# Simulation of composite raft mount system for onboard centrifugal pump used in ships

**Dr. G. Vijay Kumar[1], K.Sudheer[2], K.Sai Krishna[2], A.Naga Raviteja[2], G.Ashok Babu[2]**

[1]Professor, Dept. of Mechanical Engineering, PVP Siddhartha Institute of Technology, A.P., India.
[2]Student, Dept. of Mechanical Engineering, PVP Siddhartha Institute of Technology, A.P., India

*Abstract:* The noise induced by centrifugal pump is one of the main concerns in the design and optimization of on-board equipment for war ships and submarines. In this work, an attempt has been made to estimate the natural frequency of onboard centrifugal pump used in ship with an ordinary conventional steel material foundation using ANSYS simulation. Further, the conventional foundation is replaced with composite material with carbon reinforcement and vibration studies are accomplished by using FE software ANSYS 20.1. The natural frequencies of centrifugal pump with steel foundation are compared with natural frequencies of centrifugal pump with composite raft foundation. It is observed that the natural frequencies are enhanced by providing composite material foundation. Weight of the overall foundation also can be reduced utilizing the proposed composite raft mount.

***IndexTerms* – composite raft mount, simulation, onboard ship equipment, weight reduction**.

## I. INTRODUCTION

Centrifugal pumps are widely used in the industry, especially in the oil and gas sector for fluids transport. Classically, these are designed to transfer single phase fluids (e.g., water) at high flow rates and relatively low pressures when compared with other pump types. As part of their constructive feature, centrifugal pumps rely on seals to prevent air entrapment into the rotor during its normal operation. Although this is a constructive feature, water should pass through the pump inlet even when the inlet manifold is damaged. Modern pumps are integrated in pumping units which consist of a drive (normally electric motor), a transmission (when needed), an electronic package (for monitoring and control), and the pump itself. The unit also has intake and outlet manifolds equipped with valves.

Stealth is the common name for not any specific system, but a collective term for a range of military low observability techniques. These techniques focus on reducing the signature of "stealthy" platforms of aircraft, missiles, ships, and ground vehicles for not being detected by hostile sensors. Ships have reduced observability compared to earlier. It is not seriously expected that an enemy would not know the ship was present, but its radar signature is sufficiently reduced to make it very hard for the final guidance radar of an anti-shipping missile to discriminate the ship from decoys and electronic self-protection signals intended to confuse the missile.

Ships are full of equipment generating significant acoustic noise which can be detected underwater when the sound is transmitted through the hull. Sound can travel very long distances under water so that ships can be detected by submarines, torpedoes and sonar dipped from helicopter. The greatest threat to a warship is from the silent submarine and sonar. For a modern warship, the importance of minimizing emitted sound cannot be underestimated. Any piece of machinery incorrectly mounted can provide tell-tale signs of its presence to the experienced sonar operator. In a combat situation, an active pulse can be detected and hence, passive sonar is used instead. Propellers are the noisiest part of a warship. As the propeller blades spin, they create a region of partial vacuum at the trailing edges.

While some of the techniques are highly classified, some of the methods known are covering the decks with rubberized tiles that absorb acoustic energy. The noisier parts of the propulsion system, such as reactor pumps and turbines, are mounted on acoustic isolation "rafts" using both springs and elastics. Extreme design and maintenance are applied to propellers to reduce cavitation, a phenomenon in which bubbles form, and then noisily collapse. There is a trend to move to pump-jets and other means of transferring power to water, much less noisily than with propellers. While it is not a stealth technique, both surface ships and submarines use acoustic decoys, often noisier than the ship itself.

The ultimate goal of a warship designer must be to make the ship's signature vanish into the background noise of the sea which comprises contributions from the weather, marine life and also other shipping from a wide geographical area. This leads to an important distinction in the types of noise that are generated by the various components of ship. These are termed self- and radiated noise. Self-noise which is the noise, from all shipboard sources, generated by the subject vessel. Radiated noise which is the noise generated by the ship and experienced at some point distant from the ship, by which its detection or recognition could be initiated. Designing the systems to minimize all these energies is called Stealth Technology. Out of these, the majority of the energy is released through Vibro-Acoustical Energy. Three sources of acoustical energy (noise) present in the underwater systems are 1) Machinery noise 2) Flow noise and 3) Propeller noise.

Machinery noise occurs due to the vibrations generated inside the machinery. Vibrations occur inside the systems of machineries due to various reasons such as the unbalances inside the machines of the power plants and engines, misalignment of the shafts, bearing defects etc. These vibrations can be reduced by suitable mounting systems. They can be further reduced by multi-level Isolation, Constrained and Free layer damping treatment, Tuned mass Damper, Magneto-Rhelogical Damper, Active and passive vibration control techniques.

## II. Literature review

Cavitation is an essential problem that occurs in any pump. It highly contributes to deteriorating the performance of the pump. In industrial applications, it is important to detect and decrease the effect of the cavitation in pumps. In this work, detecting and diagnosing the cavitation phenomenon within centrifugal pumps using vibration technique was investigated [1]. Dynamic performance for a vertical pumping unit was evaluated and enhanced by replacing The original electric motor of the pump unit by another one different in design and weights. Vibration has been increased greatly after installing the new motor. Consequently, it is necessary to estimate the change in the vibration characteristics owing to the difference in the boundary conditions of the new motor. Measured vibration levels and frequency analysis were dangerous at 1× due to resonance problem. Finite Element Analysis was used to model the motor structure in order to find its natural frequencies and mode shapes[2]. A method to optimize the design of a typical multistage centrifugal pump based on energy loss model and Computational Fluid Dynamics (ELM/CFD) was proposed. Different grid numbers, turbulence models, convergence precisions, and surface roughness are calculated for a typical multistage centrifugal pump. External characteristic experiments are also conducted to benchmark the numerical simulation. Based on the results, the ELM/CFD method was established including various kinds of energy loss in the pump, such as disk friction loss, volumetric leakage loss, interstage leakage loss as well as the hydraulic loss, which occurred at inlet section, outlet section, impeller, diffuser and pump cavity, respectively. The interactive relationships among the different types of energy losses were systematically assessed [3]. With the demand of reducing vibration and underwater noise caused by the air compressor, the design of an isolation device with the floating raft is described based on the comprehension of the excitation source. FEA models of the vibration isolation system in case of test bank status and installation in ship were established separately by means of the software ANSYS. The vibration modes, the transmission characteristics of the system as well as the shock response were analyzed [4].

Flow-induced noise is a significant concern for the design and operation of centrifugal pumps. The negative impacts of flow-induced noise on operating stability, human health and the environment have been shown in many cases. A comprehensive review of the flow-induced noise study for centrifugal pumps to synthesize the current study was presented. First, the generation mechanism and propagation route of flow-induced noise are discussed. Then, three kinds of study methodologies, including the theoretical study of hydrodynamic noise, numerical simulation and experimental measurement study, are summarized. Subsequently, the application of the three study methodologies to the analysis of the distribution characteristics of flow-induced noise is analyzed from aspects of the noise source identification and comparison, the frequency response analysis, the directivity characteristics of sound field and the noise changing characteristics under various operating conditions. After that, the analysis of the noise optimization design of centrifugal pumps were summarized[5]. The characteristics of flow instabilities as well as the cavitation phenomenon in a centrifugal pump operating at low flow rates were studied by experimental and numerical means, respectively. Specially, a three-dimensional (3D) numerical model of cavitation was applied to simulate the internal flow through the pump and suitably long portions of the inlet and outlet ducts[6]. Pumps play a significant role in industrial plants and need continuous monitoring to minimise loss of production. To date, there is limited published information on the application of acoustic emission (AE) to incipient pump cavitation. This paper presents a case study where AE has been applied for detecting incipient cavitation and determining the best efficiency point (BEP) of a 60 kW centrifugal pump. Results presented are based on net positive suction head (NPSH) and performance tests[7].

An experimental study about the application of acoustic emission (AE) techniques in the monitoring of cavitation erosion mass loss in small Francis turbines was presented. A vertical Francis turbine test bench is specially devised to perform some experiments designed to evaluate the influence of small surface mass losses on turbine blades in the acoustic emission signals. An AE wideband transducer is employed in the test bench instrumentation system. In order to evaluate the AE levels associated with the turbine erosion stages, a small defect is introduced into the turbine runner. This defect is intended to simulate a small mass loss in the turbine runner[8]. The noise induced by centrifugal pump is one of the main concerns in the design and optimization of onboard equipment for ships and submarines. In this work, an attempt has been made to estimate the natural frequency of centrifugal pump with an ordinary conventional steel material foundation. Later, the conventional foundation is replaced with composite material with carbon reinforcement. The vibration studies are accomplished by using FE software ANSYS 18.1. The natural frequencies of centrifugal pump with steel foundation are compared with natural frequencies of centrifugal pump with composite raft foundation. It is observed that the natural frequencies are enhanced by providing composite material foundation[9].

## III. MODELLING

### 3.1 GEOMETRY MODAL OF CENTRIFUGAL PUMP:

Centrifugal pump modal is created in Pro-e software and the IGS file is imported into ANSYS Workbench 2020R1 as shown in figure 3.1.

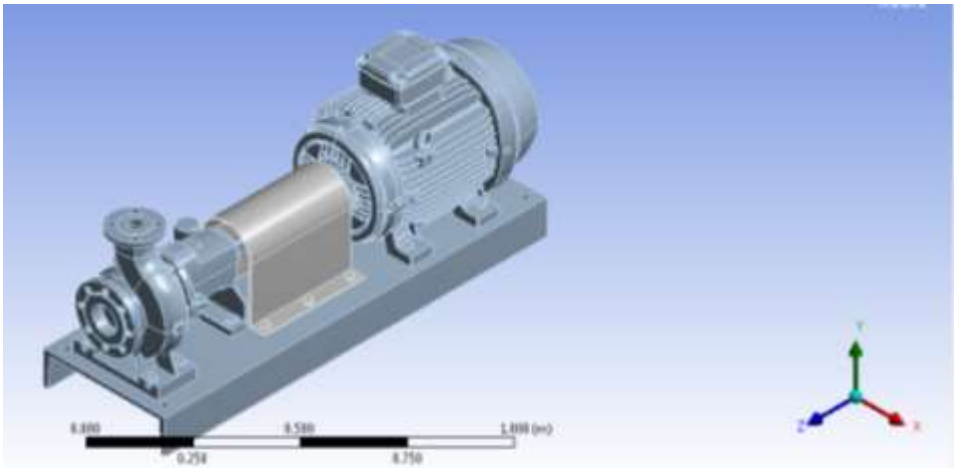


Figure 3.1. Centrifugal Pump Geometrical Model

### 3.2 MATERIAL PROPERTIES:

The engineering Data Source is the ANSYS Workbench tool used for this purpose and steel properties are shown in figure 3.2.

Properties of Outline Row 3: Structural Steel

| | A | B | C | D | E |
|---|---|---|---|---|---|
| 1 | Property | Value | Unit | | |
| 2 | Material Field Variables | Table | | | |
| 3 | Density | 7850 | kg m^-3 | | |
| 4 | Isotropic Secant Coefficient of Thermal Expansion | | | | |
| 6 | Isotropic Elasticity | | | | |
| 7 | Derive from | Young's Modul... | | | |
| 8 | Young's Modulus | 2E+11 | Pa | | |
| 9 | Poisson's Ratio | 0.3 | | | |
| 10 | Bulk Modulus | 1.6667E+11 | Pa | | |
| 11 | Shear Modulus | 7.6923E+10 | Pa | | |

Figure 3.2. Properties of Structural Steel

### 3.3 MESHING:

Creating the most appropriate mesh is the foundation of engineering simulations. ANSYS Meshing is aware of the type of solutions that will be used in the project and has the appropriate criteria to create the best suited mesh. ANSYS Meshing is automatically integrated with each solver within the ANSYS Workbench environment. For a quick analysis or for the ne and infrequent user, a usable mesh can be created with one click of the mouse. ANSYS Meshing chooses the most appropriate options based on the analysis type and the geometry of the modal.

A mesh is a network of line elements and interconnecting nodes used to modal a structural system and numerically solve the system for its simulated behavior under applied loads. The results are calculated by solving the relevant governing equations numerically at each of the nodes of the mesh.

The default mesh control that the ANSYS program uses may produce a mesh that is adequate for the modal you are analyzing. In this case, you will not need to specify any mesh controls. Mesh controls allows to establish such factors as the element shape, mid-side node placement, the element size to be used in meshing the solid modal. The step is one of the most important of r entire, analysis, for the decisions you make at this stage in your modal development ill profoundly affect the accuracy and economy of analysis.

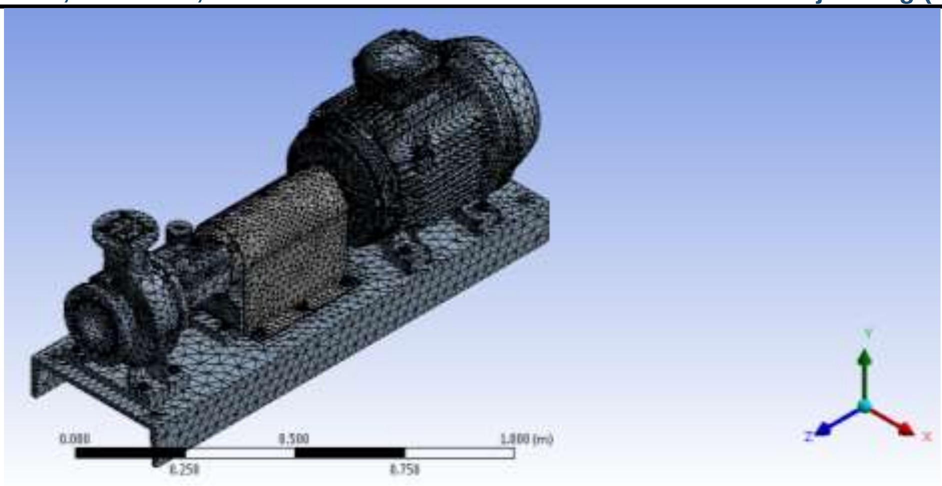


Figure 3.3. Geometric model after Meshing

### 3.4 MODAL ANALYSIS:

Modal analysis is performed to find natural frequency of a structure and mode shape of the structure at each frequency. Modal analysis assumes that the structure vibrates in the absence of any excitation and damping. Here the base of the pump is fixed and model with predefined engineering data was assumed.

Table 3.1. Natural frequency of the centrifugal pump

| mode no | frequency (Hz) |
|---|---|
| 1 | 109.88 |
| 2 | 343.65 |
| 3 | 378.43 |
| 4 | 395.88 |
| 5 | 464.87 |
| 6 | 484.43 |

## IV. SIMULATION STUDIES

### 4.1 GEOMETRY MODAL OF CENTRIFUGAL PUMP WITH STEEL MOUNT:

after simulating the structure without mount in the previous section, a steel base is considered in this section. The geometric model of centrifugal pump with steel base is as shown in figure 4.1 and the material properties considered are shown in figure 4.2.

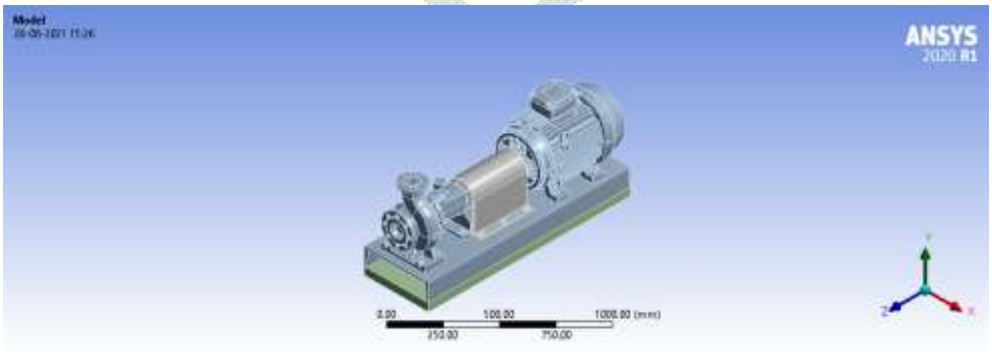
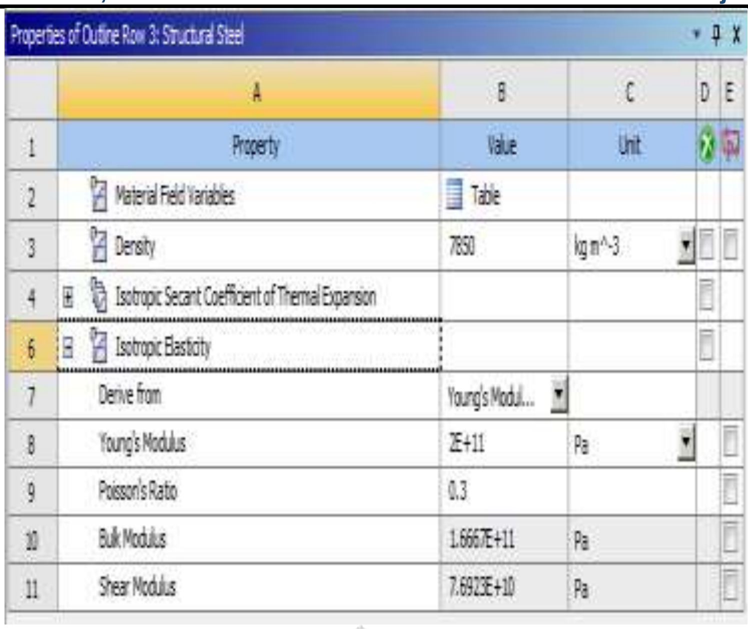



Figure 4.1. Geometry model of centrifugal pump with steel base



Properties of Outline Row 3: Structural Steel

| | A | B | C | D | E |
|---|---|---|---|---|---|
| 1 | Property | Value | Unit | | |
| 2 | Material Field Variables | Table | | | |
| 3 | Density | 7850 | kg m^-3 | | |
| 4 | Isotropic Secant Coefficient of Thermal Expansion | | | | |
| 6 | Isotropic Elasticity | | | | |
| 7 | Derive from | Young's Modul... | | | |
| 8 | Young's Modulus | 2E+11 | Pa | | |
| 9 | Poisson's Ratio | 0.3 | | | |
| 10 | Bulk Modulus | 1.6667E+11 | Pa | | |
| 11 | Shear Modulus | 7.6923E+10 | Pa | | |

Figure 4.2. Properties of Structural Steel for structure and mount

The meshed model of centrifugal pump with steel base is shown in figure 4.3

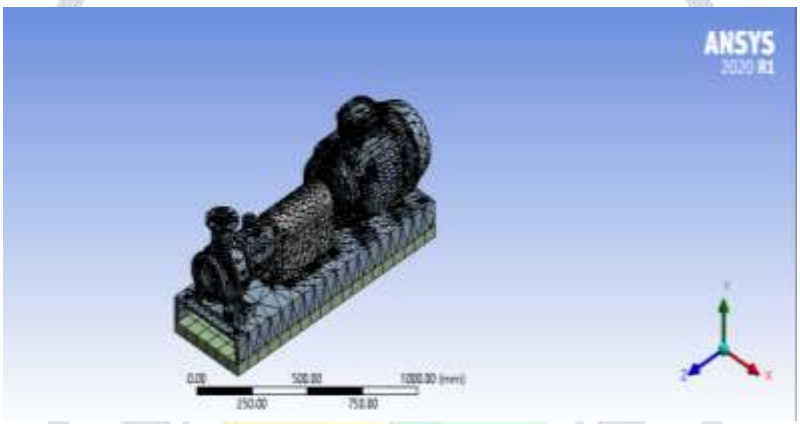



Figure 4.3 Geometric model with steel mount after Meshing

To conduct the simulation studies with steel mount, different thinness values are considered ranging from 10 mm to 50 mm. first six modes are considered in modal analysis and table 4.1 indicates the natural frequencies.

Table 4.1. Natural frequencies of centrifugal pump with steel base

| Mount Thickness (mm) | NATURAL FREQUENCY (Hz) | | | | | |
|---|---|---|---|---|---|---|
| | ω1 | ω2 | ω3 | ω4 | ω5 | ω6 |
| 10 | 88.13 | 150.7 | 184.8 | 250.27 | 298.31 | 324.57 |
| 20 | 88.73 | 150.45 | 183.88 | 250.29 | 296.72 | 323.84 |
| 30 | 88.77 | 149.9 | 183.54 | 249.67 | 296.55 | 323.06 |
| 40 | 89.733 | 150.35 | 183.98 | 249.91 | 298.23 | 324.38 |
| 50 | 92.232 | 150.2 | 183.88 | 249.62 | 298.77 | 324.11 |

It is observed that there is not much variation in natural frequencies for mount thickness 40mm and 50mm and hence the composite mount is designed for 40mm and 50mm thickness for comparison with the steel mount.

4.2 **GEOMETRY MODEL OF CENTRIFUGAL PUMP WITH COMPOSITE MOUNT**

To analyze the effect of natural vibration of mount when it is replaced with composite mount, composite with its fiber aligned in x-direction and y-direction are considered and the material properties are shown in figure 4.4

Properties of Outline Row 4: composit

| | A | B | C | D | E |
|---|---|---|---|---|---|
| 1 | Property | Value | Unit | | |
| 3 | Density | 1610 | kg m^-3 | | |
| 4 | Orthotropic Elasticity | | | | |
| 5 | Young's Modulus X direction | 1.9E+05 | MPa | | |
| 6 | Young's Modulus Y direction | 9900 | MPa | | |
| 7 | Young's Modulus Z direction | 9900 | MPa | | |
| 8 | Poisson's Ratio XY | 0.35 | | | |
| 9 | Poisson's Ratio YZ | 0.02 | | | |
| 10 | Poisson's Ratio XZ | 0.35 | | | |
| 11 | Shear Modulus XY | 7800 | MPa | | |
| 12 | Shear Modulus YZ | 7800 | MPa | | |
| 13 | Shear Modulus XZ | 7800 | MPa | | |

Fig 4.4. Material properties of composite base

The geometric model of centrifugal pump with composite base is as shown in figure 4.5 and the meshed model is shown in figure 4.6.

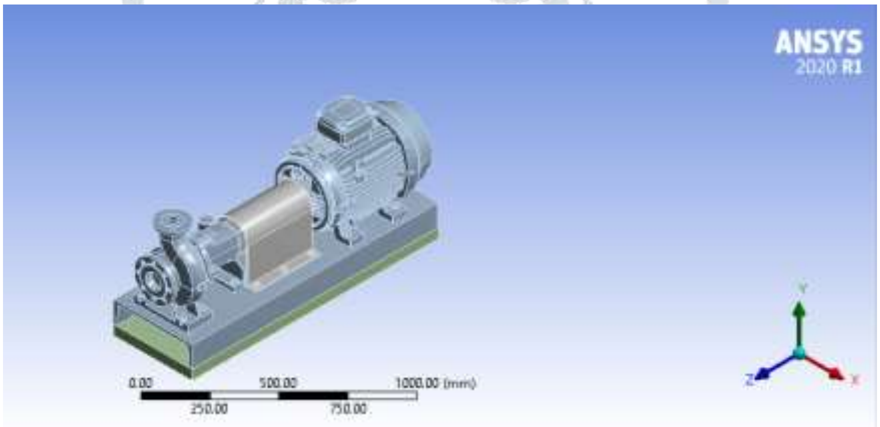



Figure 4.5. Geometry model of centrifugal pump with composite base

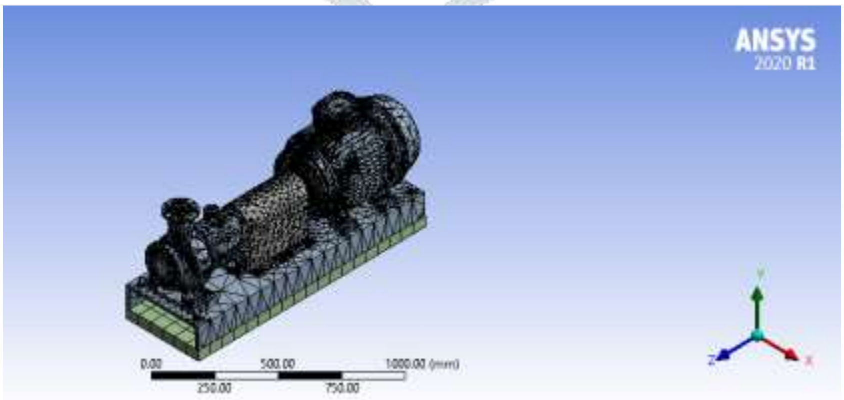



Figure 4.6. Geometric model with composite mount after Meshing

The natural frequencies for composite mount fiber in X-direction and Composite mount fiber in Y-direction are listed in table 4.2 and 4.3. first six mode shapes for model with 50mm thick composite mount are sown in figure 4.7.

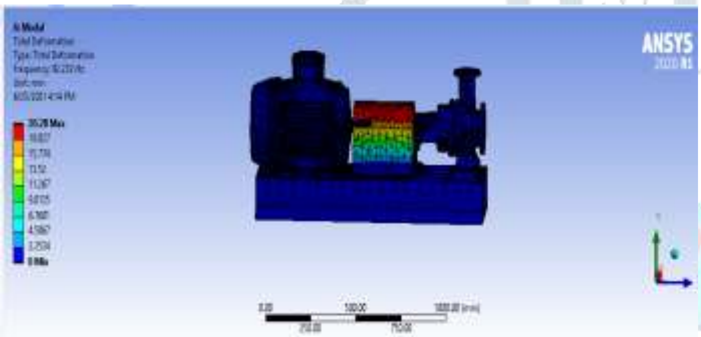
Table 4.2: Natural frequency of the centrifugal pump with composite mount fiber in X-direction

| Mount Thickness (mm) | Natural frequency (Hz) | | | | | |
|---|---|---|---|---|---|---|
| | ω1 | ω2 | ω3 | ω4 | ω5 | ω6 |
| 40 | 90.5 | 152.07 | 186.98 | 250.41 | 300.8 | 330.3 |
| 50 | 94.5 | 154.3 | 187.5 | 250.45 | 299.7 | 338.23 |

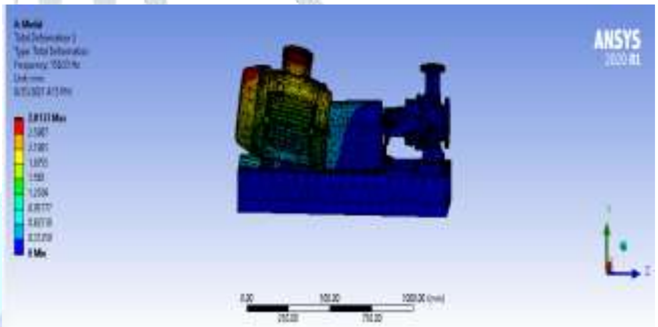
Table 4.3. Natural frequency of the centrifugal pump with Composite mount fiber in Y-direction

| Mount Thickness (mm) | Natural frequency (Hz) | | | | | |
|---|---|---|---|---|---|---|
| | ω1 | ω2 | ω3 | ω4 | ω5 | ω6 |
| 40 | 93.4 | 152.23 | 190.14 | 253.41 | 305.8 | 335 |
| 50 | 96.2 | 157.3 | 192.5 | 255.5 | 304.8 | 338.2 |

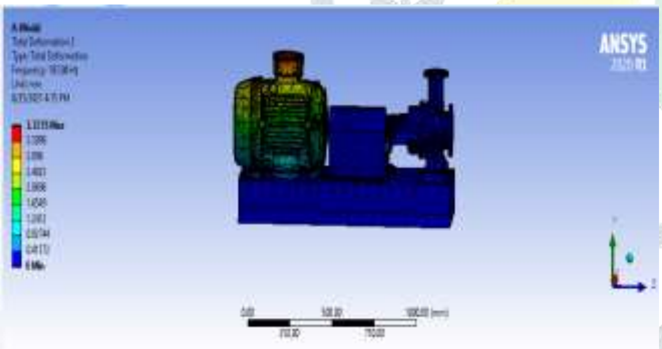
First six mode shapes of centrifugal pump with composite base at 50mm thickness and fibers in Y-direction are shown in figure 4.7



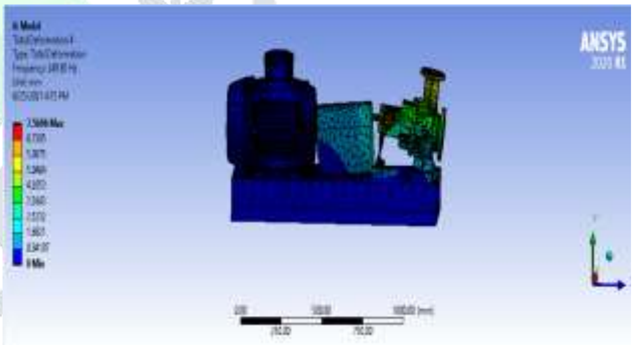
Mode 1



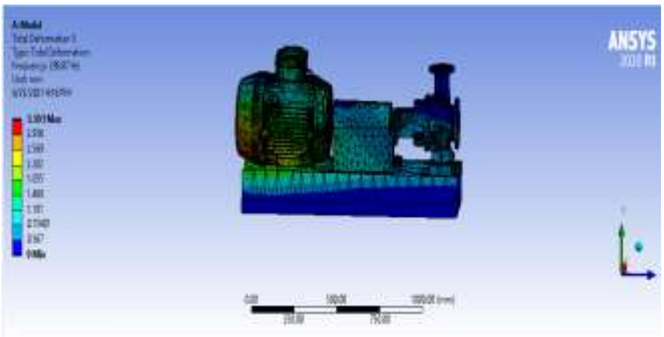
Mode 2



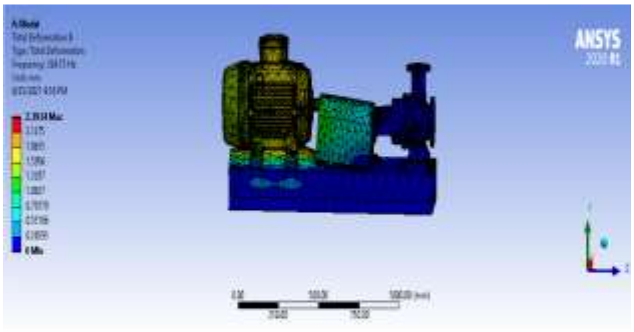
Mode 3



Mode 4



Mode 5



Mode 6

Figure 4.7. Modes base 50mm thickness in Composite Y-direction mount

# V. RESULTS AND DISCUSSION

## 5.1 COMPARISON OF NATURAL FREQUENCIES

The Comparison of natural frequency in centrifugal pump with 40 mm and 50 mm thickness mount of steel and composite raft mounts with fiber in x and y directions are presented in table 5.1 and 5.2 respectively, and it is observed that for a base with composite fibers in y-direction better results are obtained.

Table 5.1. Comparison of natural frequency in centrifugal pump with 40 mm thickness

| mode number | Steel mount Frequencies (Hz) | Composite mount with fibers in X-direction Frequencies (Hz) | Composite mount with fibers in Y- direction Frequencies (Hz) |
|---|---|---|---|
| 1 | 89.713 | 90.5 | 93.4 |
| 2 | 150.35 | 152.07 | 152.23 |
| 3 | 183.98 | 186.98 | 190.14 |
| 4 | 249.91 | 250.41 | 253.41 |
| 5 | 298.23 | 300.8 | 305.8 |
| 6 | 324.38 | 330.3 | 335 |

Table 5.2. Comparison of natural frequency in centrifugal pump with 50 mm thickness

| mode number | Steel mount Frequencies (Hz) | Composite mount with fibers in X-direction Frequencies (Hz) | Composite mount with fibers in Y- direction Frequencies (Hz) |
|---|---|---|---|
| 1 | 92.232 | 94.5 | 96.2 |
| 2 | 150.2 | 154.3 | 157.3 |
| 3 | 183.38 | 187.5 | 192.5 |
| 4 | 249.62 | 250.45 | 255.5 |
| 5 | 298.77 | 299.7 | 304 |
| 6 | 324.11 | 338.23 | 338.2 |

## 5.2 WEIGHT REDUCTION CALCULATION:

Density of steel material = 7850 kg/m$^3$

Density of composite material = 1610 kg/m$^3$

Weight reduction = (Weight of steel – Weight of composite) / Weight of steel

Weight of steel = mass * acceleration of gravity

W = m*g

Density = mass/volume

Mass = density* volume

Weight (W) = density * acceleration of gravity*volume

W =ρ* g*V

Acceleration of gravity is constant in both materials

Weight Volume of the base is constant

Weight reduction = (density of steel – density of composite)/ density of steel

Weight reduction = (7850 – 1610)/ 7850

= 0.7949 *100

= 79.49%

By using composite base mount we reduce the weight of the centrifugal pump is 79.49% compared to eight of centrifugal pump with steel mount.

## VI. CONCLUSIONS

Analysis on the centrifugal pump natural frequencies was conducted out by using ANSYS. The following are the conclusions derived from the present study

- The natural frequency of centrifugal pump with composite foundation is improved when compared to steel foundation.
- The natural frequency of centrifugal pump with composite base with fibres in Y-direction shown better results when compared to composite base fibres in X-direction.
- The weight of composite base is less. compare to the steel base
- Composite mount is suggested for reducing the vibration of centrifugal pump.
- The composite mount can be used in ships and submarines to reduce the vibration signature and improve stealth.

## REFERENCES

[1] Ahmed Ramadhan-Al-Obaidi, "Investigation of effect of pump rotational speed on performance and detection of cavitation within a centrifugal pump using vibration analysis", Heliyon, Volume 5, Issue 6, June 2019, pp 1-19.

[2] Dalia M. El-Gazzar, "Finite element analysis for structural modification and control resonance of a vertical pump", Alexandria Engineering Journal, 2017, 56, pp 695-707.

[3] ChuanWang, WeidongShi, XikunWang, XiaopingJiang, YangYang, WeiLiLingZhou, "Optimal design of multistage centrifugal pump based on the combined energy loss modal and computational fluid dynamics", Applied Energy, Volume 187, 1 February 2017, pp 10-26.

[4] WANG Guo-zhi, FANG Yuan-yuan, "Design of Vibration Isolation System for Air Compressors in Ship and Analysis of Its Anti-Shock Properties", Noise and Vibration Control, 2005.

[5] Chang Guo, Ming Gao and Suoying He, A Review of the Flow-Induced Noise Study for Centrifugal Pumps, Appl. Sci. 2020, 10, 1022; doi:10.3390/app10031022.

[6] Y Fu, J Yuan, S Yuan, G Pace, L D'Agostino, P Huang and X Li, (2014), "Numerical and Experimental Analysis of Flow Phenomena in a Centrifugal Pump Operating Under Low Flow Rates." J. Fluids Eng. 137(1), 011102.

[7] L. Alfayaz, D. MBA, G. Dyson, "The application of acoustic emission for detecting incipient cavitation and the best efficiency point of a 60KW mono block centrifugal pump", NDT&E International 38 (2005) 354- 358.

[8] Beyer, G. Smith, "Acoustic emission characteristics of a modal Francis turbine under cavitating and non-cavitating condition", Proceeding international symposium on cavitation noise, ASME, (1982) 87-92

[9] G.Vijay Kumar and P.Phani Prasanthi, "Analysis of composite raft mount for vibration reduction of a centrifugal pump", IOP Conf. Ser.: Mater. Sci. Eng. 1112 012001