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Technical Memorandum

Internal Stresses in the R7021 Transport Container

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1. PURPOSE AND SCOPE

The R7021 is a Type B transport package designed to transport both Special Form and non-SF solid radioactive material. This document analyses the thermal and mechanical stresses and strains generated in the principal structural elements of the R7021 flask under various extremes of regulatory environmental conditions. Internal pressures and the resultant stresses are functions of the design and its heat generating contents. Environmental conditions include maximum ambient temperature, insolation, reduced ambient pressure and immersion. The resulting stresses are quantified and compared with the material design limits (certain stress combinations are considered for worst-case conditions). The risk of fatigue failure in the flask and closure fixings is also assessed.

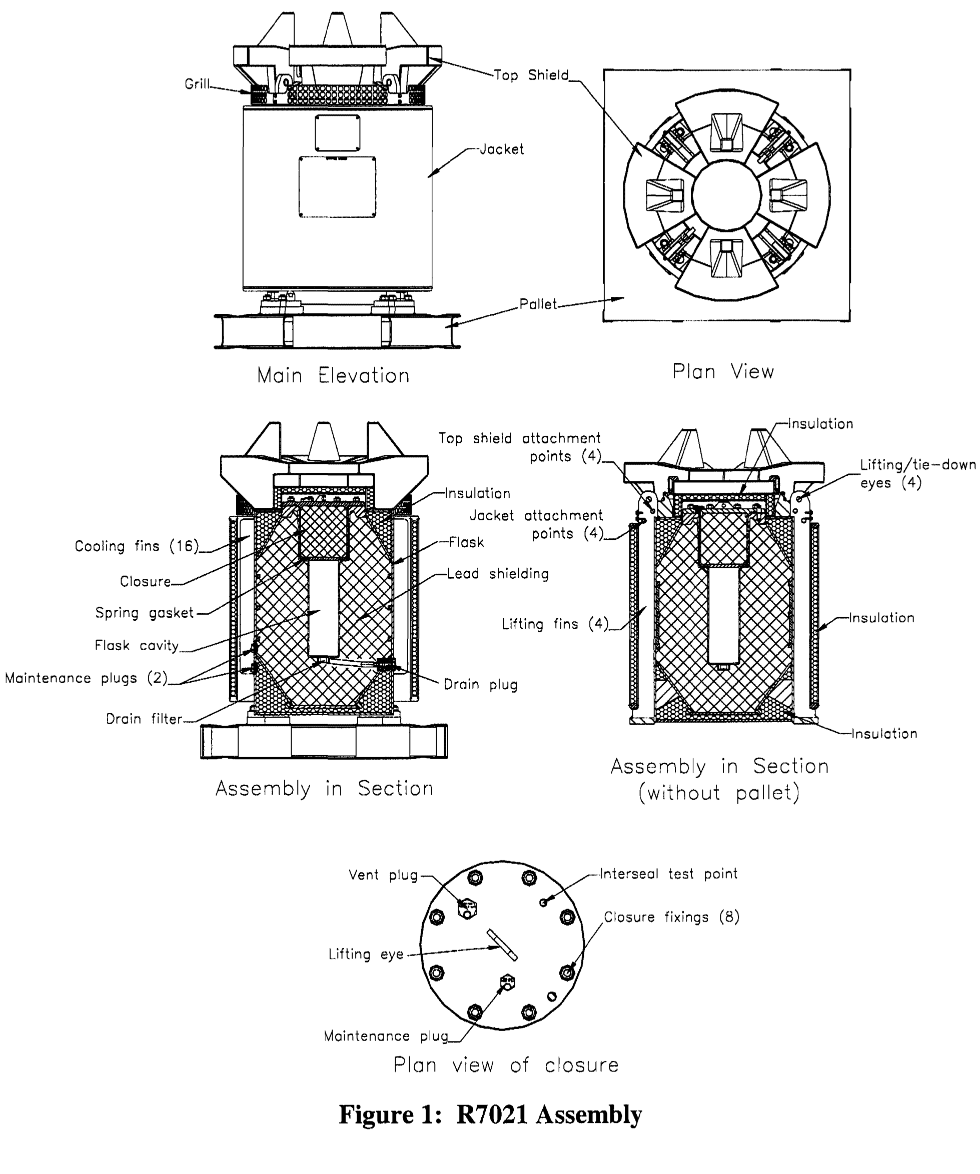

Figure 1: R7021 Assembly

2. DESCRIPTION

The design consists of a shielded, stainless steel flask mounted on a pallet and protected from heat and impact by a jacket and top shield (Figure 1). The flask is an upright, cylindrical fabrication closed with a removable shield plug, the closure, at the top. As it is designed to ship non-Special Form material the closure, vent and drain plugs are sealed with elastomer Orings and therefore the potential exists for the contents to heat the internal atmosphere and create a pressure differential to its environment.

3. Criteria

- Maximum Normal Operating Pressure (MNOP), i.e. at equilibrium loaded with 200 kCi (7.40 PBq) ^{60}Co in an ambient of 38°C with full insolation, shall not exceed 700 kPa gauge (para 662, TS-R-1).
- Stresses in the closure fixings or flask inner wall shall not exceed 10% of the design strength (yield) at the maximum normal conditions temperature as a result of:
  - internal pressure, or
  - a reduction in external pressure to 5 kPa (para 619, TS-R-1) or
  - the combination of the above.
- Stresses in the closure fixings or flask inner or outer walls resulting from internal pressures shall not exceed 10% of the design strength (yield) at the maximum accident conditions temperature.
- Compressive stresses in the flask outer wall shall not exceed the yield strength when a flask is immersed to a depth of 15m (para 729, TS-R-1).
- Stress levels shall be such that fatigue failure is not credible during the design life of 50 years.

4. Analysis

This analysis calculates the stresses generated in the R7021 flask and closure fixings under a variety of regulatory conditions. It also examines the likelihood of fatigue failure from thermal cycling or repeated tightening during its design life.

4.1 Assumptions

- Gas temperature within the containment system is taken to be the capsule temperature, i.e. the pressure in the cavity is the same as in the capsule.
- Gas temperature within the flask shielding volume is taken to be the cavity wall temperature.
- The flask is assumed closed at normal room temperature, though in practice this would be impossible to achieve given the significant time necessary to load the flask, fit the closure and purge the interior.
- Fixings strength at elevated temperature is reduced in the same proportion as the material into which they are screwed as that is the weaker of the two.
- The load on the closure fixings exerted by cavity pressure will be counteracted by the weight of the closure. The analysis will ignore this effect and consider the closure weightless.
- The pressure in the shielding space will counteract the pressure in the cavity. The analysis will ignore this effect.
- At 15m immersion depth the external pressure will be 0.150 N/mm^2 and the flask is assumed to be at the water temperature, i.e. with no internal pressure to counteract the external pressure.
- Stresses in the vent and drain plugs from the pressure differentials are ignored due to the very small area encompassed within the O-ring.
- For simplicity, the yield strength in compression is taken to be the same as in tension.

4.2 DATA

4.2.1 Temperature Maxima (RTM 120)

Component	Prior To Transport	Normal Conditions (MNOP)	Accident Conditions
Capsules	409	411	471
Closure fixings	141	150	270
Cavity wall	201	205	316
Flask wall	149	153	287

4.2.2 Design Stresses

These are taken from the pressure vessel standard, PD 5500:

Maximum Design Stress (N/mm^2)						
Temperature (°C)	20	141-150	153	201-205	270-287	316
304S11 *	200	155	141	133	126	120
Closure studs **	600	466	-	-	379	-

* Yield strength data is taken from BS EN 10088-2 for 1.4307 (304L) plate and reduced, by proportion, using the reduction in design strength cited in PD 5500 for a similar grade steel (304-S11).

** A4-80, BS EN ISO 3506-1, reduced as for 304S11.

4.2.3 Internal Pressures

- Prior to shipment: The gas inside the flask cavity expands as it is heated and exerts a pressure on the underside of the closure. According to the gas laws the flask cavity pressure, Pcv_1, is:

$$Pcv_1 = \left(\left(\frac{273 + Tc_1}{273 + Ta} \right) - 1 \right) \text{x Pa}$$

where

Tc_1 = capsule temperature prior to shipment = 409°C

Pa = atmospheric pressure at time of closing = 0.101 MPa

Ta = ambient temperature at time of closing = 20°C

thus

$$Pcv_1 = \left(\left(\frac{273 + 409}{273 + 20} \right) - 1 \right) \text{x } 0.101 = 0.134 \text{ MPa (gauge).}$$

The gas inside the shielding volume also expands as it is heated and exerts a pressure on the flask outer wall. The shielding pressure, Ps_1, therefore is:

$$Ps_1 = \left(\left(\frac{273+Tcv_1}{273+Ta}\right)-1\right) x\ Pa$$

where
Tcv_1 = cavity wall temperature prior to shipment = 201°C
thus

$$Ps_1 = \left(\left(\frac{273+201}{273+20}\right)-1\right) x\ 0.101 = 0.062\ N/mm^2 \equiv 0.062\ MPa\ (gauge).$$

- MNOP (as above but including insolation):

$$Pcv_2 = \left(\left(\frac{273+Tc_2}{273+Ta}\right)-1\right) x\ Pa$$

where
Tc_2 = capsule temperature = 411°C
thus

$$Pcv_2 = \left(\left(\frac{273+411}{273+20}\right)-1\right) x\ 0.101 = 0.135\ MPa\ (gauge).$$

and

$$Ps_2 = \left(\left(\frac{273+Tcv_2}{273+Ta}\right)-1\right) x\ Pa$$

where
Tcv_2 = cavity wall temperature = 205°C
thus

$$Ps_2 = \left(\left(\frac{273+205}{273+20}\right)-1\right) x\ 0.101 = 0.064\ N/mm^2 \equiv 0.064\ MPa\ (gauge).$$

- Accident conditions:

$$Pcv_3 = \left(\left(\frac{273+Tc_3}{273+Ta}\right)-1\right) x\ Pa$$

where
Tc_3 = capsule temperature = 471°C
thus

$$Pcv_3 = \left(\left(\frac{273+471}{273+20}\right)-1\right) x\ 0.101 = 0.155\ MPa\ (gauge).$$

and

$$Ps_3 = \left(\left(\frac{273+Tcv_3}{273+Ta}\right)-1\right) x \; Pa$$

where

Tcv_3 = cavity wall temperature = 316°C

thus

$$Ps_3 = \left(\left(\frac{273+316}{273+20}\right)-1\right) x \; 0.101 = 0.102 \text{ N/mm}^2 \equiv 0.102 \text{ MPa (gauge)}.$$

Internal Pressures Summary (MPa)			
Component	Prior To Transport	Normal Conditions (MNOP)	Accident Conditions
Cavity (Pcv)	0.134	0.135	0.155
Shielding space (Ps)	0.062	0.064	0.102

4.3 STRESS CALCULATIONS

4.3.1 Internal Pressure

- Closure Fixings Tensile Stress (Sf_1)
  The weight of the closure, which would normally counteract any pressure in the cavity, is ignored here.

$$Sf_1 = \frac{D^2.Pcv}{N.d^2}$$

where

D = O-ring internal diameter = 279 mm

Pcv = cavity pressure (see 4.2.3).

N = number of fixings = 8

d = fixings effective tensile diameter = 17.7 mm (M20, BS 3643)

Environment	Prior To Transport	Normal Conditions (MNOP)	Accident Conditions
Pressure, Pcv (MPa)	0.134	0.135	0.155
Fixings Stress, Sf_1 (N/mm^2)	4.16	4.19	4.81

- Cavity Wall Hoop Stress (Scv_h)
  The pressure in the shielding space, which would normally counteract any pressure in the cavity, is ignored here.

$$Scv_h = \frac{Pcv.R}{t}$$ (Table 13.1, Case No 1c, Roark)

where

R = internal wall radius = 75 mm

t = wall thickness = 6.2 mm
thus

Environment	Prior To Transport	Normal Conditions (MNOP)	Accident Conditions
Pressure, Pcv (MPa)	0.134	0.135	0.155
Hoop Stress, Scv_h (N/mm^2)	1.62	1.63	1.88

- Cavity Wall Axial Stress (Scv_a)

$$Scv_a = \frac{Pcv.R}{2t}$$ (Table 13.1, Case No 1c, Roark)

thus

Environment	Prior To Transport	Normal Conditions (MNOP)	Accident Conditions
Pressure, Pcv (MPa)	0.134	0.135	0.155
Axial Stress, Scv_a (N/mm^2)	0.810	0.817	0.938

- Outer Wall Hoop Stress

R = mean radius = 352 mm
t = wall thickness = 10 mm

thus the hoop stress, Ss_{h1}, is as follows:

Environment	Prior To Transport	Normal Conditions (MNOP)	Accident Conditions
Pressure, Ps (MPa)	0.062	0.064	0.102
Hoop Stress, Ss_{h1} (N/mm^2)	2.18	2.25	3.59

- Outer Wall Axial Stress

The axial stress, Ss_{a1}, is as follows:

Environment	Prior To Transport	Normal Conditions (MNOP)	Accident Conditions
Pressure, Ps (MPa)	0.062	0.064	0.102
Axial Stress, Ss_{a1} (N/mm^2)	1.09	1.13	1.80

4.3.2 Reduced External Pressure

- Closure Fixings Tensile Stress (Sf_2)

$$Sf_2 = \frac{D^2.p}{N.d^2}$$

where
p = pressure differential (95 kPa) = 0.095 N/mm^2

thus

$$Sf_2 = \frac{279^2 \times 0.095}{8 \times 17.7^2} = 2.95 \text{ N/mm}^2$$

- Outer Wall Hoop Stress, Ss_{h2}:

$$Ss_{h2} = \frac{p.R}{t} \qquad \text{(Table 13.1, Case No 1c, Roark)}$$

where

R = mean wall radius = 352 mm

t = wall thickness = 10 mm

thus

$$Ss_{h2} = \frac{0.095 \times 352}{10} = 3.34 \text{ N/mm}^2$$

- Outer Wall Axial Stress

$$Ss_{a2} = \frac{p.R}{2\,t}$$

thus

$$Ss_{a2} = \frac{0.095 \times 352}{2 \times 10} = 1.67 \text{ N/mm}^2$$

Note: there are no stresses in the containment boundary because the flask wall is leak-tight. See OP381 for leak-testing requirements.

4.3.3 15m Immersion

- Flask Wall Hoop Stress

$$Ss_{h3} = \frac{p.R}{t} \qquad \text{(Table 13.1, Case No 1c, Roark)}$$

where

p = pressure = -0.150 N/mm^2 (external)

R = mean wall radius = 352 mm

t = wall thickness = 10 mm

thus

$$Ss_{h3} = \frac{-0.150 \times 352}{10} = -5.28 \text{ N/mm}^2 \text{ (compressive)}$$

- Flask Wall Axial Stress

$$Ss_{a3} = \frac{p.R}{2\,t}$$

thus

$$Ss_{a3} = \frac{-0.150 \times 352}{2 \times 10} = -2.64 \text{ N/mm}^2 \text{ (compressive)}$$

Note: there are no stresses in the containment boundary because the flask wall is leak-tight. See OP381 for leak-testing requirements.

4.4 RESULTS SUMMARY

4.4.1 Prior to Transport

	Stress (N/mm^2)				
Location	Closure Fixings	Cavity Wall		Outer Wall	
Stress Type	Tensile	Hoop	Axial	Hoop	Axial
Internal pressure	4.16	1.62	0.810	2.18	1.09
Design Stress	466	133		155	
Proportion (%)	0.893	1.22	0.609	1.41	0.703

4.4.2 Normal Conditions

- Individual load conditions

	Stress (N/mm^2)				
Location	Closure Fixings	Cavity Wall		Outer Wall	
Internal pressure	4.19	1.63	0.817	2.25	1.13
5 kPa pressure	2.95	-	-	3.34	1.67
Maximum	4.19	1.63	0.817	3.34	1.67
Design Stress	466	133		141	
Proportion (%)	0.899	1.23	0.614	2.37	1.18

- Load combination

	Stress (N/mm^2)				
Location	Closure Fixings	Cavity Wall		Outer Wall	
Internal pressure	4.19	1.63	0.817	2.25	1.13
5 kPa pressure	2.95	-	-	3.34	1.67
Total	7.14	1.63	0.817	5.59	2.80
Design Stress	466	133		141	
Proportion (%)	1.53	1.23	0.614	3.96	1.99

4.4.3 Accident Conditions

	Stress (N/mm^2)				
Location	Closure Fixings	Cavity Wall		Outer Wall	
Internal pressure	4.81	1.88	0.938	3.59	1.80
Design Stress	379	120		126	
Proportion (%)	1.27	1.57	0.782	2.85	1.43
15m immersion	-	-	-	-5.28	-2.64
Design Stress	-	-		-126	
Proportion (%)	-	-	-	4.19	2.10
Maximum (%)	1.27	1.57	0.782	4.19	2.10

5. FATIGUE

5.1 THERMAL FATIGUE

Thermal stresses are determined by the temperature difference and the coefficient of thermal expansion. The temperature difference is determined by the heat flux, the conductivity of the material and its thickness. Heat flow is predominantly in the radial direction and therefore the highest temperature differences are across the inner and outer flask walls.

5.1.1 Thermal stress in outer flask wall

For thin-walled cylinders (inner radius/wall thickness > 10) with a temperature difference across the wall:

$$\text{Max. stress} = \frac{\Delta T.\gamma.E}{2(1-\nu)} \quad \text{(Roark, p762)}$$

where:

γ = coefficient of thermal expansion = 8.55×10^{-6} °F^{-1} (Table TE-1, ASME II, Part D) = 1.54×10^{-5} °C^{-1}

E = Young's modulus = 200 GPa = 200×10^3 N/mm^2 (PD 5500, Table 3.6-3)

ΔT = temperature difference

ν = Poisson's ratio = 0.285

From Heat Transfer:

$$q = -\frac{kA}{\Delta x}(T_2-T_1)$$

where:

q = heat = 3,074W (RTM 120)

k = thermal conductivity = 9.4 Btu/h.ft.°F = 16.3 W/m.°C (Machinery's Handbook, p378, S30400)

A = surface area of wall = $\pi \times d \times l = \pi \times 0.693 \times 1.02 = 2.22$ m^2

Δx = thickness of wall = 0.010 m

T_2-T_1 = temperature difference across wall = ΔT

rearranging gives:

$$\Delta T = \frac{q\Delta x}{kA} = \frac{3{,}074 \times 0.010}{16.3 \times 2.22} = 0.849°C$$

therefore:

$$\text{Max. stress} = \frac{0.849 \times 1.54 \times 10^{-5} \times 200 \times 10^3}{2(1-0.285)} = 1.83 \text{ N/mm}^2$$

5.1.2 Thermal stress in cavity wall

As before, assuming all the heat flows through the cavity wall (worst case):

$$\Delta T = \frac{q\Delta x}{kA}$$

where:

A = surface area of wall = $\pi \times 0.150 \times 0.476 = 0.224$ m^2

Δx = thickness of wall = 0.0062 m

therefore:

$$\Delta T = \frac{3{,}074 \times 0.0062}{16.3 \times 0.224} = 5.22^{\circ}C$$

therefore:

$$\text{Max. stress} = \frac{4.18 \times 1.54 \times 10^{-5} \times 200 \times 10^{3}}{2(1 - 0.285)} = 11.2 \text{ N/mm}^2$$

5.1.3 Thermal fatigue

An R7021 is unlikely to be used more than twelve times in a year, which represents a maximum of twenty-four heating and cooling cycles. With a nominal design life of fifty years the flask will be subject to a maximum of 1,200 thermal cycles. Using equation C-5 in PD 5500, Annex C, paragraph 3.1.2, the stress range for 1,200 cycles is 509 N/mm^2. It is evident therefore that the flask is not at risk from thermal fatigue failure during the design life.

5.2 FASTENERS

The key R7021 fixings are those retaining the closure to the flask (M20, st/st), the jacket and top shield to the flask (M16, c/st) and the flask to the pallet (M24, c/st). As the safe fatigue life is determined by the tensile stress level it can be seen that, as all the fixings are tightened to the same torque (OP 381), the smallest fixings (M20, st/st and M16, c/st) will have the highest tensile stress. The stress, S_t, from the preload is obtained from Machinery's Handbook, p178, as follows:

$$F = Q \times \frac{p + 6.2832\mu r}{6.2832r - \mu p} \times \frac{r}{R}$$

rearranging gives:

$$Q = \frac{F.R}{r} \times \frac{6.2832r - \mu p}{p + 6.2832\mu r}$$

where:

Q = load (kgf)

$F.R$ = torque (kgf.m)

r = pitch radius of screw (m)

p = thread pitch (m)

μ = coefficient of friction

5.2.1 Stainless Steel Fasteners

F.R = 15 kgf.m (OP 381)
r = 0.0092 m (BS 3643)
p = 0.0025 m (BS 3643)
μ = 0.16 (for lubricated threads, Machinery's Handbook, p173)

thus:

$$Q = \frac{15}{0.0092} \times \frac{6.2832 \times 0.0092 - 0.16 \times 0.0025}{0.0025 + 6.2832 \times 0.16 \times 0.0092} = 7{,}970 \text{ kg} = 78{,}200 \text{ N}$$

The tensile stress area of an M20 thread is 245 mm^2 (BS 3643). The tensile stress, S_t, in the bolt is therefore:

$$S_t = \frac{Q}{A} = \frac{78{,}200}{245} = 319 \text{ N/mm}^2$$

Section C.3.1.3 of Appendix C in PD 5500 uses the fatigue design curve of Fig. C4 to determine the number of cycles to failure for a given stress range in bolting materials where,

$$S_r = \frac{S_t \times E \times n}{2.09 \times 10^5}$$

Using the default fatigue strength reduction factor, n, of 4 (para. C.3.3.4, PD 5500) and a Young's modulus, E, of 200×10^3 N/mm^2 (PD 5500, Table 3.6-3) the stress range S_r is given by:

$$S_r = \frac{319 \times 200 \times 10^3 \times 4}{2.09 \times 10^5} = 1{,}220 \text{ N/mm}^2$$

Stainless steel is an inherently ductile material. Therefore, in the stainless steel studs, any localised stress concentration (in this case in the thread roots) exceeding yield will therefore deform plastically until the stress is reduced to approximately the yield value (p28.10, Standard Handbook of Machine Design). From the fatigue design curve, the maximum allowable number of operating cycles for a stress range of yield (600 N/mm^2, section 4.2.2) is not less than 3,000 cycles.

5.2.2 Carbon Steel Fasteners

5.2.2.1 M16 Shoulder Bolts

F.R = 6 kgf.m (OP 381)
r = 0.0074 m (BS 3643)
p = 0.0020 m (BS 3643)
μ = 0.16 (for lubricated threads, Machinery's Handbook, p173)

thus:

$$Q = \frac{6}{0.0074} \times \frac{6.2832 \times 0.0074 - 0.16 \times 0.0020}{0.0020 + 6.2832 \times 0.16 \times 0.0074} = 3{,}970 \text{ kg} = 38{,}900 \text{ N}$$

The tensile stress area of an M16 thread is 157 mm^2 (BS 3643). The tensile stress, S_t, in the bolt is therefore:

$$S_t = \frac{Q}{A} = \frac{38{,}900}{157} = 248 \text{ N/mm}^2$$

Section C.3.1.3 of Appendix C in PD 5500 uses the fatigue design curve of Fig. C4 to determine the number of cycles to failure for a given stress range in bolting materials where,

$$S_r = \frac{S_t \text{ x E x n}}{2.09 \text{ x } 10^5}$$

Using the default fatigue strength reduction factor, n, of 4 (para. C.3.3.4, PD 5500) and a Young's modulus, E, of 209×10^3 N/mm^2 (PD 5500, Table 3.6-3) the stress range S_r is given by:

$$S_r = \frac{248 \text{ x } 209 \text{ x } 10^3 \text{ x } 4}{2.09 \text{ x } 10^5} = 992 \text{ N/mm}^2$$

From the fatigue design curve, the maximum allowable number of operating cycles for a stress range of 992 N/mm^2 (section 4.2.2) is not less than 1,500 cycles.

5.2.2.2 M24 Studs

F.R	= 15 kgf.m (OP 381)
r	= 0.0110 m (BS 3643)
p	= 0.0030 m (BS 3643)
μ	= 0.16 (for lubricated threads, Machinery's Handbook, p173)

thus:

$$Q = \frac{15}{0.0110} \text{ x } \frac{6.2832 \text{ x } 0.0110 - 0.16 \text{ x } 0.0030}{0.0030 + 6.2832 \text{ x } 0.16 \text{ x } 0.0110} = 6{,}660 \text{ kg} = 65{,}300 \text{ N}$$

The tensile stress area of an M24 thread is 353 mm^2 (BS 3643). The tensile stress, S_t, in the bolt is therefore:

$$S_t = \frac{Q}{A} = \frac{65{,}300}{353} = 185 \text{ N/mm}^2$$

Section C.3.1.3 of Appendix C in PD 5500 uses the fatigue design curve of Fig. C4 to determine the number of cycles to failure for a given stress range in bolting materials where,

$$S_r = \frac{S_t \text{ x E x n}}{2.09 \text{ x } 10^5}$$

Using the default fatigue strength reduction factor, n, of 4 (para. C.3.3.4, PD 5500) and a Young's modulus, E, of 209×10^3 N/mm^2 (PD 5500, Table 3.6-3) the stress range S_r is given by:

$$S_r = \frac{185 \text{ x } 209 \text{ x } 10^3 \text{ x } 4}{2.09 \text{ x } 10^5} = 740 \text{ N/mm}^2$$

From the fatigue design curve, the maximum allowable number of operating cycles for a stress range of 740 N/mm^2 (section 4.2.2) is not less than 2,500 cycles.

5.2.3 Fatigue

An R7021 is unlikely to be used more than twelve times in a year, which represents twenty-four tightening operations. With a nominal design life of fifty years the fasteners will be subject to a maximum of 1,200 tightening cycles. It is evident that the fasteners are not at risk from fatigue failure during the design life.

6. CONCLUSIONS

- Maximum Normal Operating Pressure (MNOP) does not exceed 700 kPa (gauge).
- Stresses in the R7021 flask cavity and outer walls and closure fixings do not exceed 10% of the design stress at MNOP, as a result of:
    - internal pressure, or
    - a reduction in external pressure to 5 kPa, or
    - the combination of the above.
- Stresses in the R7021 flask cavity and outer walls and closure fixings as a result of internal pressure do not exceed 10% of the design stress under accident conditions of transport.
- Stresses from immersion to a depth of 15m do not exceed the levels above.
- No component is at risk of fatigue failure during the design life.

7. REFERENCES

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