad S_f = 137.9\,MPa$

(a) $\sigma_b < S_b = \begin{pmatrix} 1 \\ 0 \end{pmatrix}$

<-- we cannot use full Y2 load, even with Inconel 718 bolts (unless we assure that fast vent valve is a straight through design

(b)

(1) $S_H < 1.5S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$ S_n not applicable

(2) not applicable

(c) $S_{R\_BC} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

$S_{R\_ID} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(d) $S_T < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(e)

$$\frac{S_H + S_{R\_BC}}{2} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$$

$$\frac{S_H + S_{R\_ID}}{2} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$$

(f) not applicable

Bolt force

$F_{bolt} := \sigma_b \cdot .785 \cdot {d_b}^2$ $\quad F_{bolt} = \begin{pmatrix} 2.513 \times 10^3 \\ 4.261 \times 10^3 \end{pmatrix} lbf$

Bolt torque required

$T_{bolt\_min} := 0.2 F_{bolt} \cdot d_b$ $\quad T_{bolt\_min} = \begin{pmatrix} 17.9 \\ 30.3 \end{pmatrix} N \cdot m$ $\; T_{bolt\_min} = \begin{pmatrix} 13.2 \\ 22.4 \end{pmatrix} lbf \cdot ft$ for pressure test use 1.5x this value

Nozzle and Flange, DN75 (CF) pattern, for Vessel Nozzles

Internal radius, nozzle

$R_n := 30mm$

Nozzle wall thickness

Required nozzle wall thickness, for internal pressure is: $E_w := 1$

$$t_{rn} := \frac{MAWP \cdot (R_n)}{S_{max\_304L\_div1} \cdot E_w - 0.6 \cdot MAWP} \qquad t_{rn} = 0.41\,mm$$

We set wall thickness to be:

$t_n := 6mm$ $\qquad t_n > t_{rn} = 1$

Flange thickness:

Note: we design, if possible for standard CF bolt pattern so as to allow possibility of using CF flanges prebolted to adapter plates, on extra long screws. This allows CF flange/adapter plate to be preassembled and tested for both pressure and leak tightness prior to installing as an assembly onto pressure vessel flange. This will require utmost care to tighten nuts without loosening the CF joint. It is recommended that a torque wrench be used on the bolt heald to maintain full tightness while tightening nut on opposite side.

The flange design for Helicoflex or O-ring sealing is "flat-faced", with "metal to metal contact outside the bolt circle". This design avoids the high flange bending stresses found in a raised face flange (of Appendix 2) and will result in less flange thickness, even though the rules for this design are found only in sec VIII division 1 under Appendix Y, and must be used with the lower allowable stresses of division 1.

Flanges and shells will be fabricated from 316Ti (UNSS31635, EN 1.4571, ASME spec SA-240) stainless steel plate. The flange bolts and nuts will be either 316 SS, or Inconel 718, (UNS N77180) if required, as this is the highest strength non-corrosive material allowed for bolting.

We will attempt to design to use one Helicoflex 3mm thick gasketwith aluminum facing (softest) loaded to the minimum force required to achieve helium leak rate. Other is a double O-ring seal will be used, with a pumpout port between them

Maximum allowable material stresses, for sec VIII, division 1 rules from ASME 2010 Pressure Vessel code, sec. II part D, table 2B:

Maximum allowable design stress for flange

$S_f := S_{max\_316Ti} \qquad S_f = 137.9\,MPa$

Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

Inconel 718 (UNS N07718) $\qquad$ 316 condition/temper 2 (SA-193, SA-320)

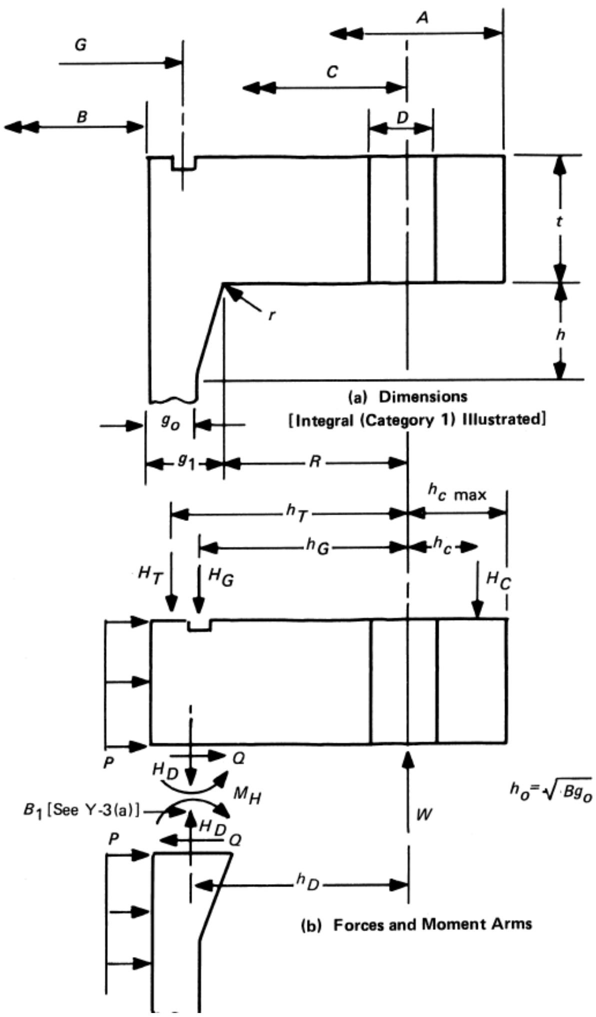
$S_{max\_N07718} := 37000psi \qquad S_{max\_316\_bolt} := 22000psi$

Bolt materials, we calculate in parallel for SS (temper 2) bolts using O-rings or very low force Helicoflex gaskets, and Inconel 718 bolts for high force Helicoflex gaskets

$$S_b := \begin{pmatrix} S_{max\_316\_bolt} \\ S_{max\_N07718} \end{pmatrix} \qquad S_b = \begin{pmatrix} 151.7 \\ 255.1 \end{pmatrix} MPa$$

From sec. VIII div 1, non-mandatory appendix Y for bolted joints having metal-to-metal contact outside of bolt circle. First define, per Y-3:

FIG. Y-3.2 FLANGE DIMENSIONS AND FORCES



(a) Dimensions
[Integral (Category 1) Illustrated]

(b) Forces and Moment Arms

hub thickness at flange (no hub)

$g_0 := t_n \quad g_1 := t_n \quad g_0 = 6\,\mathrm{mm} \quad g_1 = 6\,\mathrm{mm}$

corner radius:

$r_1 := \min(.5g_1, 5\mathrm{mm}) \quad r_1 = 3\,\mathrm{mm}$

Flange OD

$A := 13\mathrm{cm}$

Flange ID

$B := 2R_n \qquad B = 6\,\mathrm{cm}$

define:

$B_1 := B + 2g_1 \quad B_1 = 7.2\,\mathrm{cm}$

Bolt circle (B.C.) dia, C:

$C := 10.2\mathrm{cm}$

Gasket dia

$G := 2(R_n + .75\mathrm{cm}) \qquad G = 0.075\,\mathrm{m}$

Force of Pressure on head

$H := .785G^2 \cdot \mathrm{MAWP} \qquad H = 6.892 \times 10^3\,\mathrm{N}$

Sealing force, per unit length of circumference:

for O-ring, 0.275" dia., shore A 70 F= ~5 lbs/in for 20% compression, (Parker o-ring handbook); add 50% for smaller second O-ring. (Helicoflex gasket requires high compression, may damage soft Ti surfaces, may move under pressure unless tightly backed, not recommended)

Helicoflex has equiv. values of Y for the ASME force term F and gives several possible values for 3mm HN200 with aluminum jacket:

$Y_1 := 25\frac{N}{mm}$ min value for our pressure and required leak rate (He)

$Y_2 := 150\frac{N}{mm}$ recommended value is 185, we need to go a bit lower for inconel 718 bolts

for gasket diameter $D_j := G$ $D_j = 0.075\,m$

Force is then either of:

$F_m := \pi D_j \cdot Y_1$ $F_m = 5.89 \times 10^3\,N$

or

$F_j := \pi \cdot D_j \cdot Y_2$ $F_j = 3.534 \times 10^4\,N$

Helicoflex recommends using Y2 for large diameter seals, even though for small diameter one can use the greater of Y1 or Ym=(Y2*(P/Pu)). For 15 bar Y1 is greater than Ym but far smaller than Y2. Sealing is less assured, but will be used in elastic range and so may be reusable. Flange thickness and bolt load increase quite substantially when using Y2 as design basis, which is a large penalty. We plan to recover any Xe leakage, as we have a second O-ring outside the first and a sniff port in between, so we thus design for Y1 (use F_m) and "cross our fingers" : if it doesn't seal we use an O-ring instead and recover permeated Xe with a cold trap. Note: in the cold trap one will get water and N2, O2, that permeates in through the outer O-ring as well.

Number of bolts, root dia., pitch, bolt hole dia D, (these are from DN75 CF standard dimensions (VACOM catalog)

$n := 16$ $d_b := 8mm$ $p_t := 1mm$ $h_3 := .614p_t$

root dia.

$d_3 := d_b - 2h_3$ $d_3 = 6.772 \times 10^{-3}\,m$

$A_b := n \cdot \frac{\pi}{4} \cdot d_3^{\,2}$ $A_b = 5.763\,cm^2$

Check bolt to bolt clearance, for box wrench b2b spacing is 1.2 in for 1/2in bolt twice bolt dia (2.4xd_b):

$\pi C - 2.4 n \cdot d_b \geq 0 = 1$ $\pi \frac{C}{n \cdot d_b} = 2.503$

Check nut, washer clearance: $OD_w := 2d_b$ this covers the nut width across corners

$0.5C - (0.5B + g_1 + r_1) \geq 0.5OD_w = 1$

Flange hole diameter, minimum for clearance :

$D_{tmin} := d_b + 0.5mm$ $D_{tmin} = 8.5\,mm$

Set:

$D_t := 9mm$ $D_t \geq D_{tmin} = 1$

Compute Forces on flange:

$H_G := \begin{pmatrix} F_m \\ F_j \end{pmatrix}$ $H_G = \begin{pmatrix} 5.89 \times 10^3 \\ 3.534 \times 10^4 \end{pmatrix} N$ from Table 2-6 Appendix 2, Integral flanges

$h_G := 0.5(C - G)$ $h_G = 1.35\,cm$

$H_D := .785 \cdot B^2 \cdot MAWP$ $H_D = 4.411 \times 10^3\,N$

Here we add in an axial force on the nozzle from the moment of a feedthrough mounted at 45 deg

$W_{ft} := 20\text{lbf} \qquad l_{ft\_cg} := .5\text{m} \qquad M_{ft} := 0.71 W_{ft} \cdot l_{ft\_cg} \qquad M_{ft} = 31.6\,\text{N}\cdot\text{m}$

This force acts through the nozzle and can be thought of as an an additional force to be added with H_D. to calculate, we compare longitudinal stresses in nozzle from pressure to that from moment:

ID OD Moment of Inertia, nozzle

$d_{ni} := 2R_n \quad d_n := 2(R_n + t_n) \qquad I_n := \frac{\pi}{64}\left(d_n^{\,4} - d_{ni}^{\,4}\right) \qquad I_n = 68.299\,\text{cm}^4$

max. bending stress:

$$\sigma_M := \frac{M_{ft}\cdot(R_n + 0.5t_n)}{I_n} \qquad \sigma_M = 1.526\,\text{MPa}$$

max long stress from pressure:

$$\sigma_D := \frac{H_D}{2\pi(R_n + 0.5t_n)\cdot t_n} \qquad \sigma_D = 3.546\,\text{MPa}$$

then let the equivalent force be:

$$F_{ft} := \frac{\sigma_M}{\sigma_D}\cdot H_D \qquad F_{ft} = 1898\,\text{N}$$

$R := 0.5(C - B) - g_1 \qquad R = 1.5\,\text{cm}$ radial distance, B.C. to hub-flange intersection, int fl..

$h_D := R + 0.5g_1 \qquad h_D = 1.8\,\text{cm}$ from Table 2-6 Appendix 2, Int. fl.

$H_T := H - H_D \qquad H_T = 2.481\times 10^3\,\text{N}$

$h_T := 0.5(R + g_1 + h_G) \qquad h_T = 17.25\,\text{mm}$ from Table 2-6 Appendix 2, int. fl.

Total Moment on Flange (maximum value)

$$M_P := (H_D + F_{ft})\cdot h_D + H_T\cdot h_T + H_G\cdot h_G \qquad M_P = \begin{pmatrix} 235.9 \\ 633.5 \end{pmatrix}\text{N}\cdot\text{m}$$

Appendix Y Calc

$P := \text{MAWP} \qquad P = 15.4\,\text{bar}$

Choose values for plate thickness and bolt hole dia:

$t := 1.67\text{cm} \qquad D := D_t \qquad D = 0.9\,\text{cm}$

Going back to main analysis, compute the following quantities:

$$\beta := \frac{C + B_1}{2B_1} \qquad \beta = 1.208 \qquad h_C := 0.5(A - C) \qquad h_C = 0.014\,\text{m}$$

$$a := \frac{A + C}{2B_1} \qquad a = 1.611 \qquad AR := \frac{n\cdot D}{\pi\cdot C} \qquad AR = 0.449 \qquad h_0 := \sqrt{B\cdot g_0}$$

$$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}}\,\text{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right) \qquad r_B = 0.032$$

$h_0 = 0.019\,\text{m}$

We need factors F and V, most easily found in figs 2-7.2 and 7.3 (Appendix 2)

since $\frac{g_1}{g_0} = 1$ these values converge to $F := 0.90892 \quad V := 0.550103$

Y-5 Classification and Categorization

We have identical (class 1 assembly) integral (category 1) flanges, so from table Y-6.1, our applicable equations are (5a), (7)-(13),(14a),(15a),16a)

$$J_S := \frac{1}{B_1}\left(\frac{2\cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B \qquad J_S = 0.637 \qquad J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi\cdot r_B \qquad J_P = 0.43$$

(5a) $$F' := \frac{g_0^{\,2}(h_0 + F\cdot t)}{V} \qquad F' = 2.235\times 10^{-6}\,m^3 \qquad M_P = \begin{pmatrix}235.889\\633.497\end{pmatrix} N\cdot m$$

$$A = 13\,cm \qquad B = 6\,cm$$

$$K := \frac{A}{B} \qquad K = 2.167 \quad Z := \frac{K^2+1}{K^2-1} \qquad Z = 1.541$$

$f := 1$ hub stress correction factor for integral flanges, use f =1 for g1/g0=1 (fig 2-7.6)hu

$t_s := 0mm$ no spacer

$l := 2t + t_s + 0.5d_b \qquad l = 3.74\,cm$ strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

Elastic constants http://www.hightempmetals.com/techdata/hitempInconel718data.php

$$E := E_{SS\_aus} \qquad E = 193\,GPa \qquad E_{Inconel\_718} := 208GPa \qquad E_{bolt} := \begin{pmatrix}E_{SS\_aus}\\E_{Inconel\_718}\end{pmatrix}$$

Flange Moment due to Flange-hub interaction

$$M_S := \frac{-J_P\cdot F'\cdot M_P}{t^3 + J_S\cdot F'} \qquad M_S = \begin{pmatrix}-37.3\\-100.1\end{pmatrix} J \qquad (7)$$

Slope of Flange at I.D.

$$\theta_B := \frac{5.46}{E\cdot \pi t^3}(J_S\cdot M_S + J_P\cdot M_P) \qquad \theta_B = \begin{pmatrix}1.501\times 10^{-4}\\4.031\times 10^{-4}\end{pmatrix} \qquad E\cdot\theta_B = \begin{pmatrix}28.97\\77.8\end{pmatrix} MPa \qquad (7)$$

Contact Force between flanges, at h_C:

$$H_C := \frac{M_P + M_S}{h_C} \qquad H_C = \begin{pmatrix}1.419\times 10^4\\3.81\times 10^4\end{pmatrix} N \qquad (8)$$

Bolt Load at operating condition:

$$W_{m1} := H + H_G + H_C \qquad W_{m1} = \begin{pmatrix}2.697\times 10^4\\8.034\times 10^4\end{pmatrix} N \qquad (9)$$

Operating Bolt Stress

$$\sigma_b := \frac{W_{m1}}{A_b} \qquad \sigma_b = \begin{pmatrix}46.8\\139.4\end{pmatrix} MPa \qquad S_b = \begin{pmatrix}151.7\\255.1\end{pmatrix} MPa \qquad (10)$$

$$r_E := \frac{E}{E_{bolt}} \qquad r_E = \begin{pmatrix}1\\0.928\end{pmatrix}$$ elasticity factor

Design Prestress in bolts

$$S_i := \overrightarrow{\left[\sigma_b - \frac{1.159\cdot h_C^{\,2}\cdot(M_P + M_S)}{a\cdot t^3\, l\cdot r_E\cdot B_1}\right]} \qquad S_i = \begin{pmatrix}44.6\\132.9\end{pmatrix} MPa \qquad (11)$$

Radial Flange stress at bolt circle

$$S_{R\_BC} := \frac{6(M_P + M_S)}{t^2(\pi \cdot C - n \cdot D)} \qquad S_{R\_BC} = \begin{pmatrix} 24.2 \\ 65 \end{pmatrix} \text{MPa} \qquad (12)$$

Radial Flange stress at inside diameter

$$S_{R\_ID} := -\left(\frac{2F \cdot t}{h_0 + F \cdot t} + 6\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_{R\_ID} = \begin{pmatrix} 4.068 \\ 10.926 \end{pmatrix} \text{MPa} \qquad (13)$$

Tangential Flange stress at inside diameter

$$S_T := \frac{t \cdot E \cdot \theta_B}{B_1} + \left(\frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8\right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \qquad S_T = \begin{pmatrix} 6.97 \\ 18.73 \end{pmatrix} \text{MPa} \qquad (14a)$$

Longitudinal hub stress

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91 \left(\frac{g_1}{g_0}\right)^2 B_1 \cdot V} \qquad S_H = \begin{pmatrix} 15.25 \\ 40.955 \end{pmatrix} \text{MPa}$$

Y-7 Flange stress allowables: $S_b = \begin{pmatrix} 151.7 \\ 255.1 \end{pmatrix} \text{MPa}$ $\quad S_f = 137.9\,\text{MPa}$

(a) $\sigma_b < S_b = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(b)

(1) $S_H < 1.5 S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$ $\quad S_n$ not applicable

(2) not applicable

(c) $S_{R\_BC} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

$S_{R\_ID} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(d) $S_T < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(e) $\frac{S_H + S_{R\_BC}}{2} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

$\frac{S_H + S_{R\_ID}}{2} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(f) not applicable

Bolt force

$$F_{bolt} := \sigma_b \cdot .785 \cdot d_b^{\,2} \qquad F_{bolt} = \begin{pmatrix} 528.581 \\ 1.574 \times 10^3 \end{pmatrix} \text{lbf}$$

Bolt torque required

$$T_{bolt\_min} := 0.2 F_{bolt} \cdot d_b \qquad T_{bolt\_min} = \begin{pmatrix} 3.8 \\ 11.2 \end{pmatrix} \text{N} \cdot \text{m} \quad T_{bolt\_min} = \begin{pmatrix} 2.8 \\ 8.3 \end{pmatrix} \text{lbf} \cdot \text{ft}$$

for pressure test use 1.5x this value

Head Nozzle and Flange Calculation, DN40 (CF) size Nozzle

Internal radius, nozzle

$R_n := 15mm$

Nozzle wall thickness

Required nozzle wall thickness, for internal pressure is:

$E_w := 1$

$$t_{rn} := \frac{MAWP \cdot (R_n)}{S_{max\_304L\_div1} \cdot E_w - 0.6 \cdot MAWP} \qquad t_{rn} = 0.205\,mm$$

We set wall thickness to be:

$t_n := 3mm$ $\qquad t_n > t_{rn} = 1$

Flange thickness:

Note: we design , if possible for standard CF bolt pattern so as to allow possibility of using CF flanges prebolted to adapter plates, on extra long screws. this allows CF flange/adapter plate to be preassembled and tested for both pressure and leak tightness pror to installing as an assembly onto pressure vessel flange. This will require utmost care to tighten nuts without loosening the CF joint. It is recommended that a torque wrench be used on the bolt heads to maintain full tightness on CF gasket while tightening nut on opposite side.

The flange design for helicoflex or O-ring sealing is "flat-faced", with "metal to metal contact outside the bolt circle". This design avoids the high flange bending stresses found in a raised face flange (of Appendix 2) and will result in less flange thickness, even though the rules for this design are found only in sec VIII division 1 under Appendix Y, and must be used with the lower allowable stresses of division 1.

Flanges and shells will be fabricated from 304L or 316L (ASME spec SA-240) stainless steel plate. Plate samples will be helium leak checked before fabrication, as well as ultrasound inspected. The flange bolts and nuts will be inconel 718, (UNS N77180) as this is the highest strength non-corrosive material allowed for bolting.

We will design to use one Helicoflex 2mm gasket (smallest size possible) with aluminum facing (softest) loaded to the minimum force required to achieve helium leak rate.

Maximum allowable material stresses, for sec VIII, division 1 rules from ASME 2010 Pressure Vessel code, sec. II part D, table 2B:

Maximum allowable design stress for flange

$S_f := S_{max\_316Ti}$ $\qquad S_f = 137.9\,MPa$

Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

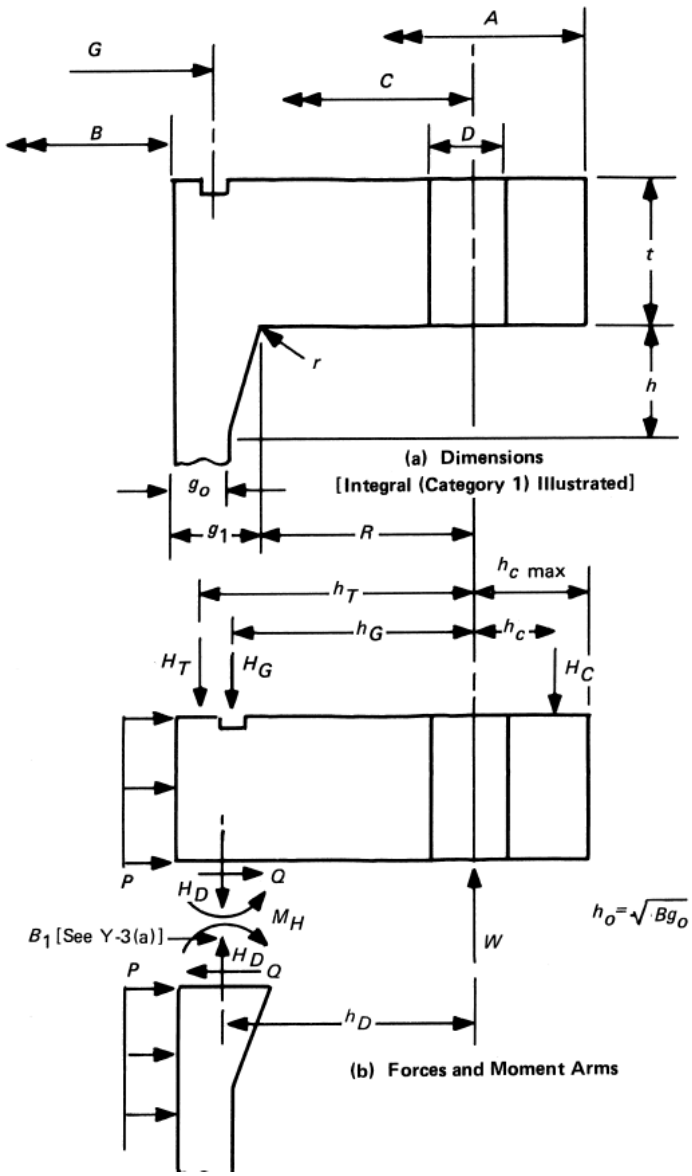
Inconel 718 (UNS N07718)	316 condition/temper 2 (SA-193, SA-320)	temper 1
$S_{max\_N07718} := 37000psi$	$S_{max\_316\_bolt2} := 22000psi$	$S_{max\_316\_bolt1} := 18800psi$

Bolt, flange materials (we use MathCAD's parallel calculation capability, with vectors)

$$S_b := \begin{pmatrix} S_{max\_316\_bolt1} \\ S_{max\_N07718} \end{pmatrix} \qquad S_b = \begin{pmatrix} 129.6 \\ 255.1 \end{pmatrix} MPa$$

From sec. VIII div 1, non-mandatory appendix Y for bolted joints having metal-to-metal contact outside of bolt circle. First define, per Y-3:

FIG. Y-3.2 FLANGE DIMENSIONS AND FORCES



hub thickness at flange (no hub) corner radius:

$g_0 := t_n \quad g_1 := t_n \quad g_0 = 3\,mm \quad g_1 = 3\,mm \quad r_1 := \min(.5g_1, 5mm) \quad r_1 = 1.5\,mm$

Flange OD

$A := 8cm$

Flange ID

$B := 2R_n \quad B = 3\,cm$

define:

$B_1 := B + 2g_1 \quad B_1 = 3.6\,cm$

Bolt circle (B.C.) dia, C:

$C := 5.9cm$ VACOM

Gasket dia

$G := 2(R_n + .5cm) \quad G = 4\,cm$

Force of Pressure on head

$H := .785G^2 \cdot MAWP \qquad H = 1.96 \times 10^3 \, N$

Sealing force, per unit length of circumference:

for O-ring, 0.275" dia., shore A 70 F= ~5 lbs/in for 20% compression, (Parker o-ring handbook); add 50% for smaller second O-ring. (Helicoflex gasket requires high compression, may damage soft Ti surfaces, may move under pressure unless tightly backed, not recommended)

Helicoflex has equiv. values of Y for the ASME force term F and gives several possible values for 3mm HN200 with aluminum jacket:

$Y_1 := 8 \frac{lbf}{in}$ min value for our pressure and required leak rate (He) $\qquad Y_2 := 175 \frac{N}{mm}$ recommended value for large diameter seals, regardless of pressure or leak rate

for gasket diameter $\quad D_j := G \qquad D_j = 0.04 \, m$

Force is then either of:

$F_m := \pi D_j \cdot Y_1$ or $F_j := \pi \cdot D_j \cdot Y_2$

$F_m = 176.057 \, N \qquad F_j = 2.199 \times 10^4 \, N$

Helicoflex recommends using Y2 for large diameter seals, even though for small diameter one can use the greater of Y1 or Ym=(Y2*(P/Pu)). For 15 bar Y1 is greater than Ym but far smaller than Y2. Sealing is less assured, but will be used in elastic range and so may be reusable. Flange thickness and bolt load increase quite substantially when using Y2 as design basis, which is a large penalty. We plan to recover any Xe leakage, as we have a second O-ring outside the first and a sniff port in between, so we thus design for Y1 (use F_m) and "cross our fingers" : if it doesn't seal we use an O-ring instead and recover permeated Xe with a cold trap. Note: in the cold trap one will get water and N2, O2, that permeates in through the outer O-ring as well.

Start by making trial assumption for number of bolts, root dia., pitch, bolt hole dia D,

$n := 12 \qquad d_b := 6mm \qquad p_t := 0.8mm \qquad h_3 := .614p_t$

root dia.

$d_3 := d_b - 2h_3 \qquad d_3 = 5.018 \times 10^{-3} \, m$

$A_b := n \cdot \frac{\pi}{4} \cdot {d_3}^2 \qquad A_b = 2.373 \, cm^2$

Check bolt to bolt clearance, for box wrench b2b spacing is 1.2 in for 1/2in bolt twice bolt dia (2.4xd_b):

$\pi C - 2.0n \cdot d_b \ge 0 = 1 \qquad \pi \frac{C}{n \cdot d_b} = 2.574$

Check nut, washer clearance: $\quad OD_w := 2d_b \quad$ this covers the nut width across corners

$0.5C - \left(0.5B + g_1 + r_1\right) \ge 0.5OD_w = 1$

Flange hole diameter, minimum for clearance :

$D_{tmin} := d_b + 0.5mm \qquad D_{tmin} = 6.5 \, mm$

Set:

$D_t := 6.5mm \qquad D_t \ge D_{tmin} = 1$

Compute Forces on flange:

$H_G := \begin{pmatrix} F_m \\ F_j \end{pmatrix}$ $\quad H_G = \begin{pmatrix} 176.057 \\ 2.199 \times 10^4 \end{pmatrix} N$ from Table 2-6 Appendix 2, Integral flanges

$h_G := 0.5(C - G) \quad h_G = 0.95\,cm$

$H_D := .785 \cdot B^2 \cdot MAWP \quad H_D = 1.103 \times 10^3\,N$

Unlike the other nozzles we will have any external forces or moments applied to these nozzles, while under pressure

$R := 0.5(C - B) - g_1 \quad R = 1.15\,cm$ radial distance, B.C. to hub-flange intersection, int fl.

$h_D := R + 0.5g_1 \quad h_D = 1.3\,cm$ from Table 2-6 Appendix 2, Int. fl.

$H_T := H - H_D \quad H_T = 857.679\,N$

$h_T := 0.5(R + g_1 + h_G) \quad h_T = 12\,mm$ from Table 2-6 Appendix 2, int. fl.

Total Moment on Flange (maximum value)

$M_P := (H_D) \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \quad M_P = \begin{pmatrix} 26.3 \\ 233.5 \end{pmatrix} N \cdot m$

Appendix Y Calc

$P := MAWP \quad P = 15.4\,bar$

Choose values for plate thickness and bolt hole dia:

$t := 1.5cm \quad D := D_t \quad D = 0.65\,cm$

Going back to main analysis, compute the following quantities:

$\beta := \frac{C + B_1}{2B_1} \quad \beta = 1.319 \quad h_C := 0.5(A - C) \quad h_C = 0.011\,m$

$a := \frac{A + C}{2B_1} \quad a = 1.931 \quad AR := \frac{n \cdot D}{\pi \cdot C} \quad AR = 0.421 \quad h_0 := \sqrt{B \cdot g_0}$

$r_B := \frac{1}{n}\left(\frac{4}{\sqrt{1 - AR^2}} \operatorname{atan}\left(\sqrt{\frac{1 + AR}{1 - AR}}\right) - \pi - 2AR\right) \quad r_B = 0.036 \quad h_0 = 9.487 \times 10^{-3}\,m$

We need factors F and V, most easily found in figs 2-7.2 and 7.3 (Appendix 2)

since $\frac{g_1}{g_0} = 1$ these values converge to $F := 0.90892 \; V := 0.550103$

Y-5 Classification and Categorization

We have identical (class 1 assembly) integral (category 1) flanges, so from table Y-6.1, our applicable equations are (5a), (7)-(13),(14a),(15a),16a)

$J_S := \frac{1}{B_1}\left(\frac{2 \cdot h_D}{\beta} + \frac{h_C}{a}\right) + \pi r_B \quad J_S = 0.813 \quad J_P := \frac{1}{B_1}\left(\frac{h_D}{\beta} + \frac{h_C}{a}\right) + \pi \cdot r_B \quad J_P = 0.539$

(5a) $F' := \frac{g_0^2(h_0 + F \cdot t)}{V} \quad F' = 3.783 \times 10^{-7}\,m^3 \quad M_P = \begin{pmatrix} 26.3 \\ 233.544 \end{pmatrix} N \cdot m$

$A = 8\,cm \quad B = 3\,cm$

$K := \frac{A}{B} \quad K = 2.667 \quad Z := \frac{K^2 + 1}{K^2 - 1} \quad Z = 1.327$

$f := 1$ hub stress correction factor for integral flanges, use f =1 for g1/g0=1 (fig 2-7.6)hu

$t_s := 0mm$ no spacer

$l := 2t + t_s + 0.5d_b \quad l = 3.3\,cm$ strain length of bolt (for class 1 assembly)

Y-6.1, Class 1 Assembly Analysis

Elastic constants http://www.hightempmetals.com/techdata/hitempInconel718data.php

$E := E_{SS\_aus} \quad E = 193\,GPa \quad E_{Inconel\_718} := 208GPa$

$$E_{bolt} := \begin{pmatrix} E_{SS\_aus} \\ E_{Inconel\_718} \end{pmatrix}$$

Flange Moment due to Flange-hub interaction

$$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'} \qquad M_S = \begin{pmatrix} -1.5 \\ -12.9 \end{pmatrix} J \qquad (7)$$

Slope of Flange at I.D.

$$\theta_B := \frac{5.46}{E \cdot \pi t^3}\left(J_S \cdot M_S + J_P \cdot M_P\right) \qquad \theta_B = \begin{pmatrix} 3.468 \times 10^{-5} \\ 3.08 \times 10^{-4} \end{pmatrix} \qquad E \cdot \theta_B = \begin{pmatrix} 6.694 \\ 59.444 \end{pmatrix} MPa \qquad (7)$$

Contact Force between flanges, at h_C:

$$H_C := \frac{M_P + M_S}{h_C} \qquad H_C = \begin{pmatrix} 2.366 \times 10^3 \\ 2.101 \times 10^4 \end{pmatrix} N \qquad (8)$$

Bolt Load at operating condition:

$$W_{m1} := H + H_G + H_C \qquad W_{m1} = \begin{pmatrix} 4.502 \times 10^3 \\ 4.496 \times 10^4 \end{pmatrix} N \qquad (9)$$

Operating Bolt Stress

$$\sigma_b := \frac{W_{m1}}{A_b} \qquad \sigma_b = \begin{pmatrix} 19 \\ 189.5 \end{pmatrix} MPa \qquad S_b = \begin{pmatrix} 129.6 \\ 255.1 \end{pmatrix} MPa \qquad (10)$$

$$r_E := \frac{E}{E_{bolt}} \qquad r_E = \begin{pmatrix} 1 \\ 0.928 \end{pmatrix}$$ elasticity factor

Design Prestress in bolts

$$S_i := \overrightarrow{\left[\sigma_b - \frac{1.159 \cdot h_C{}^2 \cdot \left(M_P + M_S\right)}{a \cdot t^3 \, l \cdot r_E \cdot B_1}\right]} \qquad S_i = \begin{pmatrix} 18.6 \\ 185.6 \end{pmatrix} MPa \qquad (11)$$

Radial Flange stress at bolt circle

$$S_{R\_BC} := \frac{6\left(M_P + M_S\right)}{t^2\left(\pi \cdot C - n \cdot D\right)} \qquad S_{R\_BC} = \begin{pmatrix} 6.2 \\ 54.8 \end{pmatrix} MPa \qquad (12)$$

Radial Flange stress at inside diameter

$$S_{R\_ID} := -\left(\frac{2F\cdot t}{h_0 + F\cdot t} + 6\right)\cdot\frac{M_S}{\pi B_1\cdot t^2} \qquad S_{R\_ID} = \begin{pmatrix} 0.411 \\ 3.65 \end{pmatrix} \mathrm{MPa} \qquad (13)$$

Tangential Flange stress at inside diameter

$$S_T := \frac{t\cdot E\cdot \theta_B}{B_1} + \left(\frac{2F\cdot t\cdot Z}{h_0 + F\cdot t} - 1.8\right)\cdot\frac{M_S}{\pi B_1\cdot t^2} \qquad S_T = \begin{pmatrix} 2.8 \\ 24.89 \end{pmatrix} \mathrm{MPa} \qquad (14a)$$

Longitudinal hub stress

$$S_H := \frac{h_0\cdot E\cdot \theta_B\cdot f}{0.91\left(\frac{g_1}{g_0}\right)^2 B_1\cdot V} \qquad S_H = \begin{pmatrix} 3.524 \\ 31.292 \end{pmatrix} \mathrm{MPa}$$

Y-7 Flange stress allowables: $S_b = \begin{pmatrix} 129.6 \\ 255.1 \end{pmatrix} \mathrm{MPa} \qquad S_f = 137.9\,\mathrm{MPa}$

(a) $\sigma_b < S_b = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(b)

(1) $S_H < 1.5S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$ $\qquad S_n$ not applicable

(2) not applicable

(c) $S_{R\_BC} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

$S_{R\_ID} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(d) $S_T < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(e) $\frac{S_H + S_{R\_BC}}{2} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

$\frac{S_H + S_{R\_ID}}{2} < S_f = \begin{pmatrix} 1 \\ 1 \end{pmatrix}$

(f) not applicable

Bolt force

$$F_{bolt} := \sigma_b\cdot .785\cdot d_b^{\,2} \qquad F_{bolt} = \begin{pmatrix} 120.552 \\ 1.204\times 10^3 \end{pmatrix} \mathrm{lbf}$$

Bolt torque required

$$T_{bolt\_min} := 0.2F_{bolt}\cdot d_b \qquad T_{bolt\_min} = \begin{pmatrix} 0.6 \\ 6.4 \end{pmatrix} \mathrm{N\cdot m} \quad T_{bolt\_min} = \begin{pmatrix} 0.5 \\ 4.7 \end{pmatrix} \mathrm{lbf\cdot ft}$$

for pressure test use 1.5x this value