**REPORT
Of The
TERAWATT LASER PRESSURE VESSEL COMMITTEE**

SEPTEMBER 25, 2000

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1.0 INTRODUCTION

In 1995 the ATF project sent out an RFP for a CO₂ Laser System having a TeraWatt output. Eight foreign and US firms responded. The Proposal Evaluation Panel on the second round selected Optoel, a Russian firm based in St. Petersburg, on the basis of the technical criteria and cost. Prior to the award, BNL representatives including the principal scientist, cognizant engineer and a QA representative visited the Optoel facilities to assess the company's capability to do the job.

The contract required Optoel to provide a x-ray preionized high pressure amplifier that included: a high pressure cell, x-ray tube, internal optics and a HV pulse forming network for the main discharge and preionizer. The high-pressure cell consists of a stainless steel pressure vessel with various ports and windows that is filled with a gas mixture operating at 10 atmospheres. In accordance with BNL Standard ESH 1.4.1 "Pressurized Systems For Experimental Use", the pressure vessel design criteria is required to comply with the ASME Boiler and Pressure Vessel Code

In 1996 a Preliminary Design Review was held at BNL. The vendor was requested to furnish drawings so that we could confirm that the design met the above criteria.

The vendor furnished drawings did not have all dimensions necessary to completely analyze the cell. Never the less, we performed an analysis on as much of the vessel as we could with the available information. The calculations concluded that there were twelve areas of concern that had to be addressed to assure that the pressure vessel complied with the requirements of the ASME code (see Appendix 7.1). This information was forwarded to the vendor with the understanding that they would resolve these concerns as they continued with the vessel design and fabrication. The assembled amplifier pressure vessel was later hydro tested to 220 psi (15 Atm) as well as pneumatically to 181 psi (12.5 Atm) at the fabricator's Russian facility and was witnessed by a BNL engineer.

The unit was shipped to the United States and installed at the ATF. As part of the commissioning of the device the amplifier pressure vessel was disassembled several times at which time it became apparent that the vendor had not addressed 7 of the 12 issues previously identified. Closer examination of the vessel revealed some additional concerns including quality of workmanship. Although not required by the contract, the vendor furnished radiographs of a number of pressure vessel welds. A review of the Russian X-rays revealed radiographs of both poor and unreadable quality. However, a number of internal weld imperfections could be observed. All welds in question were excavated and then visually and dye penetrant inspected. These additional inspections confirmed that the weld techniques used to make some of these original welds were substandard.

The applicable BNL standard, ESH 1.4.1, addresses the problem of pressure vessel non-compliance by having a committee appointed by the Department Chairman review the design and provide engineering solutions to assure equivalent safety. On January 24, 2000 Dr. M. Hart, the NSLS Chairman, appointed this committee with this charge.

This report details the engineering investigations, deliberations, solutions and calculations which were developed by members of this committee to determine that with repairs, new components, appropriate NDE, and lowering the design pressure, the vessel can be considered safe to use.

2.0 AREAS OF CONCERN

At the committee's first meeting, on Jan. 26, 2000, the following seventeen concerns were established for which engineering solutions needed to be developed. These items arose from the 12 items from the ASME calculation as well as items, which were observed by NSLS engineers during installation and testing of the Laser system:

- 2.1 Longitudinal welds between cylindrical shell and flange (both halves)
- 2.2 Welds between endplates and flanges (both halves)
- 2.3 Welds between cylindrical shell and end plate (both halves)
- 2.4 Longitudinal cylindrical seam weld

- 2.5 Presence of full penetration weld between cylindrical shell and ports for electrical feedthrus (3 places)
- 2.6 Arc strikes on cylindrical shell
- 2.7 Arc strikes on other areas
- 2.8 Discoloration of material in weld heat effected zones
- 2.9 Pressure relief valving for vessel
- 2.10 Incomplete or inaccurate drawings
- 2.11 Pressure vessel end plate thickness
- 2.12 Pressure vessel flange thickness
- 2.13 Main fasteners and tapped holes
- 2.14 Stress analysis of parts previously not dimensioned
- 2.15 Plug Weld
- 2.16 X-ray weld coverage
- 2.17 Quality of radiography

In later meetings the following additional concerns were identified:

- 2.18 Stress in the preionizer vessel
- 2.19 Pressure vessel seal material and design.

3.0 FINDINGS FROM INVESTIGATIONS AND INSPECTIONS

In order to more fully characterize some of the areas of concern, a number of detailed investigations and inspections needed to be performed. These investigations and their results are presented in this section.

3.1 Visual and Liquid Penetrant Inspections (LPT) of welds

A flanged and bolted joint longitudinally divides the cylindrical pressure vessel into two unequally sized sections. Visual and liquid penetrant inspections were made of the following welds: longitudinal shell to flange, end shell to endplate and endplate to flange. These inspections were carried out on both sections of the pressure vessel. The preponderance of defects was found on the smaller vessel section. All were repaired using an approved procedure.

3.2 Corner weld excavations

As mentioned in 3.1 above, liquid penetrant testing was done on the majority of welds. In the interior corners where the cylindrical shell, the end plate and the main flange meet, indications of defects were found. This showed up on all four corners of the smaller portion of the pressure vessel and one corner of the large portion. Excavation of these sites revealed that the corner defects were physically connected to a lack of fusion area in the weld between the end plate and the flange. Despite this defect, the weld had been pressure tight. Four material samples from the weld were tested. Three showed constituents consistent with 316 stainless steel and one was consistent with 347 stainless steel. Prior to attempting a repair, it was decided to perform finite element stress analyses on the corner and end plate/flange welds to determine the structural effects of the defects.

3.3 Boroscope investigations of port/shell weld joint.

A boroscopic inspection was performed on the cylindrical welds in port/shell joints of the vessel. This investigation was deemed necessary in order to verify that otherwise visually inaccessible welds were in fact in existence at these locations. A boroscope was borrowed with a fiber optics wand small enough and maneuverable enough to fit in the annular space between the “electrostatic shield” and the vessel. J. Skaritka and C. Czajkowski performed this inspection. The investigation disclosed that acceptable welds at these locations did exist.

3.4 “Plug” welded flange investigation

On the smaller portion of the vessel, three penetrations exist to allow tubes from the x-ray preionizer to pass through to the outside of the pressure vessel. The two larger flanged penetrations are the subject of this discussion. A cross section of this area can be seen in Appendix 7.2 as Section G-G sheet 2 of sketch ATF-SK-1217.

Upon first inspection of these flanges, there appeared to be slugs of metal welded in place to fill voids between the flanged nozzle and the cusp of the vessel penetration. The name for these apparent “plugs” or plug weldment has persisted, and would have been a problem since plug welds in this apparent form would not be allowed under the ASME pressure vessel code. Upon further inspection it was determined that the observed structure was not a plug weld. A strip of metal approximately 25mm wide by approximately 6mm thick was wrapped around the outside of the tube penetration. Fillet welds were added to secure this reinforcing strip between the nozzle’s outside wall and the outside wall of the vessel. The fillet weld ran all around the metal strip. Since the strip was not continuous, the resultant welding appeared to be a plug weld, and the strip was used to make up the space between the nozzle and an oversized hole in the shell of the vessel. Up to now the reason for this repair had been unknown, but its utility may be deduced by studying the undocumented alterations to the original design.

In the original design, an O-ring was to be used to seal the x-ray preionizer feed-through to the vessel. This sealing occurred very near the face of the flange, and it is apparent that the depth of this seal was altered. The depth of this seal slot was increased to approximately 1 inch. The center-to-center port spacing on the x-ray preionizer may have also played a role. It appears that the main vessel flanges were rebored to accommodate the as-built center-to-center distance of the preionizer feed throughs. Deep scratches were also noted on the inside surface of the nozzle just outside the pressure boundary. All these effects have produced a significant reduction in the wall thickness from the original 6mm to less than 3mm in the as-built condition. The vessel builders apparently believed the remaining wall thickness had become too thin to withstand the vessel pressure and O-ring seal loading, and therefore decided to add material to reinforce the wall in way of the deeper seal counter bore.

3.5 Metallurgical/Hardness tests

There was concern that the discolored area in the heat affected zone of a number of welds could indicate that the metal had been overheated during welding, adversely affecting its strength properties. Using an Equotip portable hardness tester, C. Czajkowski investigated the variation in hardness both outside and within the discolored zone of these welds. A significant change in hardness would indicate that the other mechanical properties had also been changed. There was no significant change, indicating that the weld was satisfactory and further that the discoloration stemmed from the welder not cleaning oxidation from the area after welding.

3.6 Russian furnished weld wire and rod

Russian supplied weld wire was sent to BNL from the vessel manufacturer for use in the repairs. This wire was received in a spool (gas metal arc wire-GMAW/MIG), with no identifying marks e.g. identification of wire type, etc. The wire was sent to BNL’s Material Science Division (R. Sabatini) for Energy Dispersive X-ray Spectroscopy (EDS) and Wavelength Dispersive X-ray Spectroscopy (WDS). This evaluation (See appendix 7.3) confirmed that the wire material was high in chromium and nickel and was similar to US austenitic stainless steel materials. This still left open the issue of the mechanical properties (tensile strength and yield strength of the wire). In order to obtain a conservative estimate of these properties the ASME Boiler and Pressure Vessel Code, Section II-Materials, Part C, 1983 Edition was consulted, specifically, Specifications SFA-5.4 “Specification for Corrosion-Resisting Chromium and Chromium-Nickel Steel Covered Welding Electrodes” and SFA-5.9 “Specification for Corrosion-Resisting Chromium and Chromium-Nickel Steel Bare and Composite Metal Cored and Standard Arc Welding Electrodes and Welding Rods”. Review of these two documents indicated that the tensile strength of 300 series austenitic weld electrodes ranged between 70ksi and 100ksi, minimum. This being the range of all austenitic materials and taking a conservative approach to the vessel repair, the lowest value of 70ksi tensile strength was used in the stress calculations. Please note, that if the Russian weld wire had been

identified or if typical Russian standards were available for review the values for tensile strength would most probably be higher. The value of 70ksi corresponds to the tensile strength of a 316 low carbon welding electrode.

3.7 Russian X-rays of lap welds between shell and end plates

After the receipt of the vessel at BNL, an "as-built" review was performed on the vessel and vessel welds. This review included both a visual inspection and a review of Russian supplied radiographs. These Russian radiographs were done during the construction of the vessel and were made to Russian standards. The radiographs did not meet US standards for either film density or radiographic quality (penetrameter). C. Czajkowski and a member of Central Shops Inspection Department performed the review. The radiographic and visual inspections disclosed a number of discontinuities in the welds, some significant arc strikes (involving incipient melting of the base material), some weld contour discrepancies, and some discoloration of the base material (possibly indicative of overheating during the welding process). After deliberation, it was determined that useful data could still be extracted from the Russian radiographs, especially in the more highly stressed welds of the vessel. It should be noted that not all of the welds radiographed needed to be radiographed. Radiography is a volumetric examination method that allows the internal structure of a weldment to be examined. Some of the welds that were radiographed were of the "partial penetration" type weld and should not have been subjected to such a tests. The reason being that a partial penetration weld will (by definition) always have "lack of penetration" at the root pass of the weld. For a radiograph of full penetration welding this condition would result in a rejection of the weld. However, for a "partial penetration weld" application the weld would be perfectly acceptable. This problem is encountered when overly conservative nondestructive testing is done for various components.

3.8 Preionizer chamber pressurization

The preionizer chamber is a vacuum vessel located inside the vessel. Accordingly, this chamber is subjected to an external pressure one atmosphere in excess of that in the main vessel. The preionizer is a cylindrical vessel approximately 10" in diameter with a rectangular window running along one side. The window is composed of a titanium foil that is supported by a perforated plate. The plate perforations are in the form of a honeycomb and allow x-rays to pass through the structure while supporting the thin titanium foil. The foil assembly has passed a pneumatic pressure test with the main vessel pressure exceeding 181 psig. This resulted in a pressure on the foil window of approximately 196 psig. The pressure test produced some plastic deformation of the foil surface, but no leaks were observed. Three tubular O-ring seal glands locate and help support the inner preionizer vessel.

4.0 DETERMINATION OF CORRECTIVE ACTIONS

This section addresses the concerns delineated in Section 2.0. Corrective actions and results are given for all concerns.

4.1 Longitudinal Welds – Cylindrical Shell to flange (4 places)

Concern - The radiographs, liquid penetrant (LP) tests and visual inspection of the longitudinal welds has shown weld undercuts and other weld contour discrepancies.

Corrective Action – The weld defects are to be removed by grinding to sound metal, weld repaired, LP tested and inspected visually with no defects allowed. A finite element analysis is to be done of the joint using the actual weld dimensions to evaluate the stress level in this area.

Results – The welds were repaired with no defects observed. See Appendix 7.4 for the weld procedure. The finite element analysis showed a maximum stress of 10,037 psi at 10 Atm. in this area, which is within the allowable for the material. See Appendix 7.5 for details of these results.

4.2 Welds – End plates to flanges (4 places)

Concern - The radiographs, liquid penetrant (LP) tests and visual inspection of the end plate to flange welds has shown discontinuities, lack of full weld penetration and weld contour discrepancies.

Corrective Action – The weld contour defects are to be removed by grinding to sound metal. Defects at the corners where the end plate, shell and flanges meet are to be excavated to determine the extent of the defect. A finite element analysis of the joint as well as the corners is to be done using actual weld dimensions to evaluate the stresses with the weld not having full penetration. Contour defects and corner excavations are to be weld repaired, LP tested and inspected visually with no defects allowed.

Results – The welds were repaired with no defects observed. See Appendix 7.4 for the weld procedure. The finite element analysis showed that even with the lack of full penetration in the weld between the endplate and the flange as well as the corners, the maximum stresses in these areas were 13,000psi, well within the allowable for the material. See Appendix 7.5 for details of these results.

4.3 Welds – Cylindrical Shell To End Plate

Concern - The radiographs, liquid penetrant (LP) tests and visual inspection of the shell to end plate welds show lack of penetration and a few weld contour discrepancies. As detailed in Section 3.7, this weld type should not have been X-rayed, because its design will always result in radiographs that show a lack of fusion, which is actually part of the weld joint design.

Corrective Action – The contour weld defects are to be removed by grinding to sound metal, weld repaired, LP tested and inspected visually with no defects allowed. A finite element analysis is to be done of the joint using the actual weld dimensions to evaluate the stress level in this area.

Results – The welds were repaired with no defects observed. See Appendix 7.4 for the weld procedure. The finite element analysis showed a maximum stress of 15,000 psi at 10 Atm. in this area, which is within the allowable for the material. See Appendix 7.5 for details of these results.

4.4 Weld – Longitudinal weld on cylindrical shell

Concern – There is a seam weld in the cylindrical pressure vessel wall for which there are no radiographic records. Moreover, there is an approximately .03" radial mismatch between the welded surfaces.

Corrective Action – Perform a FEA on the area to evaluate the stresses and a X-ray inspection of the weld seam.

Results – The finite element analysis (See Appendix 7.5) showed a stress of 10037 psi due to the offset and 10 Atm internal pressure, which is well within the allowable for the material. The weld X-rays were satisfactory. See Appendix 7.7 for the X-ray inspection report.

4.5 Completeness of weld between Cylindrical Shell and feed thru ports

Concern – The ASME Code requires welds between the cylindrical pressure vessel wall and ports to be full penetration. Several ports have corona shields on the inside which obscure views of portions of the inside weld. This precludes confirming that the welds are full penetration.

Corrective Action – A boroscopic inspection was performed on the cylindrical welds in the port/shell joint of the vessel. A boroscope was used with a fiber optics wand small enough and maneuverable enough to fit in the annular space between the corona shield and the vessel.

Results –C. Czajkowski and J. Skaritka performed this inspection. The investigation disclosed the acceptability of welds at these locations.

4.6 Arc strikes on Cylindrical Shell and other areas

Concern – Visual inspection and LP testing of arc strikes on various surfaces of the pressure vessel confirmed that some were significant (involving incipient melting of the base material), These areas can contain cracks, which could propagate with time and loading cycles.

Corrective Action – The following plan was used to evaluate and repair the arc strikes present on the vessel: Arc strikes are to be examined visually.

- 1) If there is base metal melting observed in the arc strike, then the area shall be blended down to remove the melted material and then dye penetrant inspected to verify that no discontinuities are present.
- 2) If no melting is visually noted, then the area shall be dye penetrant inspected to verify that no discontinuities are present (no metal removal).
- 3) If the arc strike repair results in metal removal, then an engineering evaluation will be required to determine if “minimum wall thickness” is still intact. Any weld repairs to re-establish minimum wall tolerances shall be made using approved procedures.

Results – The arc strikes with melting associated with them were blended out and satisfactorily dye penetrant inspected. No weld repairs for minimum wall violations needed to be performed.

4.7 Discoloration of material in weld heat effected zones

Concern – There was discoloration of the base material in the heat-affected zones of a number of welds, possibly indicative of overheating during the welding process. Overheating during welding could adversely affect strength properties.

Corrective Action – Using an Equotip portable hardness tester, C. Czajkowski investigated the variation in hardness both outside and within the discolored zone of these welds.

Results – There was no significant hardness change, indicating that the welding was satisfactory and further that the welder had not cleaned the oxidation from the area after welding.

4.8 Pressure relief valving for vessel

Concern – The ASME Code requires the presence of a relief valve attached to the pressure vessel to prevent over pressurization. This relief valve must be National Board certified. The furnished valve of Russian manufacture was probably adequate but is not National Board certified. Moreover, the information on the valve label plate was in Russian and thus in an emergency could not be understood by most emergency workers.

Corrective Action – Purchase National Board certified pressure relief devices having an appropriate relief setting.

Results – Pressure relief devices were ordered. The main vessel relief valve setting was for the maximum allowable pressure determined by the calculations. The inner preionizer relief device setting is for less than 6 psig and along with the vacuum system check valve, protects the preionizer vacuum system from over pressurization.

4.9 Incomplete and inaccurate drawings

Concern – The vendor furnished preliminary drawings of the pressure vessel at the PDR. These drawings did not show all dimensions needed for stress calculations. Optoel later furnished a CD containing AutoCad files of the final drawings. Some files, however, were corrupted. In addition, visual inspection of the vessel showed areas where the as built device departed significantly from the drawings. This cast some doubt upon the design because the dimensions used in the calculations might not reflect the as built condition.

Corrective Action – An as-built drawing was made and dimensioned based on actual measurements of the furnished pressure vessel and its components.(See Appendix 7.2) These dimensions were used to calculate stresses in parts for which no dimensions were available and to recalculate stresses in other areas using the actual dimensions.

Results – Calculations based on the as built dimensions indicated the necessity of reducing the pressure rating of the vessel. Portions of the vessel previously unanalyzed because of incomplete dimensions were checked and found to be acceptable with the reduced pressure vessel rating. (See Appendix 7.5)

4.10 Pressure vessel end plate thickness

Concern – The ASME code calculations require a certain end plate thickness to resist stresses from internal pressure and secondary stresses associated with the size and spacing of ports in the endplate. Measurements of the as-built vessel indicated that not all end plate thicknesses were adequate. These port dimensions had not all been available for the original calculations.

Corrective Action – Perform the required calculations and reduce the design pressure if required.

Results – The calculations (see Appendix 7.5) indicated that in one case the thickness of the end plate did not satisfy the ASME code and would require the design pressure to be reduced to 126 psi (8.57 Atm).

4.11 Pressure vessel flange thickness

Concern – The ASME code requires a certain flange thickness to resist stresses from internal pressure. Drawing dimensions indicated that the flange thickness was not adequate. This condition was exacerbated by the fact that the flanges had to be machined flat after welding due to distortion. As-built measurements of the flanges indicated a varying thickness less than that required by the ASME Code.

Corrective Action – Perform the required calculations using the minimum flange thickness measured and reduce the vessel design pressure accordingly.

Results – The calculations (see Appendix 7.5) confirm that the flange thickness does not satisfy the ASME code and requires the design pressure to be reduced to 116 psi (7.89 Atm).

4.12 Main Fasteners and tapped holes

Concern – The rectangular main flange of the pressure vessel has through bolting on the long sides and tapped holes and studs on the short sides. Many of the tapped hole threads are damaged and deformed. The fasteners themselves have no markings and appear to be individually fabricated. Although the material strength may be adequate, the thread damage and lack of documentation makes the use of these fasteners problematic.

Corrective Action – Replace the fasteners with certified fasteners meeting ASME criteria. (See Appendix 7.9)

Results – The existing fasteners were replaced with 1" diameter ASTM A193-B7 steel fasteners on the long sides and ¾" diameter ASTM A193-B7 steel fasteners on the short sides. The tapped holes on the short side were machined to remove the damaged threads and special eccentric washers and nut plates were used to achieve through bolting in this dimensionally restricted area.

4.13 Stress analyses of parts not previously dimensioned

Concern – The original ASME code calculations did not include a number of components because no dimensions were available. Moreover, the as-built pressure vessels had changes including additional parts not depicted on the drawings.

Corrective Action – Using the dimensions from the as-built drawing (see 4.9 above), perform the required calculations and evaluate the component design.

Results – The calculations (see Appendix 7.5) indicated that previously unanalyzed components are adequate.

4.14 "Plug" weld repair

Concern – Two of the three ports on the main pressure vessel for the preionizer chamber, appear to have a "plug" weld on them, which is not allowed by ASME Code.

Corrective Action – Investigate and perform a stress analyses on the problem area.

Results – Further investigation (see 3.4 above) revealed that a “plug” weld was not present. A strip of metal approximately 25mm wide by approximately 6mm thick was wrapped around and welded to the outside of the tube penetration as repair. Since the strip was not continuous, the resultant welding appeared to be a plug weld. The need for this repair was brought about by an undocumented change in the sealing gland design between the main pressure vessel and the preionizer feedthrus. Part of the design change may have been caused by the need to rebores the ports to match the center-to-center distance of the preionizer feedthrus. Stress analysis (see Appendix 7.5) reveals the as-built port is structurally adequate.

4.15 X-ray weld coverage and quality of radiography

Concern – Not all of the Russian radiographs were readable, and the quality of some may not reveal all weld defects resulting in areas with no indication of weld quality.

Corrective Action – Use visual and liquid penetrant inspection to find and repair weld defects. Perform finite element analysis incorporating actual weld size to determine stress levels. Hydro test the vessel to a pressure above the maximum working pressure. Also, devise an ongoing inspection plan to assure that any cracks that may propagate as a result of cyclic loading will be discovered.

Results – Weld repairs were successfully made. The finite element analysis revealed satisfactory stress levels on welds with defects. (see Appendix 7.5). A hydrostatic test was successfully performed (see Appendix 7.8). A preventive maintenance procedure was established to periodically check for cracks and other pressure vessel degradation. (See Appendix 7.7)

4.16 Pressure Vessel Seal Material and Design

Concern – The supplied seal material had inadequate documentation as to its composition and was not in good condition.

Corrective Action - Replace the existing seal with a Viton seal having the appropriate cross sectional area for the existing seal groove.

Results – The Viton seal was purchased and installed. Hydro test of the vessel was successful (see Appendix 7.8)

4.17 Weld – Preionizer chamber to support tubes

_____ Preionizer chamber subassembly structural adequacy under the external pressure load.

Corrective Action – Perform stress analysis of the vessel and components to determine their adequacy. Hydro test the vessel to a pressure above the maximum design pressure.

Results – The analysis shows the preionizer vessel assembly is satisfactory. (see Appendix 7.5). The hydro test was successful (see Appendix 7.8).

5.0 Conclusions

- 5.1 The vessel design pressure is 116 psig, which is based on the ASME Pressure Vessel Code and a flange thickness of 2.435 inches.
- 5.2 The amplifier pressure vessel relief valve must be National Board certified and relieve at 116 psig.
- 5.3 The preionizer chamber and attendant system can be internally pressurized if a failure occurs. As a minimum, it must be equipped with a National Board Certified relief device with a setpoint of at most 5 psig.
- 5.4 All piping or tubing connected directly to the pressurized portion of the vessel must be designed to withstand a minimum pressure of 150 psig.
- 5.5 Welds having defects found by NDE must be excavated to sound metal; weld repaired and reinspected using an approved procedure.

- 5.6 Main fasteners must be replaced by ¾" and 1" diameter fastener systems having a minimum tensile strength of 120,000 psi.
- 5.7 The pressure vessel flange and boltholes must be machined to remove threads and weld interference with washers.
- 5.8 Stress analysis has shown the "plug" welded flanges are adequate as is.
- 5.9 The existing seal must be replaced by an o-ring seal of suitable cross section and material.

6.0 RECOMMENDATIONS

The specification for this vessel required its design criteria to be the equivalent of the ASME Section VIII Pressure Vessel Code. This equivalency was to be manifested through the use of the appropriate materials of construction; methods of joining these materials (e.g. welding) and thickness of these materials to insure the safety of the Laser operators.

Since the materials of this vessel (both the structural and the weld) are not certified to the ASME Boiler and Pressure Vessel Codes and also some components do not meet the code required thickness at the original design conditions, the device cannot meet the code. However, the intent of the code, to insure the safety of the pressure vessel and concomitantly the personnel operating it, has been satisfied. This assurance of safety is the result of reducing the vessel design pressure, removal and repair of known defects using certified techniques and the verification of their removal by code sanctioned nondestructive inspection methodology.

Potential operating risks associated with unknown characteristics of the Russian materials of construction can and will be minimized through the use of an augmented inspection plan for the vessel. Augmented inspection programs are commonly used in the nuclear industry when a high degree of operating reliability is mandatory. For this vessel, the inspection program should be a visual inspection of all accessible inside and outside welds and repaired areas. This inspection should be done initially, whenever the vessel is opened. After a period of satisfactory operation, i.e. one year, the inspections could be reduced to once a quarter. This inspection plan should be made a part of the routine maintenance requirements for the vessel at NSLS.

7.0 APPENDICES

- 7.1 12 Areas Of Concern
- 7.2 As Built Drawing Of Pressure Vessel
- 7.3 Russian Weld Wire Metallurgical Report
- 7.4 Weld Repair Procedure
- 7.5 Stress Analyses of Pressure Vessel
- 7.6 Longitudinal Seam Weld X-ray Inspection Report
- 7.7 Preventive Maintenance Procedure
- 7.8 Hydro Test Procedure and Report
- 7.9 Bolting Sketches

Appendix 7.1

12 Areas Of Concern

BROOKHAVEN NATIONAL LABORATORY — NSLS DEPARTMENT
ENGINEERING WORKSHEET

JOB NO.
Account #
07975

DATE
9/03/96

PAGE OF
21/22

NAME Don Lynch

SECTION Mechanical Engineering

PART IV. Conclusions

The following are the items of concern generated by the analyses contained in this report.

1. There is no clear indication of pressure relief devices on this unit. Pressure relief devices which limit the maximum potential internal pressure in the outer vessel to the maximum allowable working pressure are required. The BNL cognizant engineer must verify the pressure relief capabilities of the final design.
2. The OPTOEL drawing set provided for these analyses was incomplete and inconsistent. Part I of this report presents a hierarchical listing of the parts making up the complete High Pressure CO₂ Laser Amplifier package as could best be inferred from the various drawings as possible. The BNL cognizant engineer should assure that the final design drawings are consistent with the parameters used in the calculations herein, or revisit the appropriate calculations to demonstrate compliance. The BNL cognizant engineer must see to it that an effective configuration control system is imposed on the system in accordance with the QA policies of BNL and the NSLS. All pressure vessel assembly drawings must include the design operating pressure and temperature.
3. Independent verification of the material properties used as parameters for these calculations is required in accordance with ASME Section VIII, Part UG-10 must be assured by the BNL cognizant engineer. In addition, an effective material control program in accordance with the QA policies of BNL and the NSLS must be imposed.
4. The outer cylinder end plates thickness is inadequate. These plates must be a minimum of 1.534 inches [38.96 mm] thick. In addition, the port openings on these end plates require reinforcing. Specifically the total thickness of the end plate around all of the openings must be 2.969 in. [75.40 mm] or greater with a radial thickness around all holes of at least 2.650 in [67.31 mm].
5. The main casing flange thickness is insufficient. It requires a minimum thickness of 2.748 in [69.80 mm].
6. The bolt number/bolt size for all flanges are adequate. However, the BNL cognizant engineer must verify that the assumptions made concerning the o-ring gaskets for all parts are valid. These calculations must be revisited if any changes to the design (from the assumed design analyzed herein) are made.
7. The BNL cognizant engineer must check the engagement length for all threaded fasteners to be used. There was insufficient information available to calculate these herein.
8. No information concerning the inner vacuum vessel end caps was provided. The calculations provided herein provide the minimum allowable wall thickness for this vessel. The cognizant engineer must verify that the final design is in accordance with the assumptions made herein.
9. There are four openings on the inner vacuum vessel for which information concerning their purpose was not available. The adequacy of the vessel's thickness at these openings must be verified when the design is finalized.
10. The OPTOEL drawings seem to show acceptable welds for most of the port attachments, except for some ports which indicate welding one side only. In general, ASME Section VIII, PART UW requires that ports be welded on both sides of the shell or end plate. This practice is strongly suggested. If, for any reason OPTOEL cannot weld some ports on both sides, the cognizant engineer must insure that the details of the weld satisfy ASME Section VIII by a more detailed analysis. All welds must be liquid penetrant inspected on all passes. Welders should be qualified equivalent to ASME Code qualifications; BNL Q.A. must

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verify. Welding procedures including preheat and cooldown must be in accordance with ASME Code or equivalent procedures approved by BNL Q.A.

11. The inner vacuum chamber base is too complex to analyze directly. This part will be demonstrated adequate by hydraulic pressure testing at 1.5 x maximum allowable working pressure.
12. Components for which information was missing or insufficient were not analyzed. The adequacy of these components, including the input and output window assemblies must be verified prior to final approval of the design.

Appendix 7.2

As Built Drawings Of Pressure Vessel

Appendix 7.3

Metallurgical Report – Russian Weld Wire

managed by Brookhaven Science Associates
for the U.S. Department of Energy

Memo

date: March 21, 2000

to: John Skaritka

from: Robert L. Sabatini

subject: Analysis of the materials used in TW laser amplifier vessel

The four samples were analyzed using both Energy Dispersive X-ray Spectroscopy (EDS) and Wavelength Dispersive X-ray Spectroscopy (WDS). The samples were mounted and polished.

Results:

Sample #1 Excavated Weld Mat'l

Fe	Cr	Ni	Mn	Mo	Si
64.42	19.52	11.23	1.88	2.76	0.19

Sample #2 Encased Weld M.F?

347

3.1

Fe	Cr	Ni	Mn	Nb	Si
66.56	20.66	8.91	2.43	1.22	0.21

Sample Weldrod

3.1

Fe	Cr	Ni	Mn	Mo	Si
66.07	20.81	9.33	1.60	2.03	0.16

Sample MIG Wire

Fe	Cr	Ni	Mn	Mo	Si
63.45	20.33	11.01	1.69	3.20	0.33

Appendix 7.4

Weld Repair Procedure

WPS Number: SS 1-92 Revision E Date: October 10, 1995

Base Metals (QW-403)

P-Number: 8 To P-Number: 8 and P-Number: 1
Minimum Pipe Dia. Grooves: 7/16 in. dia. Fillets: 7/16" dia.

Groove Weld Thickness Range	
Base Metal: P8 to P8	.018 - .540 in. Max Deposited Metal: .540 in.
P8 to p1	.093 - .375 in. .375 in.

Backing or Retainer: When Permitted by Engineering Instructions.
Type: Stainless Steel when used.

Base Metal	Spec. No. (SFA)	AWS No. (class)	F-No	A-No	Table
304/304L	5.9	ER308L	6	8	1
316/316L	5.9	ER316/316L	6	8	1
321/321L	5.9	ER321	6	8	1
347/347	5.9	ER347	6	8	1
304/304L	None	ER385L-M	None	None	1
304/A-36	5.9	ER309 L	6	8	2

Size of Filler: -----.045 -3/32 in.-----
 Deposited weld metal: Table 1 .540 in. Max. Table 2 .375 in. Max.
 Thickness Range of weld metal:
 Groove: Table 1 .018 to .540 in. Table 2 .093 to .375 in.
 Fillet:-----All-----
 Consumable Insert:-----N/A-----

Preheat Temperature: Minimum 50 Deg. F.
Interpass Temperature: 350 Deg. F. Maximum

Postweld Heat Treatment	None
-------------------------	------

October 10, 1995

WPS Number: SS 1-92 Rev.E (File c:\data\welding\wps\SS1-92E2)

Written By: T. Lambertson/Al Farland Approved:

GAS (QW-408)

Shielding Gas: Argon 99.99% Commercial grade. 10 - 15 cfh

Trailing Gas: None

Backing Gas: Argon or Nitrogen Commercial grade. 5 - 30 cfh

1. The inside of all internal surfaces of the joint to be welded shall be purged for a minimum of 5 inches each side of the weld with the Inert Gas (Nitrogen or Argon) before the welding starts and shall remain blanketed until the temperature of the metal has returned below 350 Deg. F. A coupon of 1/16 in. thick by 3/4 in. wide by 1/4 in. long will be used to determine if the backing gas is concentrated enough to begin welding. A weld will be made on the test coupon and the underside examined for bright shiny metal confirming the adequacy of the backing gas. The inspector will confirm appearance.

2. The flow of Argon shall be maintained on the electrode and the weld until both have returned to normal color. Care should be exercised to prevent blowing the pool of molten metal with the torch or stirring it with the filler wire. If at any time the Tungsten electrode becomes contaminated with filler or weld metal, welding should be stopped at once and the electrode cleaned or broken off and reshaped before welding is continued.

Electrical Characteristics

Current/Polarity: DCEN

Amps: 50 - 300

Volts: N/A

Tungsten Electrode: 1/16 - 3/32 2% Thoriated.

Pulsing: Not Required

Technique (QW-410)

String or Weave Bead: String

Oscillation: N/A

Gas Cup Size: No. 4 to 8

1. Degrease components within 5 hours of welding. Wire brush prior to welding and after each pass using stainless steel wire.

2. Filler wire shall be cleaned of all oil, grease and foreign matter immediately before being used. The filler wire shall be cleaned with alcohol solvent.

3. Pipe or plate to be welded shall be supported to prevent excessive stress during the welding operation.

4. Each welder who passes a Welder Qualification Test per ASME Sec. IX will be assigned a number stamp. This is to be used on all code welds or as required.

5. Remove all surface slag or rust from P-1 materials before welding.

October 10, 1995

WPS Number: SS 1-92 Rev. E (File c:\data\welding\wps\SS1-92E3)

Written By: T. Lambertson/Al Farland Approved:

5. Repair with inspection approval. Remove defects by grinding, filing, wire brushing, or plasma gouging. Repair using this procedure. NDT by Visual and Dye before proceeding.

6. Peening of root or final pass not permitted.

7. Travel speed: 1 to 20 inches per minute.

Weld Preparation Chart

Pipe/Plate Size	Thickness (T)	Root Opening	Half Angle (D)
4 in. Sched 40	.270"	3/32 to 5/32	30 to 37 ½
1 in. Sched 40	.130"	3/32 to 5/32	30 to 37 ½
7/16 in. Dia.	Under .090"	0 to 1/32	0
Plate	.187"	0 to 0	30 to 37 ½

Supporting Documents

1. PQR MMSD-1 dated Mar 17, 1972
2. PQR MMSD-1 dated May 17, 1972
3. PQR MMSD-1 dated May 15, 1987
4. PQR 12492 dated Jan 24, 1992
5. PQR 11894 dated Feb 17, 1994
6. PQR 092895 dated Oct 28, 1995

Revisions

1. Rev N/C was upgraded from WPS MMSD-1 using supporting documents listed. January 2, 1992
2. Rev A changed size of purge coupon, travel speed from 3-20 to 1-20 IPM and changed root opening from 1/8 to 5/32 in. January 24, 1992.
3. Rev. B added support document PQR 12492. February 6, 1992
4. Rev. C added support document PQR 11894. February 17, 1994. For ER385L Rod.
5. Rev. D Editorial changes in filler metals. May 16, 1994.
6. Rev. E added support document PQR 092895. Added editorial changes for "Table 2" Materials. September 28, 1995

Appendix 7.5

Stress Analyses of Pressure Vessel

**ATF/OPTOEL COMP. HIGH-PRESSURE CO₂ LASER AMPLIFIER
MECHANICAL STRESS CALCULATIONS, DR Lynch 10/3/2000
(CALCULATIONS REFLECT AS BUILT CONDITION)**

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b. Vacuum Vessel

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14. All other items

ATF/OPTOEL COMP. HIGH-PRESSURE CO₂ LASER AMPLIFIER**MECHANICAL STRESS CALCULATIONS, DR Lynch 7/20/00
CALCULATIONS REFLECT AS BUILT CONDITION****SUMMARY**

The following table summarizes the most significant results of the detailed analyses of this report.

<u>ITEM</u>	<u>MINIMUM</u> ¹	<u>ACTUAL</u> ²	<u>STRESS</u> ³	<u>MAX PRESSURE</u> ⁴
shell wall thickness	0.156 in [3.96 mm]	0.310 in [7.87 mm]	9726 psi [67.06 MPa]	292 psig [2.11 MPa]
end plate, minor, thickness	0.447 in [11.35 mm]	1.750 in [44.45 mm]	1263 psi [8.71 MPa]	2247 psig [15.60 MPa]
end plate, major, thickness	1.5205 in [38.62 mm]	2.510 in [63.75 mm]	7174 psi [49.46 MPa]	396 psig [2.83 MPa]
shell thickness between 8.200 in and 8.200 in opening pairs	0.316 in [8.03 mm]	0.310 in [7.87 mm]	19965 psi [138 MPa]	142 psig [1.08 MPa]
end plate thickness between rect. and 4.210 in opening pair	2.7254 in [69.23 mm]	2.510 in [63.75 mm]	23085 psi [159.2 MPa]	123 psig [0.95 MPa]
main casing flange thickness	2.718 in [69.03 mm]	2.435 in [61.84 mm]	24396 psi [168.2 MPa]	116.3 psig* [0.90 MPa]
main casing flange bolt stress dia.	.550 in [13.98 mm]	.652 in [16.56 mm]	17789 psi [122.7 MPa]	206 psig [1.52 MPa]
casing end cover thickness worst case, port "H"	.430 in [10.92 mm]	n/a n/a	19580 [135 MPa]	145 psig [1.10 MPa]
casing end cover bolt stress dia. worst case, port "H"	.149 in [3.78 mm]	.388 in [9.86 mm]	3678 psi [25.36 MPa]	1111 psig [7.76 MPa]
worst case stress, casing end port tube wall thickness (port "E")	.024 in [0.60 mm]	.1725 in [4.38 mm]	2486 psi [17.1 MPa]	1142 psig [7.98 MPa]
casing end flanged ports worst case stress, flange thickness (port ?)	.352 in [8.94 mm]	.715 in [18.16 mm]	4701 psi [32.41 MPa]	630 psig [4.44 MPa]
casing end flanged ports bolt worst case stress dia., port "E"	.192 in [4.88 mm]	.533 in [13.54 mm]	3628 psi [25.01 MPa]	1360 psig [9.48 MPa]
worst case stress, shell port tube thickness (based on port "B")	.043 in [1.22 mm]	.1125 in [2.86 mm]	2294 psi [15.81 MPa]	1237 psig [8.63 MPa]
worst case stress, shell port flange thickness (based on port "D")	.382 in [9.71 mm]	.472 in [11.99 mm]	12819 psi [88.4 MPa]	228 psig [1.68 MPa]
<u>ITEM</u>	<u>MINIMUM</u> ¹	<u>ACTUAL</u> ²	<u>STRESS</u> ³	<u>MAX PRESSURE</u> ⁴

worst case bolt stress, shell ports (based on port "D")	.155 in [3.94 mm]	.255 in [6.47 mm]	9212 psi [63.5 MPa]	459 psig [3.17 MPa]
type "D" port stiffening ring	n/a n/a	n/a n/a	n/a n/a	145 psig [1.10 MPa]
inner vacuum vessel shell wall thickness	.159 in [4.04 mm]	.235 [5.97 mm]	n/a n/a	300.72 psia [2.07 MPa]
inner vacuum vessel end caps	.676 in [17.17 mm]	.787 in [20 mm]	n/a n/a	216.45 psia [1.49 MPa]
inner vacuum vessel base flange	.836 in [21.23 mm]	.787 in [20 mm]	n/a n/a	141.46 psia [0.98 MPa]
inner vacuum vessel base support plate	n/a n/a	n/a n/a	11861 psi [81.78 MPa]	263.63 psia [1.82 MPa]
inner vacuum vessel base pressure boundary foil	n/a n/a	n/a n/a	74741 psi [515.3 MPa]	159.7 psia [1.10 MPa]
inner vacuum vessel base bolts	.2546 in [6.47 mm]	.0465 in [1.18 mm]	n/a n/a	4775 psia [32.92 MPa]
inner vacuum vessel shell thickness between ports	.2385 in [6.06 mm]	.235 in [5.97 mm]	n/a n/a	157.36 psia [1.08 MPa]
inner vacuum vessel end plate thickness at opening	n/a n/a	.787 in [20 mm]	7458 psi [51.42 MPa]	419.3 psia [2.89 MPa]
inner vacuum vessel shell port tube thickness (worst case: larger port)	n/a n/a	n/a n/a	n/a n/a	351.6 psig [2.53 MPa]
inner vacuum vessel shell port flange thickness (worst case, larger port)	.385 in [9.78 mm]	.25 in [6.35 mm]	n/a n/a	60.73 psig ** [0.52 MPa]
inner vacuum vessel shell port bolt stress dia., larger port	.124 in [3.15 mm]	.255 in [6.48 mm]	5927.4 psi [40.87 MPa]	612 psig ** [4.32 MPa]
inner vacuum vessel cover flange thickness	.418 in [10.62 mm]	.630 in [16 mm]	n/a n/a	362.8 psia [2.50 MPa]
seam weld	n/a [n/a]	n/a [n/a]	8052 psi [55.5 MPa]	315 psig [2.27 MPa]
shell/main flange welds, major sect.	n/a [n/a]	n/a [n/a]	9311 psi [64.2 MPa]	273 psig [1.98 MPa]
shell/main flange welds, minor sect.	n/a [n/a]	n/a [n/a]	9018 psi [62.2 MPa]	281 psig [2.04 MPa]
shell/end plate welds, major sect.	n/a [n/a]	n/a [n/a]	20047 psi [138.2 MPa]	127 psig [0.97 MPa]

ITEMMINIMUM¹ACTUAL²STRESS³MAX PRESSURE⁴

shell/end plate welds, minor sect.	n/a [n/a]	n/a [n/a]	20271 psi [139.8 MPa]	125 psig [0.96 MPa]main
flange/end plate welds, major sect.	n/a [n/a]	n/a [n/a]	15080 psi [104.0 MPa]	168 psig [0.97 MPa]
main flange/end plate welds, minor sect.	n/a [n/a]	n/a [n/a]	15397 psi [106.2 MPa]	165 psig [0.96 MPa]
main flange/end plate/shell corner welds, major sect.	n/a [n/a]	n/a [n/a]	6126 psi [42.2 MPa]	414 psig [2.96 MPa]
main flange/end plate/shell corner welds, minor sect.	n/a [n/a]	n/a [n/a]	8569 psi [59.08 MPa]	296 psig [2.14 MPa]
shell & end plate to large rectangular flange weld	n/a n/a	n/a n/a	n/a n/a	159.7 MPa [1.10 MPa]
shell to end plate weld	n/a n/a	n/a n/a	n/a n/a	159.7 MPa [1.10 MPa]
larger port tube to shell weld	n/a n/a	n/a n/a	n/a n/a	159.7 MPa [1.10 MPa]
smaller port tube to shell weld	n/a n/a	n/a n/a	n/a n/a	159.7 MPa [1.10 MPa]
larger port tube to end cap weld	n/a n/a	n/a n/a	n/a n/a	159.7 MPa [1.10 MPa]
smaller port tube to end cap weld	n/a n/a	n/a n/a	n/a n/a	159.7 MPa [1.10 MPa]

*

This item determines the actual Maximum allowable design pressure

**

The max pressure indicated is the maximum allowable pressure in the internal vacuum vessel in the event of a leak failure from the main vessel to the vacuum vessel

Other Notes:

¹This is the minimum size of the characteristic dimension (e.g. diameter, thickness) of the item that would be required to satisfy the allowable stress calculation at MAWP = 145 psig.

²This is the actual size of the characteristic dimension for the item.

³This is the calculated maximum stress in the item based on the actual size of the characteristic dimension and MAWP = 145 psig.

⁴This is the maximum pressure that the item can withstand without its maximum stress exceeding the allowable stress, based on the actual characteristic dimension

All other components analyzed, but not included, had characteristic dimensions which were substantially greater than the minimum requirement and/or were found to have maximum stresses at MAWP = 145 psig considerably less than allowable.

A. Design Criteria

Design Maximum Allowable Working Pressure(MAWP): 145 psig (1.0 MPa)

Design Pressurizing Gas: CO₂ @ 68 deg F (20 deg C)

Design External Environment: Sea Level Air (68 deg F, 14.7 psia/20 deg C , .101342 MPa)

Internal Vacuum Chamber: Maximum Allowable External Working Pressure = 145 psig/

Internal pressure = -14.7 psig ($\sim 10^{-5}$ torr.) for a total pressure differential of 159.7 psid.

B.OPTOEL Design Characteristics

Casing Cylinder OD: 29.360 in (746 mm)

Casing Cylinder Wall Thickness: .310 in (7.874 mm)

Casing Cylinder Openings (and nearest adjacent openings): 6.650 in/2.665 in spaced 13.313 in [168.9/67.7/338.1 mm]
8.200 in/8.200 in spaced 13.938 in [208.3/208.3/354 mm]
1.490 in/1.490 in spaced 13.000 in [37.9/37.9/330.2 mm]

Casing Ends, Thickness: 2.51 (major), 1.75 (minor) in (63.75/44.45 mm)

Casing Ends, Openings (and nearest adjacent openings): 4.210 in/7.784 in spaced 9.781 in [107/179.4/248.4 mm]
2.360 in/7.784 in spaced 8.689 in [60/179.4/218.3 mm]
1.490 (single opening in minor end)

(Note: For Rectangular Opening an equivalent Diameter, $De=2*\sqrt{Area/\pi}$, is used. The center of the hole for separation purposes is considered to be at either the actual center or $De/2$ away from the edge, whichever results in a smaller separation, i.e. less web.)

Casing Flange, Thickness: 2.435 in (61.85 mm)

Sealing Surface Mean Perimeter: 210.03 in (5335 mm)

Sealing Surface Nominal Width: .305 in (7.75 mm)

Casing Flange Attachment Method: 40 1-8 Bolt Studs (long sides) and 16 3/4-10 Bolt Studs (short sides) on a Rectangular Bolt Pattern on a mean perimeter of 222.25 in (5645 mm) with a maximum spacing of 4.03 in (102.8 mm). Bolt Studs are attached with 1-8 standard nuts on long ends, and 3/4-10 standard nuts the on short ends.

Flange Bolts

12 M8 Bolts on a 7.559 in (192 mm) Bolt Circle (2 places); Flange Thickness: .472 in (12 mm)

8 M8 Bolts on a 3.543 in (90 mm) Bolt Circle; Flange Thickness: .540 in (13.7 mm)

8 M16 Bolts on a 7.082 in (180 mm) Bolt Circle; Flange Thickness: .715 in (22 mm)

18 M12 Bolts on a 9.291 in (236 mm) Bolt Circle (3 places); Flange Thickness: .775 in (19.68 mm)

4 M16 Bolts on a 3.930 in (100 mm) Bolt Circle (16 places); Flange Thickness: .515 in (13 mm)

Casing End (Tapped) Cover Attachment Bolts:

24 M12 Bolts on a rectangular pattern, maximum spacing 1.578 in (37.54 mm); end thickness: 2.510 in (63.7 mm)

6 M12 Bolts on a 4.330 in (110 mm) Bolt Circle (4 places); end thickness: 2.510 in (63.7 mm)

Internal Vacuum Chamber:

Vacuum chamber shell OD: 10.16 in [258 mm]

Vacuum chamber shell min. wall thickness: .235 in [6 mm]

Vacuum chamber length: 44.81 in [1138 mm]

Vacuum chamber shell openings (diameters and nearest adjacent openings): 5.98 in/ 1.96 in spaced 13.75 in
[152/49.78/349.3 mm]

Internal Vacuum Chamber End Caps:

Vacuum chamber end plate thickness: .787 in [20 mm]
 End plate opening diameter: 6.495 in [165 mm]
 End plate cover thickness: .630 in [16 mm]
 End plate cover opening diameter: 2.360 in [60 mm]
 End plate cover opening cap thickness: .420 in [10.67 mm]

Internal Vacuum Chamber Base Plate:

Vacuum chamber base flange outer rectangular dimensions: 44.41 in [1128 mm] x 7.87 in [200 mm]
 Vacuum chamber base flange inner rectangular dimensions: 41.94 in [1065 mm] x 5.87 in [149 mm]
 Vacuum chamber base flange minimum thickness: .787 in [20 mm]

Vacuum chamber foil support plate outer rectangular dimensions: 40.25 in [1022 mm] x 5.50 in [140 mm]
 Vacuum chamber foil support plate minimum thickness: .47 in [11.94 mm]
 Vacuum chamber foil support plate hole array: triangular "honeycomb" array of 12 rows with alternating rows of 106 and 105 .315 in diameter holes spaced so that each neighboring hole center in the same row is separated from the center of its neighboring holes by .374 in [9.5 mm] and each row is separated by .325 in [8.5 mm]

Vacuum chamber pressure barrier foil rectangular dimensions: (use same dimensions as support plate)
 Vacuum chamber pressure barrier foil thickness: .003 in [.064 mm]

Note: Additional "as-built" dimensions for the internal vacuum chamber can be found on ATF-SK-1217, Sheet 4 and 5.

C. OPTOEL Selected Materials

Casing Flange Sealing Material: VITON O-Ring: Nominal sealing force 60 lbs/linear inch [10583 Nt/m] (catalog value) for larger seals used on 2.665" and 6.65" dia openings on shell, 4.21" dia opening on end plate, and main flange; sealing force 40 lbs/linear inch [7055 Nt/m] (catalog value) for smaller diameter seals used on 1.49" and 8.20" dia openings on shell and 1.49", 2.36" dia and 4.70"x10.125" rectangular openings on end plate.

Casing Cylinder and End Material: Stainless Steel 12X18H10T (Note: Properties per OPTOEL, verified at BNL by R. Sabatini):

Ultimate Strength: 78321 psi [540 MPa]
 Yield Strength: 34810 psi [240 MPa]
 Allowable Stress: 19580 psi [135 MPa] {Safety Factor =4.0 based on ult. strength}

(Same material is used for main flanges, all port tubes and flanges, all parts of inner vacuum vessel (except foil pressure barrier) and for feedthrough retaining flange.)

Inner vacuum vessel foil pressure barrier {assumed material}: titanium foil (Note: properties of the titanium foil were not supplied by OPTOEL. A sample of the foil was analyzed by Bob Sabatini of BNL Instrumentation Division and found to be consistent with commercially pure titanium, the strength properties of which vary greatly depending upon very small amounts of alloying agents, even amounts small enough for the material to be considered commercially pure.)

Main flange bolts are 1-8 UNC-2A for bolts along the long sides (40 places) and 3/4-10 UNC-2A for bolts along the short side (16 places), all are type ASTM A320 Grade L-7, and use Grade 8 nuts. All bolts are sized to have a minimum of (3) threads beyond full engagement with the nuts. (Note: these bolts selected by BNL to replace OPTOEL supplied bolts with imperfect threads and inadequate traceability.)

Ultimate Strength: 120,000 psi [827 MPa]
 Yield Strength: N/A
 Allowable Stress: 25,000 psi, [per ASME B31.1]

All other bolts are type ASTM A320 Grade L-7, and use Grade 8 nuts. All bolts are sized to have a minimum of (3) threads beyond full engagement with the nuts. Engagement length of the nuts is a minimum of 1/2 of nominal diameter of mating bolt.

Ultimate Strength: 120,000 psi [827 MPa]
 Yield Strength: N/A
 Allowable Stress: 25,000 psi, [per ASME B31.1]

D. Calculations (ASME Section VIII Methods, 1992 with revisions to Fall 1994, except where otherwise noted)

1. Cylinder wall minimum thickness {ref. Section VIII UG-27}

$t(\min)$ is greater of $t(\min) = PR / (SE - 0.6P)$ {Eqn. 1, UG-27} or $t(\min) = PR / (2SE + 0.4P)$ {Eqn. 2, UG-27}

where: $t(\min)$ is the minimum allowable wall thickness { $t(\min,1)$ & $t(\min,2)$ are calculated values of $t(\min)$ based on eqn 1 or 2 respectively}

P is the maximum working pressure= 145 psi
 R is the cylinder's inner radius= 14.37 in
 S is the allowable stress= 19580 psi
 E is a joint efficiency factor = 0.7 (for circumferential fillet welds)
 0.55 (for longitudinal welds)

(No {or inadequate} radiographic inspection, double lap joint: E= 0.55, fillet joint: E= 0.7 per Section VIII, subsection UW-12, Table UW-12)

$$t(\min,1) = (145)(14.37) / [(19580)(.70) - (0.6)(145)] = .153 \text{ in [3.89 mm]}$$

$$t(\min,2) = (145)(14.37) / [(2)(19580)(.55) + (0.4)(145)] = 0.096 \text{ in [2.45 mm]}$$

therefore $t(\min) = 0.153 \text{ in}$ (Note: add .003 for corrosion allowance, minimum cylinder wall = .156 in. [3.96 mm])
 actual wall th. = 0.310 in [7.87 mm]

{ for use in later calculations, the min wall thickness if E=1.0, $t_r = .110 \text{ in [2.79 mm]}$ }

using the actual wall thickness and 145 psig internal pressure (0 psig external pressure), Eqn.'s 1 & 2, UG-27 can be manipulated to solve for actual stress:

S (actual) is the greater of $S(\text{actual},1) = [PR + .6Pt(\text{actual})] / [Et(\text{actual})]$; $S(\text{actual},2) = [PR - .4Pt(\text{actual})] / [2Et(\text{actual})]$

$$S(\text{actual},1) = [(145)(14.37) + .6(145)(.310)] / [(70)(.310)] = 9726 \text{ psi}$$

$$S(\text{actual},2) = [(145)(14.37) - .4(145)(.310)] / [2(.55)(.310)] = 6058 \text{ psi}$$

Therefore, the maximum stress is 9726 psi [67.06 MPa]

similarly, the maximum allowable pressure for the basic shell can be determined as

$$P_{\max 1} = t(\text{actual})SE / [R + 0.6t(\text{actual})] = 292 \text{ psi [2.11 MPa]}$$

$$P_{\max 2} = 2t(\text{actual})SE / [R - 0.4t(\text{actual})] = 469 \text{ psi [3.33 MPa]}$$

therefore the max. allowable pressure is 292 psi [2.11 MPa] for the basic shell

Conclusion: The basic cylinder wall thickness is adequate.

2. Cylinder Flat Heads (End Caps) {ref. Section VIII UG-34}

$t(\min) = d \sqrt{ZCP / SE}$ {Eqn. 3, UG-34}

where: $t(\min)$ is the minimum allowable wall thickness

P is the maximum working pressure= 145 psi

d is the end cap's short dimension (major/larger section)= 19.130 in

S is the allowable stress= 19580 psi

E is the joint efficiency= 0.7
 (Assume no radiographic inspection: E= .7 per Section VIII, subsection UW-12, Table UW-12)
 C is the joint factor {Table UG-34, type e} = 0.33
 Z is a factor to account for the fact that the cover is non circular
 UG-34 eqn. (4) $Z = 3.4 - 2.4 \cdot d/D$, Z = 2.5, where D= 28.74 is the end cap's long dimension in inches
 $Z = 3.4 - 2.4 \cdot 19.130/28.74 = 1.8025$
 $t(\min) = 19.130 \cdot \sqrt{(1.8025)(0.33)(145)/(19580)(.7)} = 1.5175 \text{ inch}$
 Add .003 inch (corrosion allowance), min. end plate wall thickness = 1.5205 in [38.62 mm]
 actual thickness= 2.510 in [63.75 mm]

The minor (smaller) section flat head has d = 4.75 and D = 21.350

therefore Z=2.5
 and $t(\min) = .444 \text{ in} + .003 \text{ (corrosion allowance)} = .447 \text{ in [11.35 mm]}$
 actual thickness = 1.750 in (44.45 mm)

{ for use in later calculations, the min wall thickness if E=1.0, $t_{r,maj} = 1.273 \text{ in [32.33 mm]}$; $t_{r,min} = .374 \text{ in [9.51 mm]}$ }
 UG-34 eqn. (3) can be manipulated to solve for the maximum stress using the actual wall thicknesses

$$S = ZCP / \{E[t(\text{actual})/d]^2\}$$

for the major section flat head $S = (1.8025)(.33)(145) / \{(.7)[(2.51-.003)/(19.130)]^2\} = 7174 \text{ psi [49.46 MPa]}$

for the minor section flat head $S = (2.5)(.33)(145) / \{(.7)[(1.75-.003)/(4.75)]^2\} = 1263 \text{ psi [8.71 MPa]}$

Similarly, UG-34 eqn. (3) can be manipulated to find the maximum allowable pressure based on the actual flat head thicknesses and the max allowable stress:

$$P = SE(t(\text{actual})/d)^2 / ZC$$

for the major section flat head $P = (19580)(.7)[(2.51-.003)/19.13]^2 / [(1.8025)(.33)] = 396 \text{ psig [2.83 MPa]}$

for the minor section flat head $P = (19580)(.7)[(1.75-.003)/4.75]^2 / [(2.5)(.33)] = 2247 \text{ psig [15.6 MPa]}$

Conclusion: The basic end cap flat head wall thickness for both the major and minor sections of the vessel are adequate.

3. Cylinder Openings (ports) {ref. Section VIII UG-37}

$$A = d \cdot t_r \cdot F + 2 \cdot t_n \cdot t_r \cdot F (1 - fr1);$$

$$A1 = \text{the larger of: } A1 = d (E1 \cdot t - F \cdot t_r) - 2 \cdot t_n (E1 \cdot t - F \cdot t_r) (1 - fr1)$$

$$\text{or } A1 = 2 (t + t_n) (E1 \cdot t - F \cdot t_r) - 2 \cdot t_n (E1 \cdot t - F \cdot t_r) (1 - fr1)$$

where: A is the required area for reinforcement
 A1 is the available area for reinforcement due to oversizing of shell
 t_r is the required shell thickness if E = 1 (see above). Therefore $t_r = .110$.
 F is a factor to compensate for non-radial holes (all holes in shell are radial, F=1.0)
 t_n is the nozzle thickness of the port (note: neglect the nozzle thickness, i.e. $t_n = 0$)
 $fr1$ is a factor to compensate for different materials used for nozzle compared to shell. $fr1 = 1$ for all nozzles in shell)
 E1 is a factor to compensate for seam welds through holes. E1 =1.0 for all holes in the shell.
 t is the actual thickness of the shell = .310 in
 d is the port opening diameter (in inches)

The equations for A1 thus reduce to:

$A1 = \text{the larger of: } A1 = d(t - tr) \quad \text{or} \quad A1 = 2t(t - tr)$

for all of the shell openings, $d > 2t$, therefore use $A1 = d(t - tr)$

Therefore, setting $A1 \geq A$, putting in the above values, if true then no reinforcing is needed.

which results in $t \geq 2 * tr$. From above $t = .310$ " and $2 * tr = .220$ ".

This is true, so all holes are properly reinforced as single holes.

Now check that spacing is adequate for multiple holes. Centers of adjacent holes should be spaced at a distance greater than the sum of their diameters, else reinforcement must be greater. {ref. Section VIII UG-42}

There are three different hole pairs to be considered as indicated above.

1st pair: $6.650 \text{ in} + 2.665 \text{ in} \leq 13.313 \text{ in}$. These holes are adequately reinforced

2nd pair: $8.200 \text{ in} + 8.200 \text{ in} < 13.938 \text{ in}$. This is not true so we have to consider how much the overlap translates to required shell thickness. From UG-42, the thickness is adequate for 2 equal holes if $t \geq [d tr + (L-d) tm] / (L-d)$, where d & tr are defined above, $L = 13.938 \text{ in}$ is the distance between the 2 hole centers and tm is the min required wall thickness, calculated in calculation 1, above, $tm = .156 \text{ in}$, then $t \geq .313$. Add .003 for corrosion allowance $t \geq .316 \text{ in}$ [8.03 mm]

Since $t(\text{actual}) = .310$ these holes are not adequately reinforced.

manipulating UG-42 to solve for tm that gives the required $t (= .307)$,

$tm = .153$. This can be plugged into the cylinder min wall calculation, above, and

manipulated to solve for a new maximum allowable pressure ($= 142 \text{ psig}$ [1.08 MPa]), and similarly, can be used to find the stress at the design MAWP ($= 19965 \text{ psi}$ [138 MPa]).

Conclusion: The maximum allowable pressure for this vessel needs to be derated to 142 psig to assure that these holes are adequately reinforced.

3rd pair: $1.490 \text{ in} + 1.490 \text{ in} \leq 13.000 \text{ in}$. These holes are adequately reinforced

4. End Cap openings (ports) {ref UG-39}

$A = 0.5 d tm$; $A1 = d(t - tr)$

where: d is the hole diameter

tr is the required thickness (with $E=1.0$) $= 1.273 \text{ in}$ (.374 for minor section)

tm is the minimum required wall thickness $= 1.5205 \text{ in}$. (.447 for minor section) as found in calculation 2, above

t is the actual end cap thickness $= 2.51$ (major section, 1.75 in minor section)

therefore the single hole reinforcement is adequate if $t \geq tr + .5 tm$, or $t \geq 2.034 \text{ in}$ (for major section, $t \geq .598$ for minor section)

(Note: current design meets these requirements for both sections .)

Now consider that there are multiple openings. The minor section has only one opening at each end; therefore it is adequately reinforced per code. No further reinforcement is needed for the major section if the sum of the diameters of adjacent holes is less than the distance between the hole centers. (Note: for the rectangular opening an effective diameter of 7.784 was used.)

1st pair: $4.210 \text{ in} + 7.784 \text{ in} = 11.994 \text{ in} < 9.781 \text{ in}$. This is not true so additional reinforcement is needed.
 2nd pair: $2.360 \text{ in} + 7.784 \text{ in} = 10.144 \text{ in} < 8.689 \text{ in}$. This is not true so additional reinforcement is needed.

Section VIII UG-39 allows 2 adjacent holes on a flat end/cover to be as close as 1-1/4 the average diameter of the holes where the reinforcement between the holes shall be such that 50% of the summed individual reinforcement area is distributed in the plane section between the two openings.

1st pair: $[(4.210 \text{ in} + 7.784 \text{ in}) / 2] * 1-1/4 = 7.496 \text{ in} < 9.781$. This is true.
 2nd pair: $[(2.360 \text{ in} + 7.784 \text{ in}) / 2] * 1-1/4 = 6.340 \text{ in} < 8.689$. This is true.

Both hole pairs are sufficiently spaced to allow for reinforcement.

1st pair: Total reinforcement area needed between holes = $A_r = 0.5 * 0.5 * (d_1 + d_2) t_m = 4.559 \text{ sq in}$
 total thickness required to reinforce $(A_r / \{L - [(d_1 + d_2) / 2]\} + t_m) = 2.7254 \text{ in} [69.23 \text{ mm}]$

2nd pair: Total reinforcement area needed = $0.5 * 0.5 * (d_1 + d_2) t_m = 3.856 \text{ sq in}$
 total thickness required to reinforce $(A_r / \{L - [(d_1 + d_2) / 2]\} + t_m) = 2.587 \text{ in} [65.70 \text{ mm}]$

***** Reinforcement for both hole pairs is less than required, but since the 1st pair reinforcement requirements are greater, the 1st pair serves as the basis for the following:

Based on the first hole pair separation and ratio of hole sizes in the first hole pair, and noting that

- (a) the actual end plate is less than the calculated requirement,
- (b) the entire end plate is of uniform thickness (obviating the need for calculating the width of reinforcement)

and (c) the relationship between internal pressure and the thickness calculations : P is proportional to t^2 ,

the maximum allowable pressure can be derated by the square of the ratio of the actual end plate thickness to the calculated required thickness,

$$P (\text{allowable}) = 145 * (2.51 / 2.7254)^2 = 123.0 \text{ psig} \quad [0.95 \text{ MPa}].$$

similarly, the maximum stress in the part (based on actual thickness and MAWP) can be calculated as 23085 psi [159.17 MPa] *****

Conclusion: The maximum allowable pressure for this vessel needs to be derated to 123 psig to assure that these holes are adequately reinforced.

5. Main Casing Flange & Bolts

a. Flange thickness

The minimum main casing flange thickness is calculated by equation (5) in UG-34:

$$t(\text{min}) = d * \sqrt{ZCP/SE + 6W_h G/SELd^2}$$

where

t(min) is the minimum allowable wall thickness

P is the maximum working pressure=

145 psi

d is the flange's short dimension=

29.070 in

S is the allowable stress=

19580 psi

E is the joint efficiency=

0.7

(Assume no radiographic inspection: E= .7 per Section VIII, subsection UW-12, Table UW-12)

C is the joint factor {Table UG-34, type k} =

0.3

Z is a factor to account for the fact that the cover is non circular

(UG-34 eqn. (3) $Z = 3.4 - 2.4 \cdot d/D$, where $D = 75.945$ is the flange's long dimension in inches

$$Z = 3.4 - 2.4 \cdot 29.070/75.945; Z \leq 2.5 \quad 2.4813$$

W = Total bolt load (gasket compression load + pressure load; gasket compression load = $60 \cdot 210.03$

$$+ 145 \cdot 29.070 \cdot 75.945 = 332721 \text{ lbs [1479945 Nt]}$$

h_G is the gasket moment arm as defined in figure UG-34 (k) = 1.09 in

L is perimeter of the non-circular bolted head measured along the bolt holes =

$$222.25 \text{ inches}$$

$$t(\text{min}) = 29.070 \cdot \sqrt{(2.4813)(.3)(145)/[(19580)(.7)] + (6)(332721)(1.09)/[(19580)(.7)(222.25)(29.070)^2]}$$

$$t(\text{min}) = 2.715 \text{ in [68.96 mm]}$$

Add .003 for corrosion allowance, minimum main casing flange thickness = 2.718 in [69.03 mm]

actual thickness = 2.435 in [61.84 mm]

In addition, UG-34(d) requires that the net flange thickness under the gasket groove must be greater than (Note: in this case the moment arm on the short ends of the vessel is 2.082 in <vs. 1.09 in on the long sides> so use this as h_G

$$t(\text{net,min}) = d(6Wh_g/SLd^2)^{1/2} = 29.070 \cdot \{6 \cdot (332721)(2.082)/[(19580)(222.25)(29.070)^2]\}^{1/2} = 0.977 \text{ in} + .003$$

for corrosion allowance = 0.980 in [24.90 mm]

$$t(\text{net,actual}) = 2.260 \text{ in [57.40 mm]}$$

The eqns. of UG-34 can be manipulated to find the maximum stress in the flange at MAWP and actual flange thickness as follows:

$$S = [ZCP + 6 \cdot Wh_g/Ld^2]/[E(t/d)^2]$$

$$S = [(2.4813)(.3)(145) + 6(332721)(1.09)/\{(222.25)(29.070)^2\}]/[(0.7)((2.435-.003)/(29.070))^2] = 24396 \text{ psi}$$

[168.2 MPa]

Similarly, the eqns. of UG-34 can be manipulated to find a maximum allowable pressure based on the actual thickness as follows:

$$P = [(t/d)^2 - (6h_gFL_G)/(SELd^2)]/[(ZC)/(SE) + (6h_gdD)/(SELd^2)]$$

where

F is the gasket preload (=60 lb/inch)

L_G is the average length of the gasket groove, measured along the center of the groove, 210.03 in

and the remaining constants are as previously defined.

putting the values into this equation, get

$$P = \{((2.435-.003)/29.070)^2 - 6(1.09)(60)(210.03)/[(19580)(.7)(222.25)(29.070)^2]\}/\{(2.4813)(.3)/[(19580)(.7)] + 6(1.09)(29.070)(75.945)/[(19580)(.7)(222.25)(29.070)^2]\}$$

$$P = 116.3 \text{ psig. [0.90 MPa]}$$

Conclusion: The maximum allowable pressure for this vessel needs to be derated to 116.3 psig to assure that the flange is adequately sized.

b. Bolts

Sizing of the main flange bolts assumes that the loading calculated above is evenly distributed among all of the bolts and the bolts are loaded in pure tension with the minimum stress area for each bolt resisting the load.

For the main flange there are 40 1-8 bolts and 16 3/4-10 bolts for a total of 56 bolts

the minimum stress area for 1-8 bolts is .606 in² [391 mm²]

the minimum stress area for 3/4-10 bolts is .334 in² [215.5 mm²]

min req'd area per bolt, $A_r = W/SN$

where

A_r is the required area

W is the total bolt load, calculated above =

332721 lbs

S is the allowable stress =

25000 psi

N is the number of bolts =

56

$A_r = 332721/25000/56 =$

.238 in² [153.33 mm²]

Note: American standard nuts are sized to include a thread engagement length of approximately 7/8*the nominal diameter of the mating bolt and to have a width adequate to ensure that adequate bearing area is available to withstand loads greater than the mating bolt for bolts and nuts of like the same material. The nuts and nut plates used on the main flange and all other flanges and bolted covers on the subject vessel are supplied by BNL and meet or exceed this requirement. In addition, the length of all bolts are specified to insure full thread engagement with a minimum of 3 bolt threads beyond the end of the nut plate/nut. (REF. Black, P.H., Machine Design, 3rd Ed., McGraw-Hill Book Company, New York 1968).

Therefore, both bolts are adequately sized.

The stress in each of the bolts can be calculated by substituting the actual minimum stress areas for the bolts and the actual stress for the max allowable stress:

$$S(1-8) = W/N/A(1-8) = 332721/56/.606 = 9804 \text{ psi [67.60 MPa]}$$

$$S(3/4-10) = W/N/A(3/4-10) = 332721/56/.334 = 17789 \text{ psi [122.65 MPa]}$$

Similarly the maximum allowable pressure based on the bolts (pressure that would result in stresses equal to the allowable stress in the smaller (3/4-10) bolts) can be calculated as:

$$P = (SAN-FL_g)/A_p$$

where

S is the allowable pressure = 25000 psi

A is the actual stress area for 3/4-10 bolts = .334 in²

N is the number of bolts = 56

F is the gasket compression force = 60 lbs/in

L_g is the gasket length = 210.03 in

A_p is the pressure area = 29.07*75.945 = 2207.7 in²

$$P = ((25000)(.334)(56)-(60)(210.03))/(2207.7) = 206.1 \text{ psig [1.52 MPa]}$$

Conclusion: The bolts selected for the main flange are adequate.

6. Casing end bolted covers.

a. Cover thickness

The items which cover the end openings on this vessel are bolted to these ports and thus are not part of the pressure vessel. It is the responsibility of the BNL cognizant engineer to assure that the items attached are adequate for the pressure they will be accepting from this port, and that they are properly designed to seal the port. (Note: included in the items which are attached at these ports are the input and output windows. These windows must comply with BNL OH&S interim guide 1.4.2 "Glass and Plastic Window Design for Pressure Vessels". Compliance with this specification must be verified by the cognizant engineer.

For reference purposes, the minimum thickness for a bolted on flat cover of the same material composition as the end plate can be calculated as follows:

$$\begin{aligned} \text{FROM UG-34, type k :} \quad t(\min) &= d(CP/SE + 1.9WhG/SEd^3)^{1/2} \quad (\text{for a circular opening}) \\ \text{or} \quad t(\min) &= d(ZCP/SE + 6 WhG/SELd^2)^{1/2} \quad (\text{for the rectangular opening}) \end{aligned}$$

where: $t(\min)$ is the minimum allowable thickness for the cover

d is the pressure boundary diameter (circular) or shorter of the two rectangular cross dimensions)

Z is the non-circular correction factor: $Z = 3.4 - 2.4d/D$, $Z \geq 2.5$

where D is the longer of the two rectangular cross dimensions

C is the joint efficiency factor from figure UG-34, type $k = 0.3$

P is the MAWP = 145 psig

S is the allowable stress (assume same as end plate material for covers)

E is the weld joint efficiency factor (no welding in these areas, therefore $E=1.0$)

W is the total bolt load = gasket compression load + pressure load

(bolt seating force/in)(press. boundary perimeter) + (pressure)(pressure boundary area)

hG is the edge moment (defined in UG-34)

L is the perimeter of the rectangular opening measured along the bolt centers

= 37.876 in [962 mm]

For the 2.36 in dia opening cover

$$W = (40)PI(2.81) + (145)PI(2.81/2)^2 = 1252.3 \text{ lbs [5570 Nt]}$$

$$hG = 0.76 \text{ in}$$

$$t(\min) = (2.81)((.3)(145)/(19580)/(1.0) + 1.9(1252.3)(0.76)/(19580)/(1.0)/(2.81)^3)^{1/2} = .225 \text{ in [5.70 mm]}$$

For the rectangular opening

$$W = (40)(2(11.375+5.507)) + (145)((11.375 * 5.507)) = 10434 \text{ lb [46409 Nt]}$$

$$hG = .403 \text{ in}$$

$$Z = 3.4 - 2.4 (5.507/11.375) = 2.238$$

$$t(\min) = (5.507)((2.238)(.3)(145)/(19580)(1.0) + 6(10434)(.403)/(19580)(1.0)(37.876)(5.507)^2)^{1/2} = .430 \text{ in [10.92 mm]}$$

b. Bolts (Tapped holes in end covers)

There are three openings in each of the 2 flat ends of the pressure vessel. These openings are of 2 different sizes: a round and a rectangular opening.

The bolt load is the W calculated in the preceding section divided by the number of bolts attaching the cover. Since the actual bolt sizes are given, the stress in each bolt at MAWP and the max. allowable pressure for this component based on actual bolt size and max allowable stress can be calculated for

each of these by rearranging the above load calculations.

The minimum stress area per bolt is found as follows:

$$A(\min) = W/N/S$$

where: W is the total bolt load for the opening (calculated above)

N is the number of bolts retaining the load (=6 for the circular opening, =24 for the rect. opening)

S is the allowable stress = 25000 psi for L-7 bolts

The actual stress area for M12 bolts is .1182 in² [76.2 mm²] (for metric bolts refer to ISO 262)

The actual max. stress in each bolt (based on actual min stress area and MAWP) is found as follows:

$$S_m = W/N/A$$

The max. allowable pressure (based on actual stress area and allowable stress) on these ports is found as follows:

$$P_m = (SNA - FL_s)/A_p$$

where: S_m is the actual max .stress (based on actual bolt size and MAWP)

P_m is the max. allowable pressure (based on actual bolt size and max. allowable stress)

S is the max.allowable stress

N is the number of bolts

A is the actual stress area for M-12 bolts

F is the sealing force/inch for the flange

L_s is the seal perimeter

A_p is the seal pressure area

for the circular openings

$$A(\min) = 1252.3/6/25000 = .0083 \text{ in}^2 [5.39 \text{ mm}^2]$$

$$S_m = (1252.3)/(6)/(.1182) = 1766 \text{ psi} [12.18 \text{ MPa}]$$

$$P_m = ((25000)(6)(.1182) - (40)\pi(2.81))/(\pi(2.81)^2/4) = 2802 \text{ psig} [19.42 \text{ MPa}]$$

for the rectangular openings

$$A(\min) = 10434/24/25000 = .0174 \text{ in}^2 [11.22 \text{ mm}^2]$$

$$S = (10434)/(24)/(.1182) = 3678 \text{ psi} [25.36 \text{ MPa}]$$

$$P = ((25000)(24)(.1182) - (40)(2)(5.507+11.375))/((5.507)(11.375)) = 1111 \text{ psig} [7.76 \text{ MPa}]$$

Note: The M-12 bolts used for these ports have a nominal diameter of 12 mm (.472 in). In order to insure that the tensile stresses in the end plate threads are such that failure of the bolt would occur prior to failure of the threads. This can be accomplished by noting that the end plate threads are loaded in shear and that the allowable stress in shear is approximately 1/2 the tensile allowable stress, then if the minimum engagement length of the threads is sized so this is true, then the minimum engagement length of the threads can be taken as the length that satisfies this condition and thus, if this condition is satisfied and the bolt is found to be adequate then the thread engagement length is adequate. (REF. Black, P.H., Machine Design, 3rd Ed., McGraw-Hill Book Company, New

York 1968).

$L_e \geq .35 d (S_{a,b}/S_{a,t})$;

where $S_{a,b}$ is the allowable stress for the bolt material
 $S_{a,t}$ is the allowable stress for the end plate material

since this is an approximate relationship, use an factor of .5
 instead of .35 for added margin of safety.

$$L_e = (.5)(.472)(25000)/(19580) = .301 \text{ in [7.65 mm]}$$

The actual engagement length for these threads was measured to be minimum of 0.5 in, except that the entry section of the threads for 2 of the bolts on one of the rectangular openings were found to be damaged so that only .25 in of engagement length was available. However, considering the large margin that the bolts have when the actual stress at MAWP is compared with the allowable stress, it is concluded that, in total, the engagement length for these threads is adequate.

Conclusion: The M12 bolts used for these ports are adequate.

7. Casing end flanged ports

a. Port tube thickness {ref. Section VIII UG-27, UG-31}

$t(\min)$ is greater of $t(\min) = PR/(SE - 0.6P)$ {Eqn. 1, UG-27} or $t(\min) = PR/(2SE + 0.4P)$ {Eqn. 2, UG-27}

where: $t(\min)$ is the minimum allowable wall thickness { $t(\min,1)$ & $t(\min,2)$ are calculated values of $t(\min)$ based on eqn 1 or 2 respectively}

P is the maximum working pressure = 145 psi
 R are the tubes' inner radii = 1.9325, .6325 in
 S is the allowable stress = 19580 psi
 E is a joint efficiency factor = 0.7
 (Assume no radiographic inspection: E = .7 per Section VIII, subsection UW-12, Table UW-12)

for the 1.9325 radius tube (4.21 in dia port)

$$t(\min,1) = (145)(1.9325)/[(19580)(.7) - (0.6)(145)] = 0.021 \text{ in [.533 mm]}$$

$$t(\min,2) = (145)(1.9325)/[(2)(19580)(.7) + (0.4)(145)] = 0.010 \text{ in [.254 mm]}$$

therefore $t(\min) = 0.021 \text{ in [.533 mm]}$ add .003 for corrosion allowance, $t(\min) = .024 \text{ in [0.60 mm]}$
 actual wall th. = 0.1725 in [4.38 mm]

for the .6325 radius tube (1.49 in dia port)

$$t(\min,1) = (145)(.6325)/[(19580)(.7) - (0.6)(145)] = 0.007 \text{ in [.175 mm]}$$

$$t(\min,2) = (145)(.6325)/[(2)(19580)(.7) + (0.4)(145)] = 0.003 \text{ in [.076 mm]}$$

therefore $t(\min) = 0.007 \text{ in}$; add .003 for corrosion allowance, $t(\min) = .010 \text{ in [0.25 mm]}$
 actual wall th. = 0.1125 in [2.86 mm]

By rearranging the terms of the equation resulting in the max. wall thickness and substituting the actual wall thickness for the min wall, the maximum allowable pressure can be solved for as

$P_{max} = tSE/(R + .6t)$ where t is the actual wall thickness - .003 in for corrosion allowance

$$P_{max} = (.1695)(19580)(.7)/(1.9325 + .6(.1695)) = 1142 \text{ psig [7.98 MPa]} \text{ for the 4.21 in dia port}$$

$$P_{max} = (.1095)(19580)(.7)/(.6325 + .6(.1095)) = 2150 \text{ psig [14.92 MPa]} \text{ for the 1.49 in dia port}$$

Similarly, the max stress based on MAWP and actual wall thickness can be calculated as

$$S_{max} = P(R+.6t)/tE$$

$$S_{max} = (145)(1.9325+.6(.1695))/(.1695)(.7) = 2486 \text{ psi [17.14 MPa]} \text{ for the 4.21 in dia port}$$

$$S_{max} = (145)(.6325+.6(.1095))/(.1095)(.7) = 1321 \text{ psi [9.11 MPa]} \text{ for the 1.49 in dia port}$$

Conclusion: These port tube thicknesses are adequate.

b. Flange thicknesses {ref. Section VIII UG-34}

The minimum flange thickness is calculated by equation (2) in UG-34:

$$t(\min) = d * \sqrt{CP/SE + 1.9Wh_G/SEd^3}$$

where

t(min) is the minimum allowable wall thickness

P is the maximum working pressure=

145 psi

d is the flanges' pressure diameters =

6.375, 2.375 in

S is the allowable stress=

19580 psi

E is the joint efficiency=

1.0 (4.21 in port), 0.7 (1.49 in port)

(Assume no radiographic inspection: E= .7 per Section VIII,

subsection UW-12, Table UW-12, flange is integral

with tube for the 4.21 in port, i.e. not welded)

C is the joint factor {Table UG-34, type k} =

0.3

W = Total bolt load (gasket compression load + pressure load)

Gasket compression load = $60 \times \text{PI} \times 6.375 + 145 \times \text{PI} \times 6.375^2/4 = 5830 \text{ lbs}$, 4.21 in dia port

Gasket compression load = $40 \times \text{PI} \times 2.375 + 145 \times \text{PI} \times 2.375^2/4 = 940.8 \text{ lbs}$, 1.49 in dia port

h_G is the gasket moment arm as defined in figure UG-34 (k) = .354 in (4.21 in port), .778 in (1.49 in port)

for the 4.21 in port

$$t(\min) = 6.375 * \sqrt{(.3)(145)/[(19580)(1.0)] + (1.9)(5830)(.354)/[(19580)(1.0)(6.375)^3]}$$

$$t(\min) = .349 \text{ in} + .003 \text{ for corrosion allowance} = .352 \text{ in [8.94 mm]}$$

$$\text{actual thickness} = .715 \text{ in [18.16 mm]}$$

As above the eqn for t(min) can be rearranged to solve for the max allowable pressure based on actual flange thickness and allowable stress or the max stress based on actual flange thickness and MAWP as follows:

$$P_{max} = [(t/d)^2 - 1.9 h_G(\text{PI})dF/SEd^3]/(C/SE + 1.9(\text{PI})d^2h_G/4/SEd^3)$$

where t is the actual flange thickness - .003 in for corrosion allowance

F is the compression load of the gasket (force/unit length)

all other variables are as above

$$P_{max} = [(.712/6.375)^2 - 1.9(\text{PI})(6.375)(60)(.354)/(19580)(1.0)(6.375)^3]/[(.3)/(19580)(1.0) + 1.9(\text{PI})(6.375)^2(.354)/4/(19580)(1.0)(6.375)^3]$$

$$P_{max} = 630 \text{ psig [4.44 MPa]}$$

$$S_{max} = [CP + 1.9Wh_G/d^3]/(t/d)^2E$$

$$S_{max} = [(.3)(145) + 1.9(5830)(.354)/(6.375)^3]/(.712/6.375)^2(1.0)$$

$$S_{max} = 4701 \text{ psi [32.41 MPa]}$$

similarly for the 1.49 in port

$$t(\text{min}) = 2.375 \sqrt{[(.3)(145)/[(19580)(.7)] + (1.9)(940.8)(.778)/[(19580)(.7)(2.375)^3]}$$

$$t(\text{min}) = .246 \text{ in} + .003 \text{ for corrosion allowance} = .249 \text{ in [6.33 mm]}$$

$$\text{actual thickness} = .515 \text{ in [18.16 mm]}$$

$$P_{max} = [(.512/2.375)^2 - 1.9(PI)(2.375)(40)(.778)/(19580)(0.7)(2.375)^3]/[(.3)/(19580)(0.7) + 1.9(PI)(2.375)^2(.778)/4/(19580)(0.7)(2.375)^3]$$

$$P_{max} = 766 \text{ psig [5.38 MPa]}$$

$$S_{max} = [(.3)(145) + 1.9(940.8)(.778)/(2.375)^3]/(.512/2.375)^2(0.7)$$

$$S_{max} = 4528 \text{ psi [31.22 MPa]}$$

Conclusion: These port flange thicknesses are adequate.

c. Bolts

Using the calculated total bolt loads (from calculation 7b, above) dividing this by the number of bolts get the individual bolt loads. Using the allowable stress in each bolt of 25000 psi find the minimum bolt root areas.

for the 4.21 in port

$$A_{min} = 5830/8/25000 = .029 \text{ in}^2 [18.81 \text{ mm}^2]$$

the actual stress area for M16 bolts is .223 in²

for the 1.49 in port

$$A_{min} = 940.8/4/25000 = .009 \text{ in}^2 [6.07 \text{ mm}^2]$$

the actual stress area for M16 bolts is .223 in²

The maximum allowable pressure based on the actual stress area and allowable stress and the maximum stress in each bolt based on MAWP and actual stress area can be calculated by manipulating the above equations to get:

$$P_{max} = (SNA - F(PI)d)/[(PI)d^2/4]$$

$$S_{max} = W/N/A$$

where: N is the number of bolts
F is the gasket sealing force/unit length
d is the pressure boundary diameter for the port
all other variables are as defined above

for the 4.21 in port

$$P_{\max} = ((25000)(8)(.223) - (60)(\pi)(6.375)) / [(\pi)(6.375)^2/4] = 1360 \text{ psig [9.38 MPa]}$$

$$S_{\max} = (5830)/(8)/(.223) = 3268 \text{ psi [22.53 MPa]}$$

for the 1.499 in port

$$P_{\max} = ((25000)(4)(.223) - (40)(\pi)(2.375)) / [(\pi)(2.375)^2/4] = 4966 \text{ psig [34.25 MPa]}$$

$$S_{\max} = (940.8)/(4)/(.223) = 1055 \text{ psi [7.27 MPa]}$$

Conclusion: These bolts are adequate, for both flanges.

8. Cylinder flanged ports

There are 4 different size flanged ports on the cylindrical casing of the subject discharge cell. The data for these ports is as follows:

Cylinder Casing Flanged Ports

	Type A	Type B	Type C	Type D
O.D., in	1.490	8.200	2.665	6.650
Bolt Circle Diameter, in	3.930	9.291	3.543	7.559
Flange Thickness, in	0.515	0.775	0.540	0.472
Tube Thickness, in	0.1125	0.360	0.3075	0.372
Pressure Diameter, in	2.375	8.000	2.250	6.250
# of Bolts	4	18	8	12
Bolt Size	M16	M12	M8	M8
Bolt Root (stress) area, in ²	.2230	.1182	.0509	.0509
Calculated hg, in	.778	.646	.646	.655
Joint efficiency (tube/shell)		.7	.7	.7
Joint efficiency (tube/flange)	.7	1.0	1.0	1.0
				(tube/flange machined from one pc for ports B, C and D; welded from 2 pcs for port A)
Gasket sealing force, lb/in	40	40	60	60

a. Port tube thickness {ref. Section VIII UG-27, UG-31}

Using the methods indicated in section 7.a, above, the minimum wall thickness, max allowable pressure and maximum stress at MAWP for these tubes can be calculated. The results for these ports are tabulated below.

	<u>Type A</u>	<u>Type B</u>	<u>Type C</u>	<u>Type D</u>
t(min), in [mm]	.010 [0.25]	.043 [1.09]	.014 [0.35]	.034 [0.87]
actual wall th., in [mm]	.1125 [2.86]	.360 [9.14]	.3075 [7.81]	.372 [9.45]
max pressure, psig [MPa]	2150 [14.92]	1237 [8.63]	3456 [23.93]	1593 [11.09]
max stress, psi [MPa]	1321 [9.11]	2294 [15.82]	822 [5.66]	1782 [12.329]

Conclusion: These port tube thicknesses are adequate.

b. Flange thicknesses

Using the methods indicated in section 7.b, above, the minimum flange thickness, max allowable pressure and maximum stress at MAWP for these flanges can be calculated. The results for these flanges are tabulated below.

	<u>Type A</u>	<u>Type B</u>	<u>Type C</u>	<u>Type D</u>
total bolt load, lb [Nt]	940.8 [4185]	8294 [36891]	1001 [4451]	5627 [25027]
t(min), in [mm]	.249 [6.33]	.458 [11.64]	.201 [5.10]	.382 [9.71]
actual wall th., in [mm]	.515 [13.08]	.775 [19.69]	.540 [13.72]	.472 [11.99]
max pressure, psig [MPa]	766 [5.38]	428 [3.05]	1468 [10.23]	228 [1.68]
max stress, psi [MPa]	4528 [31.22]	6806 [46.93]	2657 [18.32]	12819 [88.38]

Conclusion: These port flange thicknesses are adequate.

c. Bolts

Using the methods indicated in section 7.c, above, the minimum bolt stress area, max allowable pressure and maximum stress at MAWP for the bolts used on these ports can be calculated. The results for these bolts are tabulated below.

	<u>Type A</u>	<u>Type B</u>	<u>Type C</u>	<u>Type D</u>
A(min), in ² [mm ²]	.009 [6.07]	.018 [11.89]	.005 [3.23]	.019 [12.10]
actual stress area, in ² [mm ²]	.223 [143.9]	.1182 [76.26]	.0509 [32.84]	.0509 [32.80]
max pressure, psig [MPa]	4966 [34.34]	1038 [7.26]	2454 [16.92]	459 [3.17]
max stress, psi [MPa]	1055 [7.27]	3898 [26.88]	2457 [16.94]	9212 [63.51]

Conclusion: The bolts for these ports are adequate.

d. Stiffening of Type "D" Port

On the "D" port, there is an undocumented addition to the port tube that on initial examination was suspected to be a plug weld, but subsequent closer examination was found to be a "stiffening" ring added to the tube wall of this port. The apparent purpose of this addition was to stiffen the wall of the tube where the interior had been counter-bored to allow for an o-ring seal, and to prevent excessive moment on the port flange. From the analyses in section 8(a), above, the minimum required wall thickness of this tube was found to be .034 in [.86 mm]. The actual wall thickness under the counter-bore is .132 in [3.36 mm]. In addition, the analyses of section 8.(b), above, indicates that the flange design is adequate. Therefore, the stiffening ring is unnecessary, but does not have any harmful affect on the port. Note: Although not used as such herein, this stiffening ring is not an acceptable method for increasing wall thickness.

Conclusion: The stiffening of type "D" port is adequate.

9. Inner vacuum vessel cylindrical shell

Allowable external pressure for a circular cylinder is determined by the method outlined in section UG-28 of 1998 ASME Section VIII Boiler and Pressure Vessel Code as follows:

a. determine $Do/t = 10.16/.235 = 43.23$

where Do is the outer diameter of the shell
t is the minimum shell thickness

b. determine $L/Do = (44.81 - 2 \cdot .787)/10.16 = 4.256$

where L is the internal length of the shell between supports (end plates)

c. since Do/t is greater than 10, UG-28 requires acquisition of factor "A" from figure G, subpart 3 of 1998 ASME Section II Part D of the Boiler and Pressure Vessel Code. From this chart find $A=.001$

d. the material under consideration is the Russian stainless steel alloy 12X18H10T. Properties for this material are not available in Section II, but this material very closely resembles US austenitic stainless steels (e.g. 321). Therefore use figure HA-2 to obtain factor "B". The modulus of elasticity for this material is assumed to be 28×10^6 psi. Using a design temperature of 100°F, from this figure find $B= 9,750$.

e. now solve for the maximum allowable pressure "Pa" using the following equation:

$$Pa = 4B/[3(Do/t)] = 4(9750)/[3(43.23)] = 300.72 \text{ psi } [2.073 \text{ MPa}]$$

To solve for the minimum allowable wall thickness (t_r , used in calculations later in this report), follow the procedure above selecting values for t in an iterative manner and solve for Pa until the t which results in a Pa equal to the MAWP is found.

The iterative process yields $A=.000156$, $B= 7800$, and $t_r = .156$ in [3.96 mm] add .003 in for corrosion allowance, $t_{min} = .159$ in [4.04 mm], for $Pa = 159.7$ psia [1.1 MPa]

Conclusion: Since the actual pressure differential across the shell wall is 159.7 psi, the actual wall thickness is adequate.

10. Inner vacuum vessel end caps

end plate thickness

The ASME Section VIII Boiler and Pressure Vessel Code defines min. thickness calculations for unstayed flat heads, covers and blind flanges in subsection UG-34, and does not differentiate based on internal/external pressure. The equation governing for minimum thickness is (note: there is no edge moment term for this configuration):

$$t_{min} = d(ZCP/SE)^{1/2}$$

where t_{min} is the calculated minimum allowable wall thickness

Z is a factor to correct for non-circularity of the end plate

$$Z = 3.4 - 2.4(d/D) \text{ and } Z \leq 2.5$$

d is the minimum cross dimension of the flat end plate (= 8.57 in [217.68 mm])

D is the long dimension of the flat end plate (=10.16 in [258.06 mm])

C is the joint configuration factor (= .33 from UG-34 (h))

S is the allowable stress (=19580 psi [135 MPa])

E is the joint efficiency factor (= 0.60 {from Table UW-12, for type 3, joint category C without radiographic inspection})

for this end plate Z the lesser of 2.5 or $3.4 - 2.4(8.57/10.16) = 1.376$

based on a design maximum pressure of 159.7 psi (1.1 MPa)

$$t_{min} = 8.57[(1.376)(.33)(159.7)/\{(19580)(.6)\}]^{1/2} = .673 \text{ in } [17.09 \text{ mm}]. \text{ Add .003 in for corrosion allowance and } t_{min} = .676 \text{ in } [17.17 \text{ mm}]$$

The actual minimum thickness of this part is .787 in. [20 mm]

Conclusion: This wall thickness is adequate.

11. Inner Vacuum vessel base (incl. bolts)

a. Base

The base of the vacuum vessel consists of several components as follows: a blind flange, a support plate for the pressure boundary foil, the foil itself, a base and cover clamping flange for the support plate, and bolts. Since this assembly can only be subjected to external pressure, the base and cover clamping flange are not subjected to forces due to pressure except to transfer the force to the blind flange from the foil and support plate, and thus need not be analyzed herein. The other components are analyzed as follows:

i. Flange

The ASME Section VIII Boiler and Pressure Vessel Code defines min. thickness calculations for unstayed flat heads, covers and blind flanges in subsection UG-34, and does not differentiate based on internal/external pressure. The equation governing for minimum thickness is (note: there is no edge moment term for this configuration):

$$t_{min} = d(ZCP/SE)^{1/2}$$

where t_{min} is the calculated minimum allowable wall thickness

Z is a factor to correct for non-circularity of the end plate

$$Z = 3.4 - 2.4(d/D) \text{ and } Z \leq 2.5$$

d is the minimum cross dimension of the flat end plate (= 7.87 in [200 mm])

D is the long dimension of the flat end plate (=44.41 in [1128 mm])

C is the joint configuration factor (= .33 from UG-34 (h))

S is the allowable stress (=19580 psi [135 MPa])

E is the joint efficiency factor (= 0.60 {from Table UW-12, for type 3, joint category C without radiographic inspection})

for this end plate Z the lesser of 2.5 or $3.4 - 2.4(7.87/44.41) = 2.97$; therefore use 2.5.

based on a design maximum pressure of 159.7 psi (1.1 MPa)

$$t_{min} = 7.87[(2.5)(.33)(159.7)/\{(19580)(.6)\}]^{1/2} = .833 \text{ in [21.17 mm]}. \text{ Add .003 in for corrosion allowance, then } t_{min} = .836 \text{ in [21.23 mm]}.$$

The actual minimum thickness of this part is .787 in. [20 mm]. Therefore the pressure rating for this component must be reduced. A quick inspection of the formula for t_{min} given above shows that t_{min} is proportional to the square root of P . Rearranging this relation indicates that the maximum allowable pressure is then proportional to t_{min} squared. Therefore

$$P_{maxallowable} = P_{design} (t_{minactual}/t_{mindesign})^2 = 159.7[(.787 -.003)/(.833)]^2 = 141.46 \text{ psia [.975 MPa]}$$

(In terms of the main vessel, the maximum allowable pressure is $141.46 - 14.7 = 126.76$ psig [.975 MPa])

Conclusion: This wall thickness is adequate if the main vessel design pressure is reduced to no more than 126.76 psig [.975 MPa].

ii. Support plate for the pressure boundary foil

This item is not covered explicitly in the Code but can be analyzed using simple engineering beam calculations. A limiting (conservative) simple case would be to calculate the maximum stress of a uniformly loaded plate fixed on the 2 long edges and free on the 2 short edges. This could be modeled (conservatively) by taking a representative rectangular section of the plate and performing an FEA. The results of this analysis are shown in figure 11-1, below.

The maximum stresses obtained in this analysis are well below the allowable stress.

Conclusion: The support plate for the pressure boundary foil is adequate.

iii. Pressure boundary foil

The pressure boundary foil is a thin titanium layer which allows x-rays to pass through while maintaining a mechanical barrier between the pressure vessel atmosphere and the internal vacuum chamber vacuum. This layer has been subjected to a pressure differential of greater than 225 psi during acceptance tests. This load caused permanent deformation of the foil into an array of spherical dimples corresponding to the array of holes in the support plate. Such permanent deformation indicates that at such pressure the stresses in the foil exceeded yield stress but not the ultimate stress for the material. It is probable that deformation of the foil served to relieve stress concentrations in the foil to the point where stresses were reduced to the yield point whereupon continued permanent deformation ceased. Since the actual operating pressure is significantly less than the above test level, stresses in the foil at operating pressure should be less than the yield stress. A simple calculation for the stresses in this foil is not feasible, but by analyzing a single dimple using finite element analysis, the expected stresses can be quantified and, with the application of appropriate safety margins, a max allowable pressure determined. Measurements indicate that the dimple depth is .012 in [.30 mm] maximum. Since the dimple represents a stretching (and correspondingly a thinning) of the foil which is calculated, the FEA model assumes that the initial thickness of the foil is reduced by the ratio of the areas of the original flat foil over the support plate hole to the surface area of a dimple in the support plate hole, which has been calculated as 0.90. The model assumes axi-symmetry and fixed support of the foil along the radiused edge of the support plate hole, and a uniform pressure loading of 159.7 psi on the inside surface of the dimple. The results are shown in figure 11-2. As can be seen, the maximum stress was found to be 74741 psi [519 MPa]. Since the allowable stress for this material is unknown, this stress analysis can not be used as an absolute measure of the adequacy of this material. However, since (A) this material has already been stressed by 150% of its maximum operating pressure, (B) the component is entirely encompassed by the main pressure vessel, (C) the range of yield stress values reported for commercially pure titanium varies from 25000 to 70000 psi, and (D) failure of the foil would not result in any personnel safety issue, this material may be assumed to be adequate for its function. The foil must be inspected periodically, however, to insure that the further yielding does not occur and that no other defects are noted.

Conclusion: The pressure boundary foil is adequate, but requires periodic inspection.

b. Bolts

There are 50 M8 bolts used to clamp the pressure boundary foil and its support plate to the large rectangular flange of the inner vacuum vessel. Since only external pressure on the foil can be present, the maximum bolt load is due only to tightening of the bolts to effect the appropriate seating forces on the flange and support plate seals.

W is the total bolt load = gasket compression load = (bolt seating force/in)(press. boundary perimeter)

The bolt sealing force is designed to be 22 lb/in [3853 Nt/m]

$$W = 22 * (2*42.65 + 2*5.71) = 2127.84 \text{ lbs [9465 Nt]}$$

The minimum stress area per bolt is found as follows:

$$A(\text{min}) = W/N/S$$

where: W is the total bolt load for the opening (calculated above)

N is the number of bolts retaining the load (= 50 for this rectangular opening)

S is the allowable stress = 25000 psi [172 MPa] for L-7 bolts

The actual stress area for M8 bolts is .0509 in² [32.8 mm²] (for metric bolts refer to ISO 262)

The actual max. stress in each bolt (based on actual min stress area and MAWP) is found as follows:

$$S_m = W/N/A$$

$$A(\text{min}) = 2127.84/50/25000 = .0017 \text{ in}^2 [.043 \text{ mm}^2]$$

$$S_m = (2127.84)/(50)/(.0509) = 836.1 \text{ psi [5.76 MPa]}$$

Note: The M-8 bolts used for this flange have a nominal diameter of 8 mm (.315 in). It is desirable to insure that the tensile stresses in the end plate threads are such that failure of the bolt would occur prior to failure of the threads. This can be accomplished by noting that the end plate threads are loaded in shear and that the allowable stress in shear is approximately 1/2 the tensile allowable stress, then if the minimum engagement length of the threads is sized so this is true, then the minimum engagement length of the threads can be taken as the length that satisfies this condition and thus, if this condition is satisfied and the bolt is found to be adequate then the thread engagement length is adequate. (REF. Black, P.H., Machine Design, 3rd Ed., McGraw-Hill Book Company, New York 1968).

$$L_e \geq .35 d (S_{a,b}/S_{a,t}); \quad \text{where } S_{a,b} \text{ is the allowable stress for the bolt material}$$

$$S_{a,t} \text{ is the allowable stress for the end plate material}$$

since this is an approximate relationship, use an factor of .5 instead of .35 for added margin of safety.

$$L_e \geq (.5)(.315)(25000)/(19580) = .201 \text{ in [5.11 mm]}$$

The actual engagement length for these threads was measured to be minimum of 0.26 in [6.60 mm],

Conclusion: The M8 bolts used for this flange are adequate.

12. Inner vacuum vessel ports (openings)

a. Shell Openings{ref. Section VIII UG-37(d)}

$$A = .5 [d t_r F + 2 t_n t_r F (1 - f_{r1})];$$

$$A1 = \text{the larger of:} \quad A1 = d (E1 t - F t_r) - 2 t_n (E1 t - F t_r) (1 - f_{r1})$$

$$\text{or } A1 = 2 (t + t_n) (E1 t - F t_r) - 2 t_n (E1 t - F t_r) (1 - f_{r1})$$

where: A is the required area for reinforcement

A1 is the available area for reinforcement due to oversizing of shell

t_r is the required shell thickness . Therefore t_r = .159.

F is a factor to compensate for non-radial holes (all holes in shell are radial, F=1.0)

t_n is the nozzle thickness of the port (note: neglect the nozzle thickness, i.e. t_n = 0)

f_{r1} is a factor to compensate for different materials used for nozzle compared to shell. f_{r1}=1 for all nozzles in shell)

E1 is a factor to compensate for seam welds through holes. E1 =1.0 for all holes in the shell.

t is the actual thickness of the shell = .235 in

d is the port opening diameter (in inches)

The equations for A1 thus reduce to:

$$A1 = \text{the larger of:} \quad A1 = d (t - t_r) \quad \text{or} \quad A1 = 2 t (t - t_r)$$

for all of the shell openings, d > 2t, therefore use A1 = d (t - t_r)

Therefore, setting A1 ≥ A, putting in the above values, if true then no reinforcing is needed.

which results in t ≥ 3/2 t_r. From above t = .235" and t_r = .159".

This is not true, For this to be true, t would have to be $\geq .2385"$. Since the required wall thickness is directly proportional to the wall minimum wall thickness, the the maximum operating pressure would have to be reduced to $P_{max} = 159.7 \cdot .235 / .2385 = 157.36$ psia [1.085 MPa]

Assuming the max operating pressure is appropriately reduced, now check that spacing is adequate for multiple holes. Centers of adjacent holes should be spaced at a distance greater than the sum of their diameters, else reinforcement must be greater. {ref. Section VIII UG-42}

There is one hole pair to be considered as indicated above.

The pair: $5.98 \text{ in} + 1.96 \text{ in.} \leq 13.750 \text{ in.}$ These holes are adequately reinforced

Conclusion: The maximum operating pressure must be reduced to 157.36 psia [1.085 MPa] in order for the inner vacuum vessel shell openings to be adequately reinforced.

b. End Plate Opening

The single opening in each of the end plates is not centrally located and the end plates are not round, therefore UG-39(c)(3) applies. This section, however, provides no specific calculation method, but rather leaves it to the design engineers judgement to provide a method that insures that the stresses in the part do not exceed those allowed by the Code. It is apparent, however, that the stresses would be comparable to those in a circular endplate of the same diameter as the circular portion of the subject end plate with a comparably sized opening, centrally located, for which UG-39(c)(1) provides a design procedure. It is reasonable, therefore, to calculate the stress in such a case and require that the end plate be sized such that the stress so calculated is less than the allowable stress as defined in 14-3(g), Appendix 14 of Section VIII. The procedure followed therein is rather complicated in that the method of Appendix 14 modifies the methods for calculating flange stresses in Appendix 2, 2-6, which are further modified for external pressure by 2-11. The flat head in consideration herein has a centrally located opening without a hub. The procedure for calculating the stresses in this case are as follows:

1. define nomenclature as follows:

P = external design pressure	= 159.7 psia [1.10 MPa]
A = outside dia of flat head and shell	= 10.16 in [258.06 mm]
B_s = inside diameter of shell	= 9.69 in [246.13 mm]
B_n = diameter of central opening	= 6.495 in [165 mm]
t = flat head minimum thickness	= .787 in [19.99 mm]
M_H = moment acting at shell/head juncture	(calculated below)
B_1 = average diameter of shell	(calculated below)
g_0 = nominal shell thickness	= .235 in [5.969 mm]
g_1 = shell thickness below flat head	= .235 in [5.969 mm]
H_T = diff. between total pressure force and pressure force on opening dia	(calculated below)
H_D = pressure force on opening dia	(calculated below)
h_0 = calculation factor	(calculated below)
h_T = radial arm from opening diameter to bolt circle dia =	.493 in [12.52 mm]
(note: for external pressure on a flat head, according to Appendix 14-3(a), h_g (as defined in 2-6) = 0 and $h_D = h_T$)	
F = integral flange factor	(from figure 2-7.2, found below)
K = integral flange factor	(calculated below)
V = integral flange factor	(from figure 2-7.3, found below)
f = integral flange factor	(from figure 2-7.6, found below)

M_o = total moment acting on the flat head	(calculated below)
$(E)^\ast$ = intermediate calculation	(calculated below)
X_1 = intermediate calculation	(calculated below)
Z = intermediate calculation	(calculated below)
Z_1 = intermediate calculation	(calculated below)
S_f = allowable stress	= 19580 psi [135 MPa]
S_{HS} = longitudinal hub stress in shell	(calculated below)
S_{RS} = radial stress at OD	(calculated below)
S_{TS} = tangential stress at OD	(calculated below)
S_{HO} = longitudinal hub stress at opening	(calculated below)
S_{RO} = radial stress at opening	(calculated below)
S_{TO} = tangential stress at opening	(calculated below)
$S_T^\ast$ = tangential flange stress	(calculated below)
$S_R^\ast$ = radial flange stress	= 0 (no hub)
$S_H^\ast$ = longitudinal hub stress	= 0 (no hub)

2. calculate operating moment

from 2-11, equation (10) (and as noted above, setting $h_g = 0$ and $h_D = h_T$)

$$M_o = (H_T + H_D)h_T = [A^2/4]P h_T = [(10.16)^2/4](159.7)(.493) = 6383 \text{ lb-in [721.16 Nt-m]}$$

3. calculate stress per 14-3(a)(2) and 2-7

as indicated above $S_H^\ast = 0$ and $S_R^\ast = 0$ for the present configuration and 2-7(b)

in accordance with 14-3(a)(2) find K

$$K = A/B_n = (10.16)/(6.495) = 1.564, \text{ then use this K, find Y from fig. 2-7.1, } Y = 4.7$$

then solve for $S_T^\ast$ in accordance with 2-7(b)

$$S_T^\ast = YM_o/[t^2 B_n] = (4.7)(6383)/[(.787)^2(6.495)] = 7458 \text{ psi [51.42 MPa]}$$

4. calculate $(E)^\ast$

from 14-3(b)(2)

$$(E)^\ast = (B_n/t)S_T^\ast = (6.495/.787)(7458) = 61550 \text{ psi [424.4 MPa]}$$

5. calculate $(E)^\ast/M_o = 61550/6383 = 9.6428$

6. calculate M_H per 14-3(d)

using:

$$h_0 = (B_s g_0)^{1/2} = ((9.69)(.235))^{1/2} = 1.509 \text{ in [39.60 mm]}$$

$$V \text{ from figure 2-7.3, with } g_1/g_0 = 1.0 \text{ (for any value of } h/h_0), = .550103$$

$$F \text{ from figure 2-7.2, with } g_1/g_0 = 1.0 \text{ (for any value of } h/h_0), = .908920$$

$$\text{and } B_1 = B_s + g_0 = 9.69 + .235 = 9.925 \text{ in [252.1 mm]}$$

$$M_H = (E)^\ast/[1.74h_0V/(g_0^3 B_1) + ((E)^\ast/M_o)(1 + Ft/h_0)]$$

$$= (61550)/[(1.74)(1.509)(.550103)/((.235)^3(9.925)) + (9.6428)(1 + (.90892)(.787)/1.509)]$$

$$= 2420.6 \text{ lb-in [273.48 Nt-m]}$$

7. calculate X_1

in accordance with 14-3(e)

$$X_1 = [M_O - M_H(1 + Ft/h_0)]/M_O$$

$$X_1 = [(6383) - (2420.6)(1 + (.90892)(.787)/(1.509))]/(6383) = .4410$$

8. calculate the head/shell juncture stresses

using the above given and calculated values, $f=1$ (from fig 2-7.6) and Z , calculated as

$$Z = (K^2 + 1)/(K^2 - 1) = (1.564^2 + 1)/(1.564^2 - 1) = 2.383$$

$$S_{HS} = (X_1) (E) (1.10)h_0f/[(g_1/g_0)^2 B_s V]$$

$$= (.4410) (61550)(1.10)(1.509)(1)/[(.235/.235)^2(9.69)(.550103)]$$

$$= 8452.4 \text{ psi [58.28 MPa]}$$

$$S_{RS} = [1.91M_H(1 + Ft/h_0)/(B_s t^2) + 0.64FM_H/(B_s h_0 t)]$$

$$= [1.91(2420.6)(1 + (.90892)(.787)/(1.509))/((9.69)(.787)^2)$$

$$+ 0.64(.90892)(2420.6)/((9.69)(1.509)(.787))]$$

$$= 1257.9 \text{ psi [8.67 MPa]}$$

$$S_{TS} = (X_1) (E) t/B_s - [0.57(1 + Ft/h_0)M_H/(B_s t^2)] + [0.64FZM_H/(B_s h_0 t)]$$

$$= (.4410) (61550) (.787)/(9.69) - [0.57(1 + (.90892)(.787)/(1.509))(2420.6)/((9.69)(.787)^2)] +$$

$$[0.64(.90892)(2.383)(2420.6)/((9.69)(1.509)(.787))]$$

$$= 2157.3 \text{ psi [14.87 MPa]}$$

9. calculate the opening head juncture stresses

using above given and calculated values and Z_1 , calculated as

$$Z_1 = (2K^2/(K^2 - 1)) = (2*1.564^2/(1.564^2 - 1)) = 3.383$$

$$S_{HO} = X_1 S_H^* = 0 \text{ (since } S_H^* = 0 \text{ per 14-3(a)(2) and 2-7(b))}$$

$$S_{RO} = X_1 S_R^* = 0 \text{ (since } S_R^* = 0 \text{ per 14-3(a)(2) and 2-7(b))}$$

$$S_{TO} = X_1 S_T^* + [0.64FZ_1 M_H/(B_s h_0 t)]$$

$$= (.4410)(7458) + [0.64(.90892)(3.830)(2420.6)/((9.69)(1.509)(.787))]$$

$$= 3702.9 \text{ psi [25.53 MPa]}$$

10. determine whether stresses are within the allowable range.

14-3(g) of the Code directs that the stresses determined above meet the allowable stresses per 2-8 of the Code. 2-8 requires that S_H^* , S_{HO} , and S_{HS} be less than $1.5S_f$, and that S_R^* , S_{RS} , S_{RO} , S_T^* , S_{TS} , and S_{TO} be less than S_f . 2-8 also requires that $(S_H^* + S_R^*)/2$ be less than S_f , $(S_{HO} + S_{RO})/2$ be less than S_f , $(S_{HS} + S_{RS})/2$ be less than S_f , $(S_H^* + S_T^*)/2$ be less than S_f , $(S_{HO} + S_{TO})/2$ be less than S_f , and $(S_{HS} + S_{TS})/2$ be less than S_f .

$$1.5 S_f = (1.5)(19580) = 29370 \text{ psi [202.5 MPa]}$$

$$S_H^* = 0 \text{ psi [0 MPa]}$$

this is less than $1.5S_f$, so this satisfies the requirements

$$S_{HS} = 8452.4 \text{ psi [58.28 MPa]}$$

this is less than $1.5S_f$, so this satisfies the requirements

$$S_{HO} = 0 \text{ [0 MPa]}$$

this is less than $1.5S_f$, so this satisfies the requirements

$$S_f = 19580 \text{ [135 MPa]}$$

$S_R^* = 0 \text{ psi [0 MPa]}$	this is less than S_f , so this satisfies the requirements
$S_T^* = 7458 \text{ psi [51.42 MPa]}$	this is less than S_f , so this satisfies the requirements
$S_{RS} = 1257.9 \text{ psi [8.67 MPa]}$	this is less than S_f , so this satisfies the requirements
$S_{TS} = 2157.3 \text{ psi [14.87 MPa]}$	this is less than S_f , so this satisfies the requirements
$S_{RO} = 0 \text{ [0 MPa]}$	this is less than S_f , so this satisfies the requirements
$S_{TO} = 3702.9 \text{ psi [25.53 MPa]}$	this is less than S_f , so this satisfies the requirements
$(S_H^* + S_R^*)/2 = 0$	this is less than S_f , so this satisfies the requirements
$(S_{H0} + S_{R0})/2 = 0$	this is less than S_f , so this satisfies the requirements
$(S_{HS} + S_{RS})/2 = (8452.4 + 1257.9)/2 = 4855.15 \text{ psi [33.47 MPa]}$	this is less than S_f , so this satisfies the requirements
$(S_H^* + S_T^*)/2 = (0 + 7458)/2 = 3729 \text{ psi [25.71 MPa]}$	this is less than S_f , so this satisfies the requirements
$(S_{H0} + S_{T0})/2 = (0 + 3702.9)/2 = 1851.45 \text{ psi [12.77 MPa]}$	this is less than S_f , so this satisfies the requirements
$(S_{HS} + S_{TS})/2 = (8452.4 + 2157.3)/2 = 5304.85 \text{ psi [36.58 MPa]}$	this is less than S_f , so this satisfies the requirements

Conclusion: The inner vacuum vessel end plate openings are adequately reinforced.

c. Port tube thickness

There are 2 sizes of port tubes on the internal vacuum chamber, a 5.98 dia port and a 1.96 dia port. Both sizes are subjected to external pressure under normal operating conditions and could be subjected to internal pressure under a failure condition.

i. external pressure

Allowable external pressure for a circular cylinder is determined by the method outlined in sections UG-28 and UG-31 of 1998 ASME Section VIII Boiler and Pressure Vessel Code as follows:

a. determine $Do/t = 5.98/.124 = 48.23$ for the larger tube; $Do/t = 1.96/0.10 = 19.60$ for the smaller tube

where Do is the outer diameter of the shell
 t is the minimum shell thickness

b. determine $L/Do = (6.93)/5.98 = 1.159$ for the larger tube; $L/Do = (6.17)/1.96 = 3.148$ for the smaller tube

where L is the maximum internal length of the tube between supported areas (end cap and shell).

c. UG-28 requires acquisition of factor "A" from figure G, subpart 3 of 1998 ASME Section II Part D of the Boiler and Pressure Vessel Code. From this chart find $A = .0035$ for the larger tube; $A = .0044$ for the smaller tube.

d. the material under consideration is the Russian stainless steel alloy 12X18H10T. Properties for this material are not available in Section II, but this material very closely resembles US austenitic stainless steels (e.g. 321). Therefore use figure HA-2 to obtain factor "B". The modulus of elasticity for this material is assumed to be 28×10^6 psi. From this figure, for the larger tube: find $B = 12720$; for the smaller tube: $B = 13200$.

e. now solve for the maximum allowable pressure "Pa" using the following equations:

equation (UG-28(c)(1) applies for both tubes:

for the larger tube,

$$P_a = 4B/[3(D_o/t)] = 4(12720)/[3(48.23)] = 351.6 \text{ psi [2.42 MPa]}$$

for the smaller tube,

$$P_a = 4B/[3(D_o/t)] = 4(13200)/[3(19.60)] = 898.0 \text{ psi [6.19 MPa]}$$

Conclusion: Since the actual pressure differential across both tube walls under external pressure loading is 159.7 psi, these wall thicknesses are adequate for the design external pressure loading.

ii. internal pressure {ref. Section VIII UG-27, UG-31}

$t(\min)$ is greater of $t(\min) = PR/(SE - 0.6P)$ {Eqn. 1, UG-27}

or $t(\min) = PR/(2SE + 0.4P)$ {Eqn. 2, UG-27}

where: $t(\min)$ is the minimum allowable wall thickness { $t(\min,1)$ & $t(\min,2)$ are calculated values of $t(\min)$ based on eqn 1 or 2 respectively}

P is the maximum working pressure=

145 psi [1.0 MPa]

R are the tubes' inner radii=

2.866 in [72.80 mm], .88 in [22.35 mm]

S is the allowable stress=

19580 psi [135 MPa]

E is a joint efficiency factor =

0.7

(Assume no radiographic inspection: E= .7 per Section VIII, subsection UW-12, Table UW-12)

for the larger tube:

$$t(\min,1) = (145)(2.866)/[(19580)(.7) - (0.6)(145)] = .031 \text{ in [.79 mm]}$$

$$t(\min,2) = (145)(2.866)/[(2)(19580)(.7) + (0.4)(145)] = .015 \text{ in [.38 mm]}$$

therefore $t(\min) = .031 \text{ in [.79 mm]}$ add .003 for corrosion allowance, $t(\min) = .034 \text{ in [0.86 mm]}$
actual wall th. = 0.124 in [3.15 mm]

for the smaller tube:

$$t(\min,1) = (145)(.88)/[(19580)(.7) - (0.6)(145)] = .009 \text{ in [.23 mm]}$$

$$t(\min,2) = (145)(.88)/[(2)(19580)(.7) + (0.4)(145)] = .005 \text{ in [.13 mm]}$$

therefore $t(\min) = 0.009 \text{ in}$; add .003 for corrosion allowance, $t(\min) = .012 \text{ in [0.31 mm]}$
actual wall th. = 0.100 in [2.54 mm]

Conclusion: These port tube thicknesses are adequate.

d. Port Flanges

The flanges on the 2 different size ports of the internal vacuum chamber are normally subjected to a 1 atm external pressure, but under certain failure conditions could be subjected to an internal pressure of 145 psig. However, the Code does not differentiate between internal and external construction for these types of construction, therefore, the flange design is based on the greater differential pressure, 145 psi [1.0 MPa]

The construction of these circular flanges is covered by UG-34(c) equation (1)

$$t(\min) = d (CP/SE)^{1/2}$$

where d is the OD of the flange = 5.98 in [151.89 mm]; =1.96 in [49.78 mm]

C is the attachment method factor = .33 for both flanges {fig UG-34(h)}

P is the differential pressure = 145 psi [1.0 MPa] for both flanges
 S is the allowable stress = 19580 psi [135 MPa] for both flanges
 E is the welded joint efficiency factor (= 0.60 {from Table UW-12, for type 3, joint category C without radiographic inspection})

for the larger port:

$$t(\min) = (5.98)[(.33)(145)/\{(19580)(.6)\}]^{1/2} = .382 \text{ in add .003 for corrosion allowance, then}$$

$$t(\min) = .385 \text{ in [9.78 mm]}$$

the actual min. wall thickness is .25, therefore the allowable pressure must be derated. Rearranging terms to solve for the max allowable pressure using the actual thickness yields:

$$P_a = [t(\min)/d]^2 SE/C = [.25-.003]/5.98^2(19580)(.6)/(.33) = 60.73 \text{ psig [.52 MPa]}$$

for the smaller port:

$$t(\min) = (1.96)[(.33)(145)/\{(19580)(.6)\}]^{1/2} = .125 \text{ in add .003 for corrosion allowance, then}$$

$$t(\min) = .128 \text{ in [3.25 mm]}$$

the actual min. wall thickness is .221 in [5.61 mm], therefore this flange thickness is adequate.

Conclusion: The port flange thickness for the smaller port is adequate; the port flange minimum wall thickness for the larger port is inadequate for the worst case failure scenario. In order to accommodate this flange the pressure relief system for the internal vacuum vessel must be designed to limit the pressure in the vacuum vessel ports from exceeding 60.73 psig [.52 MPa].

e. Bolt stress

There are 3 areas where parts are bolted onto ports on the internal vacuum vessel: the large port, the end plate opening and the end cover plate opening. For the end plate opening cover and the end plate opening cover opening cap, the cover and cap see only external pressure, so that the maximum stresses in the bolts are due only to the moment generated by seal seating. For the large port the flange bolts have their maximum loading under a failure condition during which a pressure differential between the inside of the vessel and ambient would exist. The maximum such differential pressure would be 145 psi [1.0 MPa].

W is the total bolt load = gasket compression load (bolt seating force/in)(press. boundary perimeter) + pressure load (pressure differential)(pressure area)

for the end plate cover attachment bolts (o-ring sealing force = 31 lb/in)

$$W = 31 * (6.76) + 0 = 658.4 \text{ lbs [2929 Nt]}$$

for the end plate cover cap attachment bolts (o-ring sealing force = 52 lb/in)

$$W = 52 * (2.697) + 0 = 440.6 \text{ lbs [1960 Nt]}$$

for the large port flange attachment bolts (conflat flange sealing bolt torque = 26 ft-lb which translates to a load per bolt of 155 lbs for an M8 bolt (neglecting friction).

$$W = (\text{load per bolt})(\# \text{ of bolts}) + (\text{pressure load})$$

$$= (155)(16) + (145)(\pi/4)(4.54)^2 = 2480 + 2347.3 = 4827.3 \text{ lb [21472 Nt]}$$

The minimum stress area per bolt is found as follows:

$$A(\min) = W/N/S$$

where: W is the total bolt load for the opening (calculated above)

N is the number of bolts retaining the load (= 16 for the end plate cover and the large port flange,
= 8 for the end plate cover cap.)

S is the allowable stress = 25000 psi [172 MPa] for L-7 bolts

The end plate cover and the large port flange are attached using M8 bolts, the end plate cover cap is attached using M6 bolts. The actual stress area for M8 bolts is .0509 in² [32.8 mm²]. The actual stress area for M6 bolts is .0277 in² [17.89 mm²]. (for metric bolts refer to ISO 262) The actual max. stress in each bolt (based on actual min stress area and MAWP) is found as follows:

$$S_m = W/N/A$$

Then

for the end plate cover attachment bolts

$$\begin{aligned} A(\min) &= 658.4/16/25000 &= .0016 \text{ in}^2 [1.06 \text{ mm}^2] \\ S_m &= 658.4/16/.0509 &= 808.4 \text{ psi [5.62 MPa]} \end{aligned}$$

for the end plate cover cap attachment bolts

$$\begin{aligned} A(\min) &= 440.6/8/25000 &= .0022 \text{ in}^2 [1.421 \text{ mm}^2] \\ S_m &= 440.6/8/.0277 &= 1988 \text{ psi [13.81 MPa]} \end{aligned}$$

for the large port flange attachment bolts

$$\begin{aligned} A(\min) &= 4827.3/16/25000 &= .0121 \text{ in}^2 [0.31 \text{ mm}^2] \\ S_m &= 4827.3/16/.0509 &= 5927.4 \text{ psi [40.86 MPa]} \end{aligned}$$

Note: The M-8 bolts used for this flange have a nominal diameter of 8 mm (.315 in), while the M6 bolts have a nominal dia of 6 mm (.236). It is desirable to insure that the tensile stresses in the end plate threads are such that failure of the bolt would occur prior to failure of the threads. This can be accomplished by noting that the end plate threads are loaded in shear and that the allowable stress in shear is approximately 1/2 the tensile allowable stress, then if the minimum engagement length of the threads is sized so this is true, then the minimum engagement length of the threads can be taken as the length that satisfies this condition and thus, if this condition is satisfied and the bolt is found to be adequate then the thread engagement length is adequate. (REF. Black, P.H., Machine Design, 3rd Ed., McGraw-Hill Book Company, New York 1968).

$$L_e \geq .35 d (S_{a,b}/S_{a,t});$$

where $S_{a,b}$ is the allowable stress for the bolt material
 $S_{a,t}$ is the allowable stress for the end plate material

since this is an approximate relationship, use an factor of .5 instead of .35 for added margin of safety.

for the M8 bolts

$$L_e \geq (.5)(.315)(25000)/(19580) = .201 \text{ in [5.11 mm]}$$

for the M6 bolts

$$L_e \geq (.5)(.236)(25000)/(19580) = .151 \text{ in [3.84 mm]}$$

The actual engagement length for these threads was measured to be a minimum of .280 in for the end plate cover attaching bolts, .310 in for the large port flange attaching bolts, and .25 in for the end plate cover cap attaching bolts.

Conclusion: The M8 and M6 bolts used as described in this calculation are adequate.

f. Cover Flanges {ref. Section VIII UG-34}

The end plate cover flange and the cap flange which covers its central opening are loaded by external pressure from the main vessel. Therefore, the gasket seating load is oriented in the opposite direction and does not contribute to the flange sizing. The minimum flange thickness is calculated by equation (2) in UG-34, except that the gasket moment is 0. This is conservative. The o-ring loading is opposite the pressure load.

$$t(\min) = d * \sqrt{CP/SE}$$

where

t(min) is the minimum allowable wall thickness	(calculated below)
P is the maximum working pressure differential =	159.7 psi [1.10 MPa]
d is the flanges' pressure diameters (full OD for external load) =	8.38 [213], 3.94 [100]: in [mm]
S is the allowable stress=	19580 psi [135 MPa]
E is the joint efficiency=	1.0
(no weld joints in either of these parts)	
C is the joint factor {Table UG-34, type k} =	0.3

for the end plate cover flange:

$$t(\min) = 8.38 * \sqrt{(.3)(159.7)/[(19580)(1.0)]} = .415 \text{ in}$$

$$t(\min) = .415 \text{ in} + .003 \text{ in for corrosion allowance} = .418 \text{ in [10.62 mm]}$$

$$\text{actual thickness} = .630 \text{ in [16 mm]}$$

similarly for cap flange

$$t(\min) = 3.94 * \sqrt{(.3)(159.7)/[(19580)(1)]} = .195 \text{ in [4.95 mm]}$$

$$t(\min) = .195 \text{ in} + .003 \text{ for corrosion allowance} = .198 \text{ in [5.03 mm]}$$

$$\text{actual thickness} = .420 \text{ in [10.76 mm]}$$

In addition, since both of these flanges have o-ring grooves, they need to comply with UG-34(d) (sketches j and k) which requires that the net thickness of the flange at the o-ring be no less than:

$$t(\min) = d(1.9W_h/Sd^3)$$

where: d is the average dia at the bolt circle (BC)	= 7.48 [214.5], 3.425 [87.8]: in [mm]
S is the allowable stress	= 19580 psi [135 MPa]
W is the gasket seating load	(calculated below)
h _G is the gasket/BC moment arm	= .36 [9.14], .364 [9.33]: in, [mm]

for the end plate cover flange:

$$W = (\text{o'ring seating load})(\text{o'ring average perimeter}) = W = 31 * (6.76) = 658.4 \text{ lbs [2929 Nt]}$$

$$t(\text{min}) = 7.48 * \sqrt{(1.9)(658.4)(.36)/[(19580)(7.48)^3]} = .055 \text{ in [1.41 mm]}$$

$$t(\text{min}) = .055 \text{ in} + .003 \text{ in for corrosion allowance} = .058 \text{ in [1.47 mm]}$$

actual thickness under the groove is = .526 in [13.36 mm]

similarly for cap flange:

$$W = (\text{o'ring seating load})(\text{o'ring average perimeter}) = (52) (2.697) = 440.60 \text{ lb [1960 Nt]}$$

$$t(\text{min}) = 3.425 \sqrt{(1.9)(440.6)(.364)/[(19580)(3.425)^3]} = .067 \text{ in [1.72 mm]}$$

$$t(\text{min}) = .068 \text{ in} + .003 \text{ for corrosion allowance} = .070 \text{ in [1.79 mm]}$$

actual thickness under the groove is = .293 in [8.81 mm]

Conclusion: These port flange thicknesses are adequate.

13. Welds

a. Pressure Vessel

ASME Section VIII, in general, does not require stress calculations based on the size and form of individual welds. Rather, the Code requires that welding conform to general standards for size and form based on joint type/design, with the implied assumption that for any weld joint so implemented, the stresses in the parent mating materials will substantially exceed the stresses in the welds themselves, and further implies that welds so implemented will be free from localized stress concentrations.

The actual welding performed on the subject pressure vessel was carefully examined by BNL weld experts and found to be deficient in many areas. OPTOEL provided radiographs for some of the welds, but not all of them. BNL NDE Level II personnel have determined that the radiographs supplied, in most cases, were of poor quality, substantially below U.S. standards, and were not calibrated. Accordingly, an extensive effort was made to evaluate the individual welds using non-destructive evaluation methods (NDE), and perform ASME Section VII and/or finite element analyses (FEA) to determine the maximum stresses that the welds would be subjected to under design conditions.

The following is a list of the weld joint designs involved in the pressure vessel, evaluations and analyses performed to assess the adequacy of the particular joint and the conclusions ultimately arrived at by BNL engineers for each of the weld joint designs.

- i. shell major section longitudinal seam weld - radiographed at BNL found to be acceptable to ASME Section VIII standards. FEA performed to evaluate stresses at joint, see results below
- ii. shell to main flange full penetration butt weld - examined and found to be of questionable quality - welds ground and cleaned at BNL and locally probed to assess weld form - FEA of actual weld geometry performed to evaluate structural integrity - see results below
- iii. shell to end plate double lap weld - examined and found to be of questionable quality - welds dressed and cleaned at BNL and locally probed to assess weld form - FEA of actual weld geometry performed to evaluate structural integrity - see results below
- iv. main flange to end plate butt weld - examined and found to be of questionable quality (incl. less than full penetration) - welds dressed and cleaned at BNL and locally probed to assess weld form - FEA of actual weld geometry performed to evaluate structural integrity - see results below

- v. corner welds at 8 locations where main flange, shell and end plate meet - examined and excavated and found to have a cavity extending through nearly the entire joint - FEA performed and determined that a 1/8" (minimum thickness) seal weld would provide adequate structural integrity.
- vi. port tube to end plate, port tube to shell and port tube to port flange welds - all found to be adequate to ASME Section VIII standards after dressing and cleaning, except that reinforcement welding on port type "D", not indicated on drawings required additional evaluation, see results below
- vii. non-pressure boundary welds (e.g. support structure welds) - these items were beyond the scope of the analyses reported herein.
- viii. general weld conditions - in general, the welding quality was not up to ASME Section VIII standards - multiple weld strikes, weld spatter, undercuts and other surface imperfections were found - such defects could lead to crack formation over time and as such needed to be repaired - repairs including grinding, blending and otherwise dressing of all welds was performed at BNL to bring the welds to minimal standards.

A 3D representation of the weld locations analyzed using FEA is provided in Figure 13-1.

The following are descriptions of the FEA analyses performed on the welds described in items i. thru v., above, and the calculations performed on the reinforcement welding described in item (vi), above. In the FEA analyses, all parent and weld materials are assumed to have the following properties:

Modulus of elasticity = 28,000,000 psi [193,000 MPa]

Poisson's Ratio = .305

Density = .289 lb/in³ [8000 kg/m³]

For the weld material no properties were available from Russian sources. The properties given above and a yield stress of 35,000 psi [206.8 MPa], corresponding to the low end of U.S. 300 series austenitic stainless steels, were used as a conservative approximation. Using a safety factor of 2.0 on yield, the allowable stress in all welds was set at 17,500 psi [103.4 MPa].

i. shell seam weld

The minimum seam weld dimensions were measured by BNL technicians and a 2D FEA analysis was performed using these dimensions. The measurements showed a slight misalignment of the joined sections (1/16" offset) which is reflected in the analysis. The analysis assumed quad-symmetry on orthogonal axes, plane strain, and a uniform 150 psi differential pressure across the vessel boundary (results may be linearly scaled with little error). The results are presented below in figure 13-2. (Note: figure shows only the relevant section of the model.)

The maximum stress found at the seam weld was 8330 psi [57.43 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 8052 psi [55.52 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$$P_{\max} = 315 \text{ psig} [2.27 \text{ MPa}]$$

ii. shell/main flange welds

The minimum shell to main flange weld dimensions were measured by BNL technicians and a 2D FEA analysis was performed using these dimensions. The measurements showed full penetration, and this is reflected in the analysis. The analysis assumed hemispherical symmetry, plane strain, and a uniform 150 psi differential pressure across the vessel boundary (results may be linearly scaled with little error). The results are presented below in figure 13-3 (a) and (b). (Note: figures show only the relevant section of the model for the major and minor sections of the vessel, respectively.)

The maximum stress found on the major section joint weld was 9632 psi [66.41 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 9311 psi [64.20 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$$P_{\max} = 273 \text{ psig} [1.98 \text{ MPa}]$$

The maximum stress found on the minor section joint weld was 9329 psi [64.32 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 9018 psi [62.18 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$$P_{\max} = 281 \text{ psig [2.04]}$$

iii. shell/end plate welds

The minimum shell to end plate weld dimensions were measured by BNL technicians and a 2D FEA analysis was performed using these dimensions. The measurements showed a double lap joint with zero penetration under the lap, and this is reflected in the analysis. The analysis assumed axial symmetry, and a uniform 150 psi differential pressure across the vessel boundary (results may be linearly scaled with little error). The results are presented below in figure 13-4 (a) and (b). (Note: figures show only the relevant section of the model for the major and minor sections of the vessel, respectively.)

The maximum stress found on the major section joint weld was 20738 psi [142.98 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 20047 psi [138.22 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$$P_{\max} = 127 \text{ psig} [0.97 \text{ MPa}]$$

The maximum stress found on the minor section joint weld was 20970 psi [144.58 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 20271 psi [139.76 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$$P_{\max} = 125 \text{ psig} [0.96 \text{ MPa}]$$

iv. main flange to end plate welds

The minimum main flange to end plate weld dimensions were measured by BNL technicians and a 2D FEA analysis was performed using these dimensions. The measurements showed a lack of penetration in the joints, and this is reflected in the analysis. The analysis assumed plane strain, and a uniform 150 psi differential pressure across the vessel boundary (results may be linearly scaled with little error). The results are presented below in figure 13-5 (a) and (b). (Note: figures show only the relevant section of the model for the major and minor sections of the vessel, respectively.)

The maximum stress found on the major section joint weld was 15600 psi [107.56 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 15080 psi [103.97 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$$P_{\max} = 168 \text{ psig} [1.26 \text{ MPa}]$$

The maximum stress found on the minor section joint weld was 15928 psi [109.82 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 15397 psi [106.16 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$$P_{\max} = 165 \text{ psig} [1.24 \text{ MPa}]$$

v. (main flange/end plate/shell) corner welds

The corner welds on the vessel were examined by BNL technicians and found to be lacking in penetration in the corner itself, prevented from leaking only by a thin surface weld. A 3D FEA analysis was performed using discontinuities at the corners to model these findings and providing a 1/8 inch maximum sealing bead at the interface. Adjacent welds are modeled as described above and extruded to a 3D model. The analysis assumed hemispherical symmetry, and a uniform 150 psi differential pressure across the vessel boundary (results may be linearly scaled with little error). The results are presented below in figure 13-6 (a), (b), (c) and (d).

The maximum stress found on the major section corner weld was 6337 psi [43.69 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 6126 psi [42.24 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$P_{max} = 414 \text{ psig}$ [2.96 MPa]

The maximum stress found on the minor section corner weld was 8864 psi [61.11 MPa]. This can be scaled to find the maximum stress at MAWP = 145 psig [1.10 MPa] = 8569 psi [59.08 MPa].

Scaling the stress at MAWP to the allowable stress, the maximum pressure that this component would be capable of withstanding is found to be

$P_{max} = 296 \text{ psig}$ [2.14 MPa]

Note: The maximum stresses indicated for the corner welds occurred on the interior side of the vessel. The exterior welds were also modeled and the stresses found were significantly lower.

Conclusion: All corner welds as repaired at BNL are adequate.

b. Vacuum Vessel

i. Shell and end plate to large rectangular flange weld

The type of weld used to attach the large rectangular flange to the shell and end plate corresponds to a type 3, category B weld (figure UW-3 and Table UW-12), of the type described by figure UW-13.2(d) of the ASME Boiler and Pressure Vessel Code (1992 with revisions and updates to 1995) Section VIII Subsection UW. These welds were examined visually by BNL weld experts and found to be acceptable.

Conclusion: The shell and end plate to large rectangular flange welds are adequate.

ii. Shell to end plate weld

The type of weld used to attach the shell to the end plate corresponds to a category B weld (figure UW-3 and Table UW-12), of the type described by figure UW-13.2(d) of the ASME Boiler and Pressure Vessel Code (1992 with revisions and updates to 1995) Section VIII Subsection UW. These welds were examined by BNL weld experts and found to be acceptable.

Conclusion: The shell to end plate weld is adequate.

iii. Larger ports' tube to shell

The type of welds used to attach the larger ports to the shell corresponds to a type 1, category D weld (figure UW-3 and Table UW-12), of the type described by figure UW-16.1(u) of the ASME Boiler and Pressure Vessel Code (1992 with revisions and updates to 1995) Section VIII Subsection UW. These welds were examined by BNL weld experts and found to be questionable. While externally the welds appeared to be adequate, there was no evidence of full penetration. However, these welds are

subjected to external pressure only, and as such are subject only to compressive stresses. The welds in question do not appear to be in compliance with Section VIII requirements, but since (A) the joint in question is completely enclosed within the main pressure vessel, (B) the joint is subject to external pressure (compressive stress only), (C) failure of the weld would not result in any personnel safety issue, and (D) the weld itself is without observed defect and appears to be completely fused within the limits of its penetration, this weld is considered acceptable, provided the weld is subject to periodic visual examination. Note: This joint has been loaded at least once to 150% of its design maximum load without evidence of permanent deformation or degradation.

Conclusion: The larger ports' tube to shell welds do not meet ASME Section VIII, but are judged acceptable. These welds must be periodically inspected.

iv. Smaller port's tube to shell

The type of weld used to attach the smaller port to the shell corresponds to a type 1, category D weld (figure UW-3 and Table UW-12), of the type described by figure UW-16.1(a) of the ASME Boiler and Pressure Vessel Code (1992 with revisions and updates to 1995) Section VIII Subsection UW. These welds were examined by BNL weld experts and found to be acceptable.

Conclusion: The smaller port's tube to shell welds are adequate.

v. Larger ports' tube to end cap

The type of weld used to attach the larger ports' tube to end cap corresponds to a type 3, category C weld (figure UW-3 and Table UW-12), of the type described by figure UW-13.2(d) of the ASME Boiler and Pressure Vessel Code (1992 with revisions and updates to 1995) Section VIII Subsection UW. These welds were examined by BNL weld experts and found to be acceptable.

Conclusion: The larger ports' tube to end cap welds are adequate.

vi. Smaller port's tube to end cap

The type of weld used to attach the smaller port's tube to end cap corresponds to a type 3, category C weld (figure UW-3 and Table UW-12), of the type described by figure UW-13.2(c) of the ASME Boiler and Pressure Vessel Code (1992 with revisions and updates to 1995) Section VIII Subsection UW. These welds were examined by BNL weld experts and found to be acceptable.

Conclusion: The smaller port's tube to end cap welds are adequate.

14. All other items

The items analyzed above represent all components of the vessel subjected to stresses due to internal pressure loading of the main vessel. Additional components not integral to the vessel but attached to the vessel, components attached to the components attached to the vessel, etc. and any other components normally subjected to stresses due to the pressure loading of the vessel, or which might become stressed under any possible circumstance due to the pressure loading of the vessel, including accident and/or failure of any component herein analyzed, are not covered by this report. It is the responsibility of the cognizant engineer to ensure that such other components are properly designed, selected, analyzed and documented prior to operation of the subject vessel.

Appendix 7.6

Longitudinal Seam Weld X-Ray Inspection Report

INSPECTION REPORT

Date	Acct. No.	I.L.R. No.	W.O. No.	P.O. No.	Insp. No.
4-10-00		407234	46616		0-017

DESCRIPTION

Russian Vessel	Drawing No.	No. Pieces
	—	1

VENDOR

EXPLANATION

479W	
INSPECTION REQUIRED	
☒ Complete ☐ Partial ☐ In-Process as noted	

INSPECTION FINDINGS

Radiographs taken of seam on vessel were found to be acceptable.	
Radiographer - Dennis Malloy #16907	
RADIOGRAPH #'S	
WELD # 0-033 TO 0-041	
NO. OF PIECES MEETING SPECIFICATIONS	1
NO. OF PIECES NOT MEETING SPECIFICATIONS	—

PAGE 1 OF 1 PAGES

John Skaritka

Report Forwarded To

4-10-00

Date

Dennis Malloy

Inspector

DESCRIPTION

EXPLANATION

☐ Accepted	_____ Pcs.	
☐ Rejected	_____ Pcs.	
☐ Partial acceptance as noted		
☐ To be returned to vendor	_____ Pcs.	
☐ To be reworked by BNL	_____ Pcs.	
☐ Scrapped	_____ Pcs.	

Date

Departmental Representative

Appendix 7.7

Preventive Maintenance Procedure

BROOKHAVEN NATIONAL LABORATORY NATIONAL SYNCHROTRON LIGHT SOURCE		Number: ATF-TWL-001	Revision: A
		Effective:	Page 1 of 3
Subject: Inspection Procedure for CO ₂ TeraWatt Laser Amplifier Pressure Vessel			
Prepared By: R. Beauman <i>R. Beauman</i>	Reviewed By: J. Skaritka/D. Lynch	Approved By: M. Woodle	

*Document must contain approved signatures for validity

1. SCOPE

This procedure defines the periodic inspection of the ATF CO₂ TeraWatt Laser amplifier pressure vessel. The inspection shall be carried out every 6 months or 4000 hours of pressurized time, whichever comes first. This procedure is referenced in and is part of the NSLS Preventive Maintenance Program (MARTI).

2. PREREQUISITES

The Laser Amplifier vessel will be depressurized, and disassembled at the main longitudinal flange. The two sections will be separated to allow access. The Mechanical Section Head will be notified prior to the start of inspection. Mechanical Section Head or designee shall witness all inspections.

3. INSPECTION TEST REPORT

The Inspection Test Report(s) may be compiled by the witnessing inspector; but the specific test section shall be filled out, signed and dated by the person performing the actual inspection.

Use "ATF CO₂ TeraWatt Laser Amplifier Inspection Test Report", latest revision, as needed.

Completed Inspection Test Report(s) shall be submitted to NSLS Mechanical Section Head for acceptance.

4. PROCEDURE

4.1 Main O-Ring Seal

Visually inspect the O-ring Seal for nicks cuts and abrasions. Replace O-ring if any damage is present.

NOTE: Replacement O-ring shall be new and of Viton material. O-ring shall be sized by a designee of the NSLS Mechanical Section Head.

4.2 Pressure Vessel Welds

NOTE: All Liquid Penetrant Inspections (LPI) shall be conducted by a qualified inspector per standard BNL specifications. Furthermore, at the discretion of the NSLS Mechanical Section Head, LPIs may be replaced or supplemented by other Non-Destructive Examinations (NDE).

Visually inspect all pressure vessel welds for cracks or voids. No defects are allowed. If there are any questionable areas, an initial LPI shall be performed. If a defect is found it shall be repaired as follows:

- a) Remove defects by grinding to "sound" base material.
- b) LPI the repaired area. No defects allowed.
- c) Weld repair using Central Shops division Weld Procedure SS-1-92, latest revision.
- d) Visual and final LPI shall be performed on repaired area. Repair to longitudinal seam welds shall also be inspected for adequate material thickness. No defects allowed. Any repair of seam weld must be of equal or greater thickness than surrounding base material.

5.0 SUBMITTALS

All inspection test reports shall be submitted to the NSLS Mechanical Section Head for acceptance.

6.0 RELEASE FOR OPERATIONS

NSLS Mechanical Section Head submits to NSLS ESH/Q Associate Chair copies of all appropriate reports, and an acceptance memo releasing laser for operations.

Number: ATF-TWL-001	Revision: A	Effective:	Page 3 of 3

Revision Log Table

Revision Number	Date Approved	Pages Affected	Description of Revision
A		N/A	First Issue

Date:

Date of Inspection: _____

Technician Signature: _____

Date: _____

Name: _____
(Please print legibly)

Signature: _____ Date: _____

Inspect all the following weld groups.

A separate inspection sheet shall be filled out for each weld group:

- Cylindrical shell longitudinal seal weld – both internal and external welds
- Left half – Cylindrical shell to main flange – both internal and external welds
- Right half – Cylindrical shell to main flange – both internal and external welds
- Left half – Cylindrical shell to endplate – both internal and external welds
double lap joint
- Right half – Cylindrical shell to endplate – both internal and external welds
double lap joint
- Left half – main flange to endplate – both internal and external welds
- Right half – main flange to endplate – both internal and external welds
- Port tubes to endplate welds – all locations – all welds at specific location
- Port tubes to cylindrical shell – all locations – all welds at specific location
- Port tubes to port flange – all locations – both sides of flange

ATF CO₂ Terawatt Laser Amplifier Inspection Test Report

A. Specific Weld Group Visually Inspected: Check One (1) Only

Technician Name: _____ Date of Inspection: _____
 (Please print legibly)

☐ Cylindrical shell longitudinal seal weld: internal & external	☐ Right half cyl. shell to main flange: internal & external
☐ Left half cyl. shell to main flange: internal & external	☐ Right half cyl. shell to endplate: internal & external
☐ Left half cyl. shell to endplate: internal & external	☐ Right half main flange to endplate: internal & external
☐ Left half main flange to endplate: internal & external	☐ Right half main flange to endplate: internal & external
☐ Port tubes to endplate: all locations & specific locations	☐ Port tubes to cyl. shell: all locations & specific locations
☐ Port tubes to port flange: all locations: both sides of flange	

WELDS: ☐ Acceptable, no further action required ☐ Further NDE Inspection Required – Go to Section II
 Use comment section to specify specific weld(s)

Technician Signature _____ Date: _____

II. Initial NDE of Inspection Area: ☐ Weld acceptable, no further action required ☐ Weld repair required – Go to Section III

Name of NDE Inspector: _____ Date Performed: _____
 (Please print legibly)

NDE Inspector's Signature: _____

III. Weld Repair

Welder Name: _____ Date Performed: _____
 (Please print legibly)

- a. Defects/Voids removed by grinding to "sound" base material: ☐ Acceptable
- b. NDE of repaired area performed and acceptable ☐ Work completed

Name of NDE Inspector: _____ Date Performed: _____
 (Please print legibly)

NDE Inspector's Signature: _____

- c. Weld Repair Performed:
 Specify welding wire and process performed: _____
☐ Work completed

- d. Final NDE inspection (and thickness measurement, if required) performed and acceptable: ☐ Work completed.

Name of NDE Inspector: _____ Date Performed: _____
 (Please print legibly)

NDE Inspector's Signature: _____

Welder's signature _____

Comments: _____

Appendix 7.8

Hydro Test Procedure and Report

BROOKHAVEN NATIONAL LABORATORY NATIONAL SYNCHROTRON LIGHT SOURCE		Number: LS-M-0009	Revision: A
		Effective: 5/19/2000	Page 1 of 7
Subject: HYDROSTATIC TEST PROCEDURE FOR CO₂ TERAWATT LASER PRESSURE VESSEL			
Prepared By: Ron Beauman <i>Ron Beauman</i>		Reviewed By: N/A	Approved By: Martin Woodle <i>M. Woodle</i>

*Document must contain approved signatures for validity

HYDROSTATIC TEST PROCEDURE FOR CO₂ TERAWATT LASER PRESSURE VESSEL

Number: LS-M-0009	Revision: A	Effective: 05/19/00	Page 2 of 7
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Scope:

The purpose of this procedure is to outline the hydrostatic test procedure, for the CO₂ Terawatt Laser Pressure vessel, and to define the acceptance criteria associated with this test.

Test Fluid Medium:

The test shall be conducted using clean water. The water temperature shall be less than 100°F.

Note: Domestic, potable, water is acceptable.

Test Set-up:

The set-up shall be such that a complete inspection of the vessel, while pressurized, is possible. All areas of the external vessel shall be easily accessible.

A hydrostatic pump shall be appropriately connected to the vessel.

The vessel shall have vents appropriately located such that all areas interior to the vessel may be completely vented of air.

The vessel shall have drains appropriately located such that all areas interior to the vessel may be completely drained of water at the conclusions of this test.

The vessel shall have isolation valve(s) such that, once pressurized to test conditions, the vessel may be isolated and held at pressurized conditions for the duration of the inspection period.

The vessel shall have a calibrated, pressure gauge appropriately connected and located such that the gauge may be isolated from the vessel, even under pressurized conditions. The gauge location shall be such that it is visible and readily accessible throughout the test.

The gauge shall have a range of 0-300 psi, and an accuracy better than or equal to $\pm 1\%$ full scale.

All pressure boundary penetrations plugged, prior to test.

Figure 1: Schematic of Test Set-up

Note: Remove all air, as practical, prior to pressurization.

Pressure Gauge Calibration:

The pressure gauge shall be calibrated prior to and at the completion of the hydrostatic test.

Once calibrated, prior to the test, the gauge shall not be used for any purpose other than this hydrostatic test. Also, once the test is completed, this gauge shall be calibrated prior to its use for any other purpose.

Calibration shall be conducted by an NSLS Utilities technician using the NSLS deadweight tester. A "NSLS Utilities Calibration Report" shall be submitted, for each calibration, to the NSLS Utilities engineer for approval.

Note: A non-calibrated or incorrectly maintained gauge, once calibrated, will be grounds for invalidating this test.

Testing:

All test data shall be recorded and a NSLS "Utilities Test Report" shall be completed and submitted to the NSLS Utilities engineer.

There shall be six (6) pressure tests performed: One (1) static pressure test and five (5) cyclic pressure tests.

Note: If at any point the pressure within the vessel, once isolated, cannot be maintained, the test shall stop. The faulty component replaced and the test may continue from the point at which it stopped. But if the vessel itself is the cause of the leak, then the test shall be considered a failure.

Test #1

The first test shall be at the design pressure, 116 psig. The vessel shall be taken up to 116 psig, as determined by the pressure gauge. The vessel shall be isolated and the pressure held for a minimum of ten (10) minutes, with the actual holding time being such that a complete inspection of the pressure vessel can be performed. No leakage or weepage may occur from the vessel.

Note: Any leakage or weepage is considered a test failure.

At the conclusion of this test, the pressure may be vented.

Tests #2 through #5

Any leakage or weepage, during any period of these tests, is considered a test failure. These tests shall be cyclic in nature.

Step 1: The vessel shall be taken up to design pressure, 116 psig, as determined by the pressure gauge. This pressure shall be held for ten (10) minutes. Go to Step 2.

Step 2: All the completion of the previous step, the pressure within the vessel shall be increased to 125% of design pressure, 145 psig, as determined by the pressure gauge. This pressure shall be held for ten (10) minutes. Go to Step 3.

Step 3: At the completion of Step 2, the pressure shall be lowered to design pressure, 116 psig, as determined by the pressure gauge. Go to Step 4.

Step 4: Is this the third (3rd) time Step 4 has been executed?

If Yes, go to Step 5.

If No, hold the pressure within the vessel at 116 psig, for ten (10) minutes and then go to Step 2.

Step 5: The pressure within the vessel, at 116 psig, shall be held for one (1) hour. At the conclusion of one (1) hour, a complete inspection of the pressure vessel, while pressurized at 116 psig, shall be conducted.

No leakage or weepage may occur from the vessel.

At the conclusion of this inspection period, go to Test #6.

Test #6

- Vessel is pressurized at 116 psig. The pressure shall be increased to 125% of design pressure, 145 psig, as determined by the pressure gauge. This pressure shall be maintained for a minimum of ten (10) minutes, with the actual holding time being such that a complete inspection of the pressure vessel can be performed.

No leakage or weepage may occur from the vessel.

Testing Completion

At the conclusion of Test #6, the pressure within the vessel shall be vented. The vessel shall be completely drained of water, and the test set-up may be removed from the vessel. The pressure gauge shall be calibrated at the conclusion of all testing and prior to reuse. See the section on pressure gauge calibration.

Number: LS-M-0009	Revision: A	Effective: 05/19/00	Page 6 of 7
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Test Report

At the conclusion of the testing, a NSLS "Utilities Test Report" shall be completed and submitted to the NSLS Utilities engineer.

Note: The NSLS "Utilities Test Report" is a general form; therefore, only the sections that apply need to be filled out. If needed, contact the NSLS Utilities Group.

Test Results

All test and calibration results shall be submitted by the testing personnel to the NSLS Utilities engineer. The results shall then be present by the NSLS Utilities engineer to the CO₂ Terawatt Laser Design Review Committee for their determination.

Number: LS-M-0009	Revision: A	Effective: 05/19/00	Page 7 of 7
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Revision Log Table

Revision Number	Date Approved	Pages Affected	Description of Revision
A	5/19/00	_____	First Issue

Pressure Test:

Test conducted by: JOHN SKARITKA Test Date: 5/19/00
(print name)

Page 2 of 2

Test location: ☐ In field/position ☒ Other (NOTHOF) CONDUCTED INSIDE BURNING WELD SHOP BLDG 479

Fluid medium: ☒ Water ☐ Air

Temp. <100°F = 59°F

Maximum Test Pressure: Specified: 116 psig

Actual: 116* psig

Test Duration: 10 MINUTES minutes/hours (circle correct unit)
(10 minute minimum)

No Leakage or Weepage Allowed

☒ Acceptable ☐ non-acceptable

Signature: John Skaritka Signature: _____

Witnesses (if applicable) Bill Bambina Glenn

Test Pressure was Increased/Decreased in Steps - Specify both pressure & Time Duration

Pressure Increased to Max. Test Pressure

Pressure Decreased from Pmax to lesser of Design Pres. or 100 psig.

P-psig	Time ~minutes
45	10
45	10
45	10
45	10

Max. Test Pressure
147 psig
Duration 20 Minutes
(10 minutes minimum)

P-psig	Time ~minutes
116	10
116	10
116	10
116	60

Other Test: Must Define Purpose, Procedure of Test and Criteria for Acceptance.

Use comment section.

☒ Other sheets included

Test conducted by: JOHN SKARITKA Date of Test: 5/19/00

Test location: ☒ In field/position ☐ Other INSIDE BLDG 479 (WELD SHOP)

Acceptable ☐ non-acceptable ☒ Other RESULTS GIVEN TO NLS CO. TW LASH DESIGN REVIEW COMMITTEE FOR THEIR DETERMINATION
Signature: John Skaritka Signature: Bill Bambina Signature: Glenn

Comments IN INITIAL TESTS TWO AREAS OF POROSITY OCCURRED ON THE
UP PNEUMATIZED VESSEL, SINCE THIS WAS NOT PART OF THE VESSEL THE
PROBLEM AREAS WERE PATCHED WITH HOSE CLIPS. ON THE SECOND CYCLE
1. 2 PLUG ON THE BOTTOM FLANGE GASKET, THE PLUG WAS TIGHTENED
UNTIL LEAK STOPPED. AFTER 1 HR AT 116 PSIG AND 20 MINUTE AT
145 A WET SPOT UNDER ONE BOLT WAS DETERMINED TO BE LEAK FROM
THE VESSEL BEING LEFT IN THE RAIN PRIOR TO STOPPING TESTING.

Utilities Test Report

Prepared by: JOHN SHARITKA Date: 5/19/00

Applicable: Drawing(s): _____
 Specification(s)/Procedure(s): HYDROSTATIC TEST PROCEDURE FOR CO₂ TANK WITH
UNDER PRESSURE VESSEL

Branch/Component(s) located on: ☐ HPC ☐ LPC ☐ Alum ☐ Exp.
☐ Other ☐ → specify _____

Operation(s) Conducted: ☐ Leak Test ☐ Flushing
 (check all that apply) Pressure Test → ☒ Hydrostatic ☐ Pneumatic
☐ Other → specify _____

Leak Test: Test conducted by: _____ Test date: _____
 (print name)

Test location: ☐ In field/position ☐ Other _____

Test medium: ☐ Water ☐ Gas → specify _____ ☐ Other → specify _____
 Test parameters: ☐ Domestic water ☐ Other _____ psig _____ °F

No Leakage or Weepage Allowed.

☐ Acceptable ☐ Non-acceptable
 Signature: _____ Signature: _____
☐ Other _____
 Signature: _____

Witnesses (if applicable): _____

Flushing: Conducted by: _____ Date Conducted: _____
 (print name)

Use clean water @ Temp. <100°F

Medium: ☐ Domestic Water ☐ Other → specify _____

Flushing Duration: _____ minutes/hours (circle correct units)
☐ Acceptable ☐ non-acceptable: Explain: _____
 Signature: _____

Signature: _____

Witnesses (if applicable) _____

BROOKHAVEN
NATIONAL LABORATORY

Memo

Building 725C
P.O. Box 5000
Upton, NY 11973-5000
Phone 631 344-2521
Fax 631 344-3029
woodle1@bnl.gov

managed by Brookhaven Science Associates
for the U.S. Department of Energy

date: May 22, 2000

to: M. Hart

from: M. Woodle *Woodle*

subject: TeraWatt Laser Pressure Vessel Hydrotest Results

reference: a) LS-M-0009 RevA Hydrostatic Test procedure for CO2 Terawatt Laser Pressure Vessel

This is to inform you that the TeraWatt Laser pressure vessel has successfully passed the hydrostatic test which was performed in accordance with reference a).

Final approval for the use of this pressure vessel in the Laser system, however, is contingent on the satisfaction of all the requirements of the Terawatt Pressure Vessel Committee Report, which is due to be issued in the near future.

cc: A. Ackerman, R. Beaman, R. Casey, C. Czajkowski, I. Ben-Zvi, R. Biscardi, T. Dickinson, A. Farland, N. Gmur, R. Heese, T. Lambertson, D. Lynch, I. Pogorelsky, J. Skaritka

Appendix 7.9

Bolting Sketches

Memo

Date: April 12, 2000

To: Tom Lambertson

From: John Skaritka

Subject: Machining Surfaces and Portions of the Weld of the TW Laser Pressure Vessel

The TW Laser Pressure Vessel Committee met on April 7, 2000 to discuss progress on the analysis and repairs of the laser vessel. One topic that was discussed was the repair of the stripped threads in the end flanges of the vessel.

At a previous meeting, the decision was made to remove the threads at the ends of the flange and to use $\frac{3}{4}$ " studs to secure the two ends of the vessel. As seen in Figure 1, the Committee discussed the proposed modifications to the vessel to accommodate the $\frac{3}{4}$ " studs and washers.

The Committee hereby gives approval to BNL Central Shops to proceed on the modifications as in Figure 2 ASAP.

Approval:

Martin Woodle
Committee Chair

JS:lm

Encl: Figure 1
Figure 2

FIGURE 1

FIGURE 2

109 Qu nter Farm Road
Union, OH 45322-9796
Phone: 937-836-0961
Fax: 937-832-1274
Website: www.te-co.com

CERTIFICATE OF CONFORMANCE

TO: JOHN SKARITKA / BROOKHAVEN NATIONAL LABS

FROM: HOLLY HINES/CUSTOMER SERVICE REPRESENTATIVE

SUBJECT: CUSTOMER REQUEST FOR CERTIFICATE OF CONFORMANCE

DATE: MAY 18, 2000

IT IS HEREBY CERTIFIED THAT TE-CO STUDS ARE IN COMPLIANCE WITH
STANDARDS STATED IN CATALOG #LIT23. THE ABOVE ITEMS HAVE BEEN
INSPECTED BY A QUALIFIED INSPECTOR USING QUALIFIED INSPECTION EQUIPMENT.

Thank you,

A handwritten signature in cursive script, appearing to read 'Holly Hines', written over a horizontal line.

Holly Hines , Customer Service Representative

*STUD MATERIAL IS ASTM A193-B7