Fatigue and Structural Analysis of Azimuth Thruster Assembly

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Abstract**—** *This paper primarily concerns with the determination of life of bracket assembly of Azimuth Thruster under fatigue and structural considerations. An Azimuth thruster is an arrangement of marine propellers placed in pods which can be rotated to any horizontal angle (azimuth). Joseph Becker, invented the Z-drive azimuth propeller first in 1950, whereas this kind of propulsion was first patented by Pleuger in 1955. The ships fitted with this system give better maneuverability than a fixed propeller and rudder system. As thrust acts on the assembly, the well is supported by structures like brackets, struts etc. After brief research on the optimization of design of propellers, performances, there is a scope of improvisation in the design of supporting structures related to the assembly. In this paper, problems while assembling the structure are discussed. In order to mitigate these problems, its expected life under working conditions is found out. Also, structural analysis is carried out to compare as-designed and as-fitted assembly under given thrust loading conditions.*

Keywords**— *Azimuth Thruster, Brackets, Fatigue, Structural.***

I. INTRODUCTION

Azimuth Thruster is the configuration, which is used in marine vessels to provide necessary thrust in desired direction which give ships better maneuverability than fixed propellers and rudder systems [1]. Ship Propulsion system is changing rapidly and so is the propulsion mechanism system. These thrusters are primarily used in dynamic positioning of vessels to maintain the position by counteracting environmental obstacles such as wind and waves. Now days these are also used for propulsion. Azimuth Thruster is an arrangement in which the propeller is placed in pods that can be rotated in any direction in the horizontal plane. Azimuth is an angular measurement in a spherical coordinate system. This thruster is capable of rotating itself in 360˚about its vertical axis, which provides more flexibility, dynamic positioning [2] [3] and optimal thrust in every direction to the system unlike fixed pitch thrusters. For a single vessel, there can be more than one thrusters mounted right underneath the hull of the vessel i.e. near bow and stern [4].

The azimuth thruster using the Z-drive transmission was invented in 1950 by Joseph Becker, the founder of Schottel in Germany, and marketed as rudder propeller [5]. Later on, relevant literature was studied and many modifications were suggested in the design of the thrusters.

II. PROBLEM DEFINITION

Azimuth thruster is located between frame 3 & 9, with 3^0inclination to the vertical for MSV Yard. The azimuth thrusters mounted within a 20 mm thick tunnel which is supported by one set each of internal and external radial brackets. There are 12 Original Equipment Manufacturer (OEM) supplied internal brackets and 16 fabricated external brackets for the Azimuth thruster tunnel. As per design, twelve of the sixteen brackets are to be mounted coplanar with the twelve internal radial brackets for ensuring smooth load transfer. However, the yard had practical problems in aligning these two sets of brackets in the same plane for the given mounting configuration of the thruster tunnel. This was primarily because the thruster tunnel is mounted at an angle of 3^0 to the vertical. Consequently, the tunnel brackets are out of plane to the ship's structural floor plate by a maximum of 80mm. The details of bracket alignment in designed and as-fitted cases are given in following figures. Out of these 12 OEM Brackets, 6 (BKT No. 1, 2, 3, 7, 8, 9) are out of plane to external fabricated brackets. Therefore, there is a requirement to ascertain if the as fitted bracket alignment in deviation to the original design would result in failure of the structure for the given loads.

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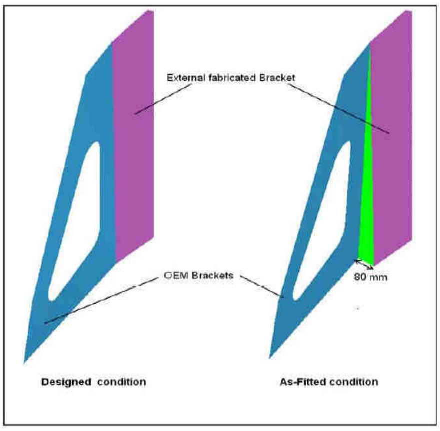

Fig 1. Assembly Problem

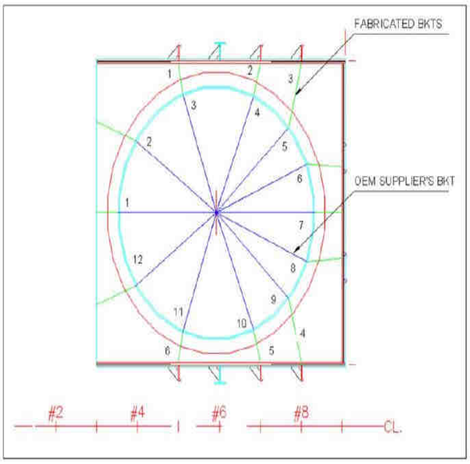

Fig 2. Azimuth Well- Bottom View

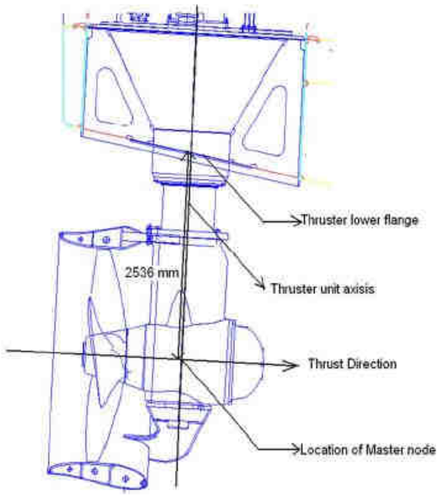

Fig 3. Master Node Location

III. THEORY

1. Material Selection:

Material and its grade depends on the thickness of the plate which is used for manufacturing of hull structure. Thickness wise grading of steel material is given in following table:

Table 1. Grades of steel for Marine Structures

Thickness of Plate	Grade
< 20.5 mm	A
20.5- 25.5 mm	B
25.5- 40 mm	C
>40 mm	D

Since the steel plates which are being used in this case are of mean thickness of **20mm**,**GradeA** structural steel is used for construction.

Grade A steel plate is used for shipbuilding's hull structure and platform. The shipbuilding steel plate grade A is the common tensile strength steel. It has good toughness properties and higher strength, strong corrosion-resistance, the processing properties, and welding properties. **ASTM A131 Grade A steel** plate can be used in the manufacture of the ship's hull structure whose weight is below 10000 tons, and usually do navigation around coastal and river area.These grade A have their tensile strength in the region of **400–520 MN/m^2** as compared to the normal grades which have tensile strength in the region of **235 MN/m^2**.

2. SN curve:

The fatigue design is based on use of S-N curves which are obtained from fatigue tests. The design S-N curves which follow are based on the mean-minus-two-standard-deviation curves for relevant experimental data. The S-N curves are thus associated with a 97.6% probability of survival.The S-N curves are applicable for normal and high strength steels used in construction of hull structures. The S-N curves for welded joints include the effect of the local weld notch. They are also defined as hot spot S-N curves. Thus these S-N curves are compatible with calculated stress that does not include the notch stress due to the weld. This also means that if a butt weld is machined or grind flush without weld overfill a better S-N curve can be used.The basic design S-N curve is given as[6]

$$\textbf{Log N} = \log \bar{a} - m\log\Delta\sigma$$

With S-N curve parameters given in Table following tables [6],

Table 2. SN curve Parameters

SN	Material	$N \leq 10^7$	$N > 10^7$

Curve		$\log \overline{a}$	m	$\log \overline{a}$	m
I	Welded Joints	12.164	3.0	15.606	5.0
II	Basic Material	15.117	4.0	17.146	5.0

N = Predicted number of cycles to failure for stress range $\Delta\sigma$
Δσ = stress range
m = negative inverse slope of S-N curve
log a = intercept of log N-axis by S-N curve

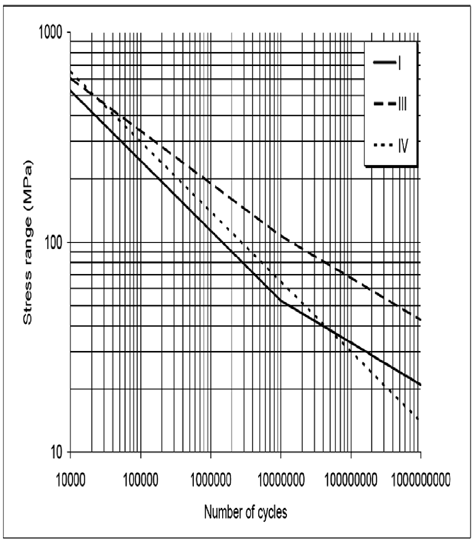

Fig 4. SN curve

3. Considerations:

Following 16 direction are considered for analysis which are recommended in specifications. [6]

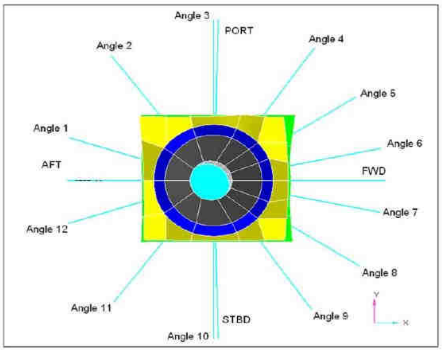

Fig 5. Thrust direction Considered for Analysis

Table 3. Thrust Direction angles

Sl. No.	Thrust Direction	Description
1	Aft-For	0° to fore-aft(ahead)
2	Fore-Aft	0° to fore-aft(astern)
3	P to S	90° to fore-aft (-y direction)
4	S to P	90° to fore-aft (y direction)
5	Angle1	17.5° to Fore-Aft
6	Angle2	52.5° to Fore-Aft
7	Angle3	91.5° to Fore-Aft
8	Angle4	125.5° to Fore-Aft
9	Angle5	148° to Fore-Aft
10	Angle6	169° to Fore-Aft
11	Angle7	-169° to Fore-Aft
12	Angle8	-148° to Fore-Aft
13	Angle9	-125.5° to Fore-Aft
14	Angle10	-91.5° to Fore-Aft
15	Angle11	-52.5° to Fore-Aft
16	Angle12	-17.5° to Fore-Aft

IV. MODELLING

1. Software tool used:

The geometrical modeling was carried out using HyperMesh module which is a high performance finite element pre-processor that provides a highly interactive and visual environment to analyze a wide spectrum of problems encountered in engineering applications.

The tunnel assembly structure was modeled in the pre-processor of HyperWorks v13.0 Student edition using nodes, lines, splines and surfaces. The computational domain considered along with the geometric models of the designed is depicted in Figure.

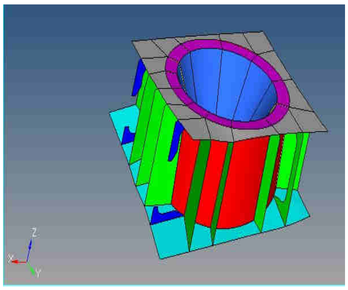

Fig 6. As Designed Assembly

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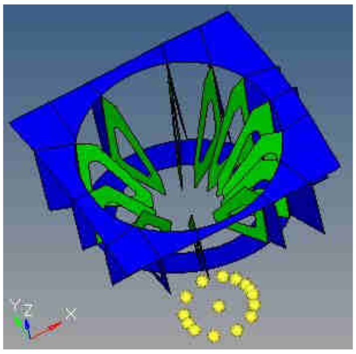

Fig 7. As Fitted Assembly

2. Meshing:

The geometric model was meshed using the Automesh option of HyperMesh. The Finite elements were created by using second order 2D quadrilateral (CQUAD8, 8 noded) and triangular (CTRIA6, 6 noded) shell elements. And the number of triangular elements is negligible as compared to that of quadrilateral elements. Element size of 20x20 mm was taken for the discretization of entire structure. The thicknesses of shell elements were assigned as per detailed structural drawings.

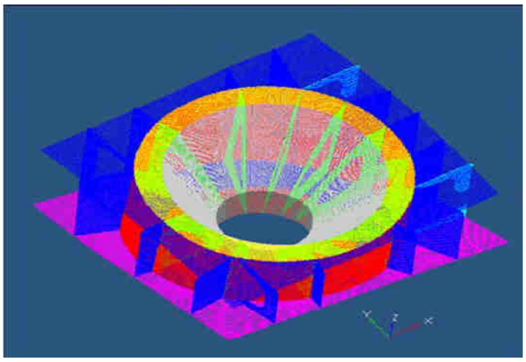

Fig 8. Meshing

3. Boundary conditions:

3.1. Fixed boundary condition is applied at,

1. Shell
2. Deck
3. External fabricated brackets; that are welded to the bulkhead boundaries of thruster unit.

3.2.The following Fixed boundary condition is applied on the FE model.

1. Linear Translation in X, Y and Z direction=0.(**Ux=Uy=Uz= 0**)
2. Rotational Translation in X, Y and Z direction=0.(**Rx=Ry=Rz= 0**).

4. Load details:

- Self-weight: 405 KN
- Thrust Load: 420 KN

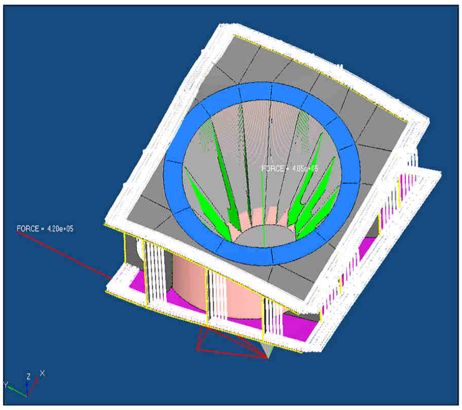

Fig 9. Loading Details and Boundary Condition

V. ANALYSIS

Thrust generated is assumed to act along the axis of rotation of propeller which is 2.536 m below from the lower thruster flange. The load imposed on the thruster assembly by the action of thrust through the propeller axis is simulated using rigid link in FE.

Table 4. Von Mises Stresses

Sr. No.	Thrust Direction	Von Misses Stresses (N/mm^2)
1	Aft-For	52.88
2	For- Aft	54.31
3	**P-S**	**69.87**
4	S-P	46.91
5	Angel 1	53.13
6	Angel 2	66.96
7	Angel 3	69.51
8	Angel 4	68.74
9	Angel 5	58.35
10	Angel 6	54.93
11	Angel 7	52.93
12	**Angel 8**	**46.64**
13	Angel 9	48.43
14	Angel 10	49.67
15	Angel 11	48.07
16	Angel 12	49.07

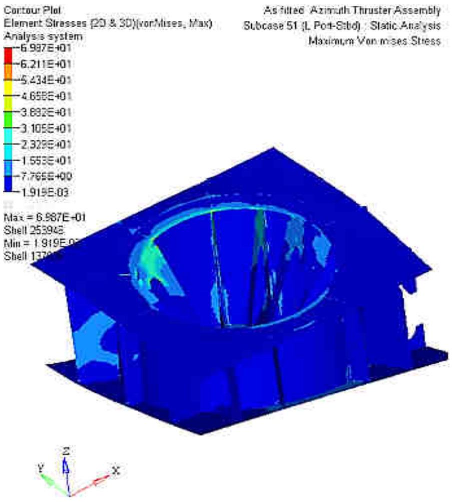

Fig 10. Max Stress Value: 69.87 MPa at P-S

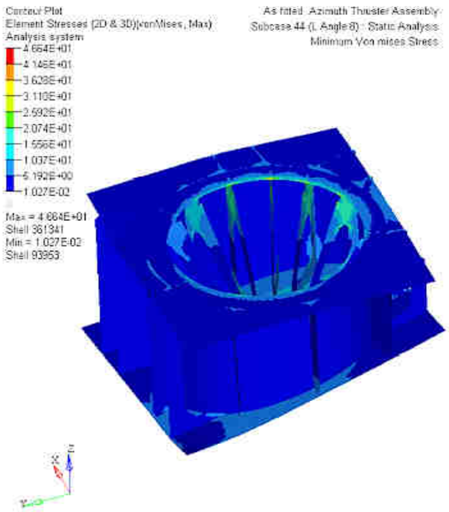

Fig 11. Min Stress Value: 47.64 MPa at angle 8

VI. STRESS RANGE (Δσ) CONSIDERED FOR ANALYSIS

- **Expected life: 25 years**
- **Max von mises stress, σ_{max} = 69.87 MPa**
- **Min von mises stress σ_{min} = 46.64 MPa.**

1. Sea Going Condition

Stress Range (Δσ) = σ_{max} - σ_{min} = 23.23 MPa, Considering a safety factor of 1.5, stress range for seagoing condition,

Δσ = 34.85 MPa

2. Harbor Condition

Stress Range (Δσ) = 2x σ max = 139.74 MPa,

Δσ = 139.44Mpa.

VII. PREDICTED LIFE (Ni) AS PER S-N CURVE

1.Sea Going Condition

- Stress Range(Δσ) = 34.85 MPa
- $\log(N_i) = \log \bar{a} - m \log(\Delta\sigma)$

=15.606 - 5 log (34.85)

=7.895

- Life(N_i) =**7.86 x10^7 Cycles**
- Cycles per year: 525600
- Life in years = $\frac{7.86 \times 10^7}{525600}$

= **150 years**

2. Harbor Condition

- Stress Range (Δσ) = 139.44 MPa
- $\log(N_i) = \log \bar{a} - m \log(\Delta\sigma)$

=12.164 - 3 log (139.44)

=5.73

- Life(N_i) = **5.38 x10^5 Cycles**
- Cycles per year: 12000
- Life in years = $\frac{5.38 \times 10^5}{12000}$

= **45 years**

3.Damage factor

$$\mathbf{D} = \sum_{i=1}^{k} \frac{n_i}{N_i} = \frac{1}{\bar{a}} \sum_{i=1}^{k} n_i (\Delta\sigma_i)^m \leq \eta$$

Where,

n_i= No of cycles applied (During designed life)

N_i= No of cycles to failure at a constant stress range

$\bar{a}$,m=SN Parameters

μ = Usage factor,

Accepted usage factor is defined as **1.0**

3.1.Sea Going Condition

Damage = $\frac{525600}{7.86 \times 10^7}$

= **0.006**

3.2. Harbor Condition

Damage = $\frac{12000}{5.38 \times 10^5}$

= **0.023**

VIII. STRUCTURAL ANALYSIS

Following loads are considered for structural analysis.

1. Self-weight of seating: Self-weight of the assembly was applied as inertial force by assigning acceleration due to gravity (g = -9810 mm/S^2)

2. Thruster unit weight: Total weight of the thruster unit is 41.3 tonnes, which is applied as concentrated nodal force on the master node.

3. Thrust generated by the thruster: Azimuth thruster is capable of generating thrust in all directions by its 360^0rotatable nozzle unit. The maximum thrust generated by the thruster unit is 420 KN. Depending on the thrust direction, all 16 possible load directions are considered for the analysis.Considering unstable flow of water, 1.3 times the thrust value, 540 KN is taken for analysis.

Linear static analysis was carried out using the Altair Optistruct solver.The maximum Von-Mises stresses and shear stress induced in thethruster assembly is taken from the stress contours. The thruster assemblycomponents are made of structural steel of NV A-grade and their permissible stresslimits are calculated as shown below.

4. Permissible Stress Status

For both Design and As-Fitted,

$[\sigma] = 0.707 \text{ x } \sigma_y$
$= 0.707 \text{ X } 235$
$= \mathbf{166\ MPa}$

5. Permissible Shear Stress Status

For both Design and As-Fitted,

$[\tau] = 0.45 \text{ x } \sigma_y$
$= 0.45 \text{ X } 235$
$= \mathbf{105\ MPa}$

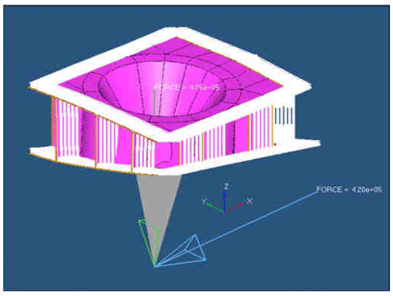

Fig 12. Loading Details

IX. RESULTS

Table 5. Von Mises Stress developed in as Designed & as Fitted Assemblies

Thrust directions	Von Mises stress in MPa		Status
	As Designed	As Fitted	
Aft-For	123.6	114.6	Ok
For-Aft	110.3	122.8	Ok
P-S	118.4	139.5	Ok
S-P	139.4	122.2	Ok
Angle 1	117.5	116.6	Ok
Angle 2	116.2	118.6	Ok
Angle 3	119.6	121.3	Ok
Angle 4	121.5	127.2	Ok
Angle 5	126.0	128.7	Ok
Angle 6	118.6	122.4	Ok
Angle 7	102.9	103.4	Ok
Angle 8	115.9	119.0	Ok
Angle 9	129.7	133.4	Ok
Angle 10	138.2	138.0	Ok
Angle 11	**140.0**	**148.1**	Ok
Angle 12	136.8	134.9	Ok

Table 6. Shear Stress developed in as Designed & as Fitted Assemblies

Thrust directions	Shear stress in MPa		Status
	As Designed	As Fitted	
Aft-For	62.25	66.32	Ok
For-Aft	47.86	51.41	Ok
P-S	60.47	67.96	Ok
S-P	**71.96**	58.48	Ok
Angle 1	59.40	63.15	Ok
Angle 2	57.82	56.44	Ok
Angle 3	61.02	58.08	Ok
Angle 4	60.96	57.74	Ok
Angle 5	56.97	57.01	Ok
Angle 6	51.52	54.01	Ok
Angle 7	46.12	49.34	Ok
Angle 8	55.16	53.44	Ok
Angle 9	67.51	64.81	Ok
Angle 10	71.37	68.31	Ok
Angle 11	71.48	68.60	Ok
Angle 12	68.62	**67.92**	Ok

X. CONCLUSION

The predicted life for Seagoing and Harbor conditions are well above the designed life of 25 years. Damage calculated for seagoing and harbor conditionsare very less as compared to Usage Factor 1.0.

Also for structural analysis, the maximum Von Mises stress induced in the Designed and As-fitted assemblies are well within the permissible limit of 166 MPa. The maximum Shear stress induced in the Designed and As-fitted assemblies are well within the permissible limit of 105.75 MPa.

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