Study & Analysis of Transportation Skid

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Abstract—Transportation skid plays very important role in various industries. Offshore skids play a vital role in transportation of heavy pumps, engines and blender units used during manufacturing treatments at the well site. For universal acceptance and usage of these skids worldwide, the offshore design should meet various applicable codes and regulations, such as Bureau Veritas, Lloyd’s, ABS, or Det Norske Veritas (DNV) design standards. The designing of skid plays important role to ensure its use for offshore work. The stress analysis of skid is one of the key factor which gives idea about its sustainability to the desired load.

Index Terms—Introduction, Theoretical calculation, conclusion, Future scope, references.

I. INTRODUCTION

Transportation skid plays very important role in various industries. Offshore skids play a vital role in transportation of heavy pumps, engines and blender units used during manufacturing treatments at the well site. For universal acceptance and usage of these skids worldwide, the offshore design should meet various applicable codes and regulations, such as Bureau Veritas, Lloyd’s, ABS, or Det Norske Veritas (DNV) design standards. The designing of skid plays important role to ensure its use for offshore work. The stress analysis of skid is one of the key factor which gives idea about its sustainability to the desired load.

DNV is an autonomous and independent foundation created in 1864 in Norway. Its main objective is to safeguard life, property, and the environment both on and offshore. This involves the establishment of rules and guidelines regarding classification, quality assurance, and certification of sea-going vessels, structures, and other installations. Like other standards, DNV certification implies that a structure or an item of equipment has been reviewed against a certain set of requirements and furthermore that a document has been issued stating that the item is in compliance with the requirement. DNV certified skids are designed as structural frames that provide good continuity under different loading and lifting conditions. All primary structural members of a skid should qualify the criteria of allowable stresses and member deflection as per DNV design guidelines.

The challenges are geometry of skid assembly is complex, the location of CG is not symmetric. The skid designed to sustained load of 12 tonnes & the acceptance criteria for the design is as per the international standard DNV 2.7-3.

RTS SKID III
THEORETICAL ANALYSIS OF EXISTING 12 TONNE SKID (RTS-III) AS PER DNV 2.7.3

DESIGN LOAD CALCULATION ACCORDING TO DNV 2.7.3

RTS-III skid classified as:
PO Unit type: Class A
Risk level: High
Operational class: R45

ACCORDING TO DNV 2.7.3, SEC. 3.5 DESIGN LOADS - LIFTING

### Design Factor (DF) calculation

<table>
<thead>
<tr>
<th>Operational Class</th>
<th>MGW&lt; 50 tonnes</th>
<th>MGW ≥ 50 tonnes</th>
</tr>
</thead>
<tbody>
<tr>
<td>R60</td>
<td>1.4 + 0.8 x ( \sqrt{50/\text{MGW}} )</td>
<td>2.2</td>
</tr>
<tr>
<td>R45</td>
<td>1.4 + 0.6 x ( \sqrt{50/\text{MGW}} )</td>
<td>2.0</td>
</tr>
<tr>
<td>R30</td>
<td>1.4 + 0.4 x ( \sqrt{50/\text{MGW}} )</td>
<td>1.8</td>
</tr>
</tbody>
</table>

According to DNV 2.7-3 clause number 3.2.1 only the primary structure shall be included in the design calculations. Strength of frame members may be calculated using manual calculation & finite element Analysis.

Design criteria: Stress In the members shall not exceed than that \( \sigma \)

Allowable stress (\( \sigma_e \)) = 0.85 x \( \sigma_y \)

Whereas,
\( \sigma_y \) = Yield strength of material
MGW = Maximum gross weight of RTS-III i.e. 12 tonne.

### MATERIAL USED FOR PRIMARY STRUCTURAL ELEMENTS:

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield Strength in Mpa (( \sigma_y ))</th>
<th>Material assigned to part</th>
</tr>
</thead>
<tbody>
<tr>
<td>Norsok M120, Y05</td>
<td>355</td>
<td>Pivot, Link arm, Diagonal beam, Lower beam, Top beam</td>
</tr>
<tr>
<td>S165 M</td>
<td>620</td>
<td>Bolts</td>
</tr>
<tr>
<td>Norsok M120, Y30</td>
<td>420</td>
<td>Padeye, Hinge</td>
</tr>
</tbody>
</table>

### ALLOWABLE LOAD (\( \sigma_e \)) CALCULATION TABLE:

<table>
<thead>
<tr>
<th>Material assigned to part</th>
<th>Yield strength (( \sigma_y ))</th>
<th>Allowable strength (( \sigma ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pivot, Link arm, Diagonal beam, Lower beam, Top beam</td>
<td>355</td>
<td>301.75</td>
</tr>
<tr>
<td>Bolts</td>
<td>620</td>
<td>527</td>
</tr>
<tr>
<td>Padeye, Hinge</td>
<td>420</td>
<td>357</td>
</tr>
</tbody>
</table>

AS PER DNV 2.7-3 CLAUSE 3.5 THE DESIGN LOAD (F) ON THE PRIMARY STRUCTURE SHALL BE TAKEN AS:

\[ F= DF \times \text{MGW} \times g \]

Where \( DF = 1.4 + 0.6 \times \sqrt{50/\text{MGW}} \)

\[ = 2.6247 \]

So, \( F = 2.6247 \times 12000 \times 9.81 = 308979.68 \text{ N} \)
THEORETICAL CALCULATION OF THE PRIMARY STRUCTURAL ELEMENT

A. Top Padeye

For pad-eyes, as per DNV 2.7-3 Appendix APadeye Calculations. Following

BEARING PRESSURE

\[ \sigma_b = 0.045 \times \frac{\sqrt{RSF \times E}}{Dh \times t} \]

where, \( \sigma_e \) Allowable stress of padeye material in MPa, = 357 MPa
E : Elastic modulus = 210 000 MPa
Dh : Diameter of pinhole (mm) = 43.5 mm
t : Total thickness of padeye at hole including cheek plates (mm) = 50 mm

RSF CALCULATION

It is explained in DNV clause 3.5.4. The in plane design load for a lifting point is equal to the resultant sling force (RSF) on the padeye. In our case single lifting point is used.

So,

\[ RSF = 1.4 \times F \]

RSF Padeye in line design load. = 407853.18 N
Therefore:

\[
\sigma_b = 0.045 \sqrt{\frac{407853.18 \times 210000}{43.50 \times 50}}
\]

\[
\sigma_b = 282.38 \text{ MPa}
\]

\[\sigma \gg \sigma_b \text{ (Bearing Pressure)} \rightarrow \text{Design is safe}\]

**Tear Out**

A tear out check is normally considered sufficient to check the padeye material above (i.e. in the load direction) the hole. The following criterion shall be fulfilled:

\[
\sigma_t = \frac{\text{RSF}}{(R_{pad} - R_{h}) \times t}
\]

where, \(\sigma_e\): Allowable stress of padeye material in MPa.

\(DH\): Diameter of pinhole (mm) = 43.5 mm

\(t\): Total thickness of padeye at hole including cheek plates (mm)

\(\text{RSF}\): Padeye in line design load = 407853.18 N

\(R_{pad}\): Radius of padeye, taken as: \(R_{pad} = 75\) mm

\[\sigma_t \text{ (tear out)} = 153.2 \text{ Mpa}\]

\[\sigma_t \gg \sigma_e \text{ (tear out)} \rightarrow \text{Design is safe}\]
**B. Hinge - Top Hole**

![Diagram of hinge-top hole]

**Bearing Pressure**

\[
\sigma_b = 0.045 \sqrt{\frac{RSF \times E}{Dh \times t}}
\]

where, \( \sigma_e \) Allowable stress of padeye material in MPa, \( = 357 \) MPa

- **E**: Elastic modulus = 210 000 MPa
- **Dpin**: Diameter of shackle pin (mm) = 48 mm
- **Dh**: Diameter of pinhole (mm) = 50 mm
- **t**: Total thick. of padeye at hole (mm) = 60 mm
- **Rh**: \( \frac{Dh}{2} \)

**RSF Calculation**

It is explained in DNV clause 3.5.4. The in plane design load for a lifting point is equal to the resultant sling force (RSF) on the padeye. In our case single lifting point is used.

So, 

\[
RSF = 1.4 \times \frac{F}{2} \quad \text{----------- (F= Design load)}
\]

RSF Padeye in line design load. = 203926.59 N

Therefore,

Bearing pressure \( \sigma_b \) will be,

\[
\sigma_b = 170 \text{ MPa}
\]
σ >>σb (Bearing Pressure)--------Design is safe

TEAR OUT

A tear out check is normally considered sufficient to check the padeye material above (i.e. in the load direction) the hole. The following criterion shall be fulfilled:

\[ \sigma_t = \frac{\text{RSF}}{(R_{\text{pad}} - R_h) \times t} \]

where, \( \sigma_t \) Allowable stress of padeye material in MPa,
\( DH \) : Diameter of pinhole (mm) = 50 mm
\( t \) : Total thickness of padeye at hole including cheek plates (mm) = 60

RSF Padeye in line design load. = 203926.59 N
Rpad Radius of padeye, taken as: Rpad = 75 mm

\[ \sigma_t \text{ (tear out)} = 67.97 \text{ Mpa} \]

\[ \sigma >> \sigma_t \text{ (tear out)} \]--------Design is safe

C. LINK ARM CALCULATIONS ACCORDING TO DNV 2.7-3

\[ \sigma_b = 0.045 \times \sqrt{\frac{\text{RSF} \times E \times D_h \times t}{D_h}} \]

where, \( \sigma_b \) Allowable stress of padeye material in MPa, = 301.75 MPa
E : Elastic modulus = 210000 MPa
Dpin : Diameter of shackle pin (mm) = 55 mm
Dh : Diameter of pinhole (mm) = 57 mm
\( t \) : Total thick. of padeye at hole including cheek plates (mm) = 35 mm
Rh : Dh/2

RSF Padeye in line design load. = 203926.59

RSF CALCULATION

It is explained in DNV clause 3.5.4. The in plane design load for a lifting point is equal to the resultant sling force (RSF) on the padeye. In our case single lifting point is used.

So,
\[ \text{RSF} = 1.4 \times F/2 \] (F= Design load)
RSF Padeye in line design load. = 203926.59 N

$\sigma_b = 208.5 \text{ Mpa}$

$\sigma_b >> \sigma \text{ (Bearing Pressure)} \text{-------------Design is safe}$

**Tear Out**

A tear out check is normally considered sufficient to check the padeye material above (i.e. in the load direction) the hole. The following criterion shall be fulfilled:

$$\sigma_t = \frac{\text{RSF}}{(R_{pad} - R_h) \times t}$$

where, $\sigma_t$ Allowable stress of padeye material in MPa, 
$DH$ : Diameter of pinhole (mm) = 57 mm
$t$ : Total thick of padeye at hole including cheek plates (mm) = 35 mm

RSF Padeye in line design load.

$R_{pad}$ Radius of padeye, taken as: $R_{pad} = 67.5 \text{ mm}$

$\sigma_t (\text{tear out}) = 149.4 \text{ Mpa}$

$\sigma_t >> \sigma_a \text{ (tear out)} \text{-------------Design is safe}$

**C. Top Beam**

MGW = 12000 Kg
$\sigma_y = 355 \text{ Mpa}$
$b = 280$
$h = 270$
$h_1 = 244 \text{ mm}$
$tw = 13 \text{ mm}$
$g = 9.81$

Design force $(F) = 2.5 \text{ MGW} \times g = 294.300 \text{ KN}$
Length of beam = \( L = 2959 \)

Peak Moment = \( M_{\text{max}} = \frac{F \times L}{4} = \frac{0.294 \times 2959}{4} = 217.7 \text{ KN-m} \)

\[
I(\text{total}) = \frac{bh^3 - h_1^3(b-tw)}{12}
\]

\[= 12.99 \times 10^7 \text{ mm}^4\]

\( e_{\text{max}} = \frac{b}{2} = 140 \text{ mm} \)

Section Modulus (W) = \( \frac{I(\text{total})}{e_{\text{max}}} = 9.28544 \times 10^5 \text{ mm}^3 \)

Bending Stress = \( \sigma_b = \frac{M_{\text{max}}}{W} = 234.46 \text{ Mpa} \)

Maximum Shear force \( F\tau = F/2 = 147 \text{ KN} \)

Shear Stress \( \tau = \frac{S \cdot \overline{A Y}}{1b} = 66.231 \text{ Mpa} \)

\( A \bar{Y} = 1.17 \times 10^5 \text{ mm}^2 \)

Von Mises Stress \( \sigma_{vm} = \sqrt{\sigma_b^2 + 3\tau^2} \)

\[= 261.02 \text{ Mpa} \]

Accept criteria \( \sigma_{vm} < 0.85 \sigma_y \)

\(0.85 \sigma_y = 301.75 \text{ Mpa} \)

**D. Pivot Bolts**

![Diagram of pivot bolt system]

MGW = 12000Kg

\( \sigma_y = 355\text{Mpa} \)

Number of bolts (Nb) = 2

Diameter \( D = 55 \text{ mm} \)

\[\text{Area} \quad A = \frac{\pi \times D^2}{4} = 1810 \text{ mm}^2\]
Design Force = \( \frac{2.5 \times \text{MGW} \times g}{\text{nb}} \) = 0.147 MN

Shear Stress = \( \frac{F}{A} \) = 62 Mpa

\( \tau \ll \sigma \) Design is safe

**3D Modeling of Transportation Skid**

**Conclusion:**

As per the theoretical calculation the skid is meeting all design requirements. All primary structural elements are well within the allowable stress limit.

**Future Scope**

Further to this study FEA analysis of all primary structural elements could be carried out to validate the theoretical results.

**References**

[1]. Atul B. Bokane1, Micah Stewart, Nitinkumar P. Katke1, and Siddharth Jain3A Ramchandra, B kandagaly paper presents design, analysis, and field test results for an offshore container, such as a skid, as per DNV regulations.

[2]. Design Implementation of Offshore Skid in Compliance with DNV Regulations presented by Atul B. Bokane1, Micah Stewart, Nitinkumar P. Katke1, and Siddharth Jain3

[3]. DNV 2.7-1 Standard “Standard For Certification, Offshore Containers,”
[4]. DNV 2.7-3 STANDARD (2011) “PORTABLE OFFSHORE UNITS”.

[5]. Pandhare A. P., Chaskar S. T., Patil J. N., Jagtap A. S., Bangal P. M presented paper on Skid Base Frame is a structural assembly consisting of beams of various cross sections and dimensions. The designed frame was analysed with Finite Element Method.

[6]. Rachakulla SaiKrishn and P V Anil Kumar The objective of project is to perform the design calculations for the lifting beam for a capacity of 350 Tonnes as per the specifications. Create 3D model as per the design calculations in UNIGRAPhICS.

[7]. SadafAkhtar and Mohammad Abbas the main objective of the work is to carry out the failure reduction and also attempt have been made on weight reduction and cost optimization of the lift arm.