JUSTIFICATION OF THE MECHANICAL STRENGTH OF PACKAGE MODEL SCREWED ASSEMBLIES UNDER ROUTINE TRANSPORT CONDITIONS

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SUMMARY

This document presents the justification of the mechanical strength of package model screwed assemblies under routine transport conditions, specifically:

- lid to body attachment screws (item 150);
- cover to body attachment screws (item 151):
- orifice plugs to lid attachment screws (item 350)

Specific calculations are carried out for trunnion screws, as presented in section 1-3 in the safety analysis report.

Mechanical properties of materials at 120°C are selected to cover the maximum temperature of the cavity (shell) under normal transport conditions (see chapter 2).

The maximum axial force on the screws is calculated from the vertical acceleration under normal transport conditions. Moreover, for containment screws (lid screws and plug screws), the internal pressure and the force induced by the presence of seals are taken into account in the calculation of the maximum axial force.

The maximum axial force applied to each screw type is less than the minimum preload of screws due to the tightening torque. Thus, it is guaranteed that there will be no disengagement of clamped parts including parts of the containment, under routine transport conditions.

Von Mises stresses in screws and shear stresses in screw threads and tapped threads are less than allowable criteria. Differential expansion of screwed systems is taken into account.

Stresses due to pressure under the head are also below allowable criteria.

Therefore the mechanical strength of the lid, shock absorbing cover and plug screws is guaranteed under routine transport conditions.

1. INTRODUCTION

The purpose of this chapter is to justify the mechanical strength of the TN-MTR package model screwed assemblies under routine transport conditions.

Thus, stresses in the screws and the tapped threads are calculated and compared with the mechanical properties of materials. It is also checked that screwed assemblies do not disengage under forces resulting from routine transport conditions, in other words forces related to:

- a conservative maximum internal pressure of 7 bars (according to <3>);
- a vertical upwards acceleration equal to 2 g;
- temperature variations (differential expansion);
- compression of seals.

This chapter applies to the following screws:

- lid to body attachment screws (item 150), fitted with washers (item 153), screwed into the packaging flange (item 105);
- shock absorbing cover to body attachment screws (item 151), fitted with washers (item 152), screwed into the packaging flange (item 105);
- orifice plug to lid attachment screws (item 350), screwed into the top disk of the lid (item 307).

Specific calculations are carried out for trunnion screws, as presented in section 1-3 in the safety analysis report.

2. BASIC DATA

2.1. Temperature considered

Conservatively, the mechanical properties are considered at 120°C because this temperature upper-bounds the cavity temperature under normal transport conditions (NTC) ([redacted] according to chapter 2).

2.2. Packaging components

The following table presents components of the packaging considered in the calculations in this chapter: the component with the screw tapped thread and the clamped part are specified for each screw type, with their items.

Screw type	Tapped thread	Clamped part
Lid screws (150) + washers (153)	Flange (105)	Lid upper disk (307)
Shock absorbing cover screws (151) + washers (152)	Flange (105)	Screw support ring (722)
Plug screws (350)	Lid upper disk (307)	Plugs (330 and 333)

2.3. Properties of materials

2.3.1. Screw materials

The mechanical properties of screw materials used in this chapter are presented in the following table. They are taken from chapter 0.

Screw	Lid screws	Cover screws	Plug screws
Material	Class 10.9		
Yield stress R_e at 20°C (MPa)	≥ 900		
Ultimate strength R_m at 20°C (MPa)	≥ 1,000		
Coefficient of thermal expansion α_1 (/°C)	11.5×10^{-6}		
Young's modulus (MPa)	212 000		

According to Appendix B in <1>, there is no significant variation in the mechanical properties of screws for temperatures varying up to 150°C. Therefore the mechanical properties specified in this table are used for the temperature considered in this study (see section 2.1).

2.3.2. Material used for tapped threads and clamped parts

The following table gives the mechanical properties of steels from which the tapped threads and clamped parts used in this chapter are made.

Packaging component	Lid upper disk (307)	Flange (105)	Ring (722)	Plugs (330/333)
Steel	**Type K**	**Type J**	**Type A**	**Type B**
Yield stress Re at 120°C (MPa)	≥ 430	≥ 216	≥ 146	≥ 353
Ultimate strength R_m at 120°C (MPa)	≥ 650	≥ 486	≥ 431	≥ 616
Coefficient of thermal expansion α_2 (/°C)	16×10^{-6}			
Young's modulus (MPa)	200 000			

These mechanical properties are interpolated linearly from the mechanical properties given in chapter 0.

2.4. Tightening torques

Tightening torques for the different screws are shown in the table below: These data are taken from Chapter 0.

Screw	Tightening torque (N.m)
Lid screws	660 $\pm$ 10%
Shock absorbing cover screws	660 $\pm$ 10%
Plug screws	40 $\pm$ 10%

2.5. Screw geometry

The geometric properties of screws used in this chapter are given in table 1-10.1 and are taken from chapter 0 and <2>.

2.6. Criteria

Criteria guaranteeing the mechanical strength of screws are determined from the mechanical properties of materials (see section 2.3) at the temperature defined in section 2.1.

The following criteria are considered for the different calculations performed in this study:

- Von Mises stresses in screws are compared with yield stresses of steels from which the screws considered are made;
- shear stresses in screw threads and tapped threads are compared with yield stresses of steels used for screws and threaded holes, multiplied by a factor $\frac{1}{\sqrt{3}}$;
- pressures under the head are compared with the value of $\frac{R_e + R_m}{2}$ (see Appendix B in <8>), for materials from which clamped parts are made.

All criteria used in this chapter are summarised in the following table:

		Criterion (MPa)		
Stress / Pressure		Lid screws	Cover screws	Plug screws
(Von Mises)		900		
Shearing of threads	Screw	520		
	Tapped threads	125	125	248
Pressure under head	Clamped part	540	289	484

3. MECHANICAL ANALYSIS OF THE SCREWS

3.1. Loading conditions

The loading conditions considered in this analysis of screwed assemblies are as follows:

3.1.1. Pressure conditions

The internal pressure of the packaging applies a force to the screws of the containment (lid screws and plug screws). The internal pressure used for the calculation corresponds conservatively to a pressure of 7 bars absolute, which is greater than the pressure reached under normal conditions of transport (see Chapter 3A).

The force in the screws due to this internal pressure is given by:

$$F_{pressure} = \frac{\pi}{4} \frac{P D_p^{\ 2}}{n}$$

Where:

- P: the internal pressure of the packaging (Pa), assumed equal to 7 x 10^5 Pa (7 bars according to <3>);
- D_p: the average diameter of the seal delimiting the area to which the pressure is applied (internal seal, see table 1-10.2) (m);
- n: the number of screws in the assembled part (n = 36 for the lid and n = 4 for the plugs).

Forces in the screws due to the internal pressure are given in the following table:

Screw	Lid screws	Plug screws
Force $F_{pressure}$ (kN)	17	1

3.1.2. Accelerations related to routine conditions of transport

Accelerations due to braking, starting, turning and simple gravity create forces in the screws. These forces are related to the mass of components of the packaging and that of its content. According to <3>, the maximum acceleration in the upwards vertical direction is 2 g for routine transport conditions. This acceleration is conservative: the IMO directive <4> does not specify any upwards vertical acceleration. Considering the downwards acceleration due to gravity equal to 1 g, the selected vertical acceleration is 1 g upwards.

The force in the screws due to external accelerations applied to components is given by:

$$F_{acc} = \frac{M \gamma g}{n}$$

Where:

- M: the mass of components held in place by the screws (kg);
- γ : accelerations under routine conditions of transport (1 g);
- g: acceleration due to gravity, (9.81 m/s²);
- n, the number of screws for the assembled part.

The data used to calculate this force are given in Table 1-10.3.

Forces in the lid, shock absorbing cover and plug screws are given in the following table.

Screws	Lid screws	Cover screws	Plug screws
Force F_{acc} (kN)	2	3	2 x 10^{-2}

3.1.3. Forces induced by the presence of seals

Lid seals (items 361 and 362) and plug seals (items 365 and 366) apply a force which opposes the tightening of the lid and plug screws.

The seal compression force is equal to:

$$F_{seal} = \frac{\pi}{n} \sum_{i=1}^{j} (D_i Y_i)$$

Where:

- D_i: the mean reaction diameter of seal No. i (m);
- Y_i: the force per unit length compressing seal No. (N/m);
- n: the number of screws for the assembled part.

Properties of lid and plug seals are given in table 1-10.2 (the dimensions are taken from chapter 0).

Forces due to seals applied to the screws are given in the following table:

Screws	Lid screws	Plug screws
Force F_{seal} (kN)	5	3

3.1.4. Maximum axial force

The maximum axial force applied to a screw is equal to the sum of the following forces:

- the force due to the internal pressure in the packaging $F_{pressure}$ (see section 3.1.1);
- the force due to accelerations under routine transport conditions F_{acc} (see section 3.1.2);
- the force induced by the presence of seals F_{seal} (see section 3.1.3).

Thus, the maximum axial force in a screw is equal to:

$$F_{E\max} = F_{pressure} + F_{acc} + F_{seal}$$

The maximum axial force for each screw type is summarised in the following table:

Screws	Lid screws	Cover screws	Plug screws
Force F_{Emax} (kN)	24	3	4

3.2. Preload in screws

The minimum and maximum preloads in screws can be calculated as follows based on screw tightening torques (see section 2.4), (according to <2>):

$$F_0 = \frac{C(1+\Delta C)}{0{,}16\,p + \mu(1+\Delta\mu)\times(0{,}583\,d_2 + r_m)}$$

Where:

- F_0: preload force due to the tightening torque applied to the screw (N);
- C: the screw tightening torque (N.m);
- ΔC : uncertainty on the screw tightening torque (%);
- μ : the coefficient of friction at the screw threads and under the screw head, $\mu = 0{,}0742$;
- $\Delta\mu$: the uncertainty on the coefficient of friction at the screw threads and under the screw head, $\Delta\mu = \pm 16{,}10\%$;
- p: the screw pitch (m);
- d_2: the diameter at the flank of the screw thread (m), $d_2 = d - 0.6495\,p$;
- d: nominal screw diameter (m);
- r_m : average bearing radius under the head (m), $r_m = \frac{1}{3}\left(\frac{d_e^3 - d_i^3}{d_e^2 - d_i^2}\right)$;

- d_e: outside diameter of the bearing surface under the screw head (m):
  - $d_e = d_{e\ screw}$ for screws without washer;
  - $d_e = \min (d_{e\ screw} ; d_{e\ washer})$ for screws with washer;
- $d_{e\ screw}$: outside diameter of screw head (m);
- $d_{e\ washer}$: outside diameter of washer (m);
- di: inside diameter of the bearing surface under the screw head (m):
  - $d_i = d_b$ for screws without washer;
  - $d_i = \max (d_b ; d_{i\ washer})$ for screws with washer (for screws with captive washer, $d_i = d_b$);
- d_b : diameter of the screw hole (m);
- $d_{i\ washer}$: inside diameter of washer (m);

The minimum and maximum preloads of the different screws due to the tightening torque are listed in the following table:

Screws	**Minimum preload due to the torque: $F_{0\ min}$ (kN)**	**Maximum preload due to the torque: $F_{0\ max}$ (kN)**
Lid screws	161	258
Cover screws	118	189
Plug screws	24	38

It is checked that the minimum preload in screws due to the torque is more than the maximum axial force applied to the screws (see section 3.1.4). Therefore it is guaranteed that clamped parts will not disengage.

3.3. Expansion of materials

Temperature conditions have an influence on screw preloads.

The force in a screw due to differential expansion is evaluated by the following formula (according to <5>):

$$\Delta F = \frac{(\alpha_2 - \alpha_1) \times (T_2 - T_1) \times E_1 \times A_1}{1 + \dfrac{E_1 \times A_1}{E_2 \times A_2}}$$

Where:

- ΔF : the force in the screw due to differential expansion between the screw and the clamped part (N);
- α_2: the coefficient of thermal expansion of the material from which the clamped part is made (α_2 = 16 x 10^{--6} C^{-1});
- α_1: the coefficient of thermal expansion of the material from which the screw is made (α_1 = 11.5 x 10^{--6} C^{-1});
- T_2: the maximum temperature under routine transport conditions T_2 = 120°C according to section 2.1);

- T_1: the ambient temperature at tightening, assumed to be equal to 20°C;
- E_1: Young's modulus of the material from which the screw is made (E_1 = 212 000 MPa);
- A_1: the stress area of the screw(m²), $A_1 = \frac{\pi}{4} d_s{}^2$;
- d: the stress diameter of the screw, $d_s = \min(d_{eq}\,;\, d_{dec})$;
- d_{eq}: equivalent screw diameter (m), $d_{eq} = \frac{d_2 + d_3}{2}$;
- d_3 the diameter at the root of the screw thread (m), $d_3 = d - 1.2268\, p$;
- d_{dec} : the screw shank machined diameter ;
- E_2: Young's modulus of the material from which the clamped part is made (E_2 = 200 000 MPa);
- A_2: bearing area between the screw head and the clamped part or the washer (m²), $A_2 = \frac{\pi}{4}(d_e{}^2 - d_i{}^2)$;
- d_e outside diameter of the contact surface under the screw head (m):
- d_i inside diameter of the contact surface under the screw head (m):

The thickness of the washer is ignored in this calculation.

Forces due to differential expansion for the lid, shock absorbing cover and plug screws are given in the following table. The maximum preload of the screws is then calculated such that:

$$F_{\max} = F_{0\max} + \Delta F$$

The results obtained for expansion of materials between 20°C and 120°C are presented in the table below:

Screws	**Differential expansion $\Delta F_{120°C}$ (kN)**	**Preload due to torque F_{0max} (kN)**	**Maximum preload F_{max} (kN)**
Lid screws	26	258	283
Cover screws	57	189	246
Plug screws	4	38	42

3.4. Maximum equivalent stress in the screws:

Maximum stresses in the packaging screws are calculated from the maximum preloads calculated in the previous section.

The maximum tensile stress in the screws is given by:

$$\sigma_{\max} = \frac{F_{\max}}{A_1}$$

Where:

- F_{max}: the maximum preload force (N), taking account of differential expansion between materials (see section 3.3);
- A_1: stress area of the screw, $A_1 = \frac{\pi}{4} d_s^2$ (m²).

The maximum torsion stress is given by:

$$\tau = \frac{16 C_t}{\pi d_s^3}$$

Where:

- C_t: the maximum torsion in the screw (N.m), $C_t = F_{\max}(0{,}16\, p + \mu_{\min} \times 0{,}583 d_2)$ (according to <2>) ;
- $\mu_{\min}$: the minimum coefficient of friction at the screw threads and under the screw head.

Thus, we obtain the maximum Von Mises equivalent stress:

$$\sigma_{VM} = \sqrt{\sigma_{\max}{}^2 + 3\tau^2}$$

The following table contains results for lid, shock absorbing cover and plug screws.

Stress (MPa)	Lid screws	Cover screws	Plug screws
Tension $\sigma_{\max}$	577	242	500
Torsion τ	145	57	130
Von Mises σ_{VM}	629	262	549
Criterion	*900*		

Von Mises equivalent stresses in screws are less than the criteria defined in section 2.6: therefore the mechanical strength of lid, shock absorbing cover and plug screws is guaranteed under routine transport conditions.

3.5. Shear stresses in the threads:

The shear strength of screw threads and internal tapped threads is checked using the method described in reference <6>.

Shear stresses in threads are calculated as follows:

- In the tapped thread: $\tau = \dfrac{F_{\max}}{S_t\, C_1\, C_3}$
- In the screw: $\tau = \dfrac{F_{\max}}{S_{vis}\, C_1\, C_2}$

Where:

- F_{max}: the maximum force in the screws (N), taking account of differential expansion between materials (see section 3.3);
- S_t: the stress area of tapped threads (m²), $S_t = \dfrac{7}{8}\pi\, d\, L$;
- S_{screw}: the stress area of screw threads (m²), $S_{srew} = \dfrac{3}{4}\pi\, D_1\, L$;
- L; the minimum contact length between screw threads and tapped threads (m);
- d: nominal screw diameter (m);
- D_1: inside diameter of tapped threads (m), $D_1 = d_1 = d - 1.0825\, p$;
- p: the screw pitch (m);
- C_1: expansion factor for the nut. Since the tapped threads for lid, shock absorbing cover and plug screws are in thick walls, $C_1 = 1$;
- C_2: bending factor of screw threads. If $R_S \leq 1$, $C_2 = 0.897$; otherwise,

  $C_2 = 5{,}594 - 13{,}682\, R_s + 14{,}107\, R_s^{\,2} - 6{,}057\, R_s^{\,3} + 0{,}9353\, R_s^{\,4}$;
- R_s: thread strength ratio, $R_s = \dfrac{R_{mtapping}\, S_{tapping}}{R_{mscrew}\, S_{vscrew}}$;
- $R_{m\ tapping}$ and $R_{m\ screw}$: the ultimate strengths of the tapped thread and the screw respectively;
- C_3: bending factor of tapped threads. If $R_S \geq 1$, $C_3 = 0.897$; otherwise,

  $C_3 = 0{,}728 + 1{,}769\, R_s - 2{,}894\, R_s^{\,2} + 1{,}296\, R_s^{\,3}$.

Shear stresses for screws and shells are given in the following table.

Stress (MPa)	Lid screws	Cover screws	Plug screws
Screw stress	128	57	164
Screw criterion	*520*		
Tapping stress	85	38	116
Tapping criterion	*125*		*248*

Shear stresses for screws and shells are less than allowable criteria (see section 2.6). Therefore the mechanical strength of threads is guaranteed.

3.6. Pressures under the screw head

If screws are not fitted with washers (case of plug screws), the pressure under the screw head is calculated as follows:

$$\sigma_t = \frac{F_{\max}}{\frac{\pi}{4}(d_e^2 - d_i^2)}$$

Where:

- F_{max}: the maximum preload in the screw (N), taking account of differential expansion between materials (see section 3.3);
- d_e: outside diameter of the bearing surface under the screw head;
- d_i: inside diameter of the bearing surface under the screw head.

For screws that are fitted with washers (case of lid and shock absorbing cover screws), the contact pressure between the washer and the clamped part is given by:

$$\sigma_c = \frac{F_{\max}}{\frac{\pi}{4}(d_{e\,washer}^2 - d_i^2)}$$

Where:

- $d_{e\,washer}$: outside diameter of washer;
- σ_c: the contact pressure at the contact between the washer and the clamped part (lid or shock absorbing cover) (Pa);

The pressures under the screw head are presented in the following table:

	Lid screws	**Cover screws**	**Plug screws**
Contact pressure (MPa)	451	109	378
Criterion	*540*	*289*	*484*

Contact pressures are below the criteria defined in section 2.6. Therefore there will be no bearing of clamped parts.

4. CONCLUSIONS

The maximum axial force applied to each screw type is less than the minimum preload of screws due to the tightening torque. Thus, it is guaranteed that there will be no disengagement of clamped parts including those of the containment, under routine transport conditions.

Von Mises stresses in screws and shear stresses in screw threads and threaded holes are less than allowable criteria. Differential expansion of screwed systems is taken into account.

Stresses due to pressure under the head are also below allowable criteria.

Therefore the mechanical strength of the lid, shock absorbing cover and plug screws is guaranteed under routine transport conditions.

5. REFERENCES

<1> Standard ISO 898-1, 2013– Mechanical properties of carbon steel and alloyed steel attachment elements – Part 1: Screws, studs and threaded rods with specified quality classes - coarse threads and fine threads;

<2> Standard E 25-030 - August 1984 –Attachment elements - screwed assemblies;

<3> Applicable IAEA regulations: see chapter 00;

<4> IMO/OIT/EEC-UNO directive for loading cargoes in transport equipment;

<5> Strength of Materials, S.P. TIMOSHENKO, Volume 1, Editions Dunod;

<6> <6> « Analysis and Design of Threaded Assemblies» - E. M. Alexander – Society of Automotive Engineers, Inc - 1978 – No 770420

<7> Parker O-Ring Handbook, Catalog ORD 5700A/US, Parker Seals, 2001 Edition ;

<8> Standard NF E 25-030-1 - December 2007 –Attachment elements - Screwed assemblies, Part 1: General design, calculation and erection rules.

LIST OF TABLES

TABLE 1-10.1
GEOMETRIC PROPERTIES OF PACKAGING SCREWS

Screws	Lid screws	Cover screws	Plug screws
Nominal diameter: d (mm)	30	42	12
Screw pitch: p (mm)	3.5	4.5	1.75
d_1 (mm)	26.211	37.129	10.106
Thread flank diameter: d_2 (mm)	27.727	39.077	10.863
Screw core diameter: d_3 (mm)	25.706	36.479	9.853
Screw shank machined diameter: d_{dec} (mm)	25	36	-
Equivalent diameter: d_{eq} (mm)	26.716	37.778	10.358
Screw stress diameter: d_s (mm)	25	36	10.358
Screw outside diameter: $d_{e\ screw}$ (mm)	45	63	18
Washer outside diameter: $d_{e\ washer}$ (mm)	45	70	-
Outside diameter of the bearing surface under the screw head: d_e (mm)	45	63	18
Inside diameter of the bearing surface under the screw head: d_i (mm)	35	45	13.5
Average radius under-head: r_m (mm)	20.1	27.25	7.93
Minimum coverage: L (mm) (*)	40	55	12
Stress area: A_1 (mm²)	490.9	1017.9	84.3
Contact area under screw head: A_2 (mm²)	628.3	1526.8	111.3

(*) used conservatively

TABLE 1-10.2

DATA CONCERNING SEALS

Seal	Lid		Plugs	
	Inner seal	**Outer seal**	**Inner seal**	**Outer seal**
Inner diameter (mm)	1,055	1,093	81.93	111.14
Torus diameter (mm)	7.8	7.8	5	5
Average diameter (mm)	1,062.8	1,100.8	86.93	116.14
Compression force per unit length	28,000 (1)		18,000 (2)	

According to chapter 0, the hardness of these EPDM seals is between 75 and 85 Shores. According to section 2.4.2 in <7>, the hardness is usually expressed with a tolerance of ± 5 Shores. Therefore a hardness of 80 Shores will be assumed for these seals. Furthermore, figures 2-4 to 2-8 in <7> show that the linear compression force increases with the torus diameter. Thus, these forces in the above table are:

(1) obtained by extrapolation, for a torus diameter of 7.8 mm, of the linear compression forces in figures 2-7 and 2-8 in <7>, for a compression of 30% and a hardness of 80 Shores.

(2) taken conservatively considering the linear compression force in figure 2-7 in <7>, with a compression of 30% and a hardness of 80 Shores.

TABLE 1-10.3

DATA REQUIRED TO CALCULATE THE FORCE DUE TO ACCELERATION

Screws	Lid screws	Cover screws	Plug screws
Number of screws	36	6	4
Components held in place by the screws	Lid + load	Shock absorbing cover	Plugs
Mass of components held in place by the screws (kg)	2,700 + 2,800 = 5,500	1,650	7