TECHNICAL REPORT

BALLTEC LTD.

FINITE ELEMENT ANAYLSIS AND FATIGUE ASSESSMENT OF 11300 kN ANCHOR CONNECTOR

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DET NORSKE VERITAS

TECHNICAL REPORT

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Summary:

Det Norske Veritas, Structural Integrity and Laboratories, has performed independent finite element analysis and evaluated the fatigue strength of a 11300 kN Balltec anchor connector.

The report includes:

- Finite element analyses of the anchor connector in order to determine the stresses as a function of applied load
- Calculation of fatigue strength based on stress and a given load spectrum

The conclusion is that the fatigue strength of the anchor connector is satisfactory and meets the requirements of relevant standards.

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Indexing terms

FEA

Fatigue

Anchor connector

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1 CONCLUSIVE SUMMARY

Det Norske Veritas, Structural Integrity and Laboratories, has performed independent finite element analysis and evaluated the fatigue strength of a 11300 kN Balltec anchor connector.

The report includes:

- Finite element analyses of the anchor connector in order to determine the stresses as a function of applied load
- Calculation of fatigue strength based on stress and a given load spectrum

The geometry of the models are based on the following drawings:

Balltec drawing no. 525123 rev. A, Female Receptacle, 20/09/2004

Balltec drawing no. 525127 rev. A, Mandrel, 20/09/2004

Balltec drawing no. 525126 rev. A, Mandrel Pin, 20/09/2004

Balltec drawing no. 525124 rev. A, Receptaclel Pin, 20/09/2004

Balltec drawing no. 525125 rev. A, Clevis Nut, 20/09/2004

It is concluded that the design fatigue capacity of mandrel and receptacle is acceptable to meets the requirements given in DNV Offshore Standard DNV-OS-E301 Position Mooring and API recommended practice API RP 2SK (ref. /1/ and /2/).

To qualify the Balltec mooring anchor according to the DNV Rules, the requirements given in "DNV Certification Note 2.6, Certification of offshore mooring chain" (ref. /5/) are governing. In case the mooring anchor connector is used for applications/areas with a more severe load spectrum than applied in this report, additional fatigue assessments are required.

2 FINITE ELEMENT ANALYSIS

2.1 Introduction

Finite element analyses (FEA) were carried out to determine the stresses in the anchor connector for a static load of 10 000 kN. The stresses are assumed to vary linearly within the mooring line load.

2.2 Finite Element model

Geometry

The diameters of the "elliptical" loading eye is 130 mm x 160 mm. The pin is represented by an "elliptical" 128 mm x 158 mm diameter steel bolt. The geometry of the mandrel and receptacle is shown in Figure 2-1 and Figure 2-2.

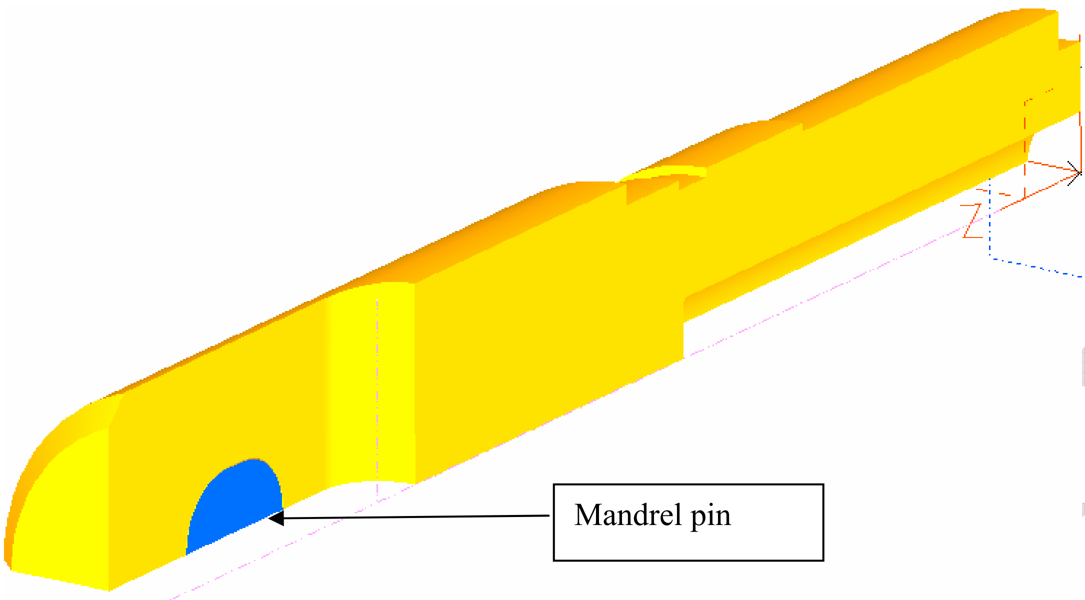

Figure 2-1 Geometry of a quarter model and boundary conditions of the mandrel.

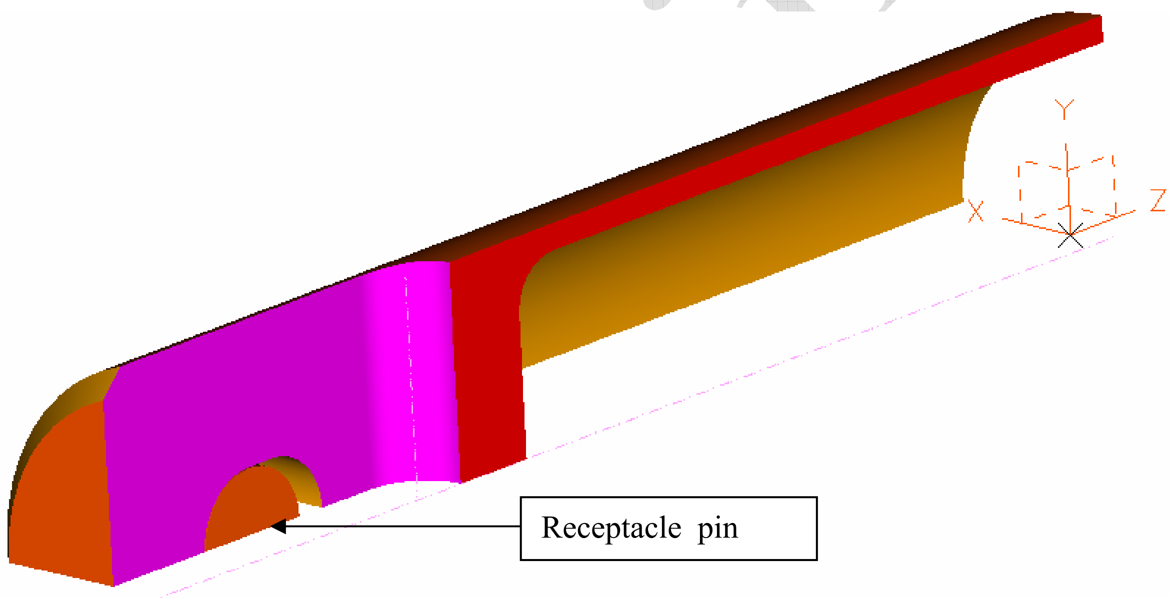

Figure 2-2 Geometry of a quarter model of the receptacle.

The interference between the loading eye and the pin surface was represented by contact elements. The initial gap is 1 mm between the loading eye and the pin. The mandrel and the receptacle can move 1 mm before contact is established against the pin, see Figure 2-3.

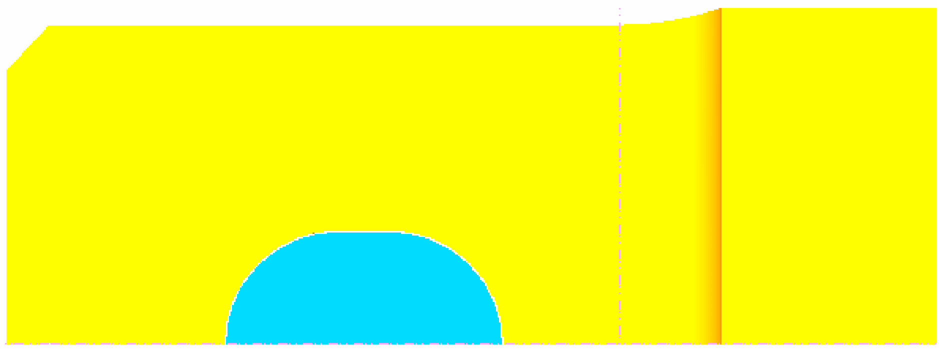

Figure 2-3. Initial 1 mm gap between pin and mandrel/receptacle.

Software

The finite element model, analysis and post processing is performed by the software I-DEAS version 10 NX.

Boundary conditions

The boundary conditions were specified as shown in Figure 2-4:

- The model has two symmetry planes and a quarter FE model is sufficient to represent the complete mandrel and receptacle
- The shackle pin is locked in position by its end-surface. The interference between the shackle pin and the loading eye is modelled as a contact surface with zero friction.

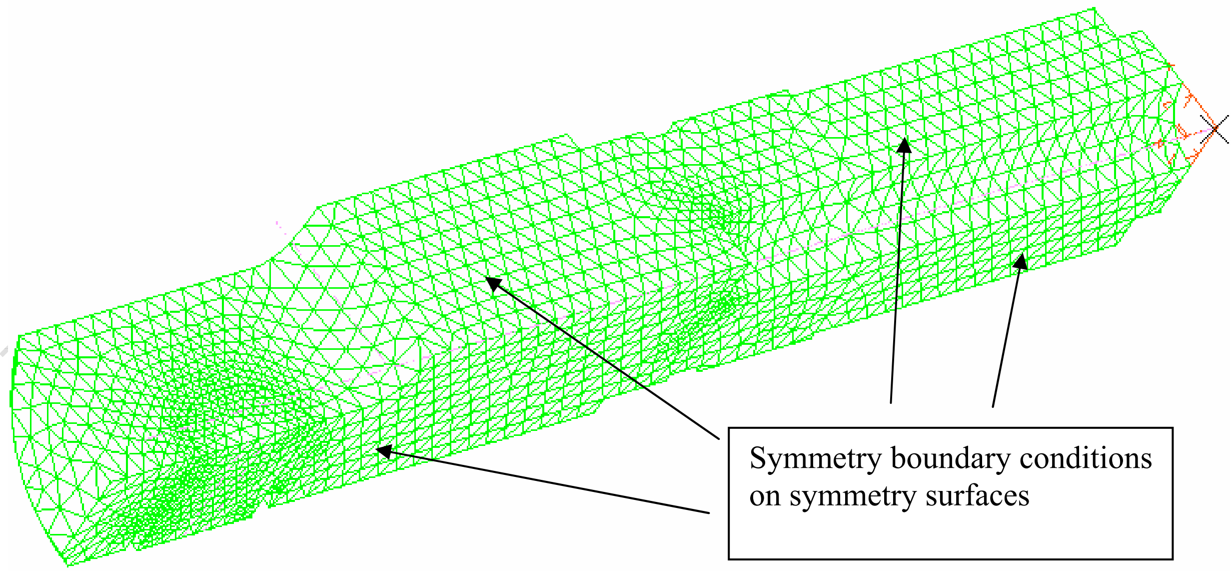

Figure 2-4 FE-model, showing mesh and boundary condition for the mandrel.

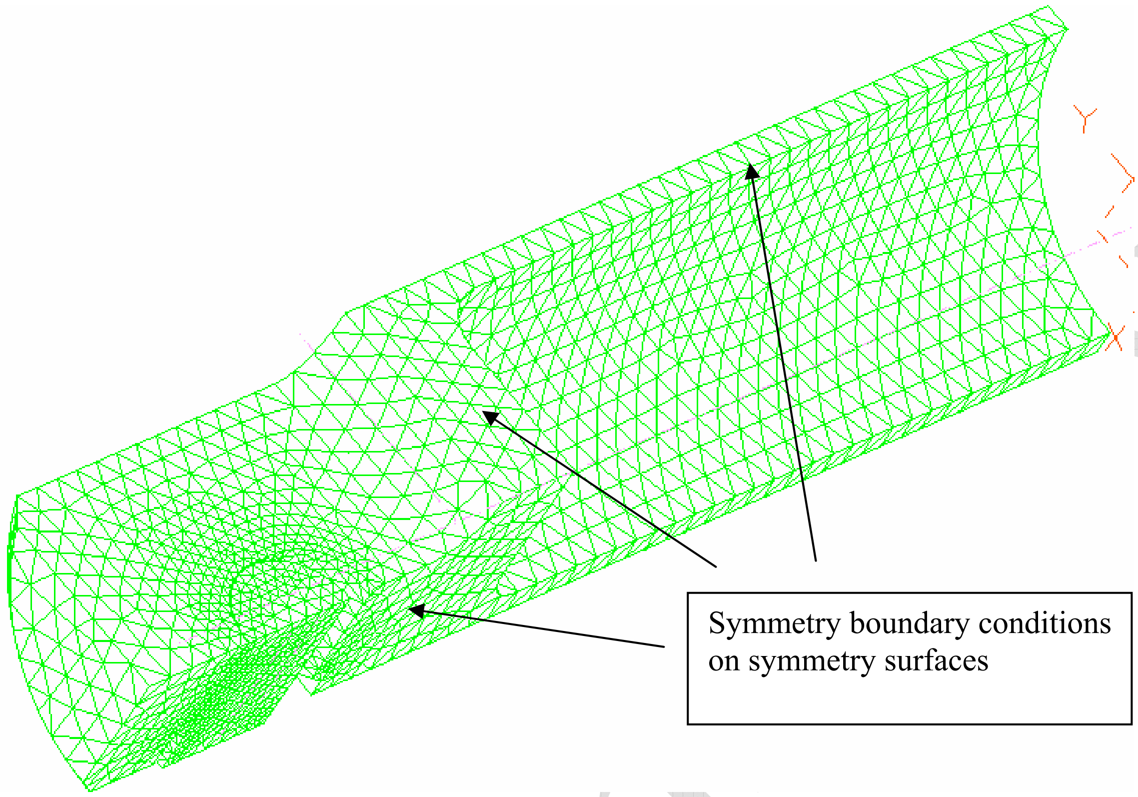

Figure 2-5 FE-model, showing mesh and boundary condition for the receptacle.

Loading

A load of 2 500 kN is applied on the quarter model. A linear relation between the applied force and stresses in the anchor connector is assumed. The only non-linearity of the assembly is the contact surfaces between the pin and loading eye. This is concluded to have negligible impact on the linearity of ratio between the considered load and stress in critical locations.

The ball locking device presses the balls outwards, giving a radial force component.

The radial force on the inside of the receptacle is calculated to be 3 times the axial load, ref. /3/. The radial force is applied as internal pressure on the receptacle and external pressure on the mandrel.

$F_{rad\ full\ model}$ = 3 x 10000 kN = 30 000 kN radial force for the full model

and for the quarter model

$F_{rad\ quarter\ model}$ = 30 000 kN / 4 = 7 500 kN.

Mesh

A 10-node quadratic tetrahedron element, featuring three active degrees of freedom per node – U1, U2 and U3) was used for the model. The element topology is shown in Figure 2-6.

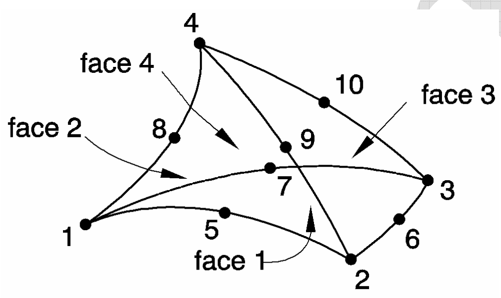

Figure 2-6 Element topology of the 10 node quadratic tetrahedron solid element used

Material

Linear elastic material was applied:

Type: ISOTROPIC

Youngs modulus: E = 207 GPa

Poisson ratio: ν = 0.29

2.3 FEA results for receptacle

The finite elemen has been carried out to determine the maximum principal stress per unit load. Figure 2-7 shows the principal stress plot for the receptacle. The figure also shows the stress in the fillet of the inside of the receptacle. The following is observed from the FEA:

- The hotspot is located at the shackle pin hole of the loading eye.
- The maximum hotspot stress calculated is 700 MPa for a load of 10 000 kN.

The calculated stress per unit load is 700 MPa / 10 000 kN = **0.070 MPa/kN**

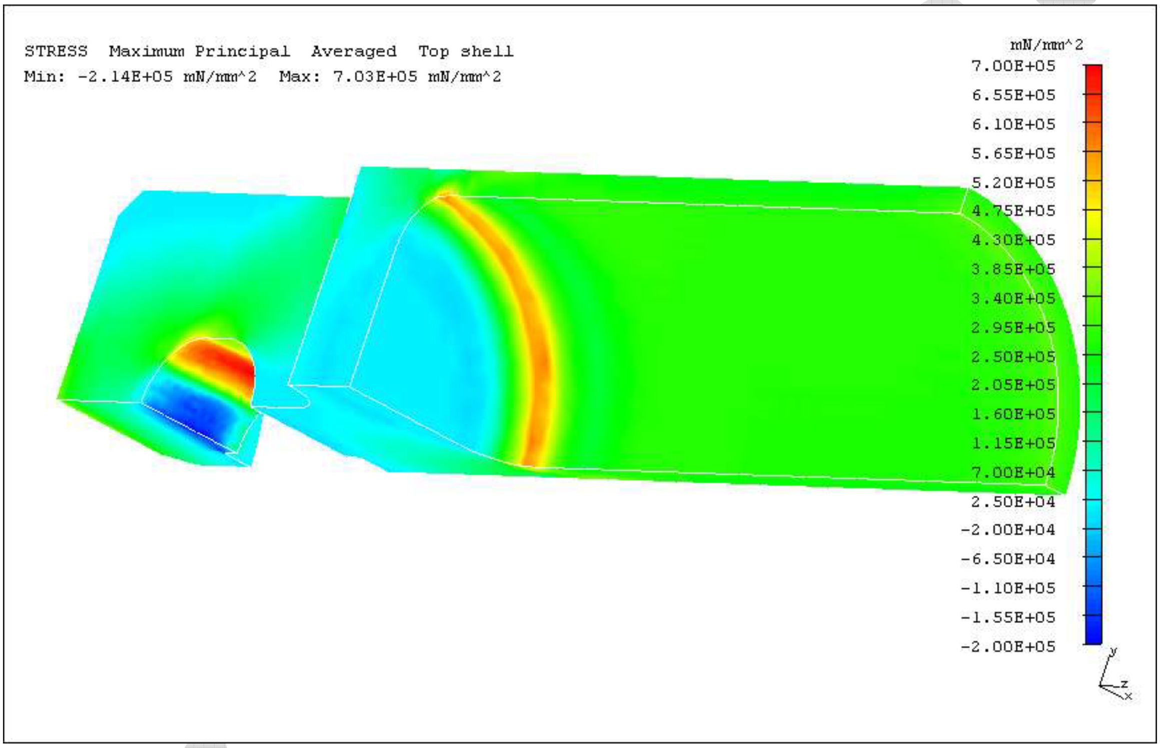

Figure 2-7. Plot of principal stresses in the receptacle. Max stress in loading eye is 700 MPa. Max stress in the fillet of the inner-most cavity of the receptacle is 580 MPa and therefore not calculated for fatigue life.

2.4 FEA results for mandrel

Figure 2-8 shows the calculated principal stresses in the mandrel. Maximum principal stress for the mandrel is 850 MPa when loaded with 10 000 kN.
The calculated stress per unit load is 850 MPa / 10000 kN = **0.085 MPa / kN.**

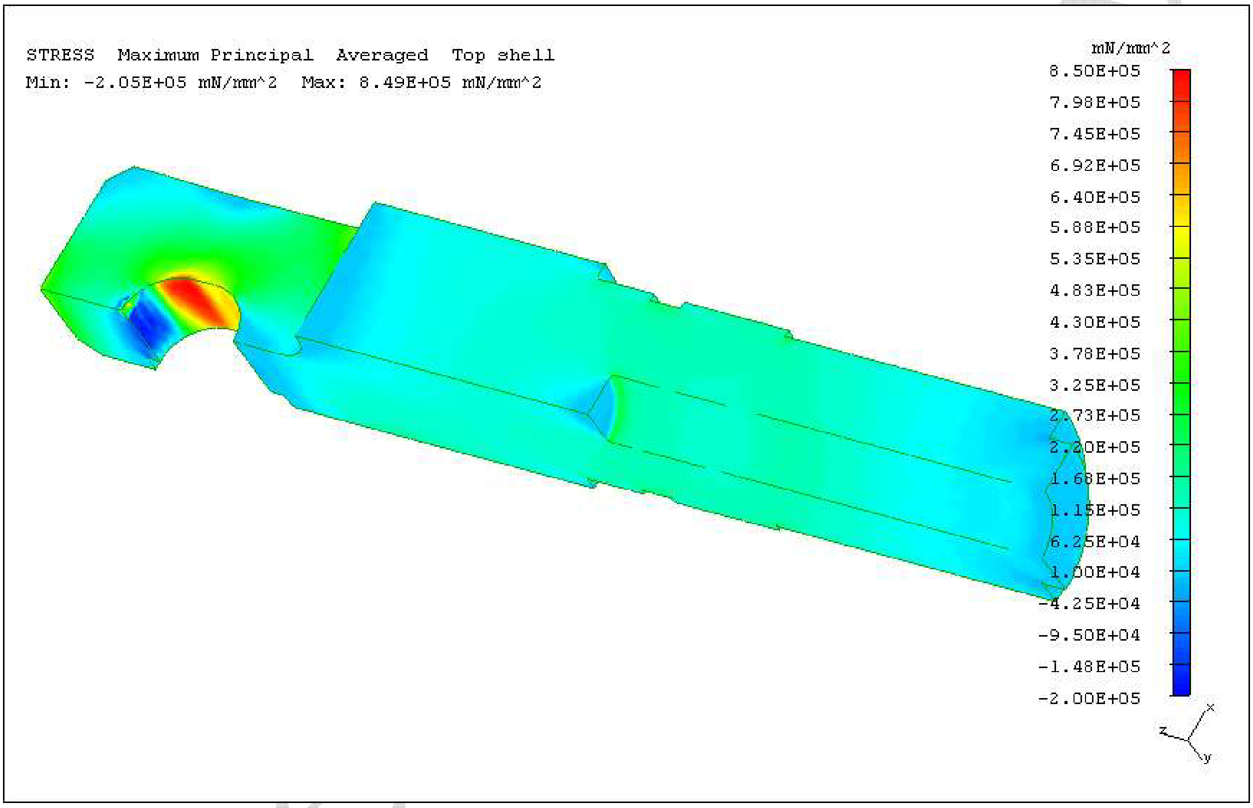

Figure 2-8. Plot of principal stresses in the mandrel.

3 FATIGUE STRENGTH

The fatigue strength evaluation is based on DNV Recommended practice RP-C203 "Fatigue Strength Analysis of Offshore Steel Structures" (Ref. /4/).

The fatigue strength is determined through the following main steps:

1. Determine the appropriate fatigue curve (i.e. S-N curve). The fatigue design S-N curves are based on the Stress range (i.e. maximum principal/stress range) versus Number of fatigue load cycles to failure.
   Ref. /4/: For members that can acquire stress concentrations due to rust pitting etc. curve C is required for machined surfaces. Further, it is assumed that the considered component is not covered by cathodic protection.

2. The load spectrum is given below for a typical North Sea Environment, scaled for an MBL of 11300 kN.

ID	Static tens	Probability	ow frequency		Wave frequency	
			Up cross p	RMS	Up cross p	RMS
	kN		s	kN	s	kN
1.0	1183.6	0.08095	232.3	2.2	9	2.9
2.0	1223.7	0.14957	232.1	8.7	9	6.2
3.0	1217.2	0.14957	232.1	7.4	9.9	7.1
4.0	1287.5	0.10996	232.2	19.4	9	10.2
5.0	1262.9	0.10996	232.2	14.5	10.4	11.8
6.0	1371.5	0.09004	232.3	34.0	9.2	15.3
7.0	1320.9	0.09004	232.2	23.8	10.6	17.4
8.0	1438.8	0.10996	232.4	47.9	10.1	23.6
9.0	1511.8	0.05498	232.8	59.7	10.7	30.2
10.0	1606.7	0.02684	232.2	74.3	11	41.2
11.0	1727.3	0.01385	232.2	91.2	11.3	52.5
12.0	1848.2	0.00909	230	126.1	11.7	72.7
13.0	1990.7	0.00303	227.7	141.5	12	91.9
14.0	2127.9	0.00130	224.4	168.2	12.4	113.6
15.0	2266.4	0.00043	220.4	198.0	2.8	153.8
16.0	2402.3	0.00043	216.8	210.2	13.2	183.2

Nomenclature

Bin: The total load spectrum is divided into 24 blocks (i.e. bin 1 – 24), where each bin represents one combination of period and load amplitude

Static tension: Static mooring line tension representative for each ID, i.e. the total mooring line tension equals "Static tension" +/- Load amplitude (denoted RMS in Appendix A)

Probability: Contribution ratio of each ID, i.e. the sum of probabilities for the 24 ID's is 1.0. The number of load cycles per ID is hence: "No. of seconds in one year" / "Period" multiplied by "Probability"

Frequency: The ship motion frequency is separated into “Low”- and Wave-frequency.

RMS: “Root Mean Square” load amplitude from the static tension. The fatigue load range is hence 2 x RMS [kN].

Combined

Spectrum: The total contribution of the ship motion from both the "Low" -and Wave-frequency.

Fatigue damage calculation:

The fatigue damage from each block of the load spectrum has been calculated in accordance with ref. /2/. The fatigue design S-N curve with thickness correction is expressed as:

$$\text{Log } N = \log a - 2s - m \log [\Delta\sigma \, (t/t_{ref})^k]$$

Where:

N	=	predicted number of cycles to failure for stress range $\Delta\sigma$
$\Delta\sigma$	=	stress range
m	=	negative inverse slope of S-N curve
a	=	constant relating to mean S-N curve
s	=	standard deviation of log N
t	=	thickness through which a crack will most likely grow.
t_{ref}	=	reference thickness.

3.1 Accumulated damage of the mandrel

The stress in the loading eye of the mandrel is 850 MPa.

For current case with the C-curve in sea water, without cathodic protection, we have

$\Delta\sigma$	= 850 MPa / (10 000) = 0.085 MPa/kN
m	= 3.0
log a – 2s	= 12.115
k	= 0.15
t	= 127.5 mm
t ref	= 25 mm

Appendix A gives the calculated fatigue strength of the considered loading eye, where

n_i = Number of load cycles per ID = 365 x 24 x 3600 / “period”

$\Delta\sigma_i$ = 0.085 x 2 x RMS (i.e. subscript i denotes each ID-number)

$N_i = 10$^[12.115 – 3 x log ($\Delta\sigma$ x $(127.5/25)^{0.15}$)]

Damage = D = n_i / N_i

Total Damage per line = $\sum (D_{dir})$ (i.e. subscript dir denotes each direction at one line)

The results are summarised in Table 3-1 below:

Table 3-1 Calculated damage

Low frequency				Wave frequency				Tot. Damage
ni [cy]	d sig [Mpa]	Ni [cy]	Damage	ni [cy]	d sig [Mpa]	Ni [cy]	Damage	
10989	0.4	1.2E+13	9.2E-10	283649	0.5	5.0E+12	5.7E-08	5.749E-08
20322	1.5	2.0E+11	1.0E-07	524093	1.1	5.3E+11	9.8E-07	1.087E-06
20322	1.3	3.1E+11	6.5E-08	476448	1.2	3.6E+11	1.3E-06	1.406E-06
14934	3.3	1.7E+10	8.6E-07	385300	1.7	1.2E+11	3.2E-06	4.077E-06
14934	2.5	4.1E+10	3.6E-07	333433	2.0	7.8E+10	4.3E-06	4.615E-06
12223	5.8	3.2E+09	3.8E-06	308641	2.6	3.6E+10	8.6E-06	1.239E-05
12229	4.0	9.5E+09	1.3E-06	267877	3.0	2.4E+10	1.1E-05	1.235E-05
14921	8.1	1.2E+09	1.3E-05	343336	4.0	9.7E+09	3.5E-05	4.825E-05
7448	10.2	6.0E+08	1.2E-05	162042	5.1	4.7E+09	3.5E-05	4.723E-05
3645	12.6	3.1E+08	1.2E-05	76948	7.0	1.8E+09	4.2E-05	5.381E-05
1881	15.5	1.7E+08	1.1E-05	38653	8.9	8.8E+08	4.4E-05	5.509E-05
1246	21.4	6.4E+07	2.0E-05	24501	12.4	3.3E+08	7.4E-05	9.339E-05
420	24.1	4.5E+07	9.3E-06	7963	15.6	1.6E+08	4.8E-05	5.778E-05
183	28.6	2.7E+07	6.8E-06	3306	19.3	8.7E+07	3.8E-05	4.477E-05
62	33.7	1.6E+07	3.7E-06	4843	26.1	3.5E+07	1.4E-04	1.417E-04
63	35.7	1.4E+07	4.6E-06	1027	31.1	2.1E+07	4.9E-05	5.404E-05
								6.321E-04

According to ref. /4/, a safety factor of 10 is required for permanent mooring components not inspected during design life of 25 years, hence;

D = 6.33e-4*10*25 = 0.158 < 1

It is concluded that the fatigue strength of the mandrel is acceptable and is found to meet the requirements given in ref. /3/ and ref. /4/.

3.2 Accumulated damage of the receptacle

The stress in the loading eye of the mandrel is 700 MPa.

For current case with the C-curve in sea water, without cathodic protection, we have

$\Delta\sigma$ = 700 MPa / (10 000) = 0.07 MPa/kN

m = 3.0

log a – 2s = 12.115

k = 0.15

t = 135 mm

t ref = 25 mm

The calculated fatigue strength of the considered loading eye, where

n_i = Number of load cycles per ID = 365 x 24 x 3600 / "period"

$\Delta\sigma_i$ = 0.07 x 2 x RMS (i.e. subscript i denotes each ID-number)

$Ni = 10\text{^}[12.115 - 3 \times \log(\Delta\sigma \times (135/25)^{0.15})]$

Damage = $D = n_i / N_i$

Total Damage per line = $\sum (D_{dir})$ (i.e. subscript dir denotes each direction at one line)

The result are summarised in Table 3-2 below:

Table 3-2 Calculated damage

Low frequency				Wave frequency				Tot. Damage
ni [cy]	d sig [Mpa]	Ni [cy]	Damage	ni [cy]	d sig [Mpa]	Ni [cy]	Damage	
10989	0.3	2.1E+13	5.3E-10	283649	0.4	8.7E+12	3.2E-08	3.300E-08
20322	1.2	3.4E+11	5.9E-08	524093	0.9	9.3E+11	5.6E-07	6.239E-07
20322	1.0	5.4E+11	3.8E-08	476448	1.0	6.2E+11	7.7E-07	8.074E-07
14934	2.7	3.0E+10	4.9E-07	385300	1.4	2.1E+11	1.8E-06	2.341E-06
14934	2.0	7.2E+10	2.1E-07	333433	1.6	1.4E+11	2.4E-06	2.650E-06
12223	4.8	5.7E+09	2.2E-06	308641	2.1	6.2E+10	5.0E-06	7.111E-06
12229	3.3	1.7E+10	7.4E-07	267877	2.4	4.2E+10	6.4E-06	7.091E-06
14921	6.7	2.0E+09	7.4E-06	343336	3.3	1.7E+10	2.0E-05	2.770E-05
7448	8.4	1.0E+09	7.1E-06	162042	4.2	8.1E+09	2.0E-05	2.712E-05
3645	10.4	5.4E+08	6.7E-06	76948	5.8	3.2E+09	2.4E-05	3.089E-05
1881	12.8	2.9E+08	6.4E-06	38653	7.4	1.5E+09	2.5E-05	3.163E-05
1246	17.7	1.1E+08	1.1E-05	24501	10.2	5.8E+08	4.2E-05	5.362E-05
420	19.8	7.8E+07	5.4E-06	7963	12.9	2.9E+08	2.8E-05	3.317E-05
183	23.6	4.7E+07	3.9E-06	3306	15.9	1.5E+08	2.2E-05	2.570E-05
62	27.7	2.9E+07	2.1E-06	4843	21.5	6.1E+07	7.9E-05	8.136E-05
63	29.4	2.4E+07	2.6E-06	1027	25.6	3.6E+07	2.8E-05	3.102E-05
								3.629E-04

According to ref. /4/, a safety factor of 10 is required for permanent mooring components not inspected during design life of 25 years, hence;

$D = 3.629e\text{-}4 * 10 * 25 = 0.091 < 1$

It is concluded that the fatigue strength of the receptacle is acceptable and is found to meet the requirements given in ref. /1/ and ref. /2/.

4 REFERENCES

/1/ DNV Offshore Standard DNV-OS-E301 Position Mooring June 2001

/2/ American Petroleum Institute Recommended Practice 2SK, December 1996 Design and Analysis of stationkeeping Systems for Floating Structures

/3/ Balltec Calculation Sheet 11300 kN MBL BREM Mooring Connector Indentation Calc, doc no. DaliaCalc001, rev A. 27/08/2004

/4/ DNV Recommended practice RP-C203 “Fatigue Strength Analysis of Offshore Steel Structures”, 2000

/5/ DNV Certification Note 2.6, Certification of offshore mooring chain, August 1995

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