RESISTANCE OF THE TN-MTR PACKAGING

TO REGULATORY PRESSURES

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REVISION STATUS

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SUMMARY

This appendix to Chapter 1 presents the verification that the TN-MTR packaging will resist pressure. In particular, this analysis is used to define the maximum allowable internal pressure (used to analyse the activity released) and the resistance to external test pressure as defined by the IAEA <1> <1'>.

The TN-MTR packaging is designed to obtain a type B(U) approval and for packages of fissile materials.

Requirements for this type and concerning the containment pressure are:

- 1. Water immersion test (<1> <1'>): resistance to an external gauge pressure of at least 150 kPa = 1.5 bars.

- 2. Water Immersion test for packages containing irradiated nuclear fuels (<1> <1'>): resistance to an external gauge pressure of at least 2 MPa = 20 bars.

The check on point 1. water immersion test is combined with the calculation of the maximum allowable internal pressure and is made by calculating maximum allowable pressures according to CODAP 95 rules <2>. This calculation shows that the allowable pressure in the cavity is 11 bars for an internal or external pressure. This value is higher than the water immersion test pressure (<1> <1'>): 1.5 bars

Considering this result concerning the maximum allowable internal pressure, the test pressure for the TN-MTR packaging is fixed at 11 bars.

The check on point 2. water immersion test for packages containing irradiated nuclear fuels, is made using analytic calculations by applying a pressure of 20 bars to the internal containment of the TN-MTR packaging and comparing the stresses in the steel (calculated maximum stress = 384 MPa) at 1.5 times the yield stress.

Conservatively, the mechanical properties of the containment steels are considered at 200°C . Chapter 2 contains calculations to demonstrate that the containment temperature is actually less than 200°C.

1. PURPOSE

This appendix to chapter 1 describes the calculation of the maximum allowable pressure according to CODAP 95 calculation rules (internal pressure) and the check on the resistance of the TN-MTR packaging to regulatory pressures defined by the IAEA <1> <1’> (external pressures).

The TN-MTR packaging is designed to obtain a type B(U) approval and for packages of fissile materials.

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- 2. Water Immersion test for packages containing irradiated nuclear fuels (<1> <1’>): resistance to an external gauge pressure of at least 2 MPa = 20 bars.

Conservatively, the mechanical properties of the containment steels are considered at 200°C . Chapter 2 contains calculations to demonstrate that the containment temperature is actually less than 200°C.

2. CALCULATION FOR THE CONTAINMENT VESSEL UNDER NORMAL CONDITIONS ACCORDING TO CODAP 95 RULES

This calculation verifies the mechanical resistance of the packaging to the test in section 629 of the IAEA rules <1> (see chapter 00-2 for <1’>).

This check is carried out by comparing the allowable pressure under normal operating conditions, as calculated using the CODAP <2> rules, with the test pressure.

2.1 – Calculation for the cylindrical containment enclosure subjected to internal pressure

The e/De ratio (thickness of the cylindrical enclosure on the external diameter of the containment) is equal to 20/1000 = 0.02 and is less than 0.16. This means that chapter C2.1 in <2> is applicable.

The allowable relative pressure in the cavity is deduced using the formula in C.2.1.4.1 general formula:

$$P = \frac{e f z}{R_i + 0{,}5e}$$

where:

. P: pressure within the cavity,

e: thickness of the cylindrical wall

. f f = nominal stress calculated for normal operating conditions.

The stress is defined in Table C.1.7.2 for austenitic stainless steels.

The criterion $f_1 = \max\left(\frac{\sigma_r}{3} \text{ is set; } \frac{\sigma_e}{1{,}5}\right)$

where:

. σ_r: ultimate strength of steel type A at 200°C = 360 MPa (see chapter O),
. σ_r: yield stress of steel type A at 200°C = 118 MPa (see chapter O),

therefore $f_1 = \max\left(\frac{360}{3}\ ;\ \frac{118}{1{,}5}\right)$ = 120 MPa.

. Z: Welding coefficient as defined in Table C.1.8.

Due to the nature of the elements transported, the construction category of the TN-MTR is B.

The coefficient, Z, is equal to 1 due to the severe production acceptance criteria .

. R: internal radius of the cylindrical enclosure = 480 mm

Thus, allowable pressures can be calculated according to CODAP 95 under normal operating conditions:

$$P = \frac{20\ x\ 120\ x\ 1}{480 + 0{,}5\,x\ 20} = 4.89 \text{ MPa or } 48.9 \text{ bars.}$$

2.2 – Calculation for the containment cylindrical enclosure subjected to internal pressure and other loads

This check is covered in Chapter C.2.4 in <2>.

Refer to section C.2.4.5 Calculation rule for cylindrical enclosures without torsion, case a): minimum thickness of a cylindrical enclosure without torsion.

$$e_{cyl} = \max \{e_C, e_1, e_2\}$$

where:

- e_{cyl}: cylinder thickness = 20 mm,
- e_C: minimum thickness calculated according to C.2.1.4, described in section 2.1 in this document

  e_C = 20 mm

- e_1: $\frac{1}{f}\left[\frac{PD_m}{4} + \frac{|F|}{\pi\, D_m} + \frac{4|M|}{\pi\, D_m^2}\right]$

  where:

  . f : calculated nominal stress already defined in section 2.1 in this document,
  f = 120 MPa

  . P: calculated pressure

  . D_m: average cylinder diameter D_m = 980 mm,

  . $|F|$: value of the force applied to the cylinder in addition to the internal pressure.

  It is assumed that 2/3 of the mass of the lid and the flange and half the mass of the shock absorbing cover is transmitted to the floor through this cylinder (the remainder of the mass being transmitted to the floor through the outer containment of the packaging body).

The result obtained using the masses given in chapter 4466-Z-0 is:

. F = mγ = $\left[\frac{2}{3}\left(2700+1500\right)+\frac{1}{2}\,1700\right]$ x 9.81 = 35 800 N.

. $|M|$: absolute value of the bending moment applied in a plane containing the enclosure centreline = 0.

$$P_1 = \left[f\, e_1 - \frac{|F|}{\pi\, D_m}\right]\frac{4}{D_m}$$

$$P = \left[120 \text{ x } 20 - \frac{35800}{\pi\, 980}\right]\frac{4}{980} = 9.75 \text{ MPa or } 97.5 \text{ bars.}$$

- $$e_2 = \sqrt{\frac{8\, D_m\left(-\frac{PD_m}{4} - \frac{F}{\pi\, D_m} + \frac{4\,|M|}{\pi\, D_m^2}\right)}{KE}}$$

Since the moment M is zero, the calculation for e_2 is null and void since the quantity under the radical is negative (section c) in C2.4.5 in <2>).

The allowable pressure under normal conditions is therefore P = min (P_e, P_1) = 48.9 bars.

2.3 Calculation of the welded circular flat bottom under pressure only under normal operating conditions (section C.3.2 in <2>)

The chosen calculation case most similar to the case of the TN-MTR is the flat bottom with discharge groove (section C 3.2.6 in <2>).

The required minimum thickness of the bottom is given by the relation:

$$e_{bottom} = \text{Max}\left\{\left(C_1\ \ D_i\ \sqrt{\frac{P}{f}}\right);\left(C_2\ D_i\ \sqrt{\frac{P}{f_{\min}}}\right)\right\} \qquad \text{C 3.2.5.a in <2>}$$

where f and f_{min}, nominal calculated stresses:

The steel used for the bottom is type B (see chapter O) with yield stress 320 MPa at 200°C and ultimate strength 580 MPa at 200°C.

$$f = \max\left\{\frac{580}{3}\,;\,\frac{320}{1{,}5}\right\} = 213 \text{ MPa.}$$

According to section 2, the nominal calculated stress f of the cylindrical enclosure is equal to 120 MPa,

$f_{min} = \min(f\,;\,f_v) = 120$ MPa.

therefore for this design case, we have:

f	f_v	f_{min}
213 MPa 2130 bars	120 MPa 1200 bars	120 MPa 1200 bars

C_1: given in graph C.3.2.4 in which the value used for f is f_{min} = 120 MPa,

e_v: thickness of the cylindrical enclosure = 20 mm

D: inside diameter of the enclosure = 960 mm

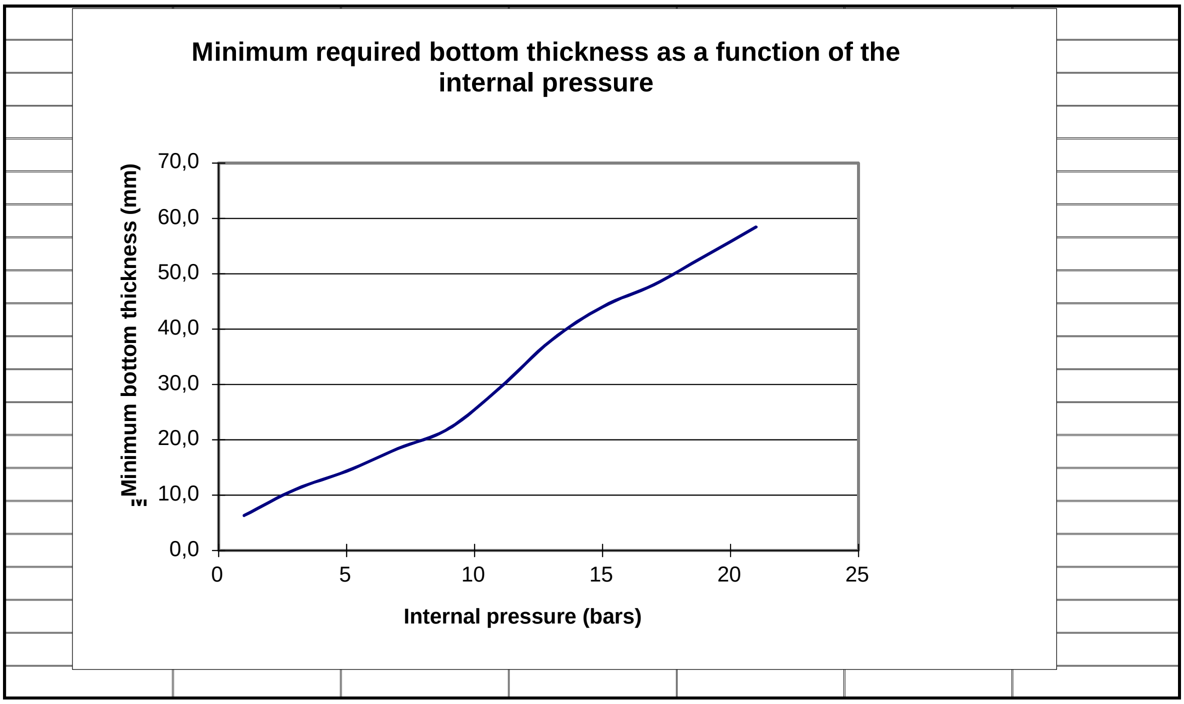
e_v/D_i = 20 / 960 = 0.021, reduced by 0.02.

The following table gives values of C_1 as a function of the internal pressure.

C_2: given by graph C.3.2.5, where f_{min} = 120 MPa, The following table gives the required minimum thickness under normal operating conditions, as a function of the internal pressure.

internal pressure (bars)	p/f_{min}	C_1	C_2	$C_1\ D_i \times (P/f)^{0.5}$ (mm)	$C2\ Di \times (P/f_{min})^{0.5}$ (mm)	minimum thickness of the bottom. (mm)
1	0.001	0.304	(1)	6.3	(1)	6.3
3	0.003	0.306	(1)	11.0	(1)	11.0
5	0.004	0.308	(1)	14.3	(1)	14.3
7	0.006	0.334	(1)	18.4	(1)	18.4
9	0.008	0.353	(1)	22.0	(1)	22.0
11	0.009	0.36	0.32	24.8	29.4	29.4
13	0.011	0.37	0.38	27.7	38.0	38.0
15	0.013	0.377	0.41	30.4	44.0	44.0
17	0.014	0.379	0.42	32.5	48.0	48.0
19	0.016	0.384	0.44	34.8	53.2	53.2
21	0.018	0.387	0.46	36.9	58.4	58.4

(1): not applicable, since C_2 is less than 0.30.



It is deduced that the maximum internal pressure under normal operating conditions and for a bottom thickness of 30 mm, is 11 bars.

2.4. Calculation for the bolted flat circular bottom subjected to internal pressure only

This calculation is described in chapter C.3.3 in CODAP 95 <2>.

The applicable section is C.3.3.4 - Calculation rules for bottoms with seals inside the bolt drilling circle.

The minimum thickness of the bottom is given by the relation:

$$e = \max \left\{ \left(e_A\right) ; \left(e_P\right) \right\}$$

in which:

+ $e_A = \sqrt{\dfrac{3\left(C - D_j\right)}{\pi\, D_j} \dfrac{F'_A}{f_A}}$

C: diameter of hole drilling circle = 1225 mm

D_j: diameter of the circle on which the seal is seated= 1130 mm,

F'_A: tensile force exerted on the bolts when the seal is seated, as defined in section C.6.1.4 in CODAP 95.

F'_A: $\dfrac{nS_r + S}{2} f_{b;A}$ (formula C 6.1.4 e)

where n: number of bolts = 36,

S_r: stressed area of an M30 bolt = 561 mm^2,

S: minimum required cross-section for all n bolts,

$S = \dfrac{F_A}{f_{b;A}}$.

F_A: minimum force to be applied by the n bolts in the situation in which seals are seated determined in section C.6.A.4

$F_A = \pi\, D_j\, Y_2$

Y_2 minimum linear force to be applied to the seal in the seal situation. According to section C6.A4, Y2 can be neglected for the layout of the seals corresponding to the case of the TN-MTR.

$Y_2 = 0$

F_A = 0 N

S = 0 mm²

$$F'_A = \frac{36 \times 561}{2} \text{ x } 178 = 1\ 797\ 450 \text{ N.}$$

f_A: nominal calculated stress in the bottom material in the seal seated situation. The flange fitted to the lid is made of type A steel (see chapter O).

Therefore we have f_A = 120 MPa (see section 2.1 in this document).

We therefore deduce:

$$e_A = \sqrt{\frac{3(1225 - 1130)}{\pi \ x\ 1130} \ x\ \frac{1797450}{120}} = 34.7 \text{ mm.}$$

This calculation shows that the thickness of the disk forming the bottom of the lid (301), which is equal to 65 mm, is sufficient because it is more than e_A = 34.7 mm.

$$+ \ e_p = \sqrt{\left[\frac{3(3+\upsilon)}{32} D_j^2 + 3\left(\frac{D_j}{4} + \frac{Y_m}{P}\right)(C - D_j)\right]\frac{P}{f}}$$

where Y_m is negligible according to section C6.A4 in <2>

$$P = \frac{ep^2 \text{ x } f}{\frac{3(3+\upsilon)}{32} D_j^2 + 3\left(\frac{D_j}{4}\right)(C - D_j)}$$

e_p: thickness of the plate of the containment lid = 65 mm.

f : nominal calculated stress for steel type B (see chapter O) = 213 MPa, according to section 2.3 in this note.

υ: Poisson's ratio for steel = 0.30 (see Table C1.6.6 in section C1.6.6 in the CODAP 95).

D_j: diameter of the circle on which the seal is seated= 1130 mm, C: diameter of hole drilling circle = 1225 mm

$$P = \frac{65^2 \times 213}{\frac{3(3+0{,}3)}{32}\,1130^2 + 3\left(\frac{1130}{4}\right)(1225-1130)}$$

P = 1.89 MPa or 18.9 bars.

+ The thickness of the region around the bottom is such that the maximum allowable pressure obtained (section C.3.3.4.b) is:

$$P = \frac{e_P^2 \; x \; f}{3\left(\frac{D_j}{4} + \frac{Y_m}{P}\right)\left(C - D_j\right)}$$ with the same notations as above.

$$P = \frac{65^2 \times 213}{3\left(\frac{1130}{4}\right)(1225-1130)}$$ = 11.2 MPa or 112 bars.

The allowable internal pressure in the cavity is determined taking account of the above two calculations, allowing for the thickness of the bottom of the lid, P = 18.9 bars.

2.5 – Calculation for the containment subjected to external pressure only

The calculation is presented in section C.4.1.5. The D_e/e ratio is $1000/20$ = 50 and is more than 10.

$$\frac{L}{D_e} = \frac{1080}{1000} = 1.08$$

$$\frac{D_e}{e} = 50$$

The coefficient A determined on the chart C.4.9.1 is A = 0.004. Coefficient B determined on chart C.4.9.2-13: austenitic stainless steels of grades type low carbon Cr-Ni and low carbon Cr-Ni with nitrogen for a temperature equal to 205°C.

B = 48.

The maximum allowable external pressure is:

$$P_a = \frac{4}{3} \frac{B}{D_e / e} K$$

where K = 1 for normal operating conditions,

$$P_a = \frac{4}{3} \, x \, \frac{48}{50} \, x \, 1 = 1{,}28 \; MPa \text{ or } 12.8 \text{ bars.}$$

2.6. CONCLUSION

The maximum allowable pressure for the containment vessel subjected to internal or external pressure loading, calculated using CODAP 95 rule criteria during normal operating conditions, is P = 11 bars.

The design basis element is the bottom of the packaging.

It is thus demonstrated that the TN-MTR packaging resists a pressure of more than 1.5 bars corresponding to the water immersion test.

The test pressure applied during acceptance of the packaging is fixed at 11 bars (see chapter 7A).

3. CALCULATION FOR THE CONTAINMENT VESSEL AT AN EXTERNAL PRESSURE OF 20 BARS.

This calculation corresponds to the test verification in section 630 of IAEA rules <1> (see chapter 00-2 for <1'>).

The stresses in disks and then the stresses in the containment shell are calculated by applying classical material strength formulas. These stresses are compared with 1.5 times the yield stress, which is justified by the accident-related nature of this test and by the requirement in section 550 in IAEA rules: if the water immersion test described in section 630 were applied to the package, the containment would not fail.

3.1 CALCULATIONS FOR BOTTOM AND LID DISKS

The calculations for the disks are made using the formula taken from <3>, corresponding to the calculations for the circular plates with a constant thickness, uniformly loaded across the whole plate and with the periphery fixed (case 10b, Table 24)

Maximum moment at the periphery of the plate: $M_{max} = -q\, a^2 / 8$

Maximum bending stress: $\sigma_{max} = - 6\ q\ a^2 / (8\ t^2)$

Maximum deflection at the centre of the plate: $y_c = - 12\ q\ a^4\ (1-\nu^2) / (64\ E\ t^3)$

Where:

q: pressure applied to the disk, q = 2 MPa

a: disk radius, variable depending on the disk being calculated

t: disk thickness, variable depending on the disk being calculated

ν: Poisson's ratio: $\nu = 0.3$

E: Modulus of elasticity of steel: E = 200,000 MPa

The allowable limit for stresses is calculated based on the criterion classically used for plasticity, i.e. 1.5 times the yield stress for bending stresses. This choice is justified by the fact that immersion is an accident case.

Calculation cases and results are shown in the table below:

Designation	Containment bottom (See Figure O.1)	Containment lid disk (see figure O.1).
Material	Type B steel	Type B steel
Yield stress (see table O.7)	320 MPa at 200°C	320 MPa at 200°C
Allowable limit for stresses (see Table O.7)	480 MPa	480 MPa
Disk radius (a)	480 mm	480 mm
Disk thickness (t)	30 mm	35 mm
Maximum stress within the disk	- 384 MPa	- 282 MPa
Maximum deflection on the disk	3.3 mm	2.1 mm

Stresses on the bottom internal disk and the lid external disk are less than the allowable limit and guarantee that the containment remains leak-tight.

3.2 CALCULATION FOR THE SHELLS

The calculations for the disks are made using the following formula taken from <3>, corresponding to the calculation for a thin wall circular shell with constant thickness, uniformly loaded over the entire surface, case 1c, Table 28, page 519;

Maximum tensile stress: $\sigma_{max} = q\ R\ /\ t$

Maximum displacement: $\Delta R = -\ q\ R^2\ (1 - \nu/2)\ /\ (E\ t)$

Where:

q: pressure applied to the disk, q = 2 MPa

t: radius of shell, t = 480 mm

t: thickness of shell, t = 20 mm

ν: Poisson's ratio: $\nu = 0.3$

E: Modulus of elasticity of steel = 200,000 MPa.

The allowable limit for the stress is the yield stress.

Calculation cases and results are shown in the table below:

Designation	Containment shell (See Figure O.1)
Material	type A steel
Yield stress (see Table O.7)	118 MPa at 200°C
Allowable limit for stresses (see Table O.7)	118 MPa
Inside radius of shell (R)	480 mm
Shell thickness (t)	20 mm
Maximum stress in the shell	48 MPa
Maximum displacement on the shell	0.1 mm

Calculated stresses are less than the allowable limit (yield stress) and leak-tightness of the containment is guaranteed.

3.3 CONCLUSION

Therefore the containment can resist an external pressure of 20 bars, corresponding to the water immersion test imposed for packages containing irradiated nuclear fuels.

4. CONCLUSION

The calculations given in this document show that the TN-MTR packaging will resist pressure tests applicable for a type B(U) packaging and for fissile material packages:

- 1. Water immersion test (<1> <1'>): resistance to an external gauge pressure of at least 150 kPa = 1.5 bars.

The allowable pressure under normal operating conditions calculated according to CODAP 95 rules <2>, is 11 bars. This pressure is used as the test pressure for the packaging.

- 2. Water Immersion test for packages containing irradiated nuclear fuels (<1> <1'>): resistance to an external gauge pressure of at least 2 MPa = 20 bars.

Stresses induced in the containment vessel by the application of this pressure to the containment are less than 1.5 times the yield stress.

5. REFERENCES

<1> IAEA Safety Collection No. 6 – Rules for the Transportation of Radioactive Materials – 1985 Edition (Amended 1990)

<1> IAEA Safety Standards Series No. TS-R-1 – Regulations for the Safe Transport of Radioactive Material – 1996 Edition (revised)

<2> CODAP 1995 edition (June 30, 1995).
Revision 96-12: January 1 1997.
SNCT - AFIAP.

<3> ROARK'S formulas for stress and strain 6th Edition - Warren C Young.