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, Lloyds, 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 ides about its sustainability to the desired load.**

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

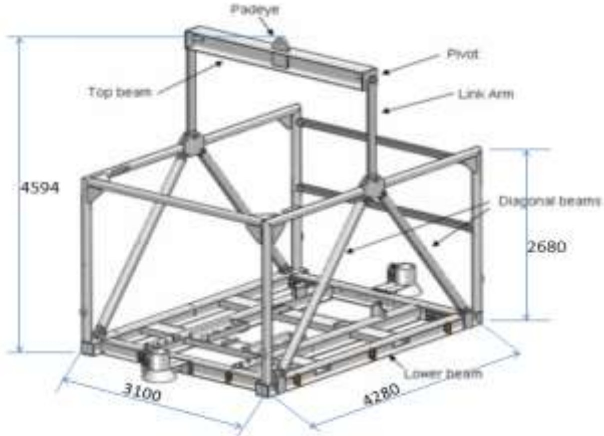
I. INTRODUCTION

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, Lloyds, 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 ides 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.

<u>RTS SKID III</u>



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		
Operational Class	MGW< 50 tonnes	MGW ≥ 50 tonnes
R60	$1.4 + 0.8 \times \sqrt{50}/\text{MGW}$	2.2
R45	$1.4 + 0.6 \times \sqrt{50}/\text{MGW}$	2.0
R30	$1.4 + 0.4 \times \sqrt{50}/\text{MGW}$	1.8

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 "σ"
Allowable stress (σ e) = 0.85 x σ y

Whereas,
σ y = Yield strength of material
MGW = Maximum gross weight of RTS-III i.e. 12 tonne.

MATERIAL USED FOR PRIMARY STRUCTURAL ELEMENTS:

Material	**Yield Strength in Mpa (σ y)**	**Material assigned to part**
Norsok M120, Y05	355	Pivot, Link arm, Diagonal beam, Lower beam , Top beam
S165 M	620	Bolts
Norsok M120, Y30	420	Padeye, Hinge

ALLOWABLE LOAD (σ E) CALCULATION TABLE:

Material assigned to part	Yield strength (σ y)	Allowable strength (σ)
Pivot, Link arm, Diagonal beam, Lower beam , Top beam	355	301.75
Bolts	620	527
Padeye, Hinge	420	357

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

F= DF x MGW x g

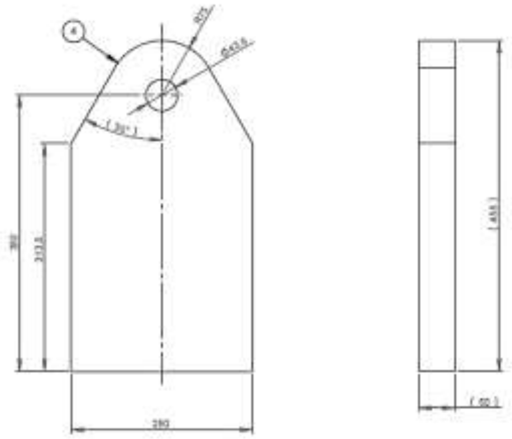
Where DF = $1.4 + 0.6 \text{X} \sqrt{50 / \text{MGW}}$

= 2.6247

So, F = 2.6247 x 12000 x 9.81= 308979.68 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

$$\sigma b = 0.045 \times \sqrt{\frac{RSF \times E}{Dh \times t}}$$

where, σe 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 x F -------------------- (F= Design load)

RSF Padeye in line design load. = 407853.18 N

Therefore:-

$$\sigma b = 0.045 \sqrt{\frac{407853.18 \times 210000}{43.50 \times 50}}$$

σb=282.38 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:

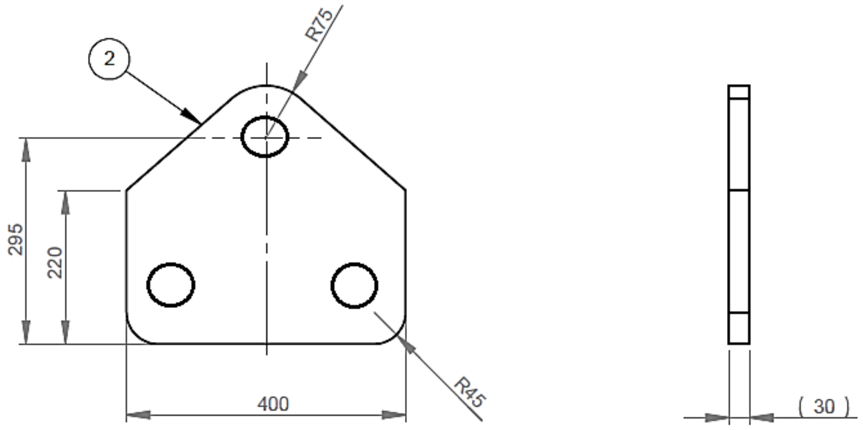
$$\sigma t = \frac{RSF}{(Rpad - Rh) \times t}$$

where, σe : 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)
RSF Padeye in line design load = 407853.18 N
Rpad Radius of padeye, taken as: Rpad = 75 mm

σt (tear out) = 153.2 Mpa

σt>>σa (tear out)--------------Design is safe

B. HINGE -TOP HOLE



BEARING PRESSURE

$$\sigma b = 0.045 \times \sqrt{\frac{RSF \times E}{Dh \times t}}$$

where, σe 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.ofpadeye at hole (mm) = 60mm
Rh	: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 x F/2 -------------------- (F= Design load)

RSF Padeye in line design load. = 203926.59 N

Therefore,

Bearing pressure σb will be,

σb= 170 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:

$$\sigma t = \frac{RSF}{(Rpad - Rh) \times t}$$

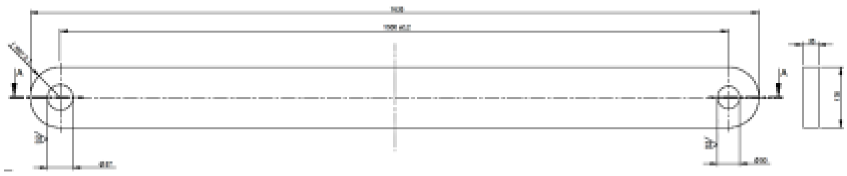
where, σe Allowable stress of padeye material in MPa,
DH : Diameter of pinhole (mm)= 50 mm
t :Total thickness of padeye at hole including cheek plates (mm)= 60

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

σt (tear out) = 67.97 Mpa

σ >>σt (tear out)--------------Design is safe

C. LINK ARM CALCULATIONS ACCORDING TO DNV 2.7-3



BEARING PRESSURE

$$\sigma b = 0.045 \times \sqrt{\frac{RSF \times E}{Dh \times t}}$$

where,σe 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.ofpadeye 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,
RSF = 1.4 x F/2 -------------------- (F= Design load)

RSF Padeye in line design load. = 203926.59 N

σb = 208.5 Mpa

σb>> σ (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:

$$\sigma t = \frac{RSF}{(Rpad - Rh) \times t}$$

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

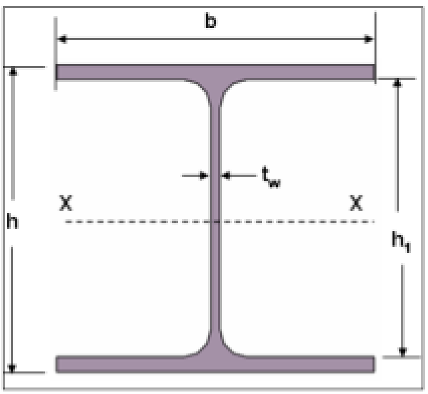
RSF Padeye in line design load.

Rpad Radius of padeye, taken as: Rpad = 67.5 mm

σt (tear out) = 149.4 Mpa

σt>>σa (tear out)--------------Design is safe

C . TOP BEAM



MGW= 12000 Kg
σy = 355 Mpa
b = 280
h = 270
h1 = 244 mm
tw = 13 mm
g = 9.81

Design force (F) = 2.5 MGW x g = 294.300 KN

Length of beam = L= 2959

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

I(total) = $\frac{bh3 - h13\ (b-tw)}{12}$

= 12.99 x 10^7 mm4

e (max) = b/ 2 = 140 mm

Section Modulus (W) = I (total) / 140 = 9.28544 x 10 ^5 mm3

Bending Stress = σ b = M (max) / W = 234.46 Mpa

Maximum Shear force Fτ = F / 2 = 147 KN

Shear Stress $\tau = \frac{S\ .\ A\ \bar{Y}}{I\ b}$ = 66.231 Mpa

$A\bar{Y}$ = 1.17 x 10 ^ 5 mm2

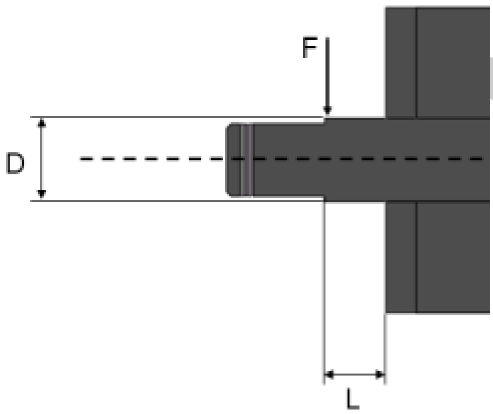
Von Mises Stress $\sigma vm = \sqrt{\sigma b2 + 3\ \tau 2}$

= 261.02 Mpa

Accept criteria $\sigma vm < 0.85\ \sigma y$

$0.85\ \sigma y$ = 301.75 Mpa

D. PIVOT BOLTS



MGW= 12000Kg

σy = 355Mpa

Number of bolts (Nb) = 2

DiameterD =55 mm.

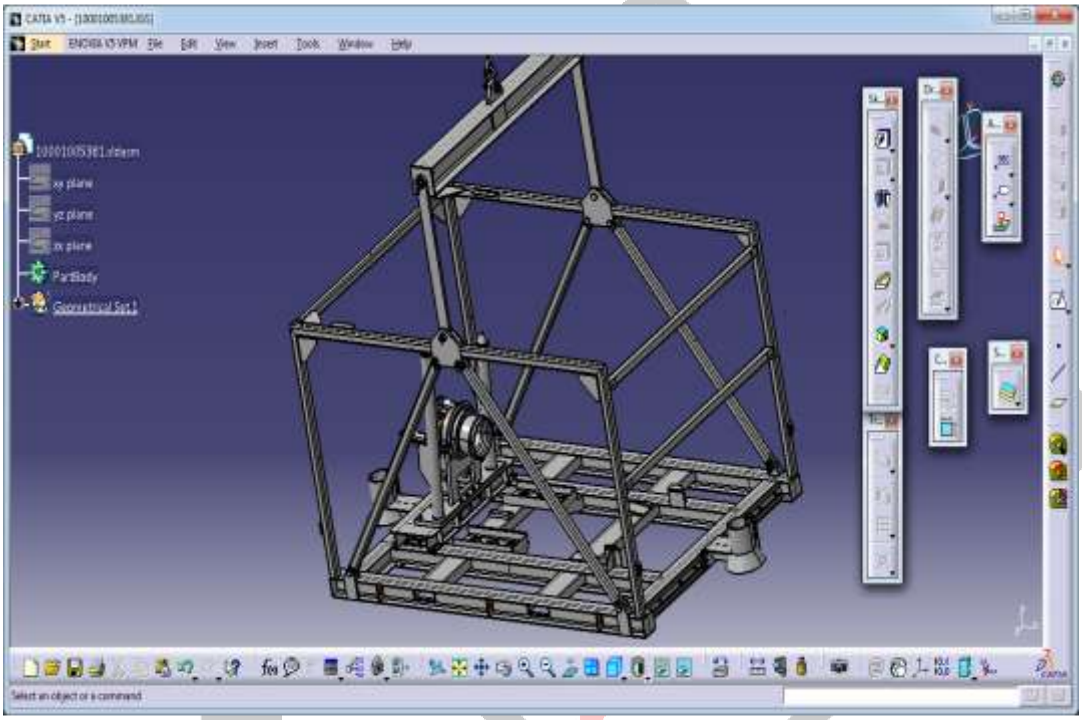
Area A = $\frac{\pi \times D2}{4}$ = 1810 mm2

$$\text{Design Force} = \frac{2.5 \times MGW \times g}{nb} = 0.147 \text{ MN}$$

$$\text{Shear Stress} = F / A = 62 \text{ Mpa}$$

$\tau << \sigma$ **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.

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