| ENERCON<br>Excellence...Every project. Every day. | CALCULATION COVER SHEET | CALC. NO. RTL-001-CALC-ST-0203<br>REV. 2<br>PAGE NO. 1 of 60 |
|---|---|---|

| Title: | RT-100 Cask Bolting Evaluation | Client: Robatel Technologies, LLC<br>Project: RTL-001 |
|---|---|---|

| Item | Cover Sheet Items | Yes | No |
|---|---|---|---|
| 1 | Does this calculation contain any open assumptions that require confirmation? (If YES, Identify the assumptions) _____ | ☐ | ☒ |
| 2 | Does this calculation serve as an "Alternate Calculation"? (If YES, Identify the design verified calculation.)<br>**Design Verified Calculation No.** _____ | ☐ | ☒ |
| 3 | Does this calculation Supersede an existing Calculation? (If YES, identify the superseded calculation.)<br>**Superseded Calculation No.** _____ | ☐ | ☒ |

**Scope of Revision:**

Added Sections 7.1.4, 7.1.4.1, 7.1.4.2, Refs. 3.20 and 3.21, and Appendix 1 Fig. 2 to address bolt plastic deformation allowed by NUREG/CR-6007 and seal integrity; corrected typo in modulus of elasticity units throughout; minor editorial changes.

**Revision Impact on Results:**

None.

**Study Calculation** ☐ **Final Calculation** ☒

**Safety-Related** ☒ **Non-Safety Related** ☐

*(Print Name and Sign)*

| | |
|---|---|
| **Originator: John Staples** John F. Staples | **Date:** 28 Nov. 2012 |
| **Design Verifier: John McFarland** | **Date:** 11/28/12 |
| **Approver: Curt Lindner** | **Date:** 11/28/12 |

| ENERCON<br>Excellence—Every project. Every day. | CALCULATION<br>REVISION STATUS SHEET | CALC. NO. RTL-001-CALC-ST-0203<br>REV. 2<br>PAGE NO. 2 of 60 |
|---|---|---|

## CALCULATION REVISION STATUS

| REVISION | DATE | DESCRIPTION |
|---|---|---|
| 0 | 8-24-2012 | Initial Issue |
| 1 | 10-8-2012 | Revised in its entirety to match SAR Review Plan Format |
| 2 | 11/28/12 | Added Sections 7.1.4, 7.1.4.1, 7.1.4.2, Refs. 3.20 and 3.21, and Appendix 1 Fig. 2 to address bolt plastic deformation allowed by NUREG/CR-6007 and seal integrity; corrected typo in modulus of elasticity units throughout; minor editorial changes. |

## PAGE REVISION STATUS

| PAGE NO. | REVISION | PAGE NO. | REVISION |
|---|---|---|---|
| 1-5 | 2 | 6 | 1 |
| 7-8 | 2 | 9-10 | 1 |
| 11-13 | 2 | 14 | 1 |
| 15-17 | 2 | 18 | 1 |
| 19-20 | 2 | 21 | 1 |
| 22-24 | 2 | 25-26 | 1 |
| 27-28 | 2 | 29 | 1 |
| 30 | 2 | 31-33 | 1 |
| 34-37 | 2 | 38 | 1 |
| 39-40 | 2 | 41 | 1 |
| 42-46 | 2 | 47-57 | 1 |
| 57A | 2 | 58-60 | 1 |

## APPENDIX REVISION STATUS

| APPENDIX NO. | PAGE NO. | REVISION NO. | APPENDIX NO. | PAGE NO. | REVISION NO. |
|---|---|---|---|---|---|
| 1 | 1-2 | 0 | 1 | 3 | 2 |
| 2 | 1-2 | 0 | | | |

| ENERCON<br>Excellence—Every project. Every day. | CALCULATION<br>DESIGN VERIFICATION<br>PLAN AND SUMMARY SHEET | CALC. NO. RTL-001-CALC-ST-0203<br>REV. 2<br>PAGE NO. 3 of 60 |
|---|---|---|

**Calculation Design Verification Plan:**

Calculation to be reviewed for correctness of inputs, design criteria, analytical methods, acceptance criteria and numerical accuracy.

Stated objectives and conclusions shall be confirmed to be reasonable and valid.

Any assumptions shall be clearly documented and confirmed to be appropriate and verified based on sound engineering principles and practices.

*(Print Name and Sign for Approval – mark "N/A" if not required)*

| Approver: Curt Lindner | Date: 11/28/12 |
|---|---|

**Calculation Design Verification Summary:**

Calculation has been designated as **Safety Related** as noted on the cover sheet.

Calculation has been verified to be mathematically correct and performed in accordance with appropriate design inputs, assumptions, analytical methods, design criteria and acceptance criteria.

The conclusions developed in the calculation are reasonable, valid and consistent with the purpose and scope.

Assumptions are appropriate and correct.

**Based On The Above Summary, The Calculation Is Determined To Be Acceptable.**

*(Print Name and Sign)*

| Design Verifier: John McFarland | Date: 11-28-12 |
|---|---|
| Others: | Date: |

| ENERCON<br>Excellence—Every project. Every day. | CALCULATION<br>DESIGN VERIFICATION<br>CHECKLIST | CALC. NO. RTL-001-CALC-ST-0203<br>REV. 2<br>PAGE NO. 4 of 60 |
|---|---|---|

| Item | CHECKLIST ITEMS | Yes | No | N/A |
|---|---|---|---|---|
| 1 | **Design Inputs -** Were the design inputs correctly selected, referenced (latest revision), consistent with the design basis, and incorporated in the calculation? | X | | |
| 2 | **Assumptions** - Were the assumptions reasonable and adequately described, justified and/or verified, and documented? | X | | |
| 3 | **Quality Assurance** - Were the appropriate QA classification and requirements assigned to the calculation? | X | | |
| 4 | **Codes, Standards, and Regulatory Requirements -** Were the applicable codes, standards, and regulatory requirements, including issue and addenda, properly identified and their requirements satisfied? | X | | |
| 5 | **Construction and Operating Experience** - Have applicable construction and operating experience been considered? | X | | |
| 6 | **Interfaces** - Have the design-interface requirements been satisfied, including interactions with other calculations? | X | | |
| 7 | **Methods** - Was the calculation methodology appropriate and properly applied to satisfy the calculation objective? | X | | |
| 8 | **Design Outputs** - Was the conclusion of the calculation clearly stated, did it correspond directly with the objectives, and are the results reasonable compared to the inputs? | X | | |
| 9 | **Radiation Exposure** - Has the calculation properly considered radiation exposure to the public and plant personnel? | | | X |
| 10 | **Acceptance Criteria** - Are the acceptance criteria incorporated in the calculation sufficient to allow verification that the design requirements have been satisfactorily accomplished? | X | | |
| 11 | **Computer Software-** Is a computer program or software used, and if so, are the requirements of CSP 3.02 met? | | | X |

COMMENTS

***(Print Name and Sign)***

| | |
|---|---|
| **Design Verifier:** John McFarland | **Date:** 11-28-12 |
| **Others:** | **Date:** |

Table of Contents

## 1.0 Purpose and Scope

Robatel Technologies is designing the RT-100 transport cask, containing radioactive waste in the form of dewatered resins and filters, to meet the requirements of 10CFR Part 71 (Ref. 3.1). The NRC requirements in Reference 3.1 state that any package must be evaluated for the normal conditions of transport and for the hypothetical accident conditions. These evaluations for the cask body, cask lid, impact limiters, etc. have been carried out in References 3.3, 3.4, 3.15, 3.16 and 3.17. The NRC requirements in Section 71.43(c) of Reference 3.1 state that a package must include a containment system that can be securely closed through the use of a positive fastening device that cannot be opened unintentionally or by a pressure that may arise within the package. The RT-100 package is designed with two sets of closure bolts: 18 M36 hex head bolts at the secondary lid and 32 M48 hex heads at the primary lid. These two sets of bolts are credited with maintaining positive closure of the package under all accident conditions. The purpose of this calculation is to structurally qualify the fully-loaded RT-100 cask for the loadings associated with the normal conditions of transport and the hypothetical accident conditions, as determined in References 3.3, 3.4, 3.15, 3.16 and 3.17.

## 2.0 Summary of Results and Conclusions

All structural bolts have a factor of safety of greater than 1.0 under the most adverse effects from the normal conditions of transport and the hypothetical accident conditions. The minimum factor of safety is 1.29 for the M36 bolts at the secondary lid. The minimum factor of safety is 1.03 for the M48 bolts at the primary lid. The results of the analysis show that the RT-100 cask can withstand the normal conditions of transport and the hypothetical accident conditions while maintaining positive closure of the containment vessel.

**Therefore, the closure bolts of the RT-100 transport cask are adequate for their design function.**

## 3.0 References

3.1 Nuclear Regulatory Commission, 10CFR Part 71, "Packaging and Transportation of Radioactive Material"
3.2 Drawing 102885 PE 1001-1 Sht. 1 & 2 Rev. E, "ROBATEL Transport Package RT100 - General Assy"
3.3 ENERCON Calculation RTL-001-CALC-ST-0201 Rev. 1, "RT-100 Lifting Structural Evaluation"
3.4 ENERCON Calculation RTL-001-CALC-ST-0202 Rev. 0, "RT-100 Tie-Down Structural Evaluation"
3.5 NUREG/CR-6007, "Stress Analysis of Closure Bolts for Shipping Casks", 1992
3.6 ASME B&PV Code, Section II, 2007
3.7 ASME B&PV Code, Section III, 2007
3.8 Nuclear Regulatory Commission Regulatory Guide 7.6 Revision 1, "Design Criteria for the Structural Analysis of Shipping Cask Containment Vessels"
3.9 EPRI "Good Bolting Practices", Volume 1, Large Bolt Manual, 1987
3.10 Joseph Edward Shigley & Larry D. Mitchell, "Mechanical Engineering Design", 4th Edition
3.11 Erik Oberg, et. al., "Machinery's Handbook", 26th Edition
3.12 ASME B1.13M-2005, "Metric Screw Threads: M Profile"
3.13 NUREG/CR-0128, "Shock and Vibration Environments for a Large Shipping Container During Truck Transport (Part II)", 1978
3.14 AISC, "Guide to Design Criteria for Bolted and Riveted Joints", 2nd Edition
3.15 ENERCON Calculation RTL-001-CALC-ST-0403 Rev. 0, "RT-100 Pin Puncture Evaluation"
3.16 ENERCON Calculation RTL-001-CALC-TH-0102 Rev. 1, "RT-100 Cask Maximum Normal Operating Pressure Calculation"
3.17 ENERCON Calculation RTL-001-CALC-ST-0401 Rev. 1, "RT100 Cask Impact Limiter Drop Evaluation"
3.18 Roark's Formulas for Stress and Strain, Sixth Edition, McGraw-Hill, New York, 1989.
3.19 AISC Guide to Design Criteria for Bolted and Riveted Joints, AMERICAN INSTITUTE OF STEEL CONSTRUCTION, 2nd Ed. 2001.
3.20 RT100 NM 1000 D - Ens emballage RT100 (Robatel RT100 Cask Bill of Materials), Rev. D
3.21 Parker O-Ring Handbook, 50th Anniversary Ed., ORD 5700, Copyright 2007, Parker Hannifin Corporation, Cleveland, OH

## 4.0 Assumptions

4.1 The weight of the cask for the analytical evaluation of the hypothetical accident is considered as the total weight of the cask and the maximum payload. The damage sustained by the cask and the impact limiters during the free drop evaluations does not result in any significant reduction in load, so no reduction is considered. This assumption is acceptable without further evaluation.

4.2 It is assumed that the bolts are replaced after 75 round trips, for a total distance of approximately 1,610,000 km (1,000,000 miles). This is consistent with standard operating practices and will be reflected in the cask operating manual. This assumption is acceptable without further evaluation.

4.3 It is possible for the center of gravity of the payload to shift ±10% of the interior dimensions of the cask containment (Ref. 3.5). This shift has no effect on the bolt loading conditions except for the end drop, corner drop and pin puncture load cases. The bolt load equations used in accordance with Reference 3.5 consider the payload weight as a pressure load distributed across the cask lid interior surface in order to maximize the combination of bolt tension and prying loads (see Section 4.6, Ref. 3.5). This load application, along with the assumptions made in the derivation of the bolt loads, is very conservative (see Section 2.2, Ref. 3.5). In

addition, if the cask-cavity radius is much larger than the cask wall thickness, the location of the applied load does not have a significant impact on the resulting average and prying loads (see Appendix III, Ref. 3.5). This shift of the center of gravity will therefore have no significant impact on the results of the analysis and the methodology of NUREG/CR-6007 (Ref. 3.5) is still valid and acceptable for use. This assumption is acceptable without further evaluation.

There are no unverified assumptions in this calculation. Other design assumptions used, if any, will be noted and referenced as needed in the body of the calculation.

## 5.0 Design Inputs

5.1 The total weight of the fully assembled cask is 33,824kg (Ref. 3.2).
5.2 The maximum payload weight is 15,000 lbs, or 6,803.9 kg (see Appendix 2). Conservatively, a value of 7,060kg will be used for the evaluation.
5.3 The weight of the upper impact limiter is 2,541kg (Ref. 3.2).
5.4 The weight of the primary lid is 3,648kg (Ref. 3.2).
5.5 The weight of the secondary lid is 857kg (Ref. 3.2).
5.6 The material properties used for the cask shell, the lead shielding and the lid bolts shall be as given in Table 1, unless noted otherwise.
5.7 All allowable stresses for the cask shell and the lid bolts shall be as given in Table 2, unless noted otherwise.
5.8 A value of 9.81 $m/s^2$ will be used for the gravitational acceleration.
5.9 A value of 0.31 will be used for the Poisson's Ratio of all stainless steel in accordance with Table NF-2 of Reference 3.6.
5.10 Nut factors (k) for bolt preloads will be in accordance with Table G of Reference 3.9. For the non-lubricated condition, the 0.30 mean value of as-received stainless steel fasteners will be used. For the lubricated condition, the 0.15 mean value of Fel-Pro N 5000 (paste) will be used.

**Table 1 - Material Properties**

| Material | Temp. (°C) | Strength (MPa) | | | Young's Modulus (GPa) | Coefficient of Thermal Expansion ($10^{-6}$ m/m) |
|---|---|---|---|---|---|---|
| | | Yield ($S_y$) | Ultimate ($S_u$) | Membrane Allowable ($S_m$) | | |
| X2CRNI19.11 (ASTM A240 Type 304L)(1) | -29 | 172.4 | 482.6 | 115.1 | 198.6 | - |
| | 20 | 172.4 | 482.6 | 115.1 | 195.2 | 8.5 |
| | 50 | 166.9 | 476.7 | 115.1 | 192.9 | 8.7 |
| | 100 | 145.7 | 451.7 | 115.1 | 189.2 | 8.9 |
| | 150 | 132.1 | 421.6 | 115.0 | 186.1 | 9.2 |
| | 200 | 121.6 | 406.1 | 109.4 | 182.3 | 9.5 |
| | 250 | 114.4 | 397.9 | 102.7 | 179.2 | 9.7 |
| Europe Grade 10.9 (ASTM A354 Gr. BD) (Lid Bolts)(1) | -29 | 896.3 | 1034.2 | 206.8 | 204.8 | - |
| | 20 | 896.3 | 1034.2 | 206.8 | 201.4 | 6.4 |
| | 50 | 879.8 | 1034.2 | 206.8 | 199.7 | 6.5 |
| | 100 | 817.8 | 1034.2 | 206.8 | 196.8 | 6.7 |
| | 150 | 792.4 | 1034.2 | 206.8 | 193.7 | 6.9 |
| | 200 | 767.5 | 1034.2 | 206.8 | 191.2 | 7.1 |
| | 250 | 736.5 | 1034.2 | 206.8 | 187.6 | 7.3 |

Notes:
1. Material properties are taken from ASME B&PV Code, Section II, Part D (Ref. 3.6) by interpolation.

**Table 2 - Allowable Stresses**

| | | | Material | |
|---|---|---|---|---|
| | | | ASTM A240 Type 304L | ASTM A354 Gr. BD |
| Yield Stress, $S_y$ | | (MPa) | 172.4 (1) | 896.3 (1) |
| Ultimate Stress, $S_u$ | | (MPa) | 482.6 (1) | 1034.2 (1) |
| Design Intensity Stress (Membrane), $S_m$ | | (MPa) | 115.1 (1) | 201.4 (1) |
| Normal Conditions | Membrane Stress | (MPa) | 115.1 (2) | 434.4 (4) |
| | Membrane + Bending Stress | (MPa) | 172.7 (2) | 651.5 (3) |
| | Peak Stress | (MPa) | 345.3 (3) | 1034.2 (4) |
| Hypothetical Accident Conditions | Membrane Stress | (MPa) | 276.2 (4) | 723.9 (4) |
| | Membrane + Bending Stress | (MPa) | 414.4 (4) | 1034.2 (4) |
| | Peak Stress | (MPa) | 965.2 (5) | 2068.4 (5) |

Notes:

1. Allowable stresses are taken from ASME B&PV Code, Section II, Part D (Ref. 3.6), by interpolation (see Table 1).
2. Allowable stresses are established per Regulatory Guide 7.6 (Ref. 3.8).
3. Allowable stresses are established per Regulatory Guide 7.6 (Ref. 3.8), Regulatory Position 4 and ASME, Section III, Division 3 (Ref. 3.7), WB-3200 criteria. The limit on this stress component is $3S_m$.
4. Regulatory Guide 7.6 does not provide criteria. ASME Section III, Appendix F (Ref. 3.7) has been used to establish these criteria.
5. Regulatory Guide 7.6 (Ref. 3.8), Regulatory Position 7 and ASME Section III, Division 3, (Ref. 3.7) WB-3221.9 criteria for limiting these stresses to $2S_a$ at ten (10) cycles results in higher allowable stresses than $2S_u$. The limits for peak stresses are therefore conservatively set at $2S_u$.

## 6.0 Methodology

The RT-100 transport cask will be a safety-related structure in accordance with 10CFR Part 71 (Ref. 3.1). The cask consists of a stainless steel containment structure with a lead shielding panel between the inner and outer cask walls, a pair of ductile steel and foam impact limiters and two sets of stainless steel closure bolts at the primary and secondary closure lids (Ref. 3.2). The bolt loadings under the various normal and accident conditions are determined, where applicable, in accordance with the recommendation of NUREG/CR-6007 (Ref. 3.5). The loads determined in accordance with this methodology are very conservative. In addition, extremely conservative dynamic amplification factors will be used for determination of the bolt loads under the accident drop load cases. Therefore, factors of safety only slightly above 1.0 are considered to have ample conservatism and are therefore acceptable. For additional information on the conservatism used in the various calculations, see the specific evaluations in the following section.

## 7.0 Calculations

**NOTE:** In many cases, calculations are made using exact values, not the rounded numbers shown. Therefore, in certain situations, the numbers displayed may not be capable of providing the final solution. Using exact numbers, however, provides the most accurate solution possible.

### 7.1 Closure Bolt Evaluation

The RT-100 package is designed with two sets of closure bolts: 18 M36 hex head bolts at the secondary lid and 32 M48 hex head bolts at the primary lid. These two sets of bolts are credited with maintaining positive closure of the package under all accident conditions. The purpose of this calculation is to structurally qualify these bolts for the loadings associated with the normal conditions of transport and the hypothetical accident conditions.

7.1.1 Methodology
Bolt loadings under the various normal and accident conditions are determined in accordance with the recommendations of NUREG/CR-6007. Stresses resulting from these loads are compared with the design criteria in Section 2.1.2.2.

7.1.2 Loads
The following loads are evaluated in this section:

- Internal pressure loads
- Temperature loads
- Bolt preload
- Impact loads
- Puncture loads
- External pressure loads
- Gasket seating load

These loads are combined per NUREG/CR-6007 in Section 7.1.3.

7.1.2.1 Internal Pressure Loads
Per NUREG/CR-6007 (Table 4.3), the forces and moments generated under the internal pressure load are a tensile load, Fap, a shear load, Fsp, a fixed edge closure force, Ffp, and a fixed edge closure moment, Mfp. These are evaluated as follows for the primary and secondary lid bolts.

7.1.2.1.1 Internal Pressure Loads for Primary Lid Closure Bolts

The tensile force per bolt due to internal pressure, Fap, is:

$$F_{ap} = \frac{\pi \times D_{lg}^2 \times (P_{li} - P_{lo})}{4 \times N_b}$$

where,

| | | | |
|---|---|---|---|
| Dlg | = | Outer Seal Diameter | |
| | = | 1835 mm | (Ref. 3-2) |
| Nb | = | Number of Bolts | |
| | = | 32 | (Ref. 3.2) |
| Pli | = | Internal Pressure | |
| | = | 35 psi = 241.3 kN/m² use 250 kN/m² | (Calc TH-102) |
| Plo | = | External Pressure | |
| | = | 0 kN/m² (conservative) | |

Thus,

$$F_{ap} = \frac{\pi \times 1.835^2 \times (250 - 0)}{4 \times 32} = 20.7 \text{ kN/bolt}$$

The shear force per bolt due to internal pressure, Fsp, is:

$$F_{sp} = \frac{\pi \times E_l \times t_l \times (P_{li} - P_{lo}) \times D_{lb}^2}{2 \times N_b \times E_c \times t_c \times (1 - N_{ul})}$$

where,

| | | | |
|---|---|---|---|
| El | = | Primary Lid Material Elastic Modulus, (SA 240 TYPE 304/304L per Table 2) | |
| | = | 195.2 GPa at 20° C | (Table 1) |
| Dlb | = | Primary Lid Bolt Circle Diameter | |
| | = | 1920 mm | (Ref. 3.2) |
| Nul | = | Primary Lid Material Poisson's Ratio, (SA 240 TYPE 304/304L per Table 2) | |
| | = | 0.31 | (Table 1) |
| Ec | = | Cask Material Elastic Modulus, (SA 240 TYPE 304/304L per Table 2) | |
| | = | 195.2 GPa at 20° C | (Table 1) |
| tl | = | Primary Lid Thickness | |
| | = | 210 mm | (Ref. 3.2) |
| tc | = | Cask Wall Thickness | |
| | = | 65 mm (neglecting lead) | (Ref. 3.2) |

The remaining terms are as previously defined. However, this expression for shear force does not apply to the RT-100 cask design because the maximum gap between the lid and cask body (just 4 mm =1741 – 1737 per Ref. 3.2, Detail 1) is less than the minimum gap between the bolt clearance holes and bolt shank ( 5.5 mm = 52.5- 47 per Ref. 3.2 and Machinery's Handbook). Thus, the RT-100 primary lid bolts will not be subjected to any shear loads. Therefore,

$$F_{sp} = 0.0 \text{ kN/bolt.}$$

The fixed edge closure force, $F_{fp}$, and moment, $M_{fp}$, are:

$$F_{fp} = \frac{D_{lb} \times (P_{li} - P_{lo})}{4} = \frac{1.920 \times (250 - 0)}{4} = 120.0 \text{ kN/m}$$

and,

$$M_{fp} = \frac{(P_{li} - P_{lo}) \times D_{lb}^2}{32} = \frac{(250 - 0) \times 1.920^2}{32} = 28.8 \text{ kN-m/m.}$$

**7.1.2.1.2 Internal Pressure Load for Secondary Lid Closure Bolts**

The secondary lid closure bolt forces and moments are determined using the same methodology and equations as shown for the primary lid bolts (Section 7.1.2.1.1) except that the secondary lid features are used.

The tensile force per bolt due to internal pressure, $F_{as}$, is:

$$F_{as} = \frac{\pi \times D_{lg}^2 \times (P_{li} - P_{lo})}{4 \times N_b}$$

where,

| | | | |
|---|---|---|---|
| $D_{lg}$ | = | Outer Seal Diameter | |
| | = | 790 mm | (Ref 3.2) |
| $N_b$ | = | Number of Bolts | |
| | = | 18 | (Ref. 3.2) |

$P_{li}$ = Internal Pressure
= 35 psi = 241.3 kN/m² use 250 kN/m² (Calc TH-102)
$P_{lo}$ = External Pressure
= 0 kN/m² (conservative)

Thus,

$$F_{as} = \frac{\pi \times 0.790^2 \times (250-0)}{4 \times 18}$$

$$= 6.8 \text{ kN/bolt}$$

The *maximum* gap between the lid and cask body (just 4 mm =748 – 744 per Ref. 3.2, Detail 2) is less than the *minimum* gap between the bolt clearance holes and bolt shank (5.5 mm = 40.5- 35 per Ref. 3.2 and Machinery's Handbook). Thus, as with the primary lid (Section 7.1.2.1.1), the shear force per bolt due to internal pressure, $F_{ss}$, is:

$$F_{ss} = 0.0 \text{ kN/bolt.}$$

The fixed edge closure force, $F_{fs}$, and moment, $M_{fs}$, are:

$$F_{fs} = \frac{D_{lb} \times (P_{li} - P_{lo})}{4}$$

and,

$$M_{fs} = \frac{(P_{li} - P_{lo}) \times D_{lb}^2}{32}$$

where ,

$D_{lb}$ = Secondary Lid Bolt Diameter
= 926 mm (Ref. 3.2)

All other terms are previously defined. Thus,

$$F_{fs} = \frac{D_{lb} \times (P_{li} - P_{lo})}{4} = \frac{0.926 \times (250-0)}{4} = 57.9 \text{ kN/m}$$

and,

$$M_{fs} = \frac{(P_{li} - P_{lo}) \times D_{lb}^2}{32} = \frac{(250-0) \times 0.926^2}{32} = 6.7 \text{ kN-m/m.}$$

**7.1.2.2 Temperature Loads**
Temperature differentials and/or differences in the thermal-expansion coefficients of the joint components induce bolt loads. These forces are evaluated Per NUREG/CR-6007 (Table 4.4).

**7.1.2.2.1 Temperature Loads for Primary Lid Closure Bolts**
The tensile force per bolt due to temperature, $F_{atp}$, is:

$$F_{atp} = 0.25 \times \pi \times D_b^2 \times E_b \times (\alpha_l \times T_l - \alpha_b \times T_b)$$

where,

$D_b$ = Nominal Bolt diameter
= 48 mm (Ref 3.2)
$E_b$ = Bolt Material Elastic Modulus, (SA 354 Grade BD per Table 2)
= 201.4 GPa at 20° C (Table 1)

| | | | |
|---|---|---|---|
| $\alpha_l$ | = | Primary Lid Material Coefficient of Thermal Expansion | |
| | = | $9.2\times10^{6}$ m/m/°C | (Table 1) |
| $\alpha_b$ | = | Bolt Material Coefficient of Thermal Expansion | |
| | = | $6.4\times10^{6}$ m/m/°C | (Table 1) |
| $T_l$ | = | Maximum Primary Lid Temperature under NCT conditions | |
| | = | 71°C (conservatively use 150 °C) | (Table 3.1-2) |
| $T_b$ | = | Minimum Bolt Temperature under NCT conditions | |
| | = | -40 °C | (10 CFR 71.71) |

Thus,

$$F_{atp} = 0.25\times\pi\times0.048^2\times201.4\times10^6\times\left(9.2\times10^{-6}\times150-6.4\times10^{-6}\times-40\right)$$
$$= 596.2 \text{ kN/bolt}$$

Shear force per bolt due to temperature, $F_{stp}$, is considered zero because the clamped components (the primary lid and the cask forged ring) have essentially the same temperature.

**7.1.2.2.2 Temperature Loads for Secondary Lid Closure Bolts**

The secondary lid closure bolt forces determined using the same methodology and equations as shown for the primary lid bolts (Section 7.1.2.2.1) with the secondary lid geometry, material properties and temperatures. Since the primary and secondary lids are constructed of the same materials and experience essentially the same temperatures, only the bolt diameter in the previous equation must be changed. The secondary bolt diameter is 36 mm. Thus, the tensile force per unit bolt due to temperature, $F_{stl}$, is:

$$F_{atl} = 0.25\times\pi\times0.036^2\times201.4\times10^6\times\left(9.2\times10^{-6}\times150-6.4\times10^{-6}\times-40\right)$$
$$= 335.4 \text{ kN/bolt}$$

As with the primary lid, the shear force per bolt due to temperature, $F_{sts}$, is considered zero because the clamped components (the primary and secondary lids) have essentially the same temperature.

**7.1.2.3 Bolt Preloads**

Tightening torques for the primary and secondary lid bolts are, respectively, 850 N-m +/-10 % and 350 +/-10 % per Ref 3.2. The method of analysis is described in NUREG/CR-6007, Table 4.1.

**7.1.2.3.1 Bolt Preload for Primary Lid Closure Bolts**

The primary lid bolt preload, $F_{pl}$, is determined as follows (NUREG/CR-6007, Table 4.1):

$$F_{pl} = \frac{T}{K_L\times D_b}$$

where,

| | | | |
|---|---|---|---|
| $D_b$ | = | Nominal Bolt diameter | |
| | = | 48 mm | (Ref 3.2) |
| K | = | Nut Factor for empirical relation between applied torque and the achieved preload | |
| | = | 0.15 (lubricated) minimum<br>0.30 (dry) maximum | (EPRI Good Bolting Practices) |
| T | = | Applied Torque | |
| | = | 850 N-m +/-10% | (Ref 3.2) |

To determine the maximum preload, $F_{plmax}$, for the primary lid bolts, the minimum nut factor, K, of 0.15 (lubricated) and maximum tightening torque of 940 N-m (conservatively bounds the 850 N-m+10% maximum):

$$F_{plmax} = \frac{T_{max}}{K_L \times D_b} = \frac{940}{0.15 \times 0.048} \times \frac{1\,kN}{1000\,N}$$
$$= 130.6 \text{ kN}$$

The residual torsion moment, $M_{rl}$, is:

$$M_{rl} = 0.5 \times T_{max} = 0.5 \times 940$$
$$= 470 \text{ N-m}$$

The residual tensile bolt force, $F_{arl}$, is

$$F_{arl} = F_{plmax} = 130.6 \text{ kN}$$

**7.1.2.3.2 Bolt Preload for Secondary Lid Closure Bolts**

The maximum secondary lid bolt preload, $F_{psmax}$, is determined in a manner similar to the primary bolt lids (Section 7.1.2.3.2). Thus,

$$F_{psmax} = \frac{T_{max}}{K_L \times D_b}$$

where,

| | | | |
|---|---|---|---|
| $D_b$ | = | Nominal Bolt diameter | |
| | = | 36 mm | (Ref 3.2) |
| T | = | Applied Torque | |
| | = | 350 N-m +/-10% | (Ref 3.2) |

Other terms are as previously defined. Using a nut factor, K, of 0.15 (lubricated) and a tightening torque of 390 N-m (conservatively bounding the 350 N-m+10% maximum) yields the maximum preload, $F_{psmax}$, for the secondary lid bolts:

$$F_{psmax} = \frac{T_{max}}{K_L \times D_b} = \frac{390}{0.15 \times 36}$$
$$= 72.2 \text{ kN}$$

The residual torsion moment, $M_{rs}$, is:

$$M_{rs} = 0.5 \times T_{max} = 0.5 \times 390$$
$$= 195.0 \text{ N-m}$$

The residual tensile bolt force, $F_{ars}$, is:

$$F_{ars} = F_{psmax} = 72.2 \text{ kN}$$

#### 7.1.2.4 Impact Loads

Maximum tension and shear loads in the closure bolts due to the regulatory impact drops are evaluated in accordance with NUREG/CR-6007. Using the NUREG terminology, the primary lid bolts are evaluated as closure bolts for an *unprotected* lid and the secondary bolts are evaluated as components of a *protected* lid. This means the primary bolt evaluation includes the impact or inertial forces of the entire cask and the secondary lid bolts are evaluated only for the forces due to the inertia of the secondary lid.

##### 7.1.2.4.1 Dynamic Load Factors

Drop impact loadings are generally considered triangular or half sine loadings and NUREG/CR-3966 presents dynamic load factor (DLF) charts for either pulse shape. For this analysis, we compare results and utilize the loading with the higher DLF.

Dynamic load factors for triangular and half sine loadings are shown in Figures 2.3 and Figure 2.15 (NUREG/CR-3966). This information is presented as graphs where the DLF is the ordinate and the abscissa is $t_d/T$. This latter quantity, $t_d/T$, is the ratio of the impact duration, $t_d$, and the natural period of the impacting object, T.

For bolt closure analyses, the period of the lids, T, is considered. T is determined from the lid lowest mode frequency.

###### 7.1.2.4.1.1 Dynamic Load Factors for Primary Lid Closure Bolts

To determine the primary lid frequency, the primary lid and secondary lid are considered a single simply-supported flat circular plate. Thus, (Roark, Table 36, Case 1la):

Resonant Frequency of Primary Lid (with secondary lid attached):

$$f_l = \frac{4.99}{2\pi}\sqrt{\frac{Dg_c}{wr^4}}$$

where,

$D$ = Lid Flexural Rigidity

$$= \frac{E_l\, t_l^3}{12\left(1 - Nu_l^{\,2}\right)}$$

$g_c$ = conversion factor

= 1000 kg-mm/s$^2$-N

w = weight per unit area

r = lid bolt radius

= $D_{lb}/2$

= 960 mm

Per 3.2, the primary lid weighs 3648 kg and the secondary lid weighs 857 kg. Thus,

$$w = \frac{(3648+857)}{\pi \cdot 960^2}$$

$$= 0.001556 \text{ kg/mm}^2$$

*D* may be determined from previously defined values:

$$D = \frac{195.2\times10^3 \cdot 210^3}{12\left(1-0.31^2\right)}$$

$$= 1.667\text{x}10^{11} \text{ N-mm}$$

The frequency of the primary lid (with attached secondary lid) is:

$$f_l = 0.7942 \cdot \sqrt{\frac{1.667 \times 10^{11} \cdot 1000}{0.001556 \cdot 960^4}}$$

$$= 282 \text{ Hz}$$

The period of the primary lid is equal to $1/f_l$, or T = 1/282 = 0.00354 s. Impact durations for the NCT and HAC impacts range from 0.012 s to 0.045 s (ST-0401). Thus, the smallest value of the ratio $t_d/T$ is 3.389 and the largest is 12.71. With these values, the maximum DLF was determined from Figures 2.3 and 2.15 (NUREG/CR-6007) to be less than 1.2. Thus, it is concluded that the DLF for the primary lid bolts may be conservatively bounded by a value of 1.2 for both NCT and HAC drops.

**7.1.2.4.1.2 Dynamic Load Factors for Secondary Lid Closure Bolts**

To determine the secondary lid frequency, the secondary lid is considered a simply-supported flat circular plate. Thus, (Roark, Table 36, Case 11a):

Resonant Frequency of Secondary Lid:

$$f_l = 0.7942\sqrt{\frac{Dg_c}{wr^4}}$$

where,

$D$ = Lid Flexural Rigidity

$$= \frac{E_l\, t_l^3}{12\left(1 - Nu_l^{\,2}\right)}$$

$g_c$ = conversion factor
= 1000 kg-mm/s$^2$-N

w = weight per unit area

r = lid bolt radius
= $D_{lb}/2$
= 463 mm

Per 3.2, the secondary lid weighs 857 kg. Furthermore,

$E_l$ = Secondary Lid Material Elastic Modulus, (SA 240 TYPE 304/304L per Table 2)
= 195.2 GPa at 20° C (Table 1)

$D_{lb}$ = Secondary Lid Bolt Circle Diameter
= 1920 mm (Ref. 3.2)

$N_{ul}$ = Secondary Lid Material Poisson's Ratio, (SA 240 TYPE 304/304L per Table 2)
= 0.31 (Table 1)

$t_l$ = Secondary Lid Thickness
= 110 mm (stainless steel only) (Ref. 3.2)

Thus,

$$w = \frac{857}{\pi \cdot 463^2}$$

$$= 0.001272 \text{ kg/mm}^2$$

$D$ may be determined from previously defined values:

$$D = \frac{195.2 \times 10^3 \cdot 110^3}{12(1 - 0.31^2)}$$

$$= 2.395 \times 10^{10} \text{ N-mm}$$

The frequency of the secondary lid is:

$$f_l = 0.7942 \cdot \sqrt{\frac{2.395 \times 10^{10} \cdot 1000}{0.001272 \cdot 463^4}}$$

$$= 508 \text{ Hz}$$

The period of the secondary lid is equal to 1/ fl, or T = 1/508 = 0.00197 s. Impact durations for the NCT and HAC impacts range from 0.012 s to 0.045 s (ST-0401). Thus, the value of td/T is 4.6 or more. With this value, the maximum DLF can be determined from Figures 2.3 and 2.15 (NUREG/CR-6007) to be approaching unity. However, for consistency with the primary lid bolt analyses, the DLF for the secondary lid bolts will be conservatively set to 1.2 for both NCT and HAC drops.

**7.1.2.4.2 End Drop Loads**

**7.1.2.4.2.1 Primary Lid Bolts**

Impact loads in the primary lid bolts due to an end drop are determined using the formulas for evaluating bolt forces/moments generated by impact load applied to an unprotected closure in NUREG/CR-6007 Table 4.5. An acceleration of 125 g is used in this analysis (which bounds the 123 g maximum reported in Section 2.13.4.2).

The non-prying tensile bolt force per primary lid bolt, $F_{tp}$, is:

$$F_{tp} = \frac{1.34 \times \sin(x_i) \times DLF \times a_i \times (W_L + W_c) \times g}{N_b}$$

where,

| | | | |
|---|---|---|---|
| $x_i$ | = | End Drop Impact Angle | |
| | = | 90° | |
| DLF | = | 1.15 | (Section 7.1.2.4.1) |
| $a_i$ | = | Maximum Impact Acceleration | |
| | = | 123 g (use 125 g) | (ST-0401) |
| $W_L$ | = | Closure Lid Weight | |
| | = | 3648 kg (use 3650 kg) | (3.2) |
| $W_c$ | = | Cask Payload Weight | |
| | = | 6804 kg (use 7000 kg) | (3.2) |
| $N_b$ | = | Number of Bolts | |
| | = | 32 | (3.2) |

Thus,

$$F_{tp} = \frac{1.34 \times \sin(90.0) \times 1.15 \times 125 \times (3650 + 7000) \times 9.81}{32} \times \frac{1\text{kN}}{1000\text{N}}$$

$$= 628.9 \text{ kN/bolt}$$

As discussed in Section 7.1.2.1.1, the RT-100 primary lid bolts will not be subjected to any shear loads. Thus,

$$F_{sp} = 0.0 \text{ kN/bolt.}$$

The fixed edge closure lid force, $F_f$, is:

$$F_f = \frac{1.34 \times \sin(x_i) \times DLF \times a_i \times (W_L + W_c) \times g}{\pi \times D_{lb}}$$

where,

$D_{lb}$ = Primary Lid Bolt Diameter
= 1920 mm (Ref. 3.2)

The remaining terms are as previously defined. Thus,

$$F_f = \frac{1.34 \times \sin(90.0) \times 1.15 \times 125 \times (3650 + 7000) \times 9.81}{\pi \times 1.920} \times \frac{1\,kN}{1000\,N}$$

= 3336.4 kN/m

The fixed edge closure lid moment, $M_f$, is:

$$M_f = \frac{1.34 \times \sin(x_i) \times DLF \times a_i \times (W_L + W_c) \times g}{\pi \times 8}$$

All other terms are as previously defined. Thus,

$$M_f = \frac{1.34 \times \sin(90.0) \times 1.15 \times 125 \times (3650 + 7000) \times 9.81}{\pi \times 8} \times \frac{1\,kN}{1000\,N}$$

= 800.7 kN-m/m

The additional tensile bolt force per bolt, Ftp, caused by the prying action of the primary lid is (NUREG/CR-6007 Table 2.1):

$$F_{tp} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left[\frac{\frac{2 \times M_f}{D_{lo} - D_{lb}} - C1 \times (B - F_f) - C2 \times (B - P)}{C1 + C2}\right]$$

where,

P = Bolt Preload per unit Length of Bolt Circle
= $F_{plmax} \times \frac{N_b}{\pi \times D_{lb}}$
B = Non-prying Tensile Bolt Force
= MAX($F_f$, P)
C1 = Force Constant
= 1.0
C2 = Second Force Constant

$$= \left(\frac{8}{3\times(D_{lo}-D_{lb})^2}\right)\times\left[\frac{E_l\times t_l^3}{1-N_{ul}}+\frac{(D_{lo}-D_{li})\times E_{lf}\times t_{lf}^3}{D_{lb}}\right]\times\left(\frac{L_b}{N_b\times D_b^2\times E_b}\right)$$

$D_{lo}$ = Closure Lid Diameter at Outer Edge
= 2016 mm (3.2)

$D_{li}$ = Closure Lid Diameter at Inner Edge
= 1730 mm (3.2)

$t_{lf}$ = Closure Lid Flange Thickness
= 120 mm (3.2)

$E_{lf}$ = Primary Lid Flange Material Elastic Modulus, (SA 240 TYPE 304/304L per Table 2)
= 195.2 GPa at 20° C (Table 1)

$L_b$ = Bolt length between the top and bottom surfaces of the closure lid at the bolt circle
= 67 mm (3.2)

Other terms are as previously defined. Thus,

$$C2 = \left(\frac{8}{3\times(D_{lo}-D_{lb})^2}\right)\times\left[\frac{E_l\times t_l^3}{1-N_{ul}}+\frac{(D_{lo}-D_{li})\times E_{lf}\times t_{lf}^3}{D_{lb}}\right]\times\left(\frac{L_b}{N_b\times D_b^2\times E_b}\right)$$

$$= \left(\frac{8}{3\times(2.016-1.920)^2}\right)\times\left[\frac{195.2\times10^6\times0.210^3}{1-0.31}+\frac{(2.016-1.730)\times195.2\times10^6\times0.120^3}{1.920}\right]\times\left(\frac{0.067}{32\times0.048^2\times201.4\times10^6}\right)$$

$$= 3.49$$

$$P = 130.6\times\frac{32}{\pi\times1.920}$$

$$= 692.9 \text{ kN/m}$$

and

$$F_{tp} = \left(\frac{\pi\times1.920}{32}\right)\times\left[\frac{\frac{2\times800.7}{2.016-1.920}-1\times(3336.4-3336.4)-3.49\times(3336.4-692.9)}{1+3.49}\right]$$

= 313.7 kN/bolt

The total tension force, $F_a$, is

$$F_a = F_t + F_{tp} = 628.9 + 313.7 = 942.6 \text{ kN/bolt}$$

The shear force, $F_s$, is 0.

The maximum bending moment generated by the applied loads, $M_{bb}$, is:

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

where,

$$Kb = \left(\frac{N_b}{L_b}\right) \times \left(\frac{E_b}{D_{lb}}\right) \times \left(\frac{D_b^4}{64}\right)$$

$$= \left(\frac{32}{0.067}\right) \times \left(\frac{201.4 \times 10^6}{1.920}\right) \times \left(\frac{0.048^4}{64}\right)$$

= 4,155 kN

$$K1 = \frac{E_l \times t_l^3}{3 \times \left[\left(1 - N_{ul}^2\right) + \left(1 - N_{ul}\right)^2 \times \left(\frac{D_{lb}}{D_{lo}}\right)^2\right] \times D_{lb}}$$

$$= \frac{195.2 \times 10^6 \times 0.210^3}{3 \times \left[\left(1 - 0.31^2\right) + \left(1 - 0.31\right)^2 \times \left(\frac{1.920}{2.016}\right)^2\right] \times 1.920}$$

= 234,960 kN

Thus,

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

$$= \left(\frac{\pi \times 1.920}{32}\right) \times \left(\frac{4,155}{4,155 + 234,960}\right) \times 800.7$$

= 2.6 kN-m

**7.1.2.4.2.2 Secondary Lid Bolts**

Impact loads in the secondary lid bolts due to an end drop are determined similarly as for the primary lid bolts in Section 7.1.2.4.2.1.

The non-prying tensile bolt force per secondary lid bolt, $F_{ts}$, is:

$$F_{ts} = \frac{1.34 \times \sin(x_i) \times DLF \times a_i \times (W_L + W_{cs}) \times g}{N_b}$$

where,

| | | | |
|---|---|---|---|
| $x_i$ | = | End Drop Impact Angle | |
| | = | 90° | |
| DLF | = | 1.15 | (Section 7.1.2.4.2.2) |
| $a_i$ | = | Maximum Impact Acceleration | |
| | = | 123 g (use 125 g) | (ST-0401) |
| $W_L$ | = | Closure Lid Weight | |
| | = | 857 kg (use 860 kg) | (3.2) |
| $W_{cs}$ | = | Payload Weight borne by Secondary Lid | |
| $N_b$ | = | Number of Bolts | |
| | = | 18 | (3.2) |

Since the payload weight is assumed to be evenly distributed across both the primary and secondary lids, the weight borne by the secondary lid can be obtained by multiplying the payload weight by the ratio of areas, i.e.,

$$W_{cs} = \frac{A_s}{A_p} \times W_c = \left(\frac{D_s}{D_p}\right)^2 \times W_c$$

$$= \left(\frac{1.000}{2.016}\right)^2 \times 7000$$

=1722 kg

Thus,

$$F_{ts} = \frac{1.34 \times \sin(90.0) \times 1.15 \times 125 \times (860 + 1722) \times 9.81}{18} \times \frac{1\,\text{kN}}{1000\,\text{N}}$$

= 266.0 kN/bolt

As discussed in Section 7.1.2.1.1.2, the RT-100 secondary lid bolts will not be subjected to any shear loads. Thus,

$$F_s = 0.0 \text{ kN/bolt}$$

The fixed edge closure lid force, $F_f$, is:

$$F_f = \frac{1.34 \times \sin(x_i) \times DLF \times a_i \times (W_L + W_{cs}) \times g}{\pi \times D_{lb}}$$

where,

| | | | |
|---|---|---|---|
| $D_{lb}$ | = | Secondary Lid Bolt Diameter | |
| | = | 926 mm | (Ref. 3.2) |

The remaining terms are as previously defined. Thus,

$$F_f = \frac{1.34 \times \sin(90.0) \times 1.15 \times 125 \times (860 + 1722) \times 9.81}{\pi \times 0.926} \times \frac{1\,\text{kN}}{1000\,\text{N}}$$

= 1646.1 kN/m

The fixed edge closure lid moment, $M_f$, is:

$$M_f = \frac{1.34 \times \sin(x_i) \times DLF \times a_i \times (W_L + W_{cs}) \times g}{\pi \times 8}$$

where all terms are as previous defined. Thus,

$$M_f = \frac{1.34 \times \sin(90.0) \times 1.15 \times 125 \times (860 + 1722) \times 9.81}{\pi \times 8} \times \frac{1\,kN}{1000\,N}$$

= 190.5 kN-m/m

The additional tensile bolt force per bolt, Ftp, caused by the prying action of the secondary lid is (NUREG/CR-6007 Table 2.1):

$$F_{tp} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left[\frac{\frac{2 \times M_f}{D_{lo} - D_{lb}} - C1 \times (B - F_f) - C2 \times (B - P)}{C1 + C2}\right]$$

where,

P = Bolt Preload per unit Length of Bolt Circle

$$= F_{plmax} \times \frac{N_b}{\pi \times D_{lb}}$$

B = Non-prying Tensile Bolt Force

= MAX($F_f$, P)

C2 = Force Constant

= 1.0

C2 = Second Force Constant

$$= \left(\frac{8}{3 \times (D_{lo} - D_{lb})^2}\right) \times \left[\frac{E_l \times t_l^3}{1 - N_{ul}} + \frac{(D_{lo} - D_{li}) \times E_{lf} \times t_{lf}^3}{D_{lb}}\right] \times \left(\frac{L_b}{N_b \times D_b^2 \times E_b}\right)$$

$D_{lo}$ = Closure Lid Diameter at Outer Edge

= 1000 mm (3.2)

$D_{li}$ = Closure Lid Diameter at Inner Edge

= 745 mm (3.2)

$t_{lf}$ = Closure Lid Flange Thickness

= 80 mm (3.2)

$E_{lf}$ = Secondary Lid Flange Material Elastic Modulus, (SA 240 TYPE 304/304L per Table 2)

= 195.2 GPa at 20° C (Table 1)

$L_b$ = Bolt length between the top and bottom surfaces of the closure lid at the bolt circle

= 43 mm (3.2)

Other terms are as previously defined. Thus,

$$C2 = \left(\frac{8}{3 \times (D_{lo} - D_{lb})^2}\right) \times \left[\frac{E_l \times t_l^3}{1 - N_{ul}} + \frac{(D_{lo} - D_{li}) \times E_{lf} \times t_{lf}^3}{D_{lb}}\right] \times \left(\frac{L_b}{N_b \times D_b^2 \times E_b}\right)$$

$$= \left(\frac{8}{3\times(1.000-0.926)^2}\right) \times\left[\frac{195.2\times10^6\times0.110^3}{1-0.31}+\frac{(1.000-0.745)\times195.2\times10^6\times0.080^3}{0.926}\right] \times\left(\frac{0.043}{18\times0.036^2\times201.4\times10^6}\right)$$

$$= 1.80$$

$$P = 72.2\times\frac{18}{\pi\times0.926}$$

$$= 446.7 \text{ kN/m}$$

and

$$F_{tp} = \left(\frac{\pi\times0.926}{18}\right) \times\left[\frac{\frac{2\times190.5}{1.000-0.926}-1.0\times(1646.1-1646.1)-1.80\times(1646.1-446.7)}{1.0+1.80}\right]$$

$$= 172.5 \text{ kN/bolt}$$

The total tension force, $F_a$, is

$$F_a = F_t + F_{tp}$$
$$= 266.0 + 172.5$$
$$= 438.5 \text{ kN/bolt}$$

The shear force, $F_s$, is 0.

The maximum bending moment generated by the applied loads, $M_{bb}$, is (NUREG/CR-6007, Table 2.2):

$$M_{bb} = \left(\frac{\pi\times D_{lb}}{N_b}\right)\times\left(\frac{Kb}{Kb+K1}\right)\times Mf$$

where,

$$Kb = \left(\frac{N_b}{L_b}\right)\times\left(\frac{E_b}{D_{lb}}\right)\times\left(\frac{D_b^4}{64}\right)$$

$$= \left(\frac{32}{0.043}\right)\times\left(\frac{201.4\times10^6}{0.926}\right)\times\left(\frac{0.036^4}{64}\right)$$

$$= 4{,}247.8 \text{ kN}$$

$$K1 = \frac{E_l \times t_l^3}{3 \times \left[ \left(1 - N_{ul}^2\right) + \left(1 - N_{ul}\right)^2 \times \left(\frac{D_{lb}}{D_{lo}}\right)^2 \right] \times D_{lb}}$$

$$= \frac{195.2 \times 10^6 \times 0.110^3}{3 \times \left[ \left(1 - 0.31^2\right) + \left(1 - 0.31\right)^2 \times \left(\frac{0.926}{1.000}\right)^2 \right] \times 0.926}$$

= 71,276.1 kN

Thus,

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

$$= \left(\frac{\pi \times 0.926}{32}\right) \times \left(\frac{4{,}247.8}{4{,}247.8 + 71{,}276.1}\right) \times 190.5$$

= 1.0 kN-m

**7.1.2.4.3 Corner Drop Evaluations**

The closure bolt evaluations for the corner drop impact are conducted very similarly to the end drop analyses in Section 7.1.4.1. The cask body acceleration is changed and the impact angle, $x_i$, is set equal to 52.5° (corresponding to a 37.5° angle between cask axis and vertical line). Additionally, an acceleration of 120 g is used in this analysis (which bounds the 116 g maximum reported in Section 2.13.4.2). Results are summarized in Table 7.1.2.4.3-1.

**Table 7.1.2.4.3-1 Closure Bolt Loads for 9.0 m Corner-Drop**

| BOLT/LOCATION | Non-Prying Tensile Force, $F_t$ (kN/bolt) | Prying Force, $F_{tp}$ (kN/m) | Bending Moment, $M_{bb}$ (kN-m/bolt) | Shear Force, Fs (kN/bolt) |
|---|---|---|---|---|
| M48x170 Bolts /Primary Lid | 479.0 | 263.1 | 2.0 | 0.0 |
| M36x120 Bolts /Secondary Lid | 202.6 | 142.5 | 0.8 | 0.0 |

**7.1.2.4.4 Side Drop Evaluations**

As shown in Sections 7.1.2.1.1 and 7.1.1.2, the gap between the cask body and the primary and second lids are smaller than the gap between the bolts and the bolt clearance holes. There-fore, no shear load are imparted to the bolts from the cask body. Since the side impact drop will primarily generate shear loads with respect to the bolts, the primary and secondary closure lid bolts will not see any significant loading from the side impact drop and are acceptable with respect to the end and corner impact drop.

**7.1.2.5 Puncture Loads**

**7.1.2.5.1 End Puncture**

Puncture loads in the primary and secondary closure lid bolts due to a puncture are determined using the formulas for evaluating bolt forces/moments in NUREG/CR-6007 Table 4.7.

**7.1.2.5.1.1 Primary Lid Bolts**

The non-prying tensile bolt force per primary lid bolt, $F_{tp}$, is:

$$F_{tp} = \frac{\sin(x_i) \times P_{un}}{N_b}$$

where,

$x_i$ = End Drop Impact Angle
= 90°
$P_{un}$ = MIN($P_{un1}$ , $P_{un2}$)
$N_b$ = Number of Bolts
= 32

The term, $P_{un}$, is the maximum impact force that can be generated by the puncture pin during a normal impact. It is the smaller of:

$$P_{un1} = 0.75 \times \pi \times D_{pb}^2 \times S_{yl}$$

$$P_{un2} = 0.6 \times \pi \times D_{pb} \times t_l \times S_{ul}$$

where,

$D_{pb}$ = Puncture bar diameter
= 150 mm (10 CFR 71.73(c)(3))
$t_l$ = Closure Lid Thickness
= 100 mm (Ref 3.2)
(the secondary lid thickness neglecting the lead)
$S_{yl}$ = Yield Strength of Closure Lid Material
(SA 240 304L per Table 2)
= 172.4 MPa at 20° C (Table 1)
$S_{ul}$ = Ultimate Strength of Closure Lid Material
(SA 240 304L per Table 2)
= 324.0 MPa at 20° C (Table 1)

Thus,

$$P_{un1} = 0.75 \times \pi \times 0.150^2 \times 172400$$

= 9,139.7 kN

$$P_{un2} = 0.6 \times \pi \times 0.150 \times 0.100 \times 482600$$

= 13,645 kN

$P_{un}$ = MIN(9,139.7, 13,645)
= 9,139.7 kN

and

$$F_{tp} = \frac{\sin(90) \times 9139.7}{32}$$

= 285.6 kN/bolt.

As shown in Sections 7.1.2.2.1.1 and 7.1.2.2.1.1, the design of the primary and secondary lids prevents shear loads being applied to the bolts. Thus,

$F_{sp}$ = 0.

It is noted that the equation given for $F_{sp}$ in NUREG/CR6007 also shows $F_{sp} = 0$.

The fixed edge closure lid force, $F_f$, is:

$$F_f = \frac{\sin(x_i) \times P_{un}}{\pi \times D_{lb}}$$

where,

$D_{lb}$ = Primary Lid Bolt Circle Diameter
= 1920 mm (Ref. 3.2)

The remaining terms are as previously defined. Thus,

$$F_f = \frac{\sin(90) \times 9139.7}{\pi \times 1.920}$$
$$= 9{,}139.7 \text{ kN/m}$$

The fixed edge closure lid moment, $M_f$, is:

$$M_f = \frac{\sin(x_i) \times P_{un}}{4 \times \pi}$$

Thus,

$$M_f = \frac{\sin(90) \times 9139.7}{4 \times \pi}$$
$$= 727.3 \text{ kN-m/m}$$

The additional tensile bolt force per bolt, Ftp, caused by the prying action of the primary lid is (NUREG/CR-6007 Table 2.1):

$$F_{tp} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left[\frac{\frac{2 \times M_f}{D_{lo} - D_{lb}} - C1 \times (B - F_f) - C2 \times (B - P)}{C1 + C2}\right]$$

where,

P = Bolt Preload per unit Length of Bolt Circle
= 692.9 kN/m (as shown in Section 7.1.2.4.2.1)

B = Non-prying Tensile Bolt Force
= MAX($F_f$, P)

C1 = Force Constant
= 1.0

C2 = Second Force Constant

$$= \left(\frac{8}{3 \times (D_{lo} - D_{lb})^2}\right) \times \left[\frac{E_l \times t_l^3}{1 - N_{ul}} + \frac{(D_{lo} - D_{li}) \times E_{lf} \times t_{lf}^3}{D_{lb}}\right] \times \left(\frac{L_b}{N_b \times D_b^2 \times E_b}\right)$$

= 3.49 (as shown in Section 7.1.2.4.2.1)

$D_{lo}$ = Closure Lid Diameter at Outer Edge
= 2016 mm (3.2)

$D_{li}$ = Closure Lid Diameter at Inner Edge
= 1730 mm (3.2)
$t_{lf}$ = Closure Lid Flange Thickness
= 120 mm (3.2)
$E_{lf}$ = Primary Lid Flange Material Elastic Modulus, (SA 240 TYPE 304L per Table 2)
= 195.2 GPa at 20° C (Table 1)
$L_b$ = Bolt length between the top and bottom surfaces of the closure lid at the bolt circle
= 67 mm (3.2)

Other terms are as previously defined. Thus,

$$F_{tp} = \left(\frac{\pi \times 1.920}{32}\right) \times \left[\frac{\frac{2 \times 727.3}{2.016 - 1.920} - 1 \times (1515.2 - 1515.2) - 3.49 \times (1515.2 - 692.9)}{1 + 3.49}\right]$$

= 515.6 kN/bolt

The total tension force, $F_a$, is

$F_a$ = $F_t + F_{tp}$
= 285.6 + 515.6
= 801.2 kN/bolt

The shear force, $F_s$, is 0.

The maximum bending moment generated by the applied loads, $M_{bb}$, is:

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

where,

$$Kb = \left(\frac{N_b}{L_b}\right) \times \left(\frac{E_b}{D_{lb}}\right) \times \left(\frac{D_b^4}{64}\right)$$

= 4.155 kN (as shown in Section 7.1.2.4.2.1)

$$K1 = \frac{E_l \times t_l^3}{3 \times \left[\left(1 - N_{ul}^2\right) + \left(1 - N_{ul}\right)^2 \times \left(\frac{D_{lb}}{D_{lo}}\right)^2\right] \times D_{lb}}$$

= 235.0 kN (as shown in Section 7.1.2.4.2.1)

Thus,

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

$$= \left(\frac{\pi \times 1.920}{32}\right) \times \left(\frac{4.155}{4.155 + 235.0}\right) \times 727.3$$

$$= 2.4 \text{ kN-m}$$

**7.1.2.5.1.2 Secondary Lid Bolts**

The non-prying tensile bolt force per secondary lid bolt, $F_{ts}$, is:

$$F_{ts} = \frac{\sin(x_i) \times P_{un}}{N_b}$$

where,

$x_i$ = End Drop Impact Angle
= 90°
$P_{un}$ = MIN($P_{un1}$ , $P_{un2}$)
$N_b$ = Number of Bolts
= 18

$P_{un}$ was evaluated in Section 7.1.2.5.1.1:

$P_{un}$ = 9,139.7 kN

As shown in Figure 7.2, the primary and secondary lids act together under the pin puncture load. Therefore, the secondary lid will only see a portion of the impact load from the pin, so $P_{un}$ will be reduced by the ratio of the secondary lid volume to the total lid volume.

$V_s$ = Secondary Lid Volume

$$= \frac{\pi}{4} \times D_{lb}^2 \times t_l$$

$V_t$ = Total Lid Volume

$$= \frac{\pi}{4} \times D_{lbp}^2 \times t_{la}$$

where,

$D_{lbp}$ = Closure Lid Bolt Diameter at Primary Lid Bolts
= 1920 mm (3.2)
$t_{lp}$ = Closure Lid Thickness at Primary Lid Bolts
= 210 mm (3.2)
$t_{la}$ = Average Lid Thickness

$$= \frac{t_l + t_{lp}}{2}$$

Thus,

$$t_{la} = \frac{110 + 210}{2}$$

$$= 160 \text{ mm}$$

$$V_s = \frac{\pi}{4} \times 0.926^2 \times 0.110$$
$$= 0.067 \text{ m}^3$$

$$V_t = \frac{\pi}{4} \times 1.920^2 \times 0.160$$
$$= 0.463 \text{ m}^3$$

$$P_{un} = P'_{un} \times \frac{V_s}{V_t}$$

$$P_{un} = 9139.7 \times \frac{0.067}{0.463}$$
$$= 1328.7 \text{ kN}$$

and

$$F_{ts} = \frac{\sin(90) \times 1328.7}{18}$$
$$= 73.8 \text{ kN/bolt.}$$

As shown in Sections 7.1.2.2.1.1 and 7.1.2.2.1.1, the design of the primary and secondary lids prevents shear loads being applied to the bolts. Thus,

$$F_{ss} = 0.$$

It is noted that the equation given for $F_{ss}$ in NUREG/CR6007 also shows $F_{ss} = 0$.

The fixed edge closure lid force, $F_f$, is:

$$F_f = \frac{\sin(x_i) \times P_{un}}{\pi \times D_{lb}}$$

where,

$D_{lb}$ = Primary Lid Bolt Circle Diameter
= 926 mm (Ref. 3.2)

The remaining terms are as previously defined. Thus,

$$F_f = \frac{\sin(90) \times 1328.7}{\pi \times 0.926}$$
$$= 456.7 \text{ kN/m}$$

The fixed edge closure lid moment, $M_f$, is:

$$M_f = \frac{\sin(x_i) \times P_{un}}{4 \times \pi}$$

Thus,

$$M_f = \frac{\sin(90) \times 1328.7}{4 \times \pi}$$
$$= 105.7 \text{ kN-m/m}$$

The additional tensile bolt force per bolt, $F_t$, caused by the prying action of the secondary lid is (NUREG/CR-6007 Table 2.1):

$$F_t = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left[\frac{\frac{2 \times M_f}{D_{lo} - D_{lb}} - C1 \times (B - F_f) - C2 \times (B - P)}{C1 + C2}\right]$$

where,

| | | |
|---|---|---|
| P | = | Bolt Preload per unit Length of Bolt Circle |
| | = | 446.7 kN/m (as shown in Section 7.1.2.4.2.2) |
| B | = | Non-prying Tensile Bolt Force |
| | = | MAX($F_f$, P) |
| C1 | = | Force Constant |
| | = | 1.0 |
| C2 | = | Second Force Constant |
| | = | $\left(\frac{8}{3 \times (D_{lo} - D_{lb})^2}\right) \times \left[\frac{E_l \times t_l^3}{1 - N_{ul}} + \frac{(D_{lo} - D_{li}) \times E_{lf} \times t_{lf}^3}{D_{lb}}\right] \times \left(\frac{L_b}{N_b \times D_b^2 \times E_b}\right)$ |
| | = | 1.80 (as shown in Section 7.1.2.4.2.2) |
| $D_{lo}$ | = | Closure Lid Diameter at Outer Edge |
| | = | 1000 mm (3.2) |
| $D_{li}$ | = | Closure Lid Diameter at Inner Edge |
| | = | 745 mm (3.2) |
| $t_{lf}$ | = | Closure Lid Flange Thickness |
| | = | 80 mm (3.2) |
| $E_{lf}$ | = | Secondary Lid Flange Material Elastic Modulus, (SA 240 TYPE 304L per Table 2) |
| | = | 195.2 GPa at 20° C (Table 1) |
| $L_b$ | = | Bolt length between the top and bottom surfaces of the closure lid at the bolt circle |
| | = | 43 mm (3.2) |

Other terms are as previously defined. Thus,

$$F_{tp} = \left(\frac{\pi \times 0.926}{18}\right) \times \left[\frac{\frac{2 \times 105.7}{1.000 - 0.745} - 1 \times (456.7 - 456.7) - 1.80 \times (456.7 - 446.7)}{1 + 1.80}\right]$$

$$= 163.9 \text{ kN/bolt}$$

The total tension force, $F_a$, is

$$F_a = F_t + F_{tp}$$

= 73.8 + 163.9
= 237.7 kN/bolt

The shear force, $F_s$, is 0.

The maximum bending moment generated by the applied loads, $M_{bb}$, is:

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

where,

$$Kb = \left(\frac{N_b}{L_b}\right) \times \left(\frac{E_b}{D_{lb}}\right) \times \left(\frac{D_b^4}{64}\right)$$

= 2.39 kN (as shown in Section 7.1.2.4.2.2)

$$K1 = \frac{E_l \times t_l^3}{3 \times \left[\left(1 - N_{ul}^2\right) + \left(1 - N_{ul}\right)^2 \times \left(\frac{D_{lb}}{D_{lo}}\right)^2\right] \times D_{lb}}$$

= 71.3kN (as shown in Section 7.1.2.4.2.2)

Thus,

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

$$= \left(\frac{\pi \times 0.926}{18}\right) \times \left(\frac{2.39}{2.39 + 71.3}\right) \times 105.7$$

= 3.8 kN-m

**7.1.2.5.2 Side Puncture**

In Section 7.1.2.1.1, the gap between the cask body and the primary lid is shown to be smaller than the gap between the M48 bolts and the bolt clearance holes. Therefore, no shear load will be imparted to the bolts from the cask body. Further, there are no other loads resulting from side puncture at the bolts. Thus, no significant loads are imparted to the primary and secondary closure lid bolts during a side puncture event.

**7.1.2.6 External Pressure**

Loads in the primary and secondary closure lid bolts due to external pressure are evaluated using the formulas for evaluating bolt forces/moments in NUREG/CR-6007 Table 4.3.

**7.1.2.6.1 Primary Lid Bolts**

The pressure outside the cask, $P_{lo}$, in the case of immersion is assumed to be 350 kPa (TH-0102). The pressure inside the cask, $P_{li}$, is conservatively taken to be 0 kPa.

The axial force per bolt due to external pressure is:

$$F_a = \frac{\pi \times D_{lg}^2 \times \left(P_{li} - P_{lo}\right)}{4 \times N_b}$$

where,

$D_{lg}$ = Outside Seal Diameter
= 1835 mm (Ref 3.2)
$N_b$ = Number of Bolts
= 32 (Ref 3.2)

Thus,

$$F_a = \frac{\pi \times 1.835^2 \times (0-350)}{4 \times 32}$$
$$= -28.9 \text{ kN/bolt}$$

Since this force is negative (inward acting), the actual resulting bolt force is Fa = 0 because the applied load is supported by the cask wall and not by the bolts.

The fixed edge closure lid force is:

$$F_f = \frac{D_{lb} \times (P_{li} - P_{lo})}{4}$$

where,

$D_{lb}$ = Bolt Circle Diameter
= 1920 mm (Ref 3.2)

Thus,

$$F_f = \frac{1.920 \times (0-350)}{4}$$
$$= -168.0 \text{ kN/m}$$

The fixed edge closure lid moment is:

$$M_f = \frac{(P_{li} - P_{lo}) \times D_{lb}^2}{32}$$
$$= \frac{(0-350) \times 1.920^2}{32}$$
$$= -40.3 \text{ kN-m/m}$$

The shear bolt force per bolt is:

$$F_s = \frac{\pi \times E_l \times t_l \times (P_{li} - P_{lo}) \times D_{lb}^2}{2 \times N_b \times E_c \times t_c \times (1 - N_{ul})}$$

All terms are as previously defined.

Thus,

$$F_s = \frac{\pi \times 195200 \times 0.210 \times (0-350) \times 1.920^2}{2 \times 32 \times 195200 \times 0.065 \times (1-0.31)}$$
$$= -296.5 \text{ kN/bolt}$$

The *maximum* gap between the lid and cask body is less than the *minimum* gap between the bolt clearance holes and bolt shank (see Section 7.1.2.1.1). Thus, the RT-100 primary lid bolts will not be subjected to any shear loads. Therefore,

$$F_s = 0.0 \text{ kN/bolt.}$$

**7.1.2.6.2 Secondary Lid Bolts**

The pressure outside the cask, $P_{lo}$, in the case of immersion is assumed to be 350 kPa (TH-0102). The pressure inside the cask, $P_{li}$, is conservatively taken to be 0 kPa.

The axial force per bolt due to external pressure is:

$$F_a = \frac{\pi \times D_{lg}^2 \times (P_{li} - P_{lo})}{4 \times N_b}$$

where,

$D_{lg}$ = Outside Seal Diameter
= 790 mm (Ref 3.2)
$N_b$ = Number of Bolts
= 18 (Ref 3.2)

Thus,

$$F_a = \frac{\pi \times 0.790^2 \times (0 - 350)}{4 \times 18}$$
$$= -9.5 \text{ kN/bolt}$$

Since this force is negative (inward acting), the actual resulting bolt force is Fa = 0 because the load is supported by the cask wall and not by the bolts.

The fixed edge closure lid force is:

$$F_f = \frac{D_{lb} \times (P_{li} - P_{lo})}{4}$$

where,

$D_{lb}$ = Bolt Circle Diameter
= 926 mm (Ref 3.2)

Thus,

$$F_f = \frac{0.926 \times (0 - 350)}{4}$$
$$= -81.0 \text{ kN/m}$$

The fixed edge closure lid moment is:

$$M_f = \frac{(P_{li} - P_{lo}) \times D_{lb}^2}{32}$$
$$= \frac{(0 - 350) \times 0.926^2}{32}$$
$$= -9.4 \text{ kN-m/m}$$

The shear bolt force per bolt is

$$F_s = \frac{\pi \times E_l \times t_l \times (P_{li} - P_{lo}) \times D_{lb}^2}{2 \times N_b \times E_c \times t_c \times (1 - N_{ul})}$$

All terms are as previously defined. Thus,

$$F_s = \frac{\pi \times 195200 \times 0.110 \times (0 - 350) \times 0.926^2}{2 \times 18 \times 195200 \times 0.065 \times (1 - 0.31)}$$

$$= -64.2 \text{ kN/bolt}$$

The *maximum* gap between the lid and cask body is less than the *minimum* gap between the bolt clearance holes and bolt shank (see Section 7.1.2.1.2). Thus, the RT-100 secondary lid bolts will not be subjected to any shear loads. Therefore,

$$F_s = 0.0 \text{ kN/bolt.}$$

**7.1.2.7 Gasket Seating Load**

A small closure force is required to maintain a positive seal between the cask lid and the cask body. However, this closure force is much less than the minimum preloads provided for the closure bolts at the primary and secondary lids. Therefore, the gasket seating load is negligible, and $F_s = 0$.

**7.1.3 Load Combinations**

The loadings in Section 7.1.2 are combined to form load cases for the closure bolt analysis per NUREG/CR-6007. The corresponding bolt stresses are obtained and compared to the criteria defined in Section 2.1.2.2. A summary of the loads on the bolts for the primary and secondary lids under the normal conditions of transport and the hypothetical accident conditions is presented in Tables 7.1.3-1 and 7.1.3-2, respectively.

Table 7.1.3-1: Primary Lid Bolt Load Summary

| Load Case | Applied Load | | Non-Prying Tensile Force $F_t$ (kN/bolt) | Torsional Moment $M_t$ (kN-m/bolt) | Prying Force $F_f$ (kN/m) | Prying Moment $M_f$ (kN-m/m) |
|---|---|---|---|---|---|---|
| Preload | Residual Torque | Minimum | 52.8 | 0.2 | 0.0 | 0.0 |
| | | Maximum | 130.6 | 0.5 | 0.0 | 0.0 |
| Gasket | Seating Load | | 0.0 | 0.0 | 0.0 | 0.0 |
| Internal Pressure | 250 kN/m² (35 psi) pressure | | 20.7 | 0.0 | 120.0 | 28.8 |
| Thermal | 150°C | | 596.2 | 0.0 | 0.0 | 0.0 |
| Puncture | Drop on 15 cm diameter pin | | 285.6 | 2.4 | 1515.2 | 727.3 |
| External Pressure | 350 kPa pressure | | 0.0 | 0.0 | -168.0 | -40.3 |
| Free Drop | Drop from 9 m height | | 628.9 | 2.6 | 3336.4 | 800.7 |

**Table 7.1.3-2: Secondary Lid Bolt Load Summary**

| Load Case | Applied Load | | Non-Prying Tensile Force $F_t$ (kN/bolt) | Torsional Moment $M_t$ (kN-m/bolt) | Prying Force $F_f$ (kN/m) | Prying Moment $M_f$ (kN-m/m) |
|---|---|---|---|---|---|---|
| Preload | Residual Torque | Minimum | 29.6 | 0.1 | 0.0 | 0.0 |
| | | Maximum | 72.2 | 0.2 | 0.0 | 0.0 |
| Gasket | Seating Load | | 0.0 | 0.0 | 0.0 | 0.0 |
| Internal Pressure | 250 kN/m² (35 psi) pressure | | 6.8 | 0.0 | 57.9 | 6.7 |
| Thermal | 150°C | | 335.4 | 0.0 | 0.0 | 0.0 |
| Puncture | Drop on 15 cm diameter pin | | 237.7 | 0.6 | 456.7 | 105.7 |
| External Pressure | 350 kPa pressure | | 0.0 | 0.0 | -81.0 | -9.4 |
| Free Drop | Drop from 9 m height | | 266.0 | 1.0 | 1646.1 | 190.5 |

**7.1.3.1 Primary Lid Closure Bolt Evaluation under Normal Conditions of Transport**

The maximum tension, shear and bolt bearing loads in the primary lid bolts due to the combined NCT loads will be evaluated in accordance with NUREG/CR-6007, with due consideration given to the prying effects on the fixed lid. Since the prying forces act inward, normal to the cask lid, an additional prying force is generated (NUREG 6007). For the NCT condition, the controlling load case is the summation of the bolt preload, the internal pressure load and the thermal expansion load. The maximum bolt tension load (Ft), shear load (Fs) and torsional moment (Mt) for the primary lid bolts are (see Table 7.1.3-1):

$$F_t = F_p + F_{ap} + F_{atp}$$

$$F_s = F_{sp} + F_{st}$$

$$M_t = M_{pt} + M_{at} + M_{st}$$

where,

$F_p$ = Maximum Bolt Preload
= 130.6 kN/bolt (Table 7.1.3-1)

$F_{ap}$ = Bolt Internal Pressure Load
= 20.7 kN/bolt (Table 7.1.3-1)

$F_{atp}$ = Bolt Temperature Load
= 0.5 kN/bolt (Table 7.1.3-1)

$F_{sp}$ = Bolt Internal Pressure Shear Load
= 0.0 kN/ bolt (Table 7.1.3-1)

$F_{st}$ = Bolt Temperature Shear Load
= 0.0 kN/bolt (Table 7.1.3-1)

$M_{pt}$ = Maximum Bolt Preload Torsional Moment
= 0.5 kN-m/bolt (Table 7.1.3-1)

$M_{at}$ = Bolt Internal Pressure Torsional Moment
= 0.0 kN-m/bolt (Table 7.1.3-1)

$M_{st}$ = Bolt Temperature Torsional Moment

= 0.0 kN-m/bolt (Table 7.1.3-1)

Thus,

$F_t$ = $F_p + F_{ap} + F_{atp}$
= $130.6 + 20.7 + 596.2$
= 747.5 kN/bolt

$F_s$ = $F_{sp} + F_{st}$
= $0.0 + 0.0$
= 0.0 kN/bolt

$M_t$ = $M_{pt} + M_{at} + M_{st}$
= $0.5 + 0.0 + 0.0$
= 0.5 kN-m/bolt

Conservatively, the fixed-edge closure lid prying is taken from the external pressure load case. This accident load case bounds all normal conditions and provides a conservative result. The additional tensile bolt force per bolt, $F_{tp}$, caused by the prying action of the primary lid is (NUREG/CR-6007 Table 2.1):

$$F_{tp} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left[\frac{\frac{2 \times M_f}{D_{li} - D_{lb}} - C1 \times (B - F_f) - C2 \times (B - P)}{C1 + C2}\right]$$

where,

$F_f$ = Fixed Edge Closure Force
= -168.0 kN/m (Table 7.1.3-1)
$M_f$ = Fixed Edge Closure Moment
= -40.3 kN-m/m (Table 7.1.3-1)

Other terms are as previously defined.

Thus,

$$F_{tp} = \left(\frac{\pi \times 1.920}{32}\right) \times \left[\frac{\frac{2 \times -40.3}{2.016 - 1.920} - 1 \times (692.9 - -168.0) - 3.49 \times (692.9 - 692.9)}{1 + 3.49}\right]$$

= -18.3 kN/bolt

Since this bolt load is less than the load generated by the minimum bolt preload (130.6 kN > -18.3 kN), the prying force generated by the external pressure is not critical with respect to bolt stress and will not result in the loss of lid closure seal.

The total tension force, $F_a$, is:

$$F_a = F_t + F_{tp}$$
$$= |747.5| + |-18.3|$$
$$= 765.8 \text{ kN/bolt}$$

The maximum bending moment generated by the applied loads, $M_{bb}$, is:

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

All terms are as previously defined. Thus,

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$
$$= \left(\frac{\pi \times 1.920}{32}\right) \times \left(\frac{4.155}{4.155 + 235.0}\right) \times -40.3$$
$$= -0.13 \text{ kN-m}$$

The average bolt stresses from the combined NCT loads are determined in accordance with Table 5.1 of NUREG/CR-6007. The bolt stress diameter, D, is:

$$D = D_b - 0.9382 \times p$$

where,

p = Bolt Pitch
= 5.0 mm (Machinery Handbook)

Thus,

$$D = D_b - 0.9382 \times p$$
$$= 0.048 - 0.9382 \times 0.0050$$
$$= 0.043 \text{ m}$$

The average tensile stress, Sba, average shear stress, Sbs, maximum bending stress, Sbb, and maximum torsional stress, Sbt, are:

$$Sba = \frac{1.2732 \times Fa}{D^2}$$
$$= \frac{1.2732 \times 765.8}{0.043^2} \times \frac{1 \text{ MPa}}{1000 \text{ kN/m}^2}$$
$$= 527.3 \text{ MPa}$$

$$Sbs = \frac{1.2732 \times Fs}{D^2}$$

$$= \frac{1.2732 \times 0.0}{0.043^{2}} \times \frac{1\ \text{MPa}}{1000\ \text{kN}/\text{m}^{2}}$$

= 0.0 MPa

$$\text{Sbb} = \frac{10.186 \times M_{bb}}{D_b^3}$$

$$= \frac{10.186 \times -0.13}{0.048^{3}} \times \frac{1\ \text{MPa}}{1000\ \text{kN}/\text{m}^{2}}$$

= -12.2 MPa

$$\text{Sbt} = \frac{5.093 \times M_t}{D_b^3}$$

$$= \frac{5.093 \times 0.5}{0.048^{3}} \times \frac{1\ \text{MPa}}{1000\ \text{kN}/\text{m}^{2}}$$

= 21.6 MPa

The allowable stresses for the bolts are:

$\sigma_{ta}$ = Allowable Tensile Stress

= $0.7 \times S_u$

$\sigma_{sa}$ = Allowable Shear Stress

= $0.42 \times S_u$

$\sigma_{ba}$ = Allowable Bending Stress

= $1.5 \times S_{mn}$

where,

$S_u$ = Primary Bolt Ultimate Stress (SA 354 Grade BD per Table 2)

= 1034.2 MPa at 20° C (Table 1)

$S_{mn}$ = Primary Bolt Membrane Stress (SA 354 Grade BD per Table 2)

= 434.4 MPa at 20° C (Table 1)

Thus,

$\sigma_{ta}$ = $0.7 \times S_u$

= $0.7 \times 1034.2$

= 723.9 MPa

$\sigma_{sa}$ = $0.42 \times S_u$

= $0.42 \times 1034.2$

= 434.4 MPa

$\sigma_{ba}$ = $1.5 \times S_{mn}$

= $1.5 \times 434.4$

= 651.6 MPa

The maximum interaction ratio for the combined shear and tension loads is therefore:

$$\text{I.R.} = \left(\frac{Sba}{\sigma_{ta}}\right)^2 + \left(\frac{Sbs}{\sigma_{sa}}\right)^2$$

$$= \left(\frac{527.3}{723.9}\right)^2 + \left(\frac{0.0}{434.4}\right)^2$$

$$= 0.5306$$

The minimum factor of safety is:

$$FS = \frac{1}{\text{I.R.}}$$

$$= \frac{1}{0.5306}$$

$$= 1.88 > 1.0$$

The maximum interaction ratio for the bending load is therefore:

$$\text{I.R.} = \left(\frac{Sbb}{\sigma_{ba}}\right)^2$$

$$= \left(\frac{-12.2}{651.6}\right)^2$$

$$= 0.00035$$

The minimum factor of safety is:

$$FS = \frac{1}{\text{I.R.}}$$

$$= \frac{1}{0.00035}$$

$$= 2869.1 > 1.0$$

The maximum stress intensity in the primary lid bolts under the combined loads, Sbi, is:

$$S_{bi} = \sqrt{(Sba + Sbb)^2 + 4 \times (Sbs + Sbt)^2}$$

$$= \sqrt{(527.3 + -12.2)^2 + 4 \times (0.0 + 21.6)^2}$$

$$= 516.9 \text{ MPa}$$

The primary lid closure bolts utilize a custom washer for the bolts with an outer diameter, $d_{ow}$, of 130 mm and a hole diameter, $d_{oh}$, of 52 mm. Therefore, the bearing stress under the bolt head, Sbrg, is:

$$Sbrg = \frac{Fa}{A_{brg}}$$

where,

$$A_{brg} = \text{Bolt Bearing Area}$$
$$= \frac{\pi}{4}\times\left(d_{ow}^2 - d_{oh}^2\right)$$
$$= \frac{\pi}{4}\times\left(0.130^2 - 0.052^2\right)$$
$$= 0.0111\ m^2$$

Thus,

$$Sbrg = \frac{765.8}{0.0111}\times\frac{1\ MPa}{1000\ kN/m^2}$$
$$= 69.0\ MPa$$

The allowable normal condition bearing stress on the lid is taken to be the yield stress of the lid material at 250 °C. The maximum interaction ratio for the bearing load is therefore:

$$I.R. = \frac{Sbrg}{S_{yl}}$$

where,

$S_{yl}$ = Primary Lid Material Yield Stress, (SA 240 TYPE 304L per Table 2)
= 114.4 MPa at 250° C (Table 1)

Thus,

$$I.R. = \frac{Sbrg}{S_{yl}}$$
$$= \frac{69.0}{114.4}$$
$$= 0.603$$

The minimum factor of safety is:

$$FS = \frac{1}{I.R.}$$
$$= \frac{1}{0.603}$$
$$= 1.66 > 1.0$$

Because the cask material is weaker than the bolting material, failure will occur at the root of the cask material threads. The minimum required thread engagement length to prevent cask material failure is determined in accordance with Reference Machinery Handbook. Since the constants in the equation assume customary units, the metric units used for the cask design will be converted for determination of the required engagement length. Thus, the minimum engagement length, $L_e$, for the cask is:

$$L_e = \frac{S_{ub} \times 2 \times A_b}{S_{ul} \times \pi \times n \times D_{s,min} \times \left[\frac{1}{2 \times n} + 0.57735 \times \left(D_{s,min} - E_{n,max}\right)\right]}$$

where,

$S_{ub}$ = Primary Bolt External Thread Tensile Strength (SA 354 Grade BD per Table 2)
= 150,000 psi (1,034.2 MPa) at 20° C (Table 1)

$S_{ul}$ = Cask Internal Thread Tensile Strength (SA 240 TYPE 304L per Table 2)
= 70,000 psi (482.6 MPa) at 20° C (Table 1)

$A_b$ = Stress Area of Primary Bolt External Threads
= 2.28 in$^2$ (1470 mm$^2$) (Machinery)

p = Bolt Pitch
= 0.197 in (5.0 mm) (Machinery)

n = Number of Threads per Inch
$= \frac{1}{p}$

$D_{s,min}$ = Minimum Major Bolt Diameter
= 1.866 in (47.399 mm) (ASME B1.13M)

$E_{n,max}$ = Maximum Pitch Diameter of Internal Thread
= 1.866 in (47.399 mm) (ASME B1.13M)

$L_{ep}$ = Provided Engagement Length
= 73.0 mm (3.2)

Thus,

$$n = \frac{1}{p} = \frac{1}{0.197} = 5.08 \text{ threads/in}$$

and

$$L_e = \frac{S_{ub} \times 2 \times A_b}{S_{ul} \times \pi \times n \times D_{s,min} \times \left[\frac{1}{2 \times n} + 0.57735 \times \left(D_{s,min} - E_{n,max}\right)\right]}$$

$$= \frac{150000 \times 2 \times 2.28}{70000 \times \pi \times 5.08 \times 1.866 \times \left[\frac{1}{2 \times 5.08} + 0.57735 \times (1.866 - 1.705)\right]} \times \frac{25.4 \text{ mm}}{\text{in}}$$

$= 43.5 \text{ mm} < L_{ep} = 73.0 \text{ mm}$

Therefore, the primary closure lid bolts are acceptable for the normal conditions of transport.

**7.1.3.2 Secondary Lid Closure Bolt Evaluation under Normal Conditions of Transport**

The maximum tension, shear and bolt bearing loads in the secondary lid bolts due to the combined NCT loads will be evaluated in accordance with NUREG/CR-6007, with due consideration given to the prying effects on the fixed lid. Since the prying forces act inward, normal to the cask lid, an additional prying force is generated (NUREG 6007). For the NCT condition, the controlling load case is the summation of the bolt preload, the internal pressure load and the thermal expansion load. The maximum bolt tension load ($F_t$), shear load ($F_s$) and torsional moment ($M_t$) for the primary lid bolts are (see Table 7.1.3-2):

$$F_t = F_p + F_{as} + F_{ats}$$

$$F_s = F_{ss} + F_{ass}$$

$$M_t = M_{pt} + M_{at} + M_{st}$$

where,

| | | | |
|---|---|---|---|
| $F_p$ | = | Maximum Bolt Preload | |
| | = | 72.2 kN/bolt | (Table 7.1.3-2) |
| $F_{as}$ | = | Bolt Internal Pressure Load | |
| | = | 6.8 kN/bolt | (Table 7.1.3-2) |
| $F_{ats}$ | = | Bolt Temperature Load | |
| | = | 0.3 kN/bolt | (Table 7.1.3-2) |
| $F_{ss}$ | = | Bolt Internal Pressure Shear Load | |
| | = | 0.0 kN/ bolt | (Table 7.1.3-2) |
| $F_{ass}$ | = | Bolt Temperature Shear Load | |
| | = | 0.0 kN/bolt | (Table 7.1.3-2) |
| $M_{pt}$ | = | Maximum Bolt Preload Torsional Moment | |
| | = | 0.2 kN-m/bolt | (Table 7.1.3-2) |
| $M_{at}$ | = | Bolt Internal Pressure Torsional Moment | |
| | = | 0.0 kN-m/bolt | (Table 7.1.3-2) |
| $M_{st}$ | = | Bolt Temperature Torsional Moment | |
| | = | 0.0 kN-m/bolt | (Table 7.1.3-2) |

Thus,

$$F_t = F_p + F_{ap} + F_{atp}$$
$$= 72.2 + 6.8 + 335.4$$
$$= 414.4 \text{ kN/bolt}$$

$$F_s = F_{sp} + F_{st}$$
$$= 0.0 + 0.0$$
$$= 0.0 \text{ kN/bolt}$$

$$M_t = M_{pt} + M_{at} + M_{st}$$
$$= 0.2 + 0.0 + 0.0$$
$$= 0.2 \text{ kN-m/bolt}$$

Conservatively, the fixed-edge closure lid prying is taken from the external pressure load case. This accident load case bounds all normal conditions and provides a conservative result. The additional tensile bolt force per bolt, $F_{tp}$, caused by the prying action of the secondary lid is (NUREG/CR-6007 Table 2.1):

$$F_{tp} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left[\frac{\frac{2 \times M_f}{D_{li} - D_{lb}} - C1 \times (B - F_f) - C2 \times (B - P)}{C1 + C2}\right]$$

where,

| | | | |
|---|---|---|---|
| $F_f$ | = | Fixed Edge Closure Force | |
| | = | -81.0 kN/m | (Table 7.1.3-2) |
| $M_f$ | = | Fixed Edge Closure Moment | |
| | = | -9.4 kN-m/m | (Table 7.1.3-2) |

Other terms are as previously defined.

Thus,

$$F_{tp} = \left(\frac{\pi \times 0.926}{18}\right) \times \left[\frac{\frac{2 \times -9.4}{0.754 - 0.926} - 1 \times (446.7 - -81.0) - 1.80 \times (446.7 - 446.7)}{1 + 1.80}\right]$$

= -24.5 kN/bolt

Since this bolt load is less than the load generated by the minimum bolt preload (72.2 kN > -24.5 kN), the prying force generated by the external pressure is not critical with respect to bolt stress and will not result in the loss of lid closure seal.

The total tension force, $F_a$, is:

$$F_a = F_t + F_{tp} = |414.4| + |-24.5| = 438.9 \text{ kN/bolt}$$

The maximum bending moment generated by the applied loads, $M_{bb}$, is:

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

All terms are as previously defined. Thus,

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf = \left(\frac{\pi \times 0.926}{18}\right) \times \left(\frac{2.39}{2.39 + 71.3}\right) \times -9.4$$

= -0.05 kN-m

The average bolt stresses from the combined NCT loads are determined in accordance with Table 5.1 of NUREG/CR-6007. The bolt stress diameter, D, is:

$$D = D_b - 0.9382 \times p$$

where,

p = Bolt Pitch
= 4.0 mm (Machinery Handbook)

Thus,

$$D = D_b - 0.9382 \times p$$
$$= 0.036 - 0.9382 \times 0.0040$$
$$= 0.032 \text{ m}$$

The average tensile stress, Sba, average shear stress, Sbs, maximum bending stress, Sbb, and maximum torsional stress, Sbt, are:

$$Sba = \frac{1.2732 \times Fa}{D^2}$$
$$= \frac{1.2732 \times 438.9}{0.032^2} \times \frac{1 \text{ MPa}}{1000 \text{ kN/m}^2}$$
$$= 545.7 \text{ MPa}$$

$$Sbs = \frac{1.2732 \times Fs}{D^2}$$
$$= \frac{1.2732 \times 0.0}{0.032^2} \times \frac{1 \text{ MPa}}{1000 \text{ kN/m}^2}$$
$$= 0.0 \text{ MPa}$$

$$Sbb = \frac{10.186 \times M_{bb}}{D_b^3}$$
$$= \frac{10.186 \times -0.05}{0.036^3} \times \frac{1 \text{ MPa}}{1000 \text{ kN/m}^2}$$
$$= -10.7 \text{ MPa}$$

$$Sbt = \frac{5.093 \times M_t}{D_b^3}$$
$$= \frac{5.093 \times 0.2}{0.036^3} \times \frac{1 \text{ MPa}}{1000 \text{ kN/m}^2}$$
$$= 21.3 \text{ MPa}$$

The allowable stresses for the bolts are as previously defined. The maximum interaction ratio for the combined shear and tension loads is therefore:

$$\text{I.R.} = \left(\frac{Sba}{\sigma_{ta}}\right)^2 + \left(\frac{Sbs}{\sigma_{sa}}\right)^2$$

$$= \left(\frac{545.7}{723.9}\right)^2 + \left(\frac{0.0}{434.4}\right)^2$$

$$= 0.568$$

The minimum factor of safety is:

$$FS = \frac{1}{\text{I.R.}}$$

$$= \frac{1}{0.568}$$

$$= 1.76 > 1.0$$

The maximum interaction ratio for the bending load is therefore:

$$\text{I.R.} = \left(\frac{Sbb}{\sigma_{ba}}\right)^2$$

$$= \left(\frac{-10.7}{651.6}\right)^2$$

$$= 0.00027$$

The minimum factor of safety is:

$$FS = \frac{1}{\text{I.R.}}$$

$$= \frac{1}{0.00027}$$

$$= 3685.3 > 1.0$$

The maximum stress intensity in the primary lid bolts under the combined loads, Sbi, is:

$$S_{bi} = \sqrt{(Sba + Sbb)^2 + 4 \times (Sbs + Sbt)^2}$$

$$= \sqrt{(545.7 + -10.7)^2 + 4 \times (0.0 + 21.3)^2}$$

$$= 123.9 \text{ MPa}$$

The primary lid closure bolts utilize a custom washer for the bolts with an outer diameter, $d_{ow}$, of 90 mm and a hole diameter, $d_{oh}$, of 40 mm. Therefore, the bearing stress under the bolt head, Sbrg, is:

$$Sbrg = \frac{Fa}{A_{brg}}$$

where,

$$A_{brg} = \text{Bolt Bearing Area}$$

$$= \frac{\pi}{4} \times \left(d_{ow}^2 - d_{oh}^2\right)$$

$$= \frac{\pi}{4} \times \left(0.090^2 - 0.040^2\right)$$

$$= 0.0051 \text{ m}^2$$

Thus,

$$Sbrg = \frac{438.9}{0.0051} \times \frac{1 \text{ MPa}}{1000 \text{ kN/m}^2}$$

$$= 86.1 \text{ MPa}$$

The allowable normal condition bearing stress on the lid is taken to be the yield stress of the lid material at 250 °C. The maximum interaction ratio for the bearing load is therefore:

$$I.R. = \frac{Sbrg}{S_{yl}}$$

All terms are as previously defined.

Thus,

$$I.R. = \frac{Sbrg}{S_{yl}}$$

$$= \frac{86.1}{114.4}$$

$$= 0.753$$

The minimum factor of safety is:

$$FS = \frac{1}{I.R.}$$

$$= \frac{1}{0.753}$$

$$= 1.33 > 1.0$$

Because the cask material is weaker than the bolting material, failure will occur at the root of the cask material threads. The minimum required thread engagement length to prevent cask material failure is

determined in accordance with Reference Machinery Handbook. Since the constants in the equation assume customary units, the metric units used for the cask design will be converted for determination of the required engagement length. Thus, the minimum engagement length, $L_e$, for the cask is:

$$L_e = \frac{S_{ub} \times 2 \times A_b}{S_{ul} \times \pi \times n \times D_{s,min} \times \left[\frac{1}{2 \times n} + 0.57735 \times \left(D_{s,min} - E_{n,max}\right)\right]}$$

where,

| | | | |
|---|---|---|---|
| $A_b$ | = | Stress Area of Primary Bolt External Threads | |
| | = | 1.27 in$^2$ (817 mm$^2$) | (Machinery) |
| p | = | Bolt Pitch | |
| | = | 0.157 in (4.0 mm) | (Machinery) |
| n | = | Number of Threads per Inch | |
| | = | $\frac{1}{p}$ | |
| $D_{s,min}$ | = | Minimum Major Bolt Diameter | |
| | = | 1.396 in (35.465 mm) | (ASME B1.13M) |
| $E_{n,max}$ | = | Maximum Pitch Diameter of Internal Thread | |
| | = | 1.270 in (32.270 mm) | (ASME B1.13M) |
| $L_{ep}$ | = | Provided Engagement Length | |
| | = | 46.0 mm | (3.2) |

All other terms are as previously defined. Thus,

$$n = \frac{1}{p} = \frac{1}{0.157} = 6.35 \text{ threads/in}$$

and

$$L_e = \frac{S_{ub} \times 2 \times A_b}{S_{ul} \times \pi \times n \times D_{s,min} \times \left[\frac{1}{2 \times n} + 0.57735 \times \left(D_{s,min} - E_{n,max}\right)\right]}$$

$$= \frac{150000 \times 2 \times 1.27}{70000 \times \pi \times 6.35 \times 1.396 \times \left[\frac{1}{2 \times 6.35} + 0.57735 \times (1.396 - 1.270)\right]} \times \frac{25.4\,\text{mm}}{\text{in}}$$

$$= 32.7 \text{ mm} < L_{ep} = 46.0 \text{ mm}$$

Therefore, the secondary closure lid bolts are acceptable for the normal conditions of transport.

**7.1.3.3 Primary Lid Closure Bolt Evaluation under Hypothetical Accident Conditions**

The maximum tension, shear and bolt bearing loads in the primary lid bolts due to the combined HAC loads will be evaluated in accordance with NUREG/CR-6007, with due consideration given to the prying effects on the fixed lid. Since the prying forces act inward, normal to the cask lid, an additional prying force is generated (NUREG 6007). For the HAC condition, the controlling load case is the summation of the bolt preload, the internal pressure load and the end drop load. Since the internal pressure load will act counter to the drop load, the internal pressure load will be considered as negative for determination of the maximum bolt tension load. The maximum bolt tension load ($F_t$), shear load ($F_s$) and torsional moment ($M_t$) for the primary lid bolts are (see Table 7.1.3-2):

$$F_t = F_p - F_{ap} + F_{ad}$$

$$F_s = F_{sp} + F_{sd}$$

$$M_t = M_{pt} + M_{at} + M_{dt}$$

where,

| | | | |
|---|---|---|---|
| $F_p$ | = | Maximum Bolt Preload | |
| | = | 130.6 kN/bolt | (Table 7.1.3-1) |
| $F_{ap}$ | = | Bolt Internal Pressure Load | |
| | = | 20.7 kN/bolt | (Table 7.1.3-1) |
| $F_{ad}$ | = | Bolt End Drop Load | |
| | = | 628.9 kN/bolt | (Table 7.1.3-1) |
| $F_{sp}$ | = | Bolt Internal Pressure Shear Load | |
| | = | 0.0 kN/ bolt | (Table 7.1.3-1) |
| $F_{sd}$ | = | Bolt End Drop Shear Load | |
| | = | 0.0 kN/bolt | (Table 7.1.3-1) |
| $M_{pt}$ | = | Maximum Bolt Preload Torsional Moment | |
| | = | 0.5 kN-m/bolt | (Table 7.1.3-1) |
| $M_{at}$ | = | Bolt Internal Pressure Torsional Moment | |
| | = | 0.0 kN-m/bolt | (Table 7.1.3-1) |
| $M_{dt}$ | = | Bolt End Drop Torsional Moment | |
| | = | 0.0 kN-m/bolt | (Table 7.1.3-1) |

Thus,

$$F_t = F_p - F_{ap} + F_{ad} = 130.6 - 20.7 + 628.9 = 738.8 \text{ kN/bolt}$$

$$F_s = F_{sp} + F_{sd} = 0.0 + 0.0 = 0.0 \text{ kN/bolt}$$

$$M_t = M_{pt} + M_{at} + M_{dt} = 0.5 + 0.0 + 0.0 = 0.5 \text{ kN-m/bolt}$$

Conservatively, the fixed-edge closure lid prying is taken from the end drop load case. The additional tensile bolt force per bolt, $F_{tp}$, caused by the prying action of the primary lid is (NUREG/CR-6007 Table 2.1):

$$F_{tp} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left[\frac{\frac{2 \times M_f}{D_{li} - D_{lb}} - C1 \times (B - F_f) - C2 \times (B - P)}{C1 + C2}\right]$$

where,

| | | | |
|---|---|---|---|
| $F_f$ | = | Fixed Edge Closure Force | |
| | = | 3336.4 kN/m | (Table 7.1.3-1) |
| $M_f$ | = | Fixed Edge Closure Moment | |
| | = | 800.7 kN-m/m | (Table 7.1.3-1) |

Other terms are as previously defined.

Thus,

$$F_{tp} = \left(\frac{\pi \times 1.920}{32}\right) \times \left[\frac{\frac{2 \times 800.7}{2.016 - 1.920} - 1 \times (3336.4 - 3336.4) - 3.49 \times (3336.4 - 692.9)}{1 + 3.49}\right]$$

= 313.7 kN/bolt

This bolt load is greater than the load generated by the minimum bolt preload (130.6 kN < 313.7 kN). However, the drop load is an inward load, which will press the closure lid against the sealing gasket. The prying force generated by the drop load will therefore not result in the loss of lid closure seal. The outward load of the internal pressure has already been evaluated in Section 7.1.3.1 and found to be acceptable. All other accident loads are acceptable by comparison.

The total tension force, $F_a$, is:

$$F_a = F_t + F_{tp} = 738.8 + 313.7 = 1052.5 \text{ kN/bolt}$$

The maximum bending moment generated by the applied loads, $M_{bb}$, is:

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

All terms are as previously defined. Thus,

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf = \left(\frac{\pi \times 1.920}{32}\right) \times \left(\frac{4.155}{4.155 + 235.0}\right) \times 800.7$$

= 2.6 kN-m

The average bolt stresses from the combined HAC loads are determined in accordance with Table 5.1 of NUREG/CR-6007. The bolt stress diameter is as defined previously. The average tensile stress, Sba, average shear stress, Sbs, maximum bending stress, Sbb, and maximum torsional stress, Sbt, are:

$$Sba = \frac{1.2732 \times Fa}{D^2} = \frac{1.2732 \times 1052.5}{0.043^2} \times \frac{1\,MPa}{1000\ kN/m^2} = 714.5\ MPa$$

$$Sbs = \frac{1.2732 \times Fs}{D^2} = \frac{1.2732 \times 0.0}{0.043^2} \times \frac{1\,MPa}{1000\ kN/m^2} = 0.0\ MPa$$

$$Sbb = \frac{10.186 \times M_{bb}}{D_b^3} = \frac{10.186 \times 2.6}{0.048^3} \times \frac{1\,MPa}{1000\ kN/m^2} = 241.6\ MPa$$

$$Sbt = \frac{5.093 \times M_t}{D_b^3} = \frac{5.093 \times 0.5}{0.048^3} \times \frac{1\,MPa}{1000\ kN/m^2} = 21.6\ MPa$$

The allowable stresses for the bolts are as previously defined. The maximum interaction ratio for the combined shear and tension loads is therefore:

$$I.R. = \left(\frac{Sba}{\sigma_{ta}}\right)^2 + \left(\frac{Sbs}{\sigma_{sa}}\right)^2 = \left(\frac{714.5}{723.9}\right)^2 + \left(\frac{0.0}{434.4}\right)^2 = 0.97$$

The minimum factor of safety is:

$$FS = \frac{1}{I.R.} = \frac{1}{0.97} = 1.03 > 1.0$$

The maximum interaction ratio for the bending load is therefore:

$$I.R. = \left(\frac{Sbb}{\sigma_{ba}}\right)^2 = \left(\frac{241.6}{651.6}\right)^2 = 0.14$$

The minimum factor of safety is:

$$FS = \frac{1}{I.R.} = \frac{1}{0.14} = 7.27 > 1.0$$

The primary lid closure bolts utilize a custom washer for the bolts with an outer diameter, $d_{ow}$, of 130 mm and a hole diameter, $d_{oh}$, of 52 mm. Therefore, the bearing stress under the bolt head, Sbrg, is:

$$Sbrg = \frac{Fa}{A_{brg}}$$

All terms are as previously defined.

Thus,

$$Sbrg = \frac{1052.5}{0.0111} \times \frac{1\ MPa}{1000\ kN/m^2} = 94.4\ MPa$$

The allowable normal condition bearing stress on the lid is taken to be the yield stress of the lid material at 250 °C. The maximum interaction ratio for the bearing load is therefore:

$$I.R. = \frac{Sbrg}{S_{yl}}$$

All terms are as previously defined. Thus,

$$\text{I.R.} = \frac{Sbrg}{S_{yl}} = \frac{94.4}{114.4} = 0.83$$

The minimum factor of safety is:

$$FS = \frac{1}{\text{I.R.}} = \frac{1}{0.83} = 1.21 > 1.0$$

Because the cask material is weaker than the bolting material, failure will occur at the root of the cask material threads. The minimum required thread engagement length to prevent cask material failure is determined in accordance with Machinery Handbook. Since the constants in the equation assume customary units, the metric units used for the cask design will be converted for determination of the required engagement length. Thus, the minimum engagement length, $L_e$, for the cask is:

$$L_e = \frac{S_{ub} \times 2 \times A_b}{S_{ul} \times \pi \times n \times D_{s,min} \times \left[\frac{1}{2 \times n} + 0.57735 \times \left(D_{s,min} - E_{n,max}\right)\right]}$$

All terms are as previously defined.

Thus,

$$L_e = \frac{S_{ub} \times 2 \times A_b}{S_{ul} \times \pi \times n \times D_{s,min} \times \left[\frac{1}{2 \times n} + 0.57735 \times \left(D_{s,min} - E_{n,max}\right)\right]}$$

$$= \frac{150000 \times 2 \times 2.28}{70000 \times \pi \times 5.08 \times 1.866 \times \left[\frac{1}{2 \times 5.08} + 0.57735 \times (1.866 - 1.705)\right]} \times \frac{25.4\,\text{mm}}{\text{in}}$$

$$= 43.5 \text{ mm} < L_{ep} = 73.0 \text{ mm}$$

Therefore, the primary closure lid bolts are acceptable for the hypothetical accident conditions.

**7.1.3.4 Secondary Lid Closure Bolt Evaluation under Hypothetical Accident Conditions**

The maximum tension, shear and bolt bearing loads in the secondary lid bolts due to the combined HAC loads will be evaluated in accordance with NUREG/CR-6007, with due consideration given to the prying effects on the fixed lid. Since the prying forces act inward, normal to the cask lid, an additional prying force is generated (NUREG 6007). For the HAC condition, the controlling load case is the summation of

the bolt preload, the internal pressure load and the end drop load. Since the internal pressure load will act counter to the drop load, the internal pressure load will be considered as negative for determination of the maximum bolt tension load. The maximum bolt tension load ($F_t$), shear load ($F_s$) and torsional moment ($M_t$) for the primary lid bolts are (see Table 7.1.3-2):

$$F_t = F_p - F_{as} + F_{ad}$$

$$F_s = F_{ss} + F_{sd}$$

$$M_t = M_{pt} + M_{at} + M_{dt}$$

where,

| | | | |
|---|---|---|---|
| $F_p$ | = | Maximum Bolt Preload | |
| | = | 72.2 kN/bolt | (Table 7.1.3-2) |
| $F_{as}$ | = | Bolt Internal Pressure Load | |
| | = | 6.8 kN/bolt | (Table 7.1.3-2) |
| $F_{ad}$ | = | Bolt End Drop Load | |
| | = | 507.8 kN/bolt | (Table 7.1.3-2) |
| $F_{ss}$ | = | Bolt Internal Pressure Shear Load | |
| | = | 0.0 kN/ bolt | (Table 7.1.3-2) |
| $F_{sd}$ | = | Bolt End Drop Shear Load | |
| | = | 0.0 kN/bolt | (Table 7.1.3-2) |
| $M_{pt}$ | = | Maximum Bolt Preload Torsional Moment | |
| | = | 0.2 kN-m/bolt | (Table 7.1.3-2) |
| $M_{at}$ | = | Bolt Internal Pressure Torsional Moment | |
| | = | 0.0 kN-m/bolt | (Table 7.1.3-2) |
| $M_{dt}$ | = | Bolt End Drop Torsional Moment | |
| | = | 0.0 kN-m/bolt | (Table 7.1.3-2) |

Thus,

$$F_t = F_p - F_{as} + F_{ad} = 72.2 - 6.8 + 277.6 = 343.0 \text{ kN/bolt}$$

$$F_s = F_{ss} + F_{sd} = 0.0 + 0.0 = 0.0 \text{ kN/bolt}$$

$$M_t = M_{pt} + M_{at} + M_{dt} = 0.2 + 0.0 + 0.0 = 0.2 \text{ kN-m/bolt}$$

Conservatively, the fixed-edge closure lid prying is taken from the end drop load case. The additional tensile bolt force per bolt, $F_{tp}$, caused by the prying action of the primary lid is (NUREG/CR-6007 Table 2.1):

$$F_{tp} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left[\frac{\frac{2 \times M_f}{D_{li} - D_{lb}} - C1 \times (B - F_f) - C2 \times (B - P)}{C1 + C2}\right]$$

where,

$F_f$ = Fixed Edge Closure Force
= 1717.6 kN/m (Table 7.1.3-2)
$M_f$ = Fixed Edge Closure Moment
= 198.8 kN-m/m (Table 7.1.3-2)

Other terms are as previously defined.

Thus,

$$F_{tp} = \left(\frac{\pi \times 0.926}{18}\right) \times \left[\frac{\frac{2 \times 198.8}{1.000 - 0.926} - 1 \times (1717.6 - 1717.6) - 1.80 \times (1717.6 - 446.7)}{1 + 1.80}\right]$$

= 178.0 kN/bolt

This bolt load is greater than the load generated by the minimum bolt preload (130.6 kN < 178.0 kN). However, the drop load is an inward load, which will press the closure lid against the sealing gasket. The prying force generated by the drop load will therefore not result in the loss of lid closure seal. The outward load of the internal pressure has already been evaluated in Section 7.1.3.2 and found to be acceptable. All other accident loads are acceptable by comparison.

The total tension force, $F_a$, is:

$$F_a = F_t + F_{tp} = 343.0 + 178.0 = 521.0 \text{ kN/bolt}$$

The maximum bending moment generated by the applied loads, $M_{bb}$, is:

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf$$

All terms are as previously defined. Thus,

$$M_{bb} = \left(\frac{\pi \times D_{lb}}{N_b}\right) \times \left(\frac{Kb}{Kb + K1}\right) \times Mf = \left(\frac{\pi \times 0.926}{18}\right) \times \left(\frac{2.39}{2.39 + 71.3}\right) \times 198.8$$

= 1.0 kN-m

The average bolt stresses from the combined HAC loads are determined in accordance with Table 5.1 of NUREG/CR-6007. The bolt stress diameter is as defined previously. The average tensile stress, Sba, average shear stress, Sbs, maximum bending stress, Sbb, and maximum torsional stress, Sbt, are:

$$Sba = \frac{1.2732 \times Fa}{D^2}$$

$$= \frac{1.2732 \times 521.0}{0.032^2} \times \frac{1\ \text{MPa}}{1000\ \text{kN/m}^2}$$

$$= 637.9\ \text{MPa}$$

$$Sbs = \frac{1.2732 \times Fs}{D^2}$$

$$= \frac{1.2732 \times 0.0}{0.032^2} \times \frac{1\ \text{MPa}}{1000\ \text{kN/m}^2}$$

$$= 0.0\ \text{MPa}$$

$$Sbb = \frac{10.186 \times M_{bb}}{D_b^3}$$

$$= \frac{10.186 \times 1.0}{0.036^3} \times \frac{1\ \text{MPa}}{1000\ \text{kN/m}^2}$$

$$= 241.6\ \text{MPa}$$

$$Sbt = \frac{5.093 \times M_t}{D_b^3}$$

$$= \frac{5.093 \times 0.2}{0.036^3} \times \frac{1\ \text{MPa}}{1000\ \text{kN/m}^2}$$

$$= 21.3\ \text{MPa}$$

The allowable stresses for the bolts are as previously defined. The maximum interaction ratio for the combined shear and tension loads is therefore:

$$I.R. = \left(\frac{Sba}{\sigma_{ta}}\right)^2 + \left(\frac{Sbs}{\sigma_{sa}}\right)^2$$

$$= \left(\frac{637.9}{723.9}\right)^2 + \left(\frac{0.0}{434.4}\right)^2$$

$$= 0.78$$

The minimum factor of safety is:

$$FS = \frac{1}{I.R.}$$

$$= \frac{1}{0.78}$$

$$= 1.29 > 1.0$$

The maximum interaction ratio for the bending load is therefore:

$$\text{I.R.} = \left(\frac{Sbb}{\sigma_{ba}}\right)^2 = \left(\frac{227.5}{651.6}\right)^2 = 0.12$$

The minimum factor of safety is:

$$FS = \frac{1}{\text{I.R.}} = \frac{1}{0.12} = 8.20 > 1.0$$

The primary lid closure bolts utilize a custom washer for the bolts with an outer diameter, $d_{ow}$, of 90 mm and a hole diameter, $d_{oh}$, of 40 mm. Therefore, the bearing stress under the bolt head, Sbrg, is:

$$Sbrg = \frac{Fa}{A_{brg}}$$

All terms are as previously defined. Thus,

$$Sbrg = \frac{521.0}{0.0051} \times \frac{1\ \text{MPa}}{1000\ \text{kN}/\text{m}^2} = 102.1\ \text{MPa}$$

The allowable normal condition bearing stress on the lid is taken to be the yield stress of the lid material at 250 °C. The maximum interaction ratio for the bearing load is therefore:

$$\text{I.R.} = \frac{Sbrg}{S_{yl}}$$

All terms are as previously defined.

Thus,

$$\text{I.R.} = \frac{Sbrg}{S_{yl}} = \frac{102.1}{114.4} = 0.89$$

The minimum factor of safety is:

$$FS = \frac{1}{I.R.} = \frac{1}{0.89} = 1.12 > 1.0$$

Because the cask material is weaker than the bolting material, failure will occur at the root of the cask material threads. The minimum required thread engagement length to prevent cask material failure is determined in accordance with Reference Machinery Handbook. Since the constants in the equation assume customary units, the metric units used for the cask design will be converted for determination of the required engagement length. Thus, the minimum engagement length, $L_e$, for the cask is:

$$L_e = \frac{S_{ub} \times 2 \times A_b}{S_{ul} \times \pi \times n \times D_{s,min} \times \left[\frac{1}{2 \times n} + 0.57735 \times \left(D_{s,min} - E_{n,max}\right)\right]}$$

All terms are as previously defined. Thus,

$$L_e = \frac{S_{ub} \times 2 \times A_b}{S_{ul} \times \pi \times n \times D_{s,min} \times \left[\frac{1}{2 \times n} + 0.57735 \times \left(D_{s,min} - E_{n,max}\right)\right]}$$

$$= \frac{150000 \times 2 \times 1.27}{70000 \times \pi \times 6.35 \times 1.396 \times \left[\frac{1}{2 \times 6.35} + 0.57735 \times (1.396 - 1.270)\right]} \times \frac{25.4\,\text{mm}}{\text{in}}$$

$$= 32.7 \text{ mm} < L_{ep} = 46.0 \text{ mm}$$

Therefore, the secondary closure lid bolts are acceptable for the hypothetical accident conditions.

### 7.1.4 Seal Integrity

The maximum stress analyses in the previous sections are based on criteria for the accident conditions intended to prevent failures by excessive plastic deformation or by the rupture of the bolt. Using the yield stress as the stress limit for average tensile bolt stress implies that a small amount (0.02%) of plastic deformation is permitted (Ref. 3.5). The following calculations show that the o-rings will continue to provide positive sealing of the closure lids even with this small plastic deformation.

#### 7.1.4.1 Primary Lid Seals

The 0.02% bolt plastic deformation permitted in NUREG/CR-6007 (Ref. 3.5) is distributed over the 67 mm bolt shank dimension shown in Detail 1 of Ref. 3.2. This may result in a separation between the primary lid and cask flange mating surfaces of 0.0134 mm (= 67mm x 0.0002). However, the primary lid seals are 12 +/-0.3 mm diameter EPDM rubber and the grooves for these seals are 9.4 +/- 0.15 mm deep (see Ref 3.2). Thus, the seal is minimally compressed 2.15 mm (= (12 - 0.3) - (9.4 + 0.15)). Considering that EPDM O-rings have a compression set of up to 45% (Appendix 1, Figure 2) at 150 °C, the minimum compression in the seal is 1.18 mm (= 2.15 – 0.45*2.15). Since the minimum seal compression greatly exceeds the separation due to possible plastic deformation, the primary lid/cask flange containment boundary will remain sealed following an HAC drop event.

#### 7.1.4.2 Secondary Lid Seals

The 0.02% bolt plastic deformation permitted in NUREG/CR-6007 (Ref. 3.5) is distributed over the 43 mm bolt shank dimension shown in Detail 1 of Ref. 3.2. This may result in a separation between the secondary and primary lid mating surfaces of 0.0086 mm (= 43mm x 0.0002). However, the secondary seals are 12 +/-0.3 mm diameter EPDM rubber and the grooves for these seals are 9.4 +/- 0.15 mm deep (see Ref 3.2). Thus, the seal is minimally compressed 2.15 mm (= (12 - 0.3) - (9.4 + 0.15)). Considering that EPDM O-rings have a compression set of up to 45% (Appendix 1, Figure 2) at 150 °C, the minimum compression in the seal is 1.18 mm (= 2.15 – 0.45*2.15). Since the minimum seal compression greatly exceeds the separation due to possible plastic deformation, the primary to secondary lid containment boundary will remain sealed following an HAC drop event.

### 7.2 Vibration

10 CFR 71.71 (c)(5) requires that "vibration normally incident to transport" be evaluated. The RT-100 package consists of thick section materials that will be unaffected by vibration normally incident to transport, such as over the road vibrations.

### 7.2.1 Vibration Evaluation of the RT-100 Cask Primary Lid Bolts

It is assumed that the cask will be traveling a total distance, $D_t$, of 1.61 million kilometers at an average speed, $V_t$, of 70 kilometers/hour during its service life (NUREG 0128). Therefore, the time during which the cask is in transit is:

$$t_t = \frac{D_t}{V_t} = \frac{1610000}{70.0} \times \frac{3600\text{sec}}{1\text{hr}} = 82{,}800{,}000 \text{ seconds}$$

Assuming that the cask package on the conveyance has a fundamental frequency, F, of 2.5 Hz (NUREG 0128), the cask will be subjected to a load cycle of:

$$C_L = F \times t_t = 2.5 \times 82800000 = 207{,}000{,}000 \text{ cycles}$$

Thus, the RT-100 package may be subjected to a cycle range typically associated with high-cycle fatigue (> $10^8$ cycles). The endurance limit of the material for the high cycle fatigue can be approximated by using a 60% reduction, $r_h$, of the ultimate tensile strength (AISC Guide to Design Criteria for Bolted and Riveted Joints) with an additional 10% reduction, $r_g$, for the connection surface (Machinery's Handbook). Thus the endurance limit for the material is:

$$S_a = (1 - r_h) \times (1 - r_g) \times S_{ub}$$

where,

$S_{ub}$ = Bolt Ultimate Stress
= 1034.2 MPa (ASTM A354 Grade B, Table 1)

$$S_a = (1 - 0.60) \times (1 - 0.10) \times 1034.2 = 372.3 \text{ MPa}$$

NUREG 0128 gives the following RMS vibration load factors for the road travel:

$f_v$ = Vertical Vibration Load Factor
= 0.52

$f_L$ = Longitudinal Vibration Load Factor
= 0.27

$f_t$ = Transverse Vibration Load Factor
= 0.19

The RT-100 package is transported in the vertical orientation. The cask lid will be subjected to vibration in the vertical direction. A notch factor, $f_N$, of 3.0 will be used, conservatively (AISC Guide to Design Criteria for Bolted and Riveted Joints). The vibration stress in the bolts is:

$$s_v = \frac{F_b \times f_N}{A_b}$$

where,

$F_b$ = Bolt Force due to Vibration

$$= \frac{f_v \times W_{Lp} \times g}{N_b}$$

$A_b$ = Bolt Stress Area
= 1470 mm$^2$ (Machinery Handbook)

$W_{Lp}$ = Cask Lid Weight
= 3648 kg, use 3650 kg (Ref. 3.2)

$N_b$ = Number of Bolts
= 32 (Ref. 3.2)

$$F_b = \frac{0.52 \times 3650 \times 9.81}{32} \times \frac{1\,\text{kN}}{1000\,\text{N}}$$

= 0.58 kN

$$s_v = \frac{0.58 \times 3.0}{0.001470} \times \frac{1\,\text{MPa}}{1000\,\text{kN/m}^2}$$

= 1.19 MPa << $S_a$ = 372.3 MPa

Since the stress in the bolts is well below the endurance limit of the material, the primary lid bolts will not be subjected to transportation-related fatigue damage during their service life.

The maximum shock loading coefficient for the three orthogonal directions is specified as 2.9 (NUREG 0128). The cask primary lid will be subjected to shock loading during transport. The primary lid closure bolts have been shown to withstand a 125 g impact load, which is much larger than the 2.9W shock loading during transport. The primary lid closure bolts are therefore acceptable for shock loading by comparison.

**7.2.2 Vibration Evaluation of the RT-100 Cask Secondary Lid Bolts**

Per Section 7.2.1, the components of the package are into high-cycle fatigue range (> $10^8$ cycles). The endurance limit of the material for the high cycle fatigue for the secondary lid bolts is the same as for the

primary lid bolts. The cask lid will be subjected to vibration in the vertical direction. A notch factor, $f_N$, of 3.0 will be used, conservatively (AISC Guide to Design Criteria for Bolted and Riveted Joints). The vibration stress in the bolts is:

$$s_v = \frac{F_b \times f_N}{A_b}$$

where,

$F_b$ = Bolt Force due to Vibration

$$= \frac{f_v \times W_{Lp} \times g}{N_b}$$

$A_b$ = Bolt Stress Area
= 817 $mm^2$ (Machinery's Handbook)

$W_{Ls}$ = Cask Lid Weight
= 857 kg (Ref. 3.2)

$N_b$ = Number of Bolts
= 18 (Ref. 3.2)

All other quantities are as defined in Section 7.2.1.

$$F_b = \frac{0.52 \times 857 \times 9.81}{18} \times \frac{1\,\text{kN}}{1000\,\text{N}}$$

$$= 0.24\ \text{kN}$$

$$s_v = \frac{0.24 \times 3.0}{0.000817} \times \frac{1\,\text{MPa}}{1000\ \text{kN}/\text{m}^2}$$

$$= 0.89\ \text{MPa} << S_a = 372.3\ \text{MPa}$$

Since the stress in the bolts is well below the endurance limit of the material, the secondary lid bolts will not be subjected to transportation-related fatigue damage during their service life.

The maximum shock loading coefficient for the three orthogonal directions is specified as 2.9 (NUREG 0128). The cask primary lid will be subjected to shock loading during transport. The secondary lid closure bolts have been show to withstand a 125g impact load (Section 2.13.4.2 proprietary), which is much larger than the 2.9W shock loading during transport. The secondary lid closure bolts are therefore acceptable for shock loading by comparison.

The RT-100 satisfies the requirements for normal vibration incident to transport as required by 10 CFR 71.71(c)(5) [Ref. 2].

# Appendix 1

Elastomer O-Ring Data

## Joints toriques

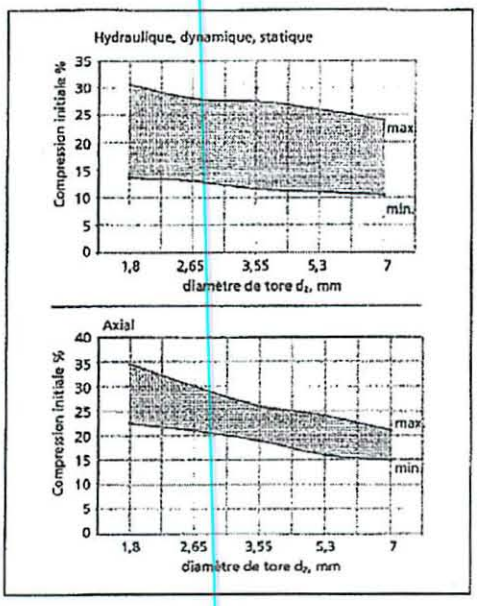


Figure 16 Plage admissible de compression initiale en fonction du diamètre de tore, montage radial statique ou axial

### Efforts de compression

Les efforts de déformation varient avec l'écrasement initial et la dureté Shore. La figure 17 montre l'effort de compression spécifique en N par cm de la circonférence du joint en fonction du diamètre de tore.

Les efforts de compression indiqués peuvent être utilisés pour estimer la force totale à appliquer pour le montage statique des joints toriques.

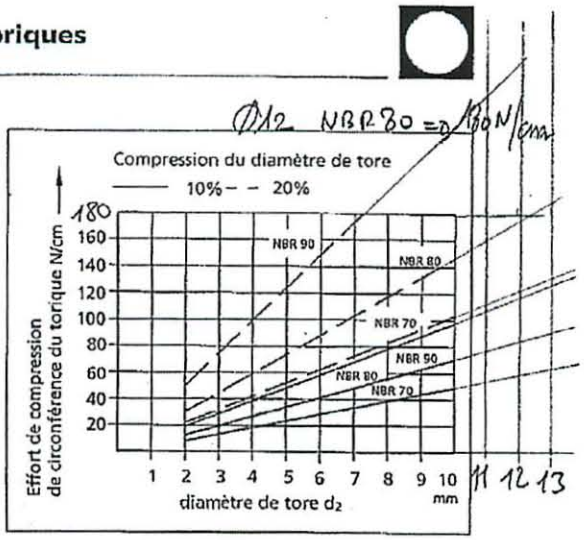


Figure 17 Efforts de compression sur la circonférence du joint torique selon le matériau

### B.2.3 Allongement- compression

Dans le cas d'une étanchéité radiale, le joint torique monté dans une gorge interne - "étanchéité extérieure" - doit être étiré sur tout le diamètre de la gorge. L'allongement maximal au montage est de 6% pour les joints toriques de diamètre intérieur > 50 mm et de 8% pour les joints toriques de diamètre intérieur < 50 mm.

Avec des gorges externes - "étanchéité intérieure" - le joint torique est comprimé de préférence sur sa périphérie. La compression périphérique maximale au montage est de 3%.

Le dépassement de ces valeurs se traduira par une augmentation ou une diminution importante du diamètre de tore du joint torique, ce qui risque de réduire la durée de vie du joint.

On peut calculer la réduction du diamètre de tore ($d_2$) à l'aide de la formule suivante

$$Reduction_{max} = \frac{d_{2min}}{10} \cdot \sqrt{6 \cdot \left( \frac{d_{3max} - d_{1min}}{d_{1min}} \right)}$$

avec $d_{1min}$ = diamètre intérieur minimal du joint torique
$d_{2min}$ = diamètre de tore minimal du joint torique
$d_{3max}$ = diamètre maximal du logement

mais on peut dire qu'il est approximativement égal à la moitié de l'allongement. Un allongement de 1% correspond à une réduction d'environ 0,5% du diamètre de tore ($d_2$).



Figure 1. Compression Force

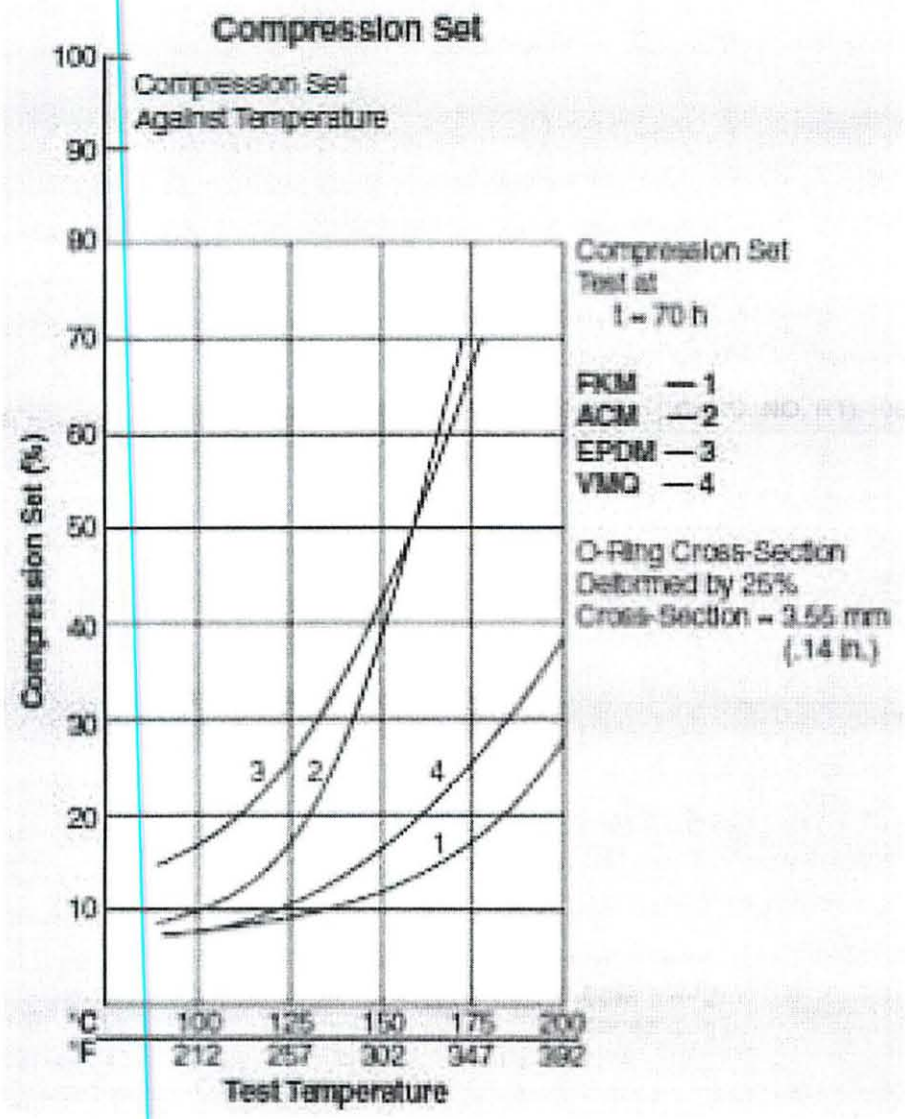


Figure 2-13: Compression Set vs. Polymer Family

Figure 2. Compression set for EPDM O-ring (Ref. 3.21)

# Appendix 2

Cask Design Input Email

**Johnathon McFarland**

**From:** Curt Lindner [clindner@robateltech.com]
**Sent:** Friday, July 06, 2012 4:04 PM
**To:** John Staples; Johnathon McFarland
**Subject:** RT-100 - Maximum Payload and Gross Weights

For the purposes of preparing the lifting, tie-down and bolting analyses, consider a maximum payload weight as 15000 lbs (6820 kg) and a maximum total weight of 41,000 kg (90,200 lbs). These values bound the sum of the maximum gross weight in the procurement agreement and the maximum cask weight of 34054 kg (74,920 lbs) per Robatel drawing RT-100 PE 1001-1, Rev. C.

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