Scientific Journal of Impact Factor (SJIF): 5.71
e-ISSN (O): 2348-4470
p-ISSN (P): 2348-6406

International Journal of Advance Engineering and Research Development

Volume 5, Issue 03, March -2018

# Design and FEA Analysis of Shafting System in a Fleet Support Vessel

J. Venkata SomiReddy[1], P.V.D. Vivek Kumar[2], Mallikarjuna Rao Dandu[3]

[1, 3] *Department of Mechanical Engineering, LBRCE*
[2] *Department of Mechanical Engineering, SIET*

---

**ABSTRACT-***Propulsion system for the ships is for transferring the developed power by the Main Engine forming a power train and deliver the power to the propeller for obtaining the required thrust. The design phase of shafting includes Material Selection, selection of couplings, bearings, and type of stern tube used.*

*The current projects use the shafting with the material CS45E. The material is of Carbon steel with grade. The design of shafting is a state of art and alignment calculations plays a vital role. In this regard shafting analysis with static and dynamic considerations including bearing reactions using software ANSYS will be conducted.*

*The ship is designed with two intermediate shafts and one tail shaft and marine lubricated stern tube and bearings. The design of shafting is done from on strength basis, and checked with Lloyd's register of shipping rules, which are standard rules to maintain quality of the products. The shaft modeling is done in CATIA. The change of material by replacing it by composite is done in HYPERMESH and results will be compared. The shafting analysis conducting the calculations arriving at the shafting system alignments are shown as results.*

---

***Keywords-*** *Propulsion shafting system, FEA, Design and Analysis, Ship shaft, ANSYS, CATIA, HYPERMESH, Composites, Carbon-epoxy, Composite shaft*

## I. INTRODUCTION

Conversion of rotary power output from prime movers to develop thrust required to impel the ship constitutes shafting system. Propeller which is also a member in the family of propulsion system to provide the thrust to vessel structure.

Shafting system includes main engine shaft, intermediate shaft, propeller shaft, couplings, bearings, stern tube, and rope guard. While in operating range of 80% to 110% of speed the main masses of the shafting system should be free from any kind of stresses. Considerations should be like such that ship is not disturbed while moving. The section in which propeller secured or attached is called tail shaft or propeller shaft. If in case the propeller is not supported the shaft running through stern tube is labelled to be stern shaft. In between shafting denoted to be intermediate shaft, and shaft that connects to prime movers is named thrust shaft or engine shaft.

### 1.1. Static Structural Analysis:

To determine the loading effects of physical erections and their workings Static structural analysis is done. Structures are which taking loads such as furniture, machine members, bridges, frames etc. Fitness of the construction to with stand the stresses and loads is determined by static analysis of structure. The present shafting system is imported to hyper mesh and static analysis is performed.

### 1.2. Modal Analysis:

Modal analysis is to check the physical system natural frequency to avoid resonance. Resonance creates vibration problem which may run system to failure. It is avital part of the design as the vibration may cause component failures which is disastrous in standings of cost, time, and human life

### 1.3. Composite Materials:

The inherent flaws and imperfections in the material makes the strength of the material to fall under theoretical strength values. Small cross-sectional fibres minimize the flaws, and hence polymeric materials they have greater strengths. High strength and stiffness in polymeric materials is result of molecular orientation of fibres.

Fibres are not in direct use as they have small cross-sections. These fibres are thus imported to matrix which holds these fibres together, and distributes the taken load to these fibres. Fibres are which are primary constituent of fibre reinforced plastics are very economical to be used. They constitute more volume fractions to matrix. They load is distributed among fibres in matrix.

@IJAERD-2018, All rights Reserved

## II. DESIGN AND MODELLING OF SHAFTS

**2.1. Description of Problem:** The shafting system has to be designed to transmit a power 18560 KW at a speed of 160.6rpm from engine to the propeller. One end of engine shaft or thrust shaft is coupled to engine by a propeller by a hydraulic flange coupling. Intermediate shaft ends are joined by hydraulic sleeve coupling. The shaft material is a forged steel of density 8740 Kg/$m^3$, with a maximum tensile strength of 600 N/$mm^2$, and a shear strength of 295 N/$mm^2$.Propeller speed=160.06 rpm, Material of the shaft = CS45E, Tensile Strength = 600 N/$mm^2$, Density of the material =8740 kg/$m^3$, Young's Modulus = 207 GPa, Shear Strength = 295 N/$mm^2$, Power of the vessel to be transmitted = 18560 kW @160.6rpm

Torque, with load of 10% for quick reversals.

$$T = \frac{60 \times P \times 1.1}{2\pi N}$$

$$T = \frac{60 \times 18560 \times 10^3 \times 1.1}{2\pi \times N}$$

$$T = 1213.94 \times 10^6 N - mm.$$

**2.2. Shafting Design Based on Pure Torsion:**

$$\frac{\tau}{r} = \frac{T}{J} = \frac{C\theta}{L}$$

Taking factor of safety F.S = 4, the allowable shear stress will be = 73.75 N/$mm^2$

Diameter of the shaft$d = \sqrt[3]{\frac{16T}{\pi \times 73.75}}$

Diameter, $d = 437.65mm.$

**2.2.1. Propeller Shaft:**With the maximum bending occurring at aft end of the stern tube, the propeller shaft takes bending moment weight of the propeller and thrust. Hence it is proper to design propeller shaft to 15% extra what it takes to the design on pure torsion.

$d_p = 1.15 \times 437.65$ mm

d =515 mm (next nearest standard diameter)

**2.2.2. Intermediate Shaft:** The intermediate shaft is exposed to bending moment end to end with twisting moment. The next standard diameter earlier to stress under allowable stress is taken. The intermediate shaft diameter **d** =450 mm

**2.3. Hydraulic Sleeve and Flange Coupling:** This type of coupling consists basically of two sleeves of high quality steel, a thin inner sleeve and a thick outer sleeve. The outer surface of the inner sleeve is slightly tapered and the bore of the outer sleeve has a corresponding taper. The inner sleeve bore is somewhat larger than the diameter of the shafts, so that the sleeve can be passed over them with ease. The coupling is mounted by driving the outer sleeve up on the taper of the inner sleeve using the hydraulic unit incorporated in the coupling. The coupling is modelled as per standard diameter of the shaft.

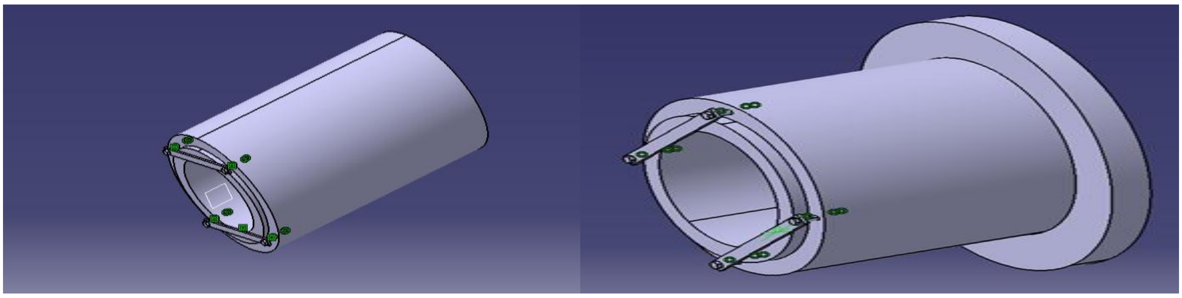


*Fig 2.1.Assembly CAT Product of Hydraulic Couplings*

**2.4. Plummer block bearing and stern tube:** A Plummer block bearing to support the coupled shaft is modelled as per the standard diameter of the shaft. Stern tube is a kind of bearings supporting the propeller shaft. It should be able to the plunge as the maximum bending moment ensues at the aft-end.

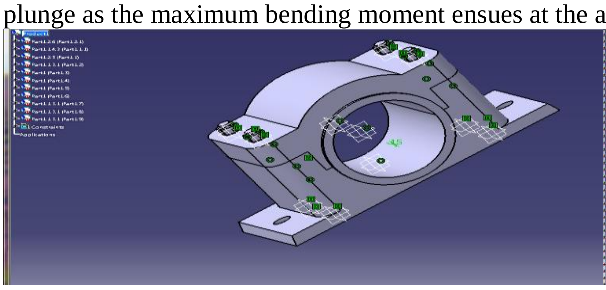


*Fig 2.2. Plummer block CAT product*

*Fig 2.3. Stern tube CAT product*

@IJAERD-2018, All rights Reserved 

**2.5. Final shaft assembly:**

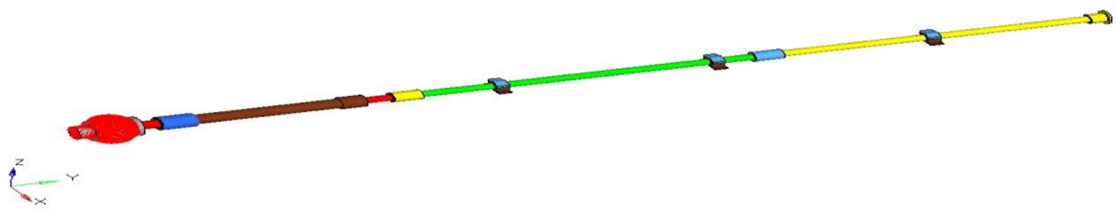


*Fig 2.4. Final Shafting System Assembly CAT Product*

## III. Static Structural Analysis for Safe Design of Shaft:

The assembled shafting system which consists of bearings, couplings, shafts, stern tube, and hub is analyzed by considering it as beam. The couplings and propeller weights are taken as point loads. The bearings are taken as fixed supports. The weight of the shaft is taken as UDL. The bending moments and bearing reactions at each key point are calculated

The value of thrust force acting on the propeller shaft

$$F_t = \frac{326 \times H \times P}{V(1-t)}$$

$$F_t = \frac{326 \times 24899.33 \times 0.73}{16 \times (1-0.2)}$$

$F_t = 462.75KN$

For the taken thrust force 10% load factor is considered for safety

Therefore, thrust force becomes. $F_t = 509KN$.

**3.1. Static-Structural Analysis in ANSYS Methodology:**

Open ANSYS APDL, set the preferences to the Structural. From pre-processor menu, add 2node188 as beam element. Take Material properties as Structural, linear, Elastic and isotropic. From section, select the common-section of beam as circular. Modelling the shaft using key points in active coordinate system using lines, then mesh using meshing. Define loads and displacement on key points, apply force and pressure

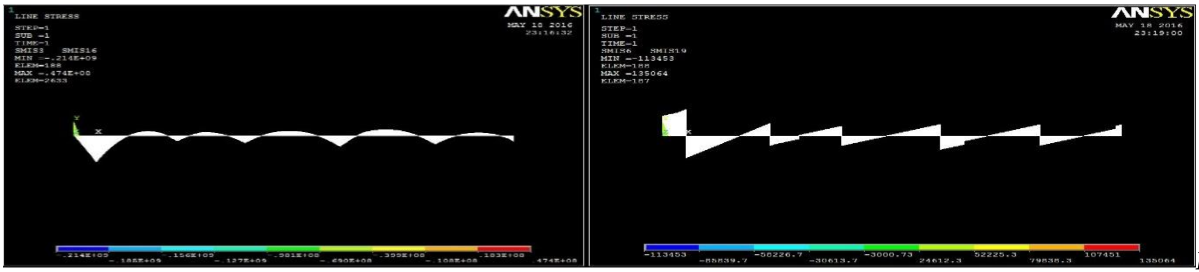



***Fig 3.1. Shear force and bending moment results in ANSYS***

Extreme bending moment for the propeller shaft is observed to be$215.018 \times 10^6 N - mm$, in analytical which is nearer to the extreme bending moment over ANSYS that is $214 \times 10^6$N-mm.

Extreme bending moment for intermediate shaft is eyed as$100.43 \times 10^6 N - mm$, which is nearer to that bending moment value from ANSYS which is$98.10 \times 10^6 N - mm$.

**3.1.1. Shafts subjected to axial loads in addition to combined torsional and bending loads:**

$T_e = \sqrt{[K_m \times \frac{\propto Fd}{8}]^2 + [K_t T]^2} k_m = 1.5, k_m = 1 \propto = 1.97$

The equivalent twisting moment is obtained as $T_e = 1273.44 \times 10^6 N - mm$

But

$T_e = \frac{\pi \tau d^3}{16}$ Therefore, $\tau = \frac{47.48N}{mm^2} < 73.75N/mm^2$ . Hence it is safe in static condition.

$$M_e = \frac{1}{2}[K_m M + \frac{\propto Fd}{8} \sqrt{[K_m M + \frac{\propto Fd}{8}]^2 + [K_t T]^2} M_e = 830245546.4$$

But $M_e = \frac{\pi \sigma_b d^3}{32}$ therefore, $\sigma_b = \frac{61.91N}{mm^2} < 150N/mm^2$. Hence propeller shaft is safe.

**3.2. Validation of design using LLYOD'S Shipping Rules:**
The minimum diameter of the propeller shaft is given by

$d_p = 100 \times k \times \sqrt[3]{\frac{P}{R}[\frac{560}{\sigma_u+160}]}$ , Where K=1.15 for keyless propeller, Rated speed R = 160.6 rpm, Rated power P = 18560 kW

$d_p = 508.44 < 515mm$ , $d_i = 439.94 < 450mm$

Hence the Shaft is determined to be Safe.

**3.3. Analysis with composite material for modelled shafting system:**
The final assembly of Shafting System which is modelled and assembled in CATIA is saved as igs file.Using the Mesh tool meshing of the model on anti-symmetric plane of the geometry. Carbon epoxy material is applied using material card. Fifteen stacks are used with 15mm and number piles used are 20. All the plies are stacked into a laminate. Properties of carbon /epoxy composite with 60% volume fraction of fibers are taken. Four designs are considered for each stack in the components depending on orientation of fibers. Design1 of 45/-45 fiber orientation, Design2 of 0 orientation, Design3 of 90/-90 orientation and Design4 is a combination of the above three designs of ply orientation.Tail end of the Shaft is constrained in x, y, z directions and rotation along the axis of shaft is left free. Bearings and Stern tube are fixed. Mass elements which are dimensionless are created on hub nodes and propeller weight is added (101372 N).Moment is applied at aft side on the hub. (214000000 N-mm).

*Fig 4.1. Fully Meshed Shafting System and element orientations*

## IV. Results and Discussion:

**4.1. Static Analysis Result of CS45E:** For the conventional material the von-misses stress observed as 58.27 $\frac{N}{mm^2}$. the hub portion is observed to be deformed and the displacement of the deformation is 1.171 mm the von-misses stress is detected to be well within the allowable stress for CS45E. The critical section is observed in the propeller shaft at aft end of the stern bearing

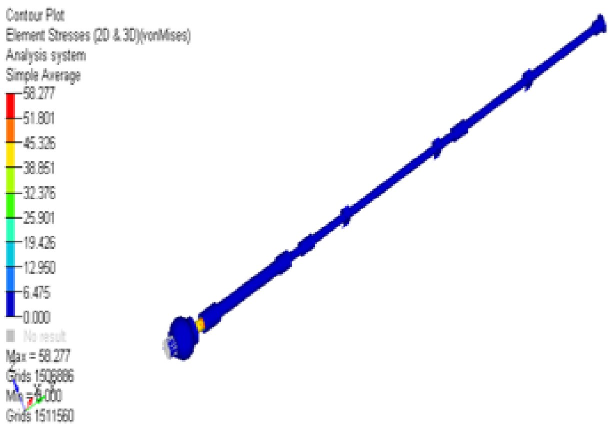



*Fig 4.2. Von-misses Stresses of Conventional Material*

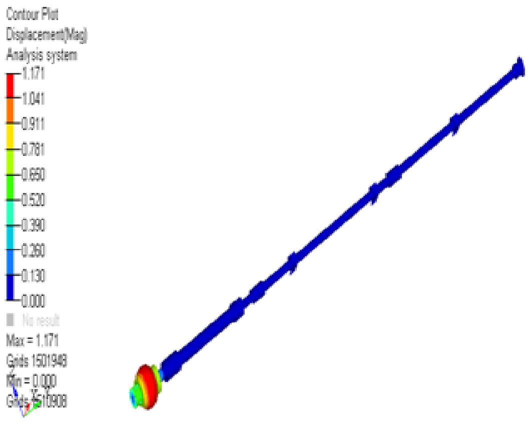



*Fig 4.3. Deformation of Conventional Material Shaft*

**4.2. Modes of Frequencies of Conventional Material System:** The natural frequency for the conventional material CS45E is observed to be 17 Hz which does not match with the operating frequency 2.67 Hz and hence avoiding resonance. Maximum amplitude of 2.6 mm is observed at hub, where the whole system is absolutely safe.

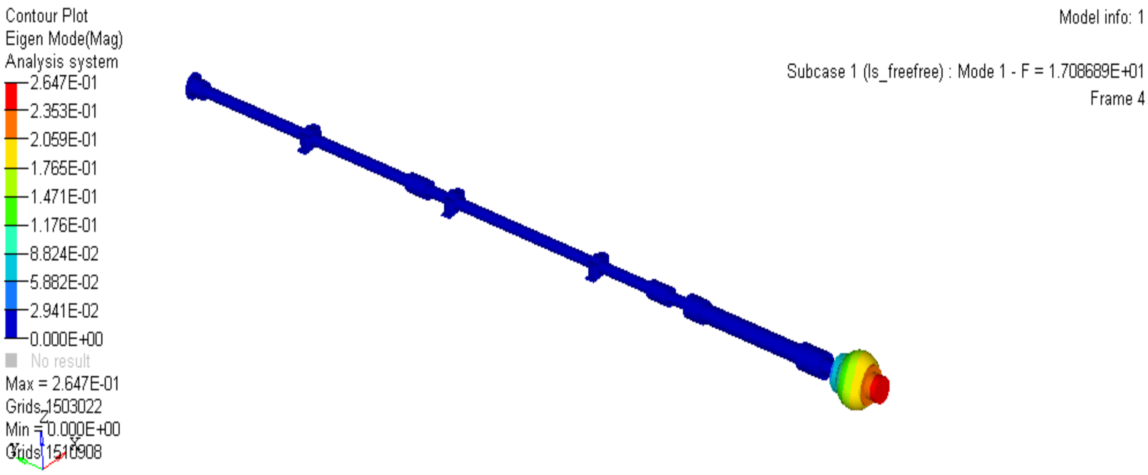
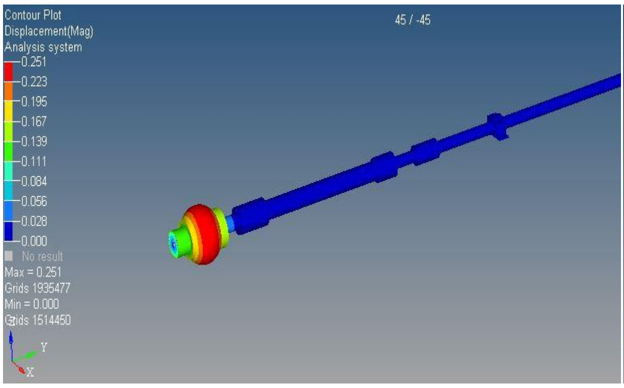



*Fig 4.4. Natural frequency of shafting system with CS45E*

**4.3. Design 1: Static Analysis Result:** The von-misses stress for the composite with orientation [45/-45] plies is observed to be 14.57$N/mm^2$ which is less than that of conventional forged steel. The displacement for design-1 is 0.251mm and its reduced compared to conventional forged steel. The hub portion which loads propeller is observed to deform by 0.251 mm. The stress is well within the safety limit in comparison to the allowable stress.



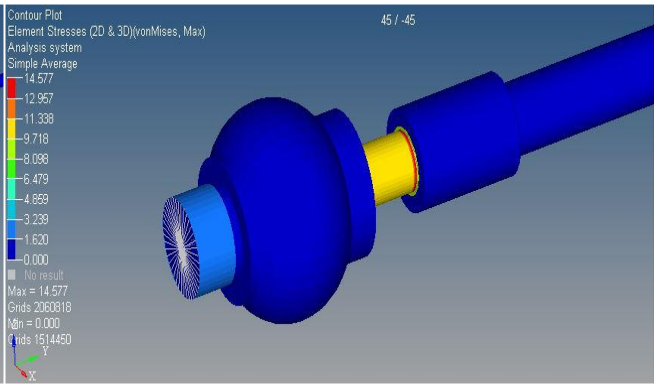




*Fig 4.5. Von-Misses Stresses for Design-1Fig 4.6. Deformation of Design-1*

**4.4. Modal analysis of Design-1:** The natural frequency for the composite of carbon epoxy having the ply design as [45/-45] is 13.57 Hz which doesn't match with the operating frequency of 2.67Hz. Hence there will no chance of resonance and vibration

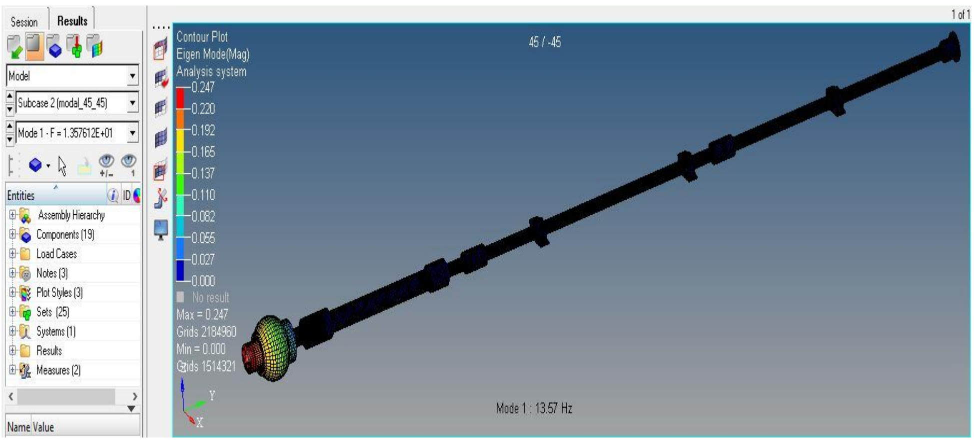
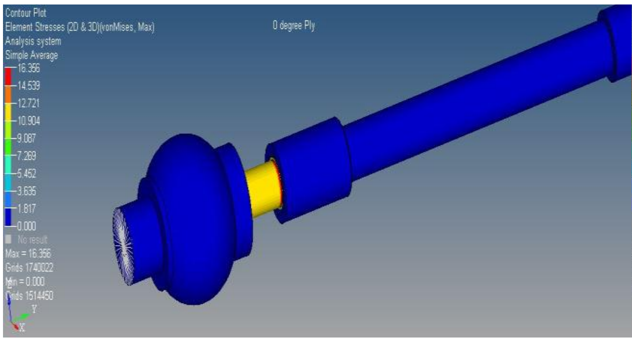



*Fig 4.7. Mode 1 frequency of Design-1*

**4.5. Design 2: Static Analysis Result:** The von-misses stress for the composite with orientation 0 degree plies is observed to be 16.35$N/mm^2$ which is less than that of conventional forged steel. The displacement for design-1 is 0.29 mm and its reduced compared to conventional forged steel. The hub portion which loads propeller is observed to deform by 0.29 mm. The stress is well within the safety limit in comparison to the allowable stress.



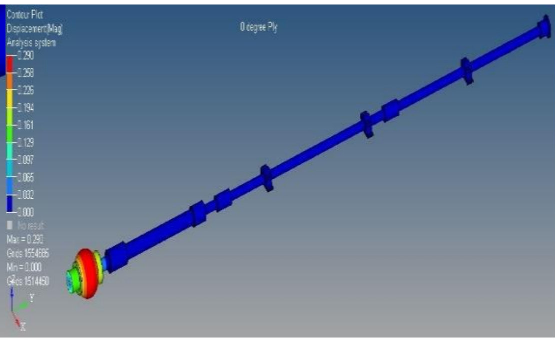




*Fig 4.8. Von-Misses Stresses of Design-*2 *Fig 4.9. Deformation of Design-2*

**4.6. Modal analysis of design-2:** The natural frequency for the composite of carbon epoxy having the ply design as zero degree is 12.4 Hz which doesn't match with the operating frequency of 2.67Hz. Hence there will no chance of resonance and vibration.

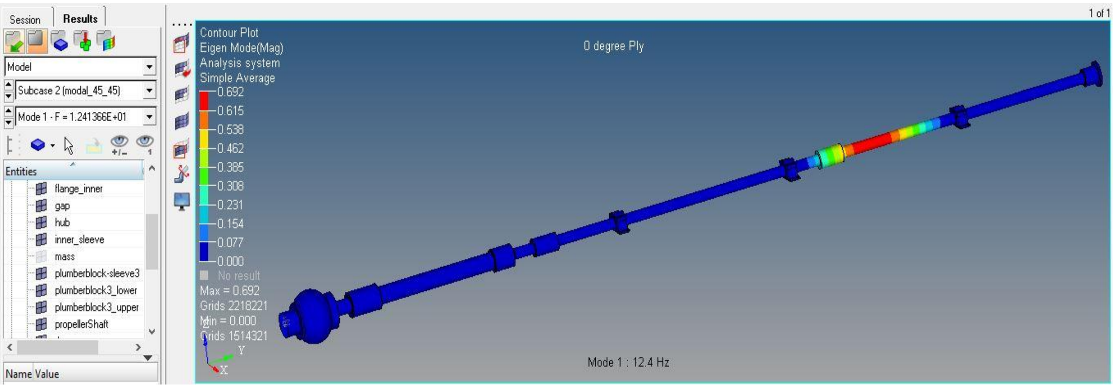
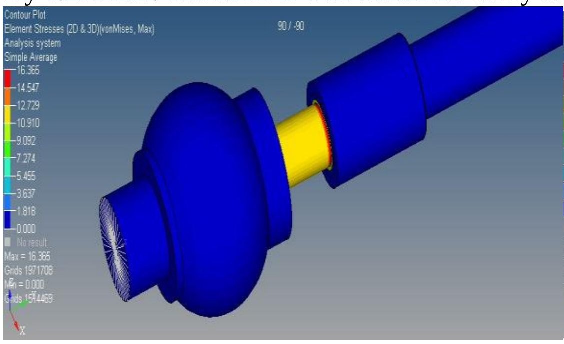



*Fig 4.10. Mode 1 Frequency of Design-2*

**4.7. Design 3: Static Analysis Result:** The von-misses stress for the composite with orientation [90/-90] plies is observed to be 16.35$N/mm^2$ which is less than that of conventional forged steel. The displacement for design-1 is 0.291mm and its reduced compared to conventional forged steel. The hub portion which loads propeller is observed to deform by 0.291 mm. The stress is well within the safety limit in comparison to the allowable stress.



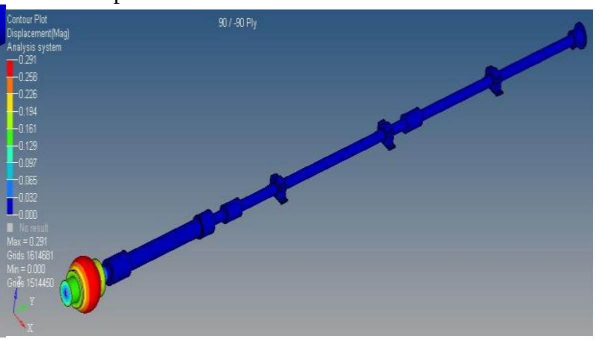




*Fig 4.11. Von-Misses Stresses of Design-3* *Fig 4.12. Deformation of design-3*

**4.8. Modal analysis of design-3:**

The natural frequency for the composite of carbon epoxy having the ply design as [90/-90] is 13.14 Hz which doesn't match with the operating frequency of 2.67Hz. Hence there will no chance of resonance and vibration.

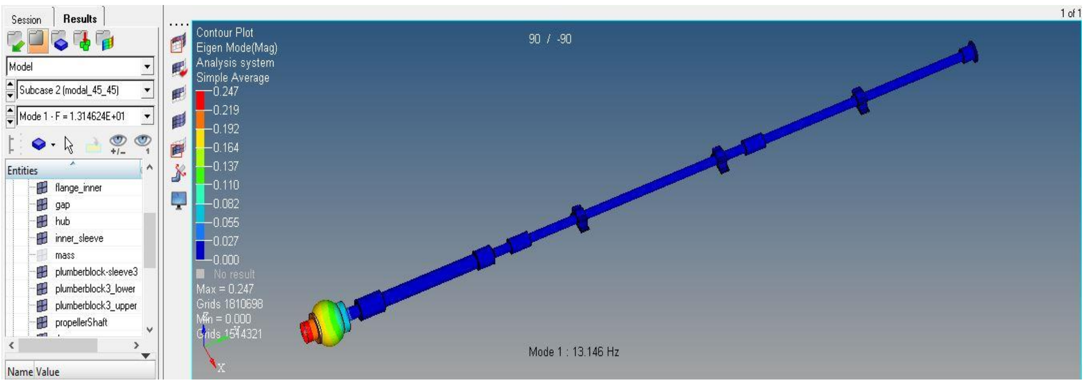
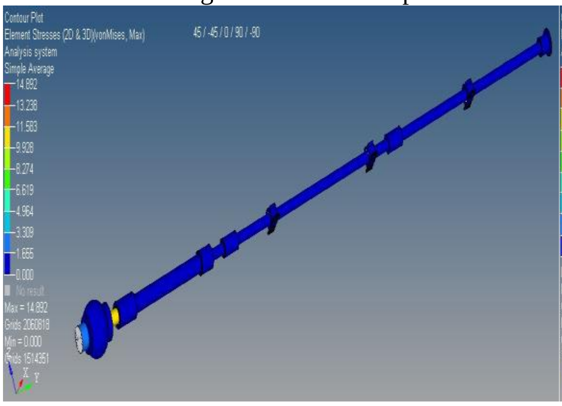



*Fig 4.13. Mode 1 frequency of Design-3*

### 4.9. Design 4: Static Analysis Result:

The von-misses stress for the composite with mixed orientation of plies is observed to be 14.89$N/mm^2$ which is less than that of conventional forged steel. The displacement for design-1 is 0.257mm and its reduced comparedto conventional forged steel. The hub portion which loads propeller is observed to deform by 0.257 mm.



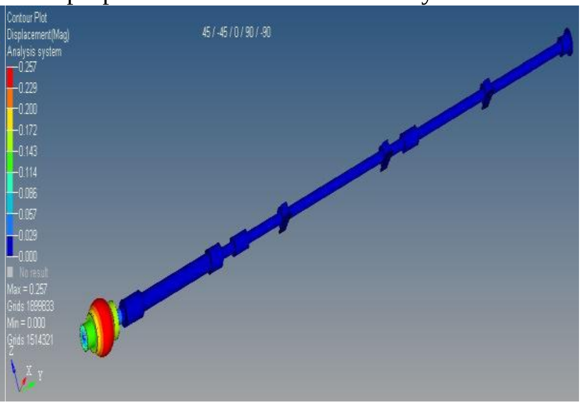




*Fig 4.14. Von-Misses Stresses of Design-4* *Fig 4.15. Deformation of Design-4*

### 4.10. Modal analysis of design-4:

The natural frequency for the composite of carbon epoxy having the ply design as [45/0/90] is 13.66 Hz which doesn't match with the operating frequency of 2.67Hz. Hence there will no chance of resonance and vibration.

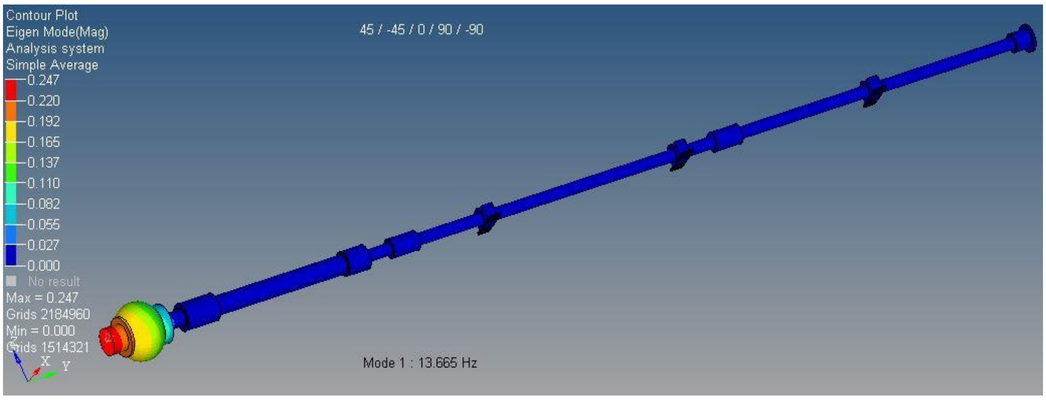



*Fig 4.16. Mode-1 frequency of design-4*

### 4.11. Comparison of Results

*Table 4.1 Comparison of stresses and deformation*

| Material | Stress (MPa) | Deformation (mm) | Mode1 Frequency (Hz) | Allowable Stress (MPa) | Weight (Kg) |
|---|---|---|---|---|---|
| CS45E | 58.27 | 1.171 | 17.08 | 600 | 52265 |
| Design-1 | 14.57 | 0.251 | 13.57 | 1100 | 8980 |
| Design-2 | 16.35 | 0.290 | 12.41 | 1100 | 8980 |
| Design-3 | 16.36 | 0.291 | 13.14 | 1100 | 8980 |
| Design-4 | 14.89 | 0.257 | 13.66 | 1100 | 8980 |

## V. CONCLUSIONS

The shaft is designed for pure torsion and diameter of intermediate shaft is obtained to be 450mm. The diameter of the propeller shaft is obtained to be 515mm. Design of shaft is checked on basis on strength (combined bending and torsional loading) and found to be safe. The diameter is checked for its quality and safety using Lloyd's shipping rules which are standard rules for maintaining quality of the product. The conventional material carbon steel is analyzed and found to have safe stress and natural frequency. Among the 4 designs, the design with plies orientation (45/-45) is found to have lower stress value 14.57 MPa, and found to have safe natural frequency. The deformation for design is 0.251mm which is less than the all designs considered. Weight of the shaft is reduced by 82.8% by using the composite material. Amplitude of vibration when considered at 1st three modes is reduced significantly in composite in comparison to the carbon steel. From all the above considerations Design1 having greater factor of safety is considered to be safe and optimum

@IJAERD-2018, All rights Reserved 

## VI. REFERENCES

[1] L. Murawski, "Shaft line alignment analysis taking ship construction flexibility and deformations into consideration", *Marine Structures*, vol. 18, no. 1, pp. 62-84, 2005.
[2] L. Shi, D. Xue and X. Song, "Research on shafting alignment considering ship hull deformations", *Marine Structures*, vol. 23, no. 1, pp. 103-114, 2010.
[3] Y. Yang, W. Tang and J. Ma, "Optimal design for a VLCC propulsion system based on torsional vibration analysis", *Procedia Engineering*, vol. 15, pp. 5378-5383, 2011.
[4] B. Bai, L. Zhang, T. Guo and C. Liu, "Analysis of Dynamic Characteristics of the Main Shaft System in a Hydro-turbine Based on ANSYS", *Procedia Engineering*, vol. 31, pp. 654-658, 2012.
[5] H. Han, K. Lee and S. Park, "Estimate of the fatigue life of the propulsion shaft from torsional vibration measurement and the linear damage summation law in ships", *Ocean Engineering*, vol. 107, pp. 212-221, 2015.
[6] E. Sevkat and H. Tumer, "Residual torsional properties of composite shafts subjected to impact loadings", *Materials & Design*, vol. 51, pp. 956-967, 2013.
[7] A. Chasalevris and F. Dohnal, "A journal bearing with variable geometry for the suppression of vibrations in rotating shafts: Simulation, design, construction and experiment", *Mechanical Systems and Signal Processing*, vol. 52-53, pp. 506-528, 2015.
[8] W. Litwin and C. Dymarski, "Experimental research on water-lubricated marine stern tube bearings in conditions of improper lubrication and cooling causing rapid bush wear", *Tribology International*, vol. 95, pp. 449-455, 2016.
[9] O. Montagnier and C. Hochard, "Optimisation of hybrid high-modulus/high-strength carbon fibre reinforced plastic composite drive shafts", *Materials & Design*, vol. 46, pp. 88-100, 2013.

@IJAERD-2018, All rights Reserved 