GLOBAL VIBRATION ANALYSIS OF A 1900 TEU CAPACITIY CONTAINER SHIP

Yaşar GÜL [1], Levent KAYDIHAN [2]

[1] Naval Architect and Marine Engineer, M. Sc, General Manager, Delta Marine Enginnering. Co.
[2] Naval Architect and Marine Engineer, M. Sc, Department Manager, Delta Marine Enginnering. Co.

Özet: *Gemilerin aşırı titreşimler barındırmadan seferlerini sürdürebilmesi, gemi yapısı dayanımı ve yolcu konforu için çok önemlidir ve dizayn aşamasında incelenmesi gereklidir. Gemi titreşimlerini tetikleyici kuvvetler esas olarak pervaneden ve ana makinadan gelen uyarıcı kuvvetlerdir. Bu çalışmada, Delta Denizcilik Mühendislik tarafından Turkon A.Ş. için dizayn edilen 1900 TEU konteyner kapasiteli bir konteyner gemisinin global titreşim frekans ve şekilleri etrafındaki su kütlesi de hesaba katılarak hesaplanmış ve uyarıcı kuvvetler ile zorlanmış titreşim hesapları yapılarak elde edilen değerler ISO standartları limitleri ile kontrol edilmiştir.Bu makale gemi sahibine verilen raporun kısaltılmış halidir ve geminin kuru haldeki titreşimlerinin ıslak haldeki titreşimleri ile karşılaştırmasını da içerir.*

1. INTRODUCTION

The design of a ship construction without any excessive vibration is an important matter and needs to be investigated through analyses in the design stage. Three dimensional finite element method is a common procedure to obtain the ship's main vibration characteristics. Therefore, resonance frequencies can be obtained and through forced vibration analysis, the peak values of the displacements, velocities or accelerations can be checked with ISO standards.

The excitations induced by the propulsion system are the main source of the ship vibrations. These excitations from the propulsion system can be seen in can be occur in several ways. Dynamic forces such as thrust and moment variations from the propeller are transmitted to the hull through shaft bearings. The propeller induced fluctuating pressures on the hull surface induces vibration on the ship structure. The main and auxiliary engines can excite the ship's natural frequencies through dynamic forces transmitted from the foundations and supports. These excitations can cause the vibration of the hull girder, deckhouses, local structures and equipment. The order of the excitation frequencies from the propulsion system can be shaft rotational frequency (RPM) , blade frequency and their harmonics. The main engine induced unbalanced excitations for the slow speed diesel engines are the first and second order external forces and moments.

The response of the hull structure may be resonant or non-resonant. The hull structure will normally vibrate in the following modes;

- Vertical bending modes,
- Horizontal bending modes,
- Torsional modes,
- Longitudinal modes,
- Coupling modes between the horizontal and torsional modes, (especially for containerships)

Major substructures such as deckhouses and superstructures, which are in direct interaction with hull structure, can significantly influence the global vibration characteristics of the ship. Excessive vibration of a major substructure may be the result of a structural resonance frequency in the substructure.

A three dimensional finite element model representing the entire ship hull, including the superstructures, deckhouses and machinery propulsion system of 1900 TEU capacity container ship is developed to evaluate the vibration characteristics. The ship was designed by Delta Marine Engineering and built by Sedef Shipyard with the hull number of 151 for Turkon Co.. The main dimensions are as given below and the general arrangement can be seen in Figure 1.

MAIN PARTICULARS

Length overall	:	182.85 m
Length perpendicular	:	171.00 m
Breadth (moulded)	:	28.00 m
Depth (moulded)	:	16.10 m
Design draught	:	10.00 m
Scatling draught	:	11.00 m
Service speed	:	19.50 knot
Deadweight (at scantling draught)	:	26200 ton

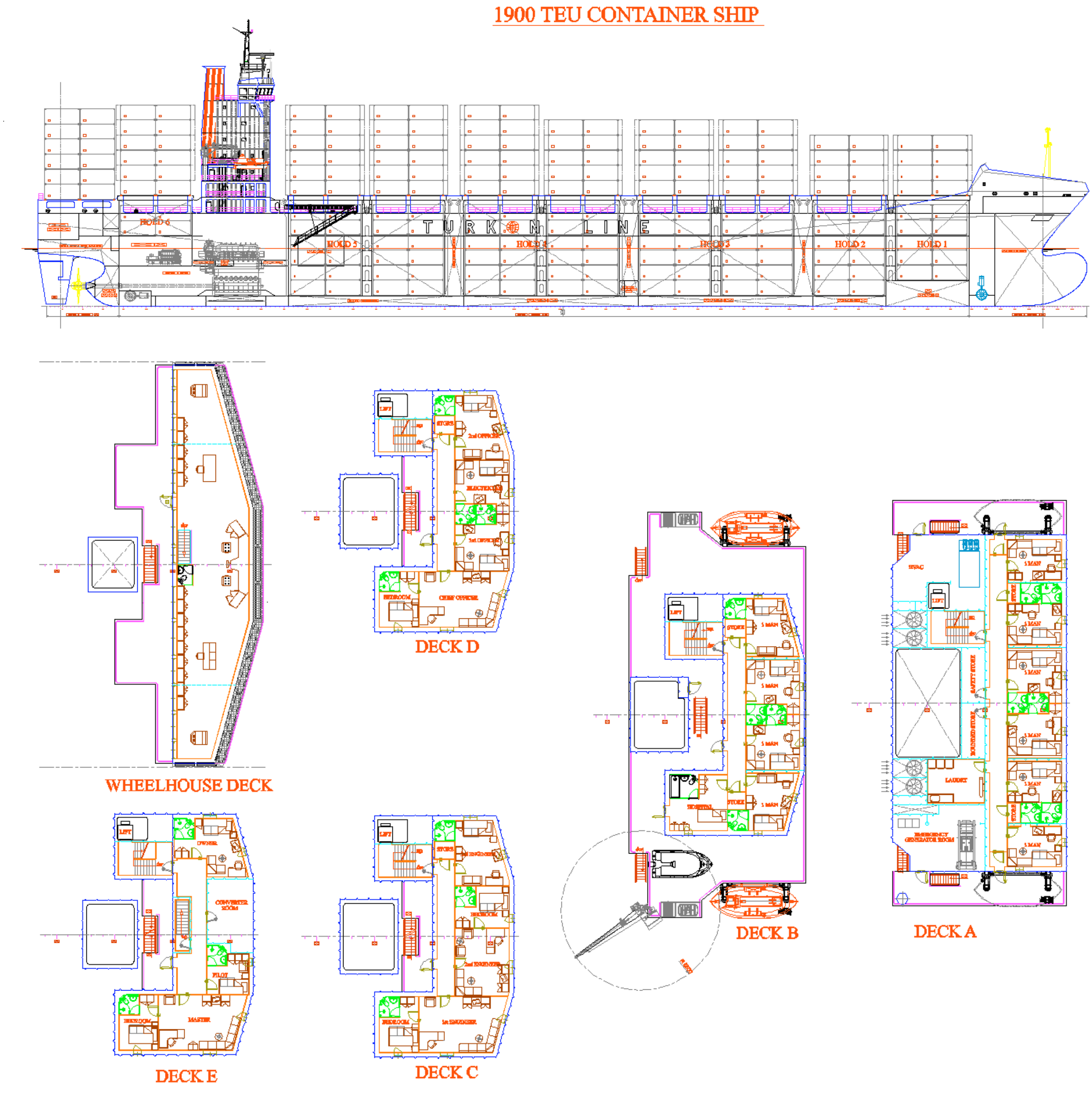

Figure 1 – General arangement

2. MODELLING STAGE

2.1 Finite Element Modelling Stage

Abaqus 6.6 general purpose finite element software is used for all modelling, analysing and post-processing operations.

Ship model is developed in two parts; aft and fore parts. Aft part consists of engine room, poop deck, aft peak and superstructure decks. Fine mesh density is used for aft part model. Fore part consists of cargo area, fore peak and forecastle deck. Due to the reason that the local vibration modes in fore part are not in concern, relatively coarse mesh density is used.

In order to include the hydrodynamic effects, surrounding seawater around the ship is also modelled. Acoustic medium in Abaqus is used to model the seawater. This element can be used for coupled acoustic structural analysis, sound propagation, emission, radiation and underwater explosion problems. The constitutive behaviour of the fluid is assumed to be inviscid with the constant density. Both the bulk modulus and density is defined for the seawater.

Tie constraint approach of the Abaqus is used to connect the different parts of the model, which are aft part, fore part and surrounding seawater parts. A tie constraint allows to fuse together two regions even though the meshes created on the surfaces of the regions may be dissimilar.

Cargo loading is applied as inertial mass elements by distributed along the cargo area's inner bottom plating and hatch coamings to obtain the specified loading condition. Ballast weights, heavy fuel oil and other detail tank weight are also applied by inertial mass element to their corresponding places to obtain the deadweight and it's longitudinal center of gravity.

Lightweight and it's longitudinal center of gravity is obtained by increasing some plating densities and adding some inertial mass element. Propeller is modelled by inertial mass element. Main engine and auxiliary engines are modelled by rectangular solid shapes which have same center of gravity and weight as its related equipment. In order to represent the weights in the superstructure such as coverings, added weights, lining and partition panels, the densities of superstructure decks are rearranged.

Stiffeners used in the aft part of structure are modelled by directly using the beam elements. While modelling the forepart stiffened plating, Abaqus Rebar reinforcement feature is used. Rebar is used to define layers of uniaxial reinforcement in membrane, shell, and surface elements. Stiffener properties of the plating are defined as physical property such as thickness and therefore, less mesh density could be used in the fore part.

Abaqus specific general purpose conventional S4 and S3 finite element are used for the plates. These elements allow transverse shear deformation. They use thick shell theory as the shell thickness increases and become discrete Kirchhoff thin shell elements as the thickness decreases; the transverse shear deformation becomes very small as the shell thickness decreases. These elements are 4 or 3 node stress/displacement shell elements and a large-strain formulation and have 6 degrees of freedom at each node.

Abaqus specific general purpose B31 element is used for stiffeners and for pillars in the aft part model. This element is 2 node linear timoshenko beam that allows for the transverse shear deformation. It can be used for thick as well as slender beams. Abaqus assumes that the transverse shear behavior of Timoshenko beams is linear elastic with a fixed modulus.The Timoshenko beams can be subjected to large axial strains. The axial strains due to torsion are assumed to be small.

Abaqus specific general purpose C3D elements are used for solid elements such as main engine. This element is 8 or 6 node linear brick stress displacement element with the 3 degrees of freedom at each node. Hourglass control, incompatible modes and reduced integration features can be chosen.

Abaqus specific acoustic element AC3D elements are used for seawater parts. Acoustic elements model the propagation of acoustic waves and are active only in dynamic analysis procedures. In general, analysis with acoustic elements should be thought of as small-displacement linear perturbation analysis, in which the strain in the acoustic elements is strictly volumetric and small. For solid structures interacting with air or water, the initial stress (if any) in the air or water has negligible physical effect on the acoustic waves. Most engineering acoustic analyses, transient or steady state are of this type. This element has only one degrees of freedom, which is acoustic pressure.

After the meshing operations are completed, it is seen that the finite element model has 176030 nodes, 176800 structural elements and 60300 acoustic elements in total. The finite element model can be seen in Figures 2 to 7. In the finite element model, x is longitudinal, y is lateral and z indicates the vertical direction.

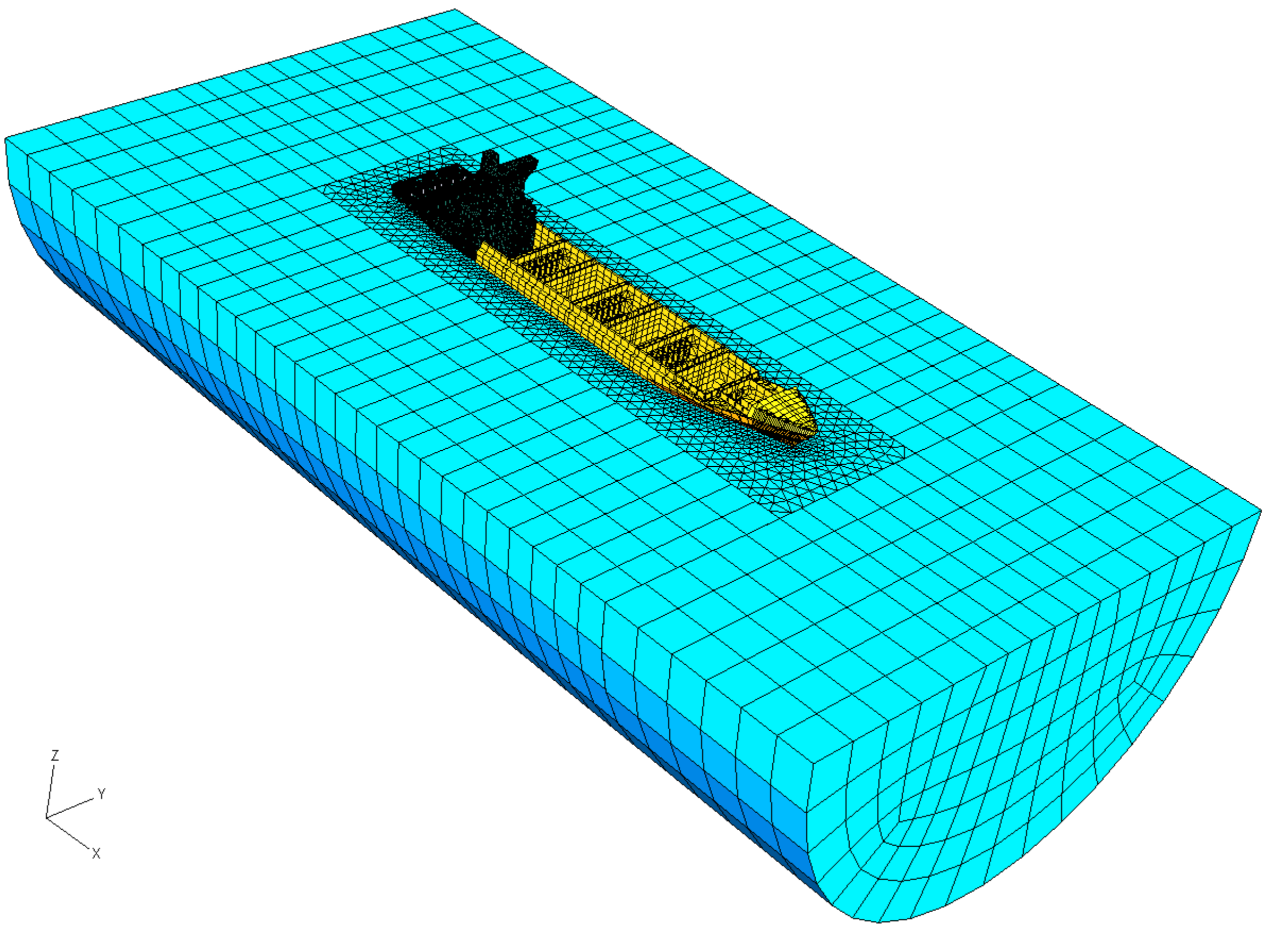

Figure 2 – Finite Element Model

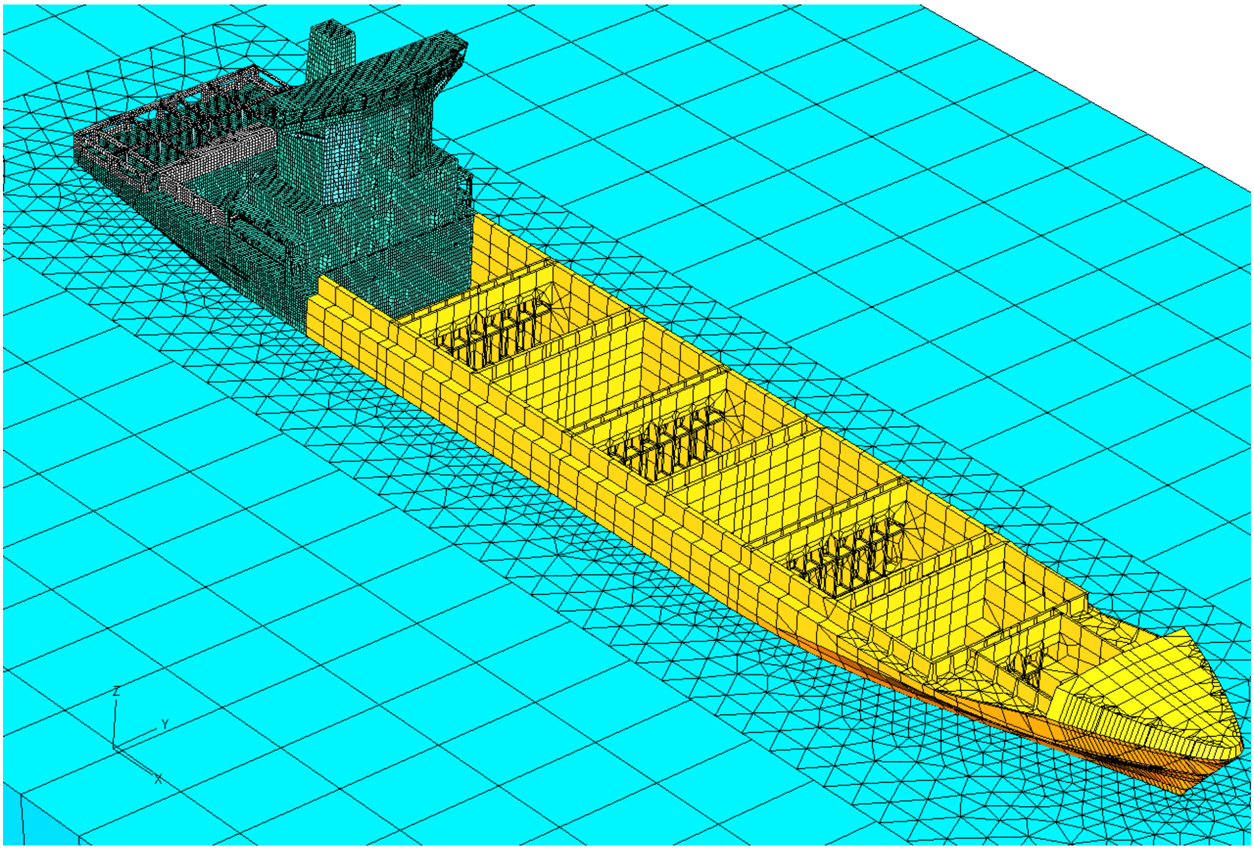

Figure 3 – Finite Element Model

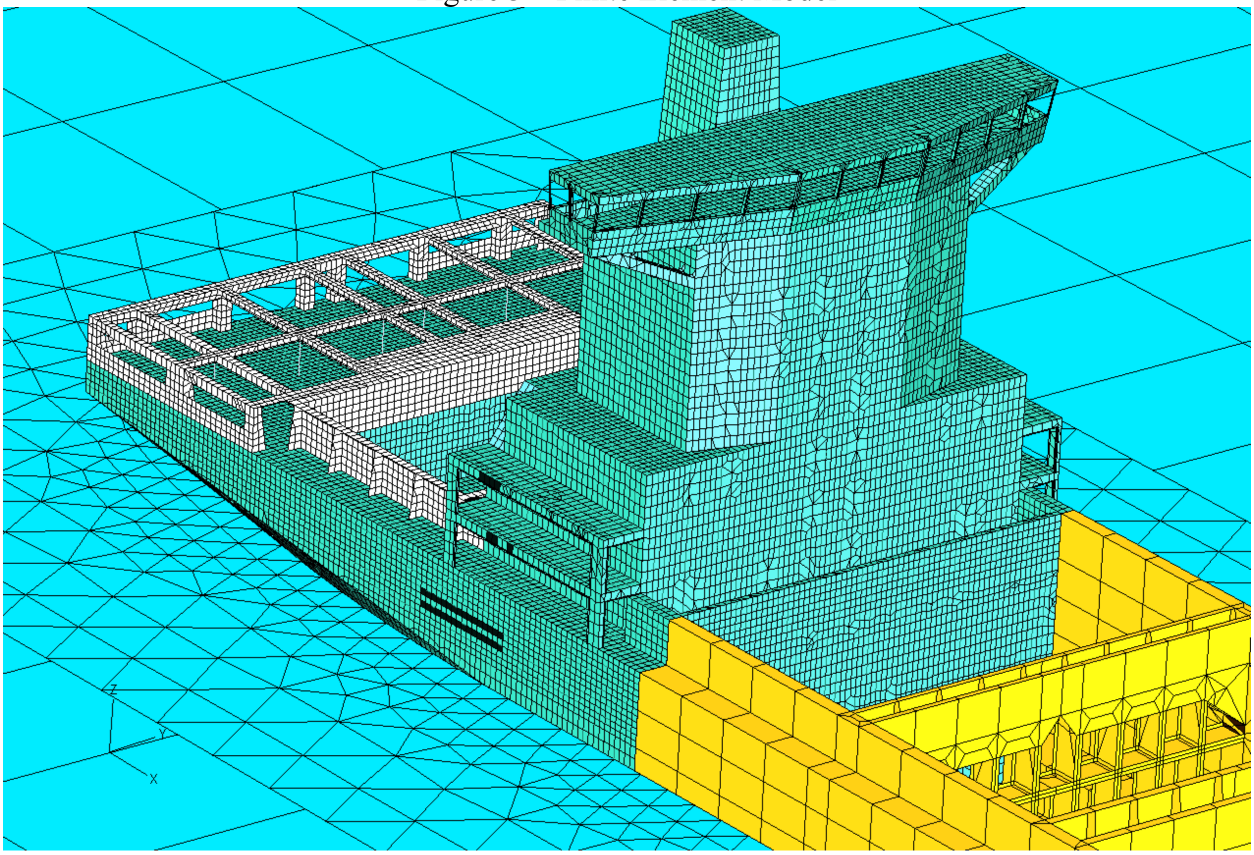

Figure 4 – Finite Element Model

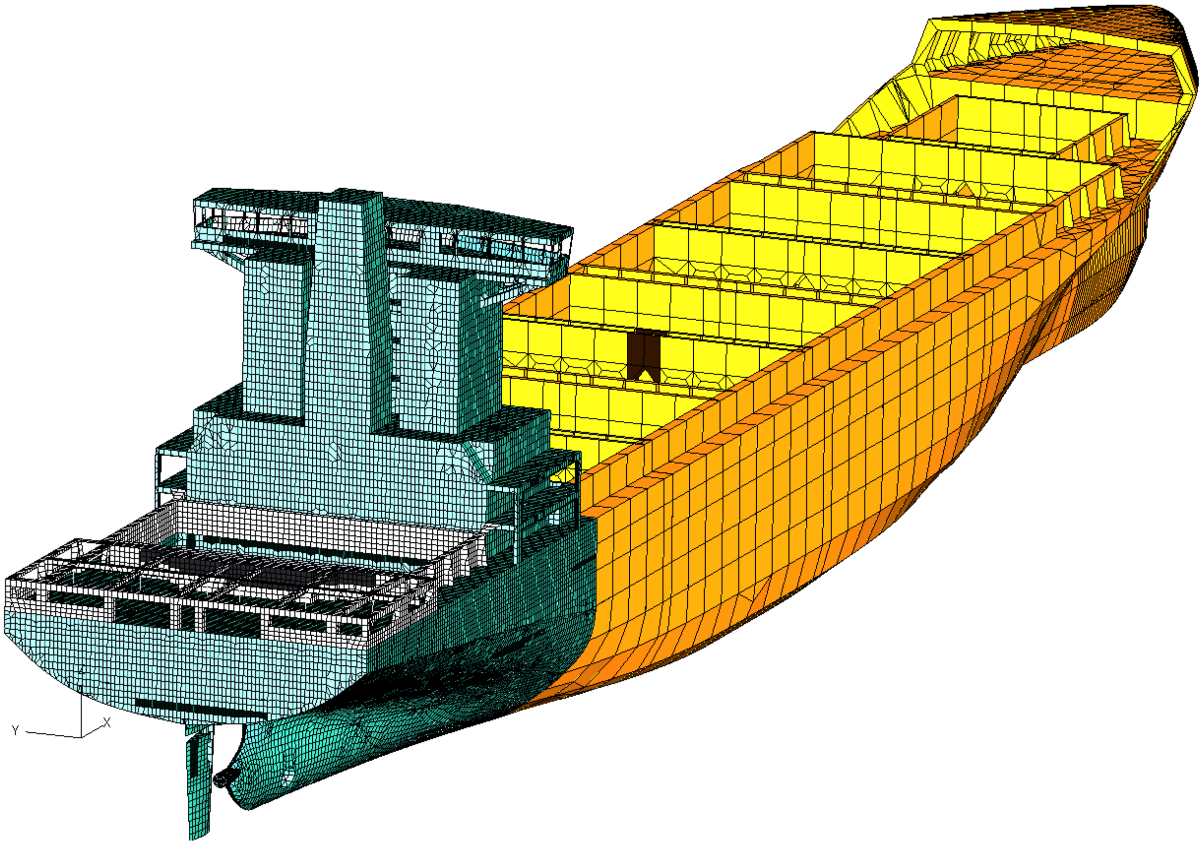

Figure 5 – Finite Element Model

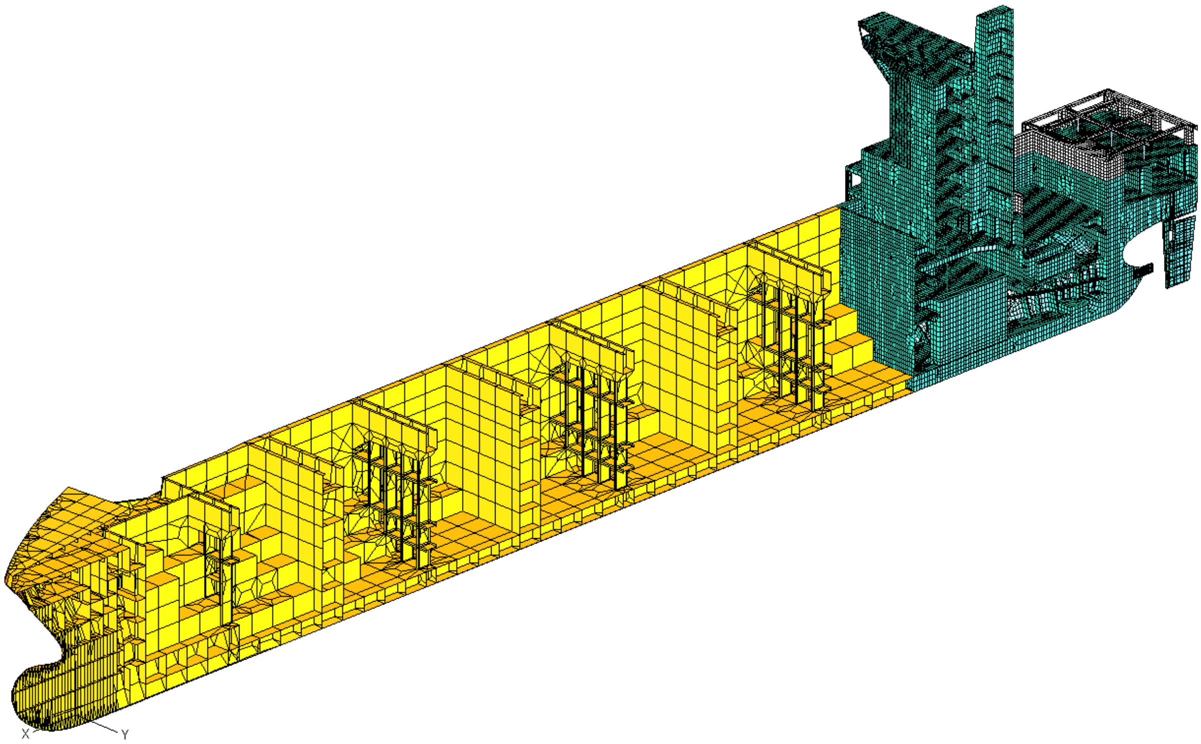

Figure 6 – Finite Element Model

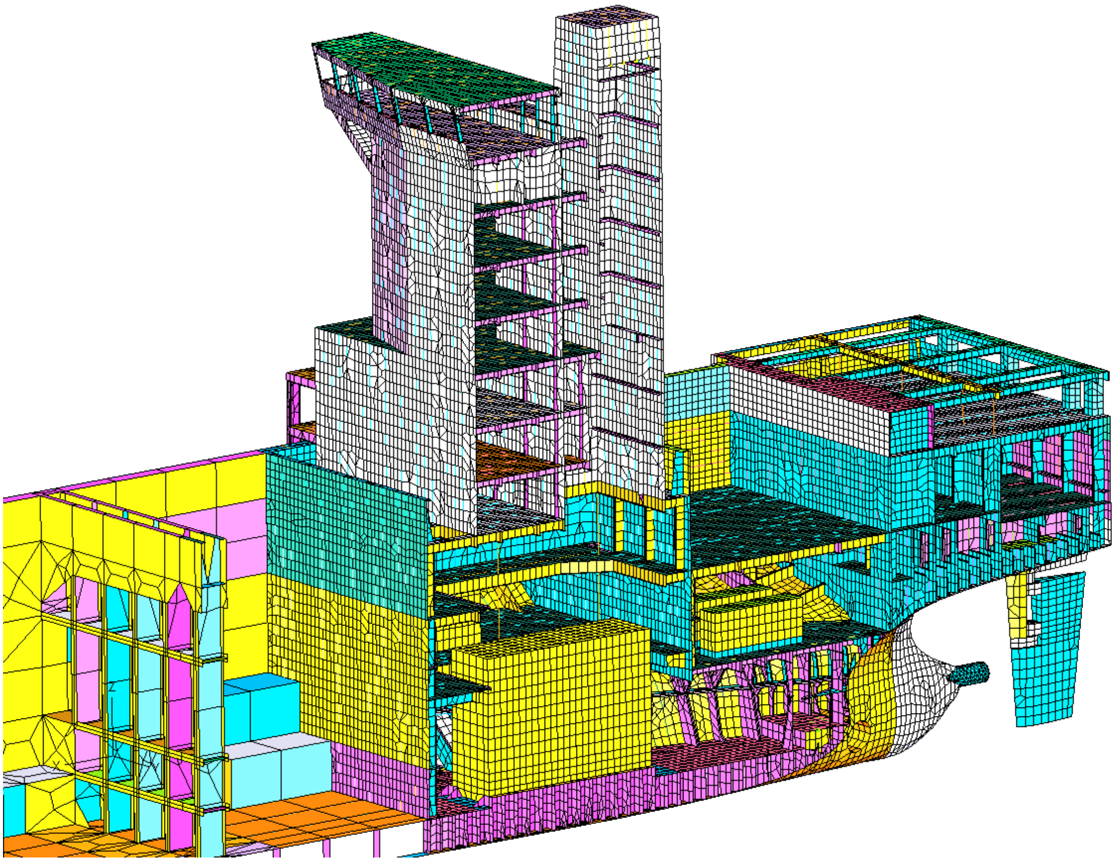

Figure 7 – Finite Element Model

2.2 Material Properties

The material for the steel used in the ship is St 42 grade shipbuilding steel and its properties are given below;

- Elasticity modulus = 210000 N/mm2
- Poisson ratio = 0.30
- Density = 7850 kg/m3

The material properties for seawater are given below;

- Bulk modulus = 2300 N/mm2
- Density = 1025 kg/m3

3. BOUNDARY & LOADING CONDITIONS

3.1 Boundary Conditions

Zero acoustic medium pressure is applied along the waterline surface of seawater parts. Setting the pressure to zero represents a "free surface" where the pressure does not vary because of the motion of the surface. Default boundary condition for the outer surfaces of seawater cylindrical parts is stationary rigid wall. No boundary conditions are applied to structural part of the finite element model.

3.2 Loading Conditions

3.2.1 Ship Weight Loading Condition for Design Draught

The ship is loaded with the loading case of "Full loading with design draught of 10 meter". This loading case can be seen in Figure 8 below. Ship's and model's weights and longitudinal center of gravity values can be seen in Table I below.

3.2.1.1 Deadweight Distribution

- Cargo loading of 17150 ton,
- Ballast weights of 3021 ton,
- Heavy fuel oil of 1886 ton,
- Marine diesel oil of 165 ton,
- Fresh water of 206 ton,
- 165 ton of other tank weights

Cargo loading is distributed along the cargo area's inner bottom plating and hatch coamings and ballast HFO, MDO, FW and other tanks weights are distributed to their corresponding places to obtain the deadweight and it's longitudinal center of gravity.

3.2.1.2 Lightweight Distribution

Lightweight and it's longitudinal center of gravity is obtained by changing necessary densities. Propeller is modelled by inertial mass element. Main engine and auxiliary engines are modelled by rectangular solid shapes which have same center of gravity and weight as its related equipment. In order to represent the weights in the superstructure such as coverings, added weights, lining and partition panels, the densities of superstructure decks are rearranged.

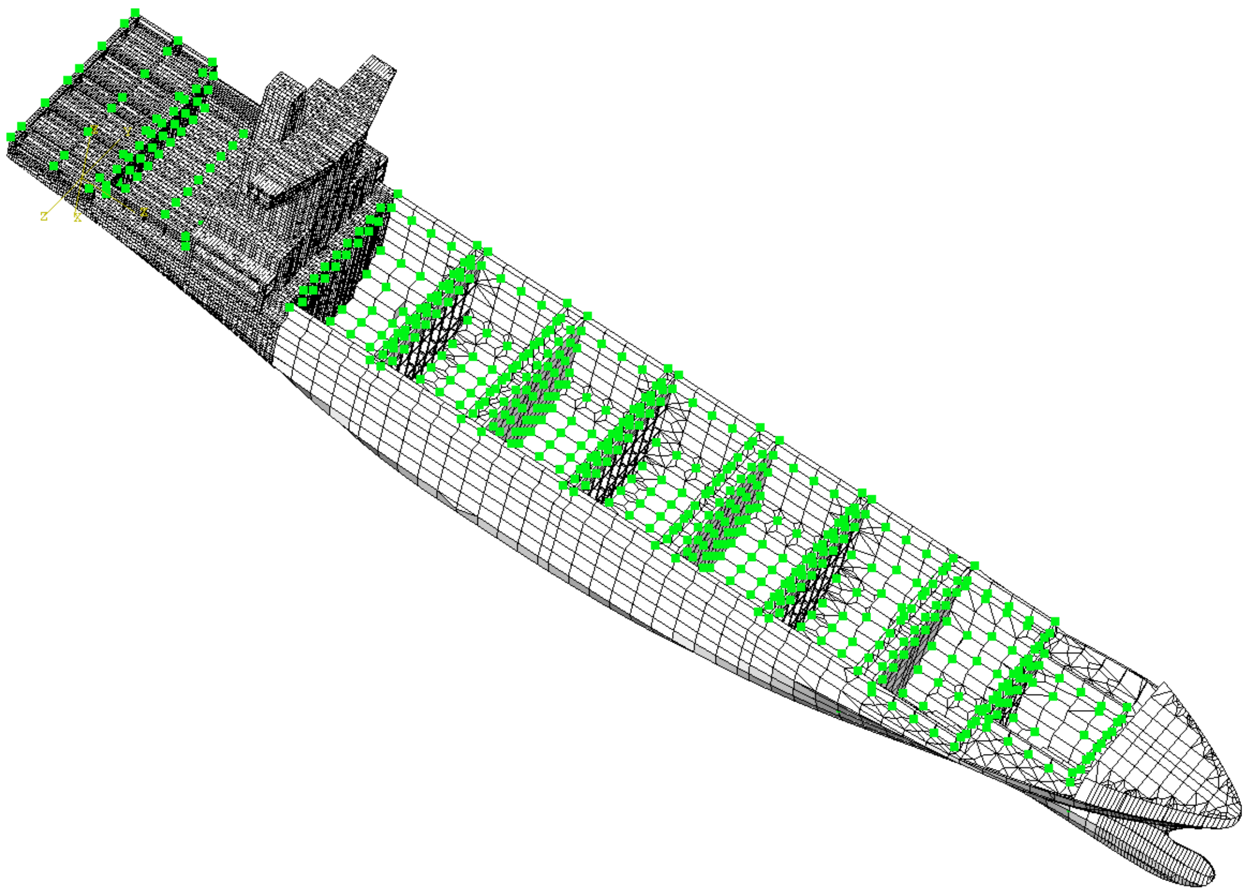

Figure 8 – Loading of Finite Element Model

Table I – Ship's and Model's Weight and LCG Values

		Weight (ton)	LCG (m)
	Lightweight	9000	75
Ship	Deadweight	22595.7	81.79
	Total weight	31595.7	79.85
FE Model	Total weight	31520	80.3

3.2.2 Loading Conditions for Forced Vibration Analysis

The resonance event occurs when the natural frequencies of structure encounter with the excitation frequencies. Main excitation sources are;

- Propeller induced pressures on the hull,
- Propeller induced longitudinal thrust variations,
- Main engine dynamic loading;

  - Main engine external moments,
  - Main engine H moments,
  - Main engine X moments.

Propeller features;

- Propeller = Rolls Royce CPP
- Diameter = 5900 mm
- Number of Blades = 4
- RPM = 127 rpm
- Pressure pulse level = 3.8 kPa Given by Rolls Royce at MCR condition (single amplitude at blade frequency).

Main Engine Features;

- Engine = MAN B&W 8S50MC-C
- Power = 13260 kW
- Engine RPM = 127 rpm

Propeller RPM is 127 rpm that is 2.116 Hz. Propeller induced pressures and thrust variations are effective at first blade frequency of 8.46 Hz and second harmonic of blade frequency of 16.93 Hz. Pressure pulse level of the propeller is given as 3.8 kPa from Rolls Royce at MCR condition. This is the single amplitude value at blade frequency. Propeller pressure is to affect to an area on the hull above the propeller. For the second harmonic of blade frequency, the pressure force value is assumed as the half of the value for the blade frequency.

Thrust variations of the propeller caused by the non-uniform wake field are taken from Reference 3 and 1. $0.03*T_0$ for blade frequency and $0.018*T_0$ for 2nd harmonic of blade frequency are accepted as thrust variation value by assuming that the propeller is working in relatively good wake field. T_0 is the thrust value of propeller.

As can be seen from the Engine guide of MAN B&W 8S50MC-C, main engine dynamic forces are most effective at 1st order (vertical and horizontal), 3rd order, 4th order and 5th order of engine rpm of 127 rpm that is 2.116 Hz. Therefore, six loading cases are obtained. Details of these loading cases can be seen below.

3.2.2.1 – 1st Order Loading Case (Vertical)

- Frequency Range to be analysed = 0 Hz < 2.116 Hz < 4 Hz,
- Main engine 1st order external vertical moment = 192 kNm,
- Main engine 1st order X type moment = 141 kNm.

3.2.2.2 – 1st Order Loading Case (Horizontal)

- Frequency Range to be analysed = 0 Hz < 2.116 Hz < 4 Hz,
- Main engine 1st order external horizontal moment = 192 kNm,
- Main engine 1st order X type moment = 141 kNm.

3.2.2.3 – 3rd Order Loading Case

- Frequency Range to be analysed = 5.5 Hz < 6.35 Hz < 7 Hz,
- Main engine 3rd order X type moment = 457 kNm.

3.2.2.4 – 4th Order Loading Case

- Frequency Range to be analysed = 7 Hz < 8.46 Hz < 9.5 Hz,
- Main engine 8th order H type moment = 269 kNm,
- Propeller blade frequency hull pressure force in vertical direction ,
- Propeller thrust variation amplitude at blade frequency in longitudinal direction.

3.2.2.5 – 5th Order Loading Case

- Frequency Range to be analysed = 9.5 Hz < 10.58 Hz < 12 Hz,
- Main engine 5th order X type moment = 689 kNm.

3.2.2.6 – 8th Order Loading Case

- Frequency Range to be analysed = 16 Hz < 16.93 Hz < 18 Hz,
- Main engine 4th order external horizontal moment = 60 kNm,
- Main engine 4th order X type moment = 279 kNm,
- Propeller 2nd blade frequency hull pressure force in vertical direction ,
- Propeller thrust variation amplitude at 2nd blade frequency in longitudinal direction.

4. ANALYSES & RESULTS

4.1 Natural Frequency Analysis

ABAQUS can compute both real and complex eigensolutions for purely acoustic or structural-acoustic systems, with or without damping.

In a coupled acoustic-structural model, real-valued coupled modes are extracted by default in an eigenfrequency extraction procedure. While all the modes extracted in a coupled structural-acoustic analysis include the effects of fluid-solid interaction, some of them may have predominantly structural contributions while others may have predominantly acoustic contributions. Extraction of the coupled acoustic-structural modes is supported only for the Lanczos eigenvalue extraction method.

Optimum length, width and depth values of the seawater acoustic medium volume, which have minimum effect on results, are used in the solutions. Optimum seawater element size around the ship structure, which have minimum effect on results, are used in the mesh of seawater acoustic medium.

Vibration modes of ship structure in frequency range of 0-25 Hz are calculated for both wetted and dry finite element model. The wetted natural frequencies could be seen in Figure 9 to 21 below. Comparison between the wetted and dry modes also can be seen in Table II below.

Table II – Comparison Between the Wetted and Dry Vibration Modes

Vibration Modes	Wetted Frequency	Dry Frequency
Mode 1 - 1st Torsional Mode (Figure 9)	0.998 Hz	1.118 Hz
Mode 2 - 1st Bending Mode (Figure 10)	1.015 Hz	1.33 Hz
Mode 3 - Horizontal Bending and Torsional Mode (Figure 11)	1.328 Hz	1.51 Hz
Mode 4 - 2 nd Bending Mode (Figure 12)	2.059 Hz	2.67 Hz
Mode 5 - 2 nd Torsional Mode (Figure 13)	2.325 Hz	2.546 Hz

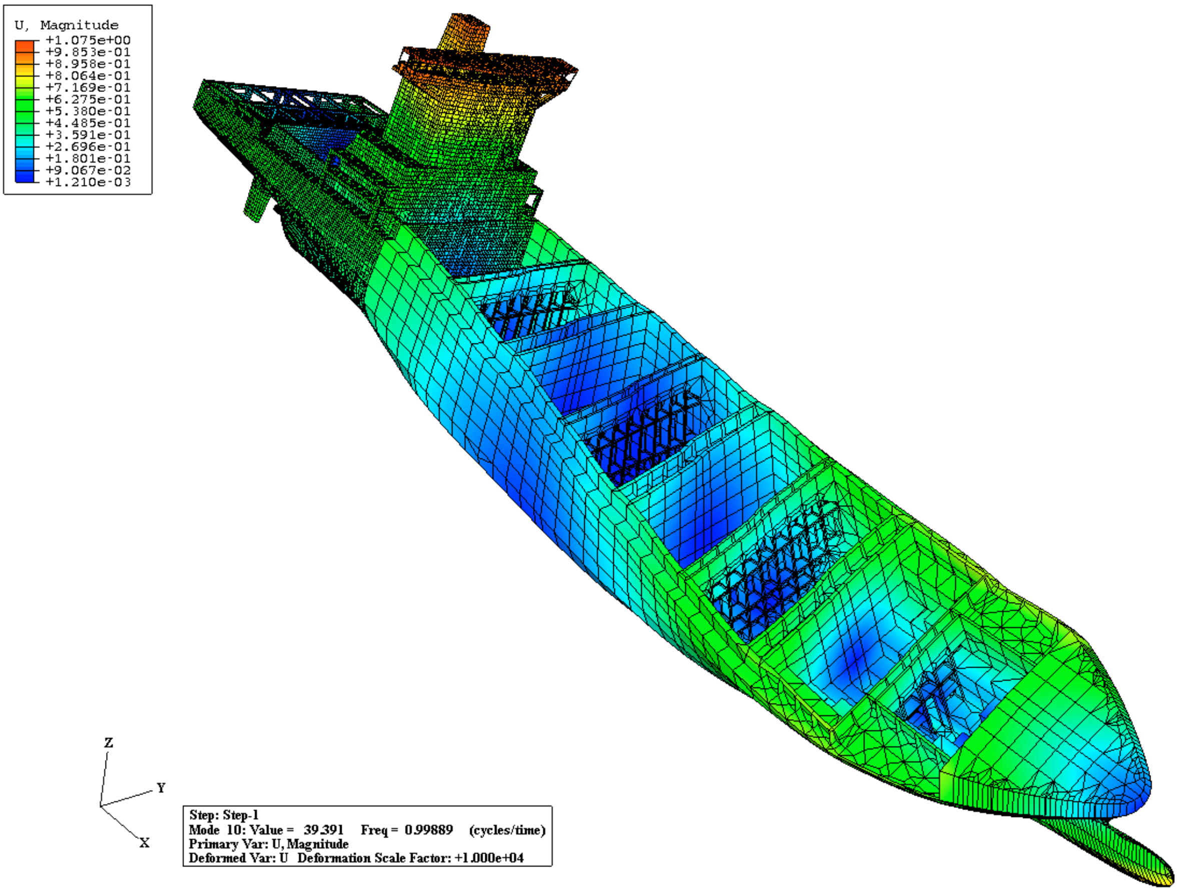

Figure 9 – Global Torsional Mode at 0.998 Hz

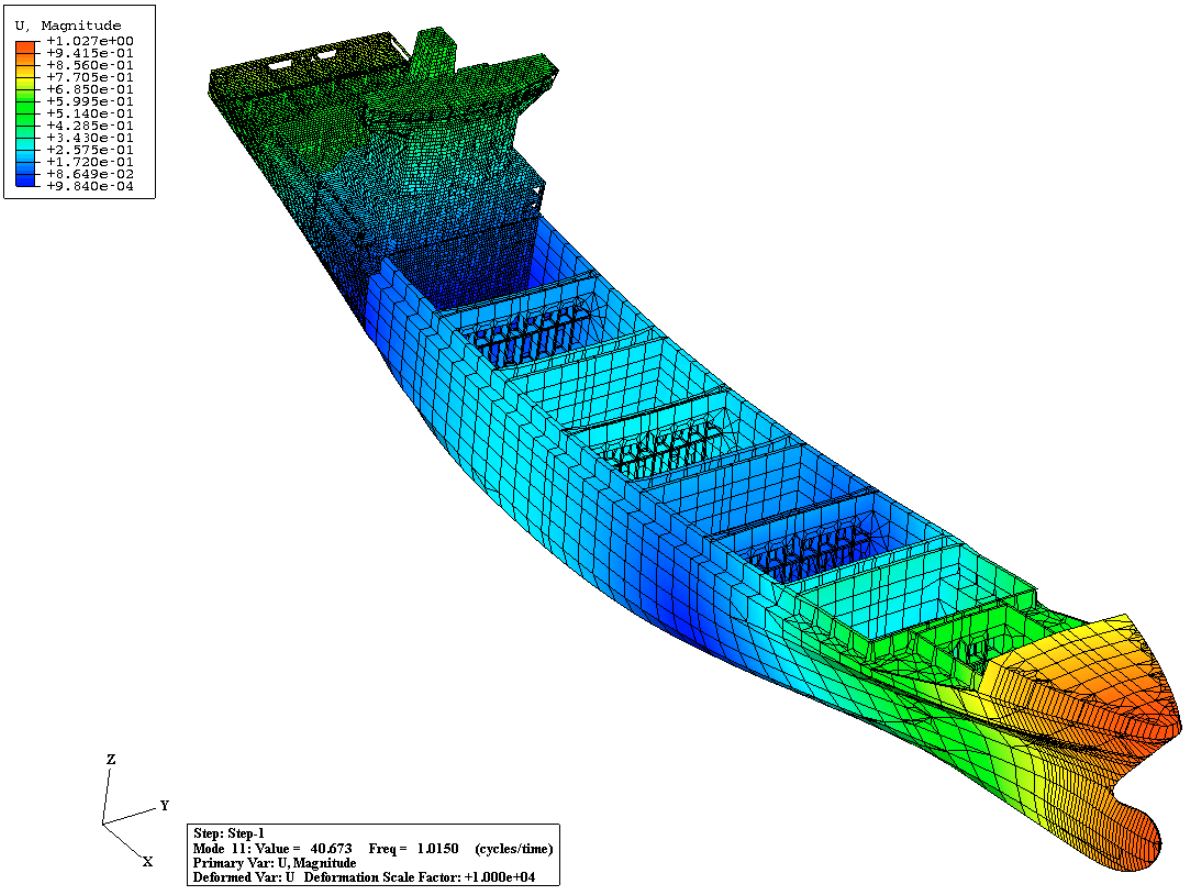

Figure 10 – Global Vertical Bending Mode at 1.015 Hz

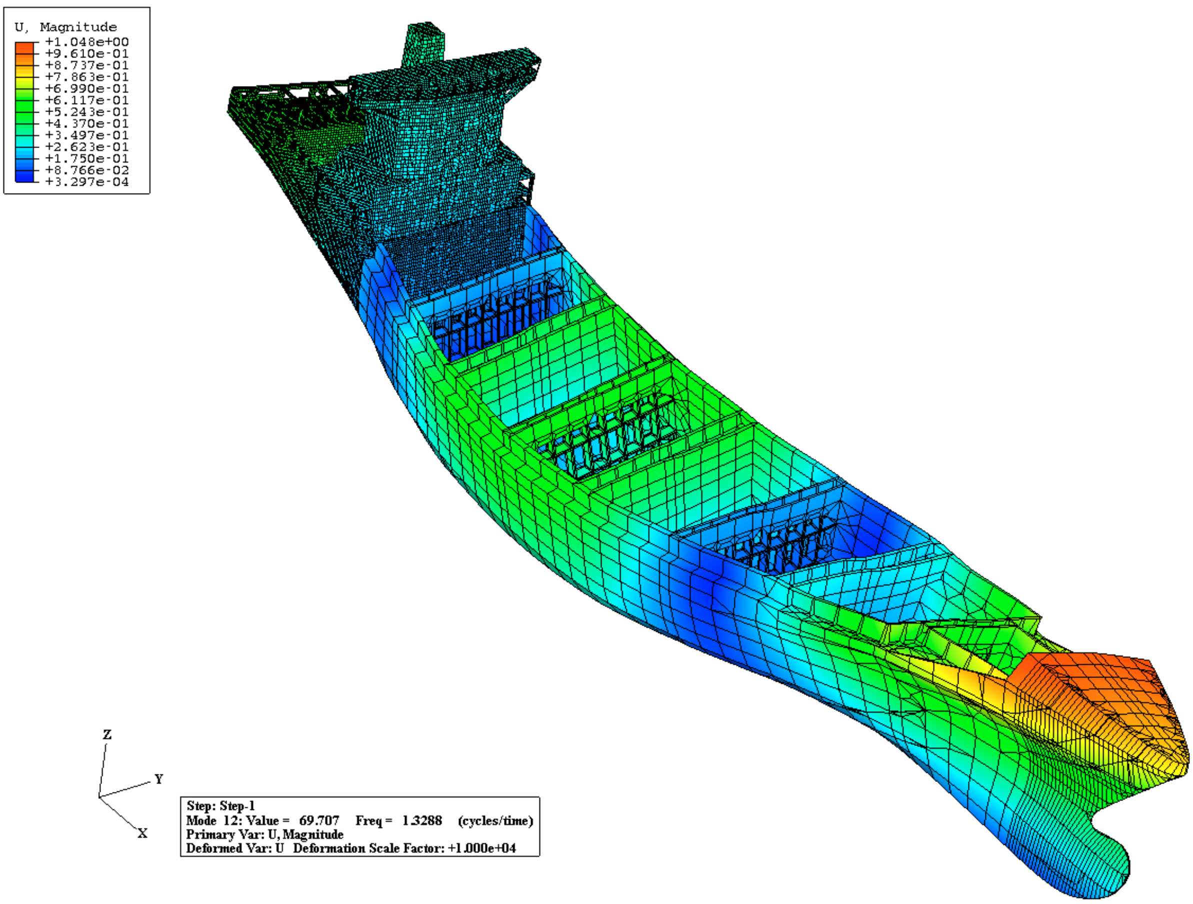

Figure 11 – Global Torsional and Horizontal Bending Mode at 1.328 Hz

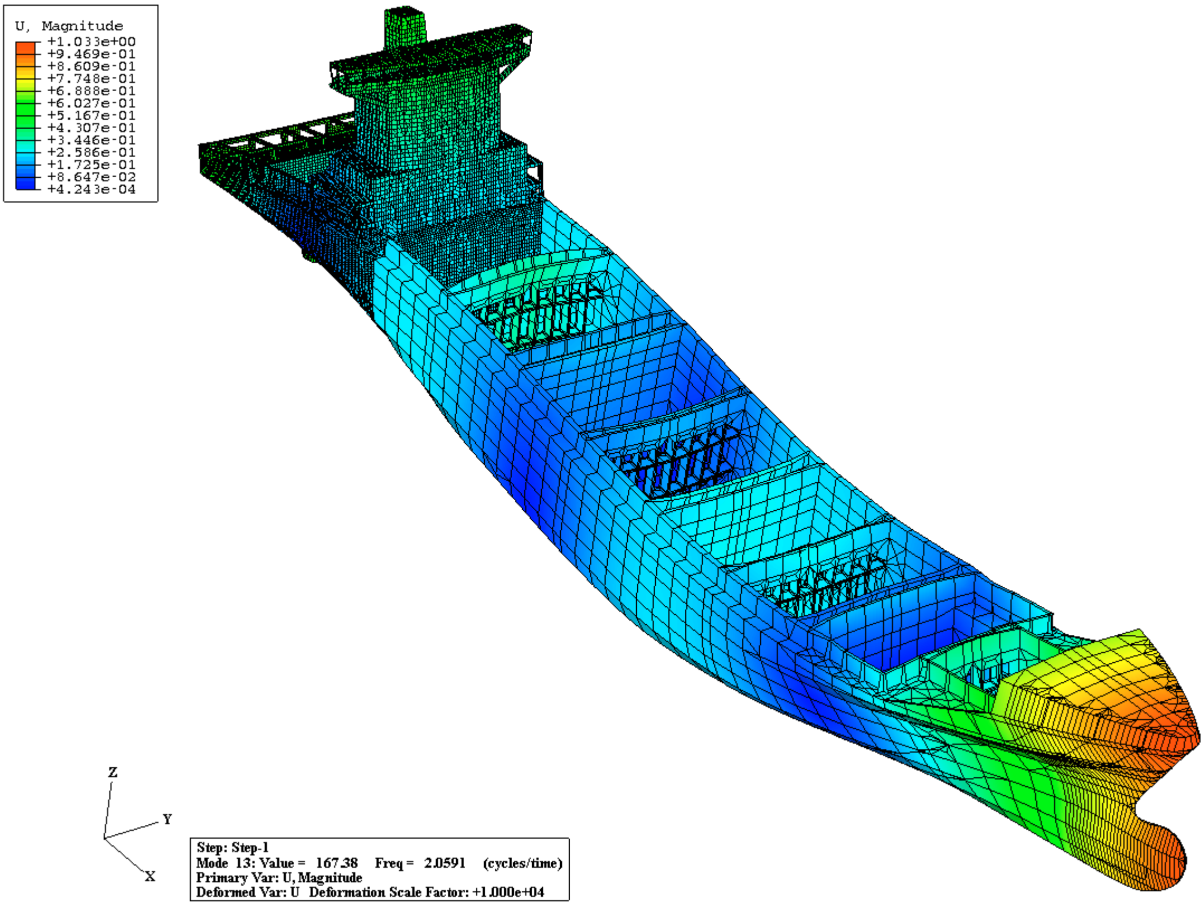

Figure 12 – Global Bending Mode at 2.059Hz

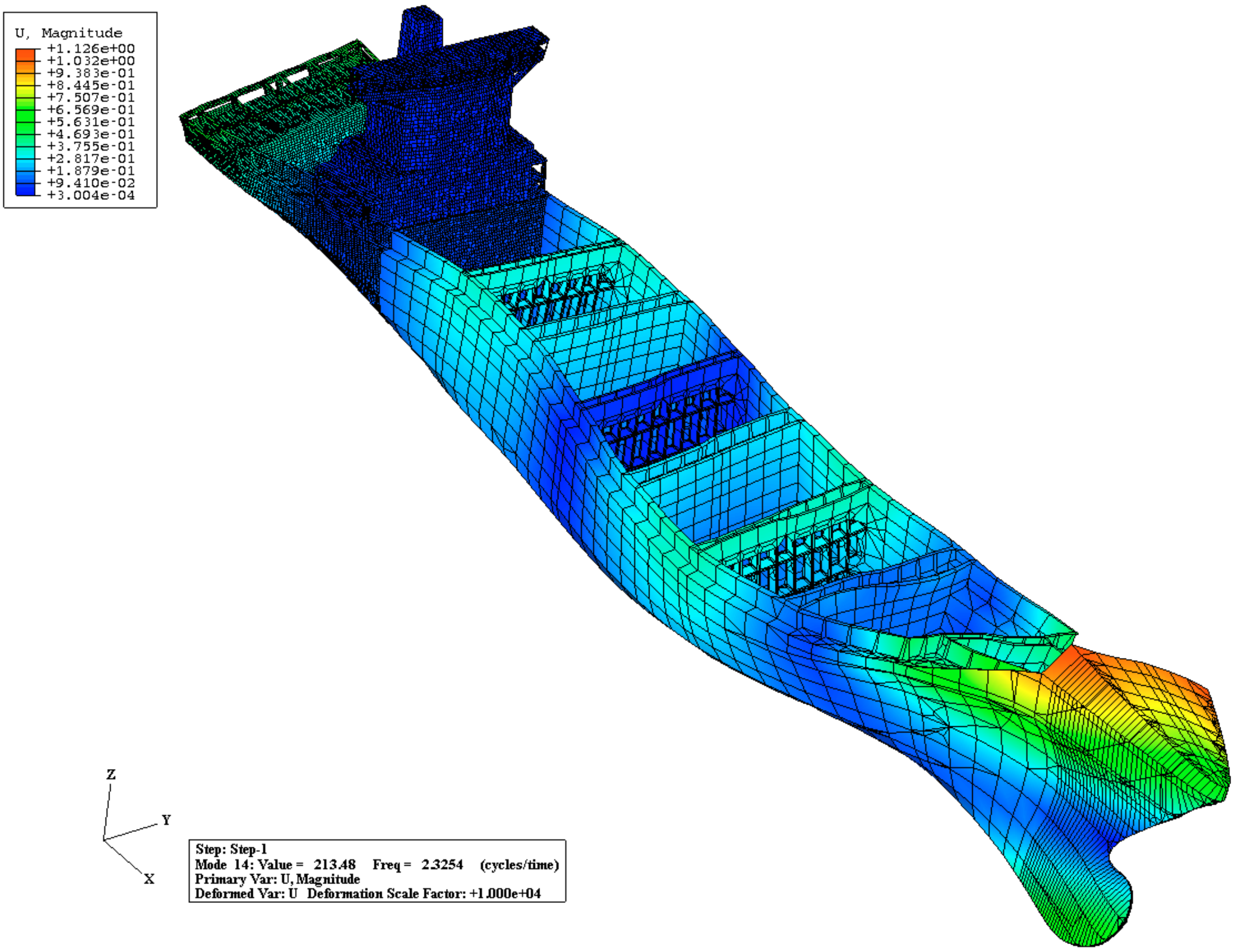

Figure 13 –Global Torsional Mode at 2.325 Hz

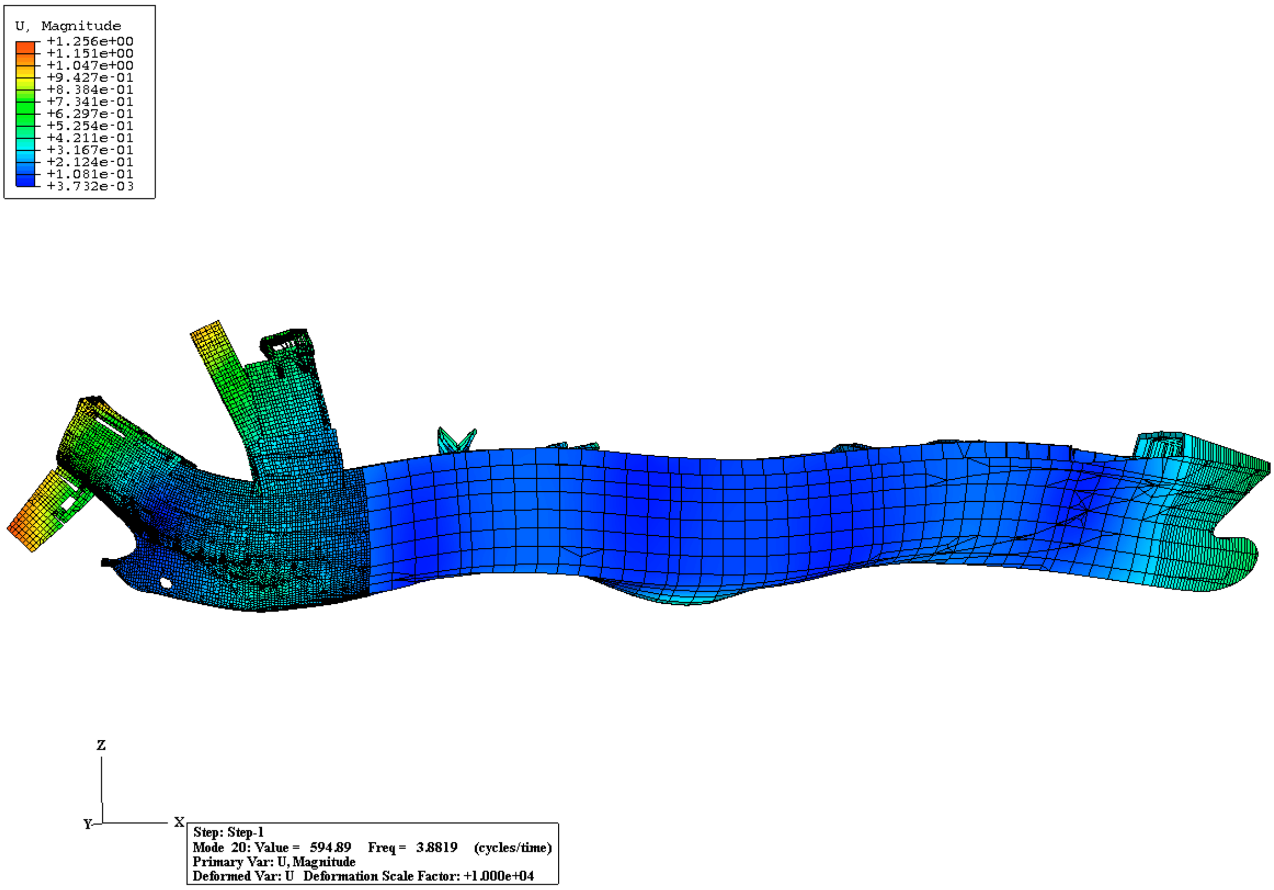

Figure 14 – Global Mode at 3.881 Hz

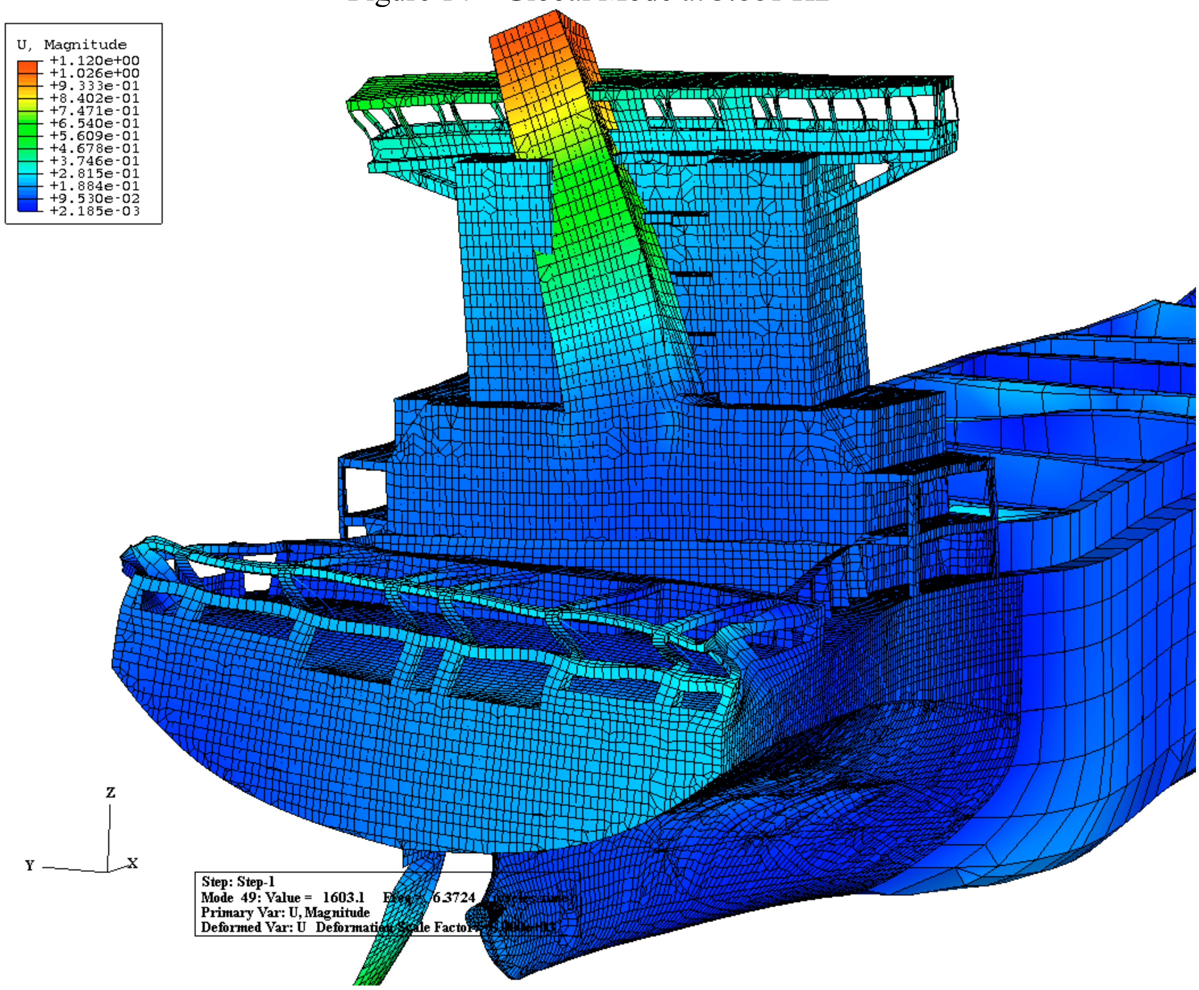

Figure 15 - Superstructure Mode at 6.354 Hz

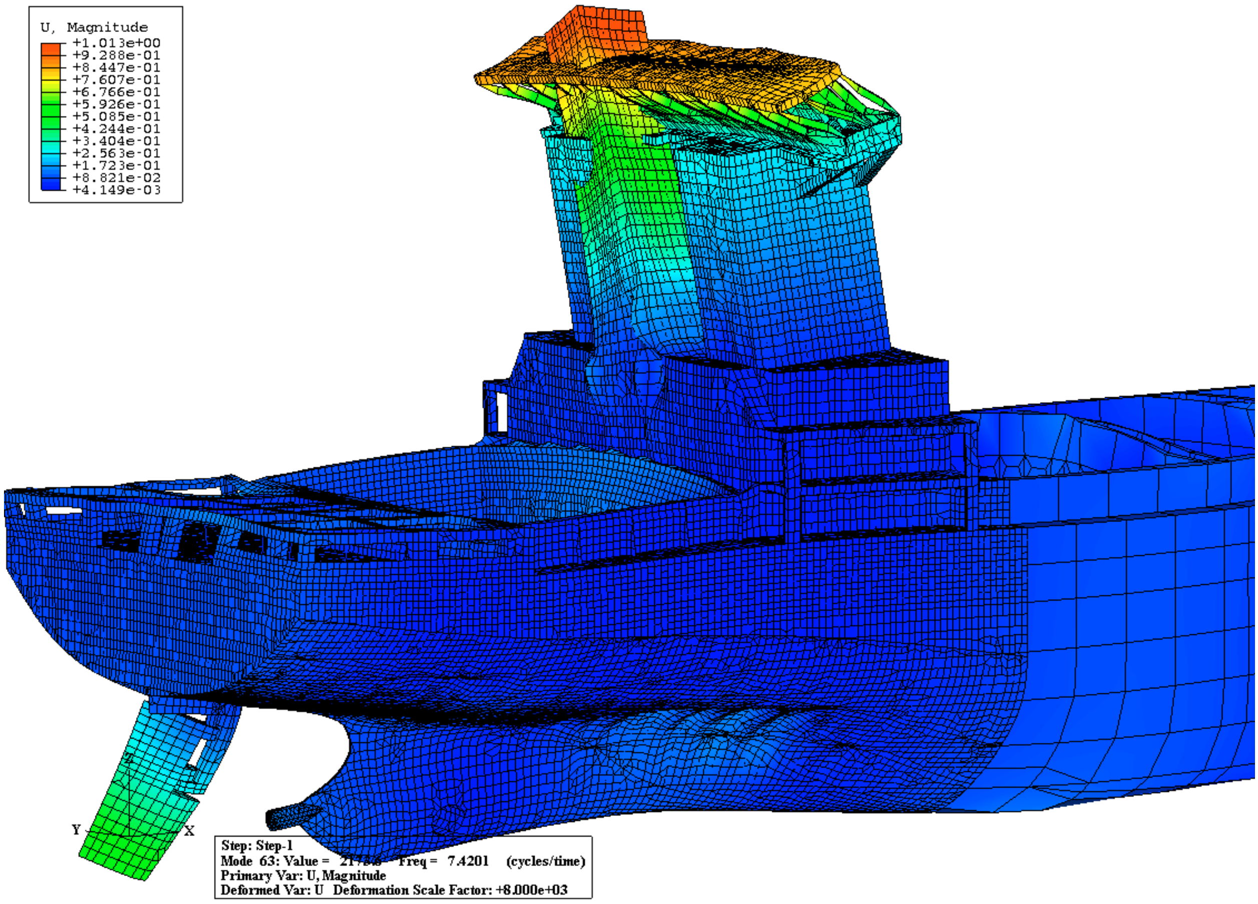

Figure 16 – Superstructure Mode at 7.420 Hz

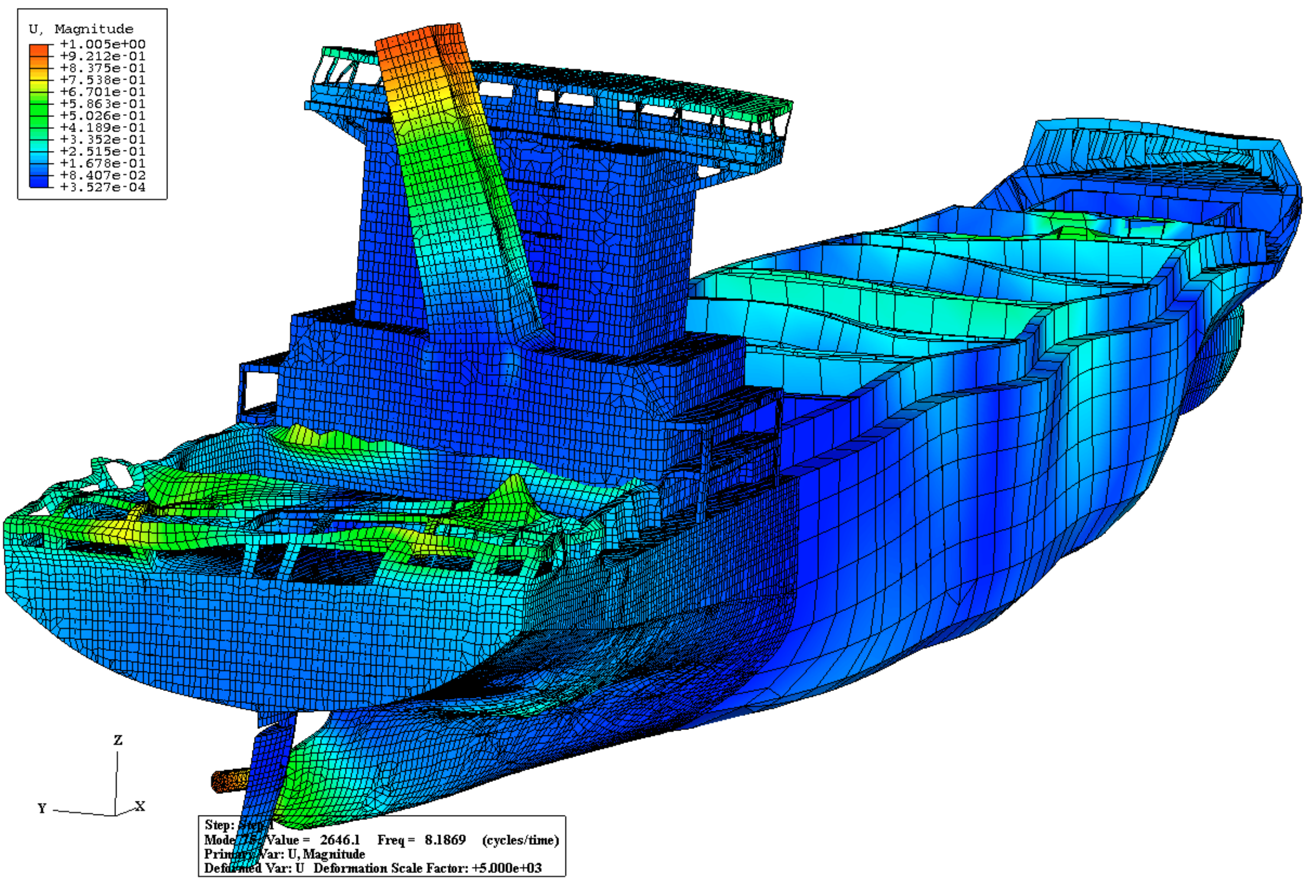

Figure 17 – Superstructure Mode at 8.186 Hz

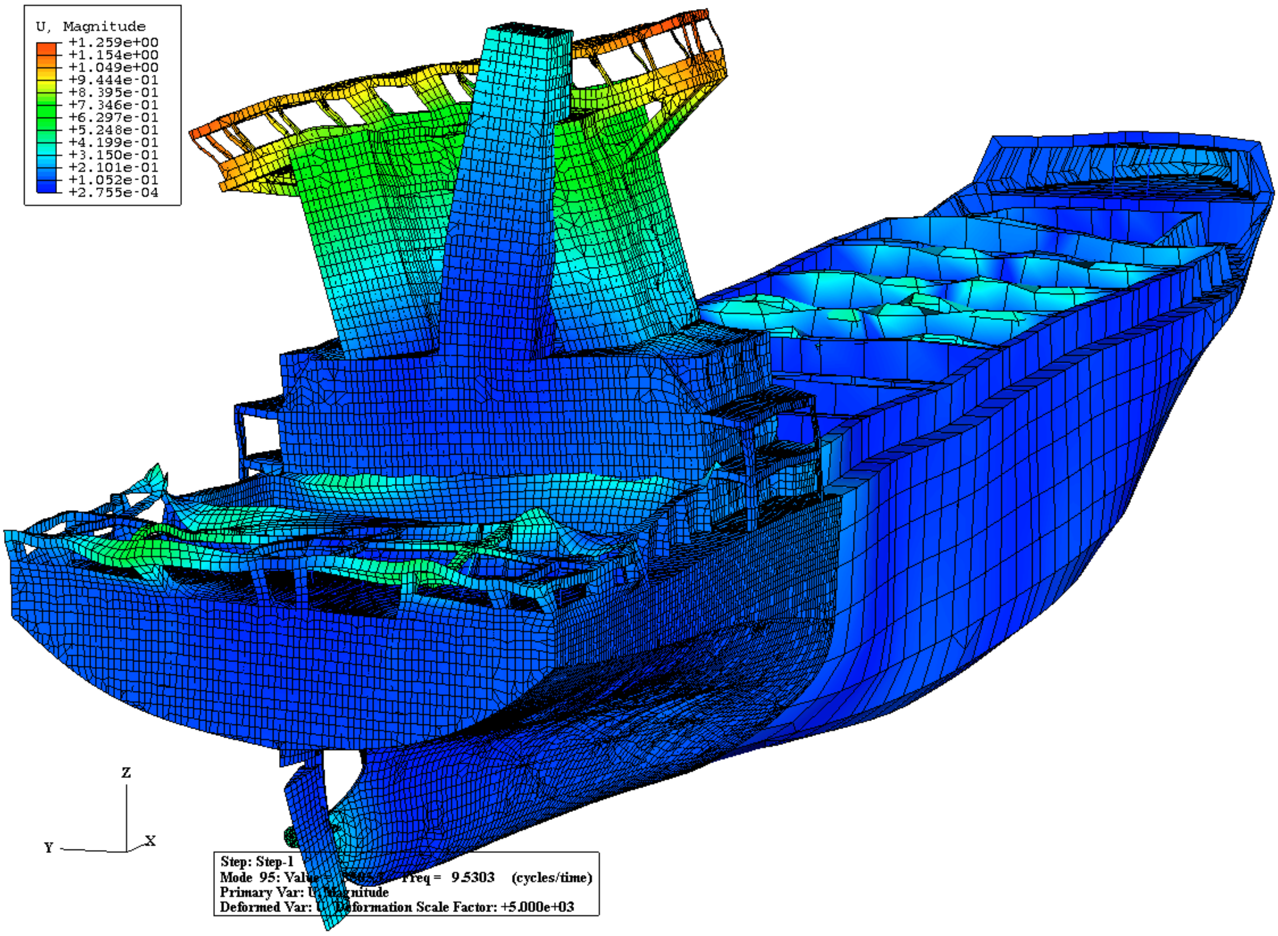

Figure 18 – Superstructure Mode at 9.530 Hz

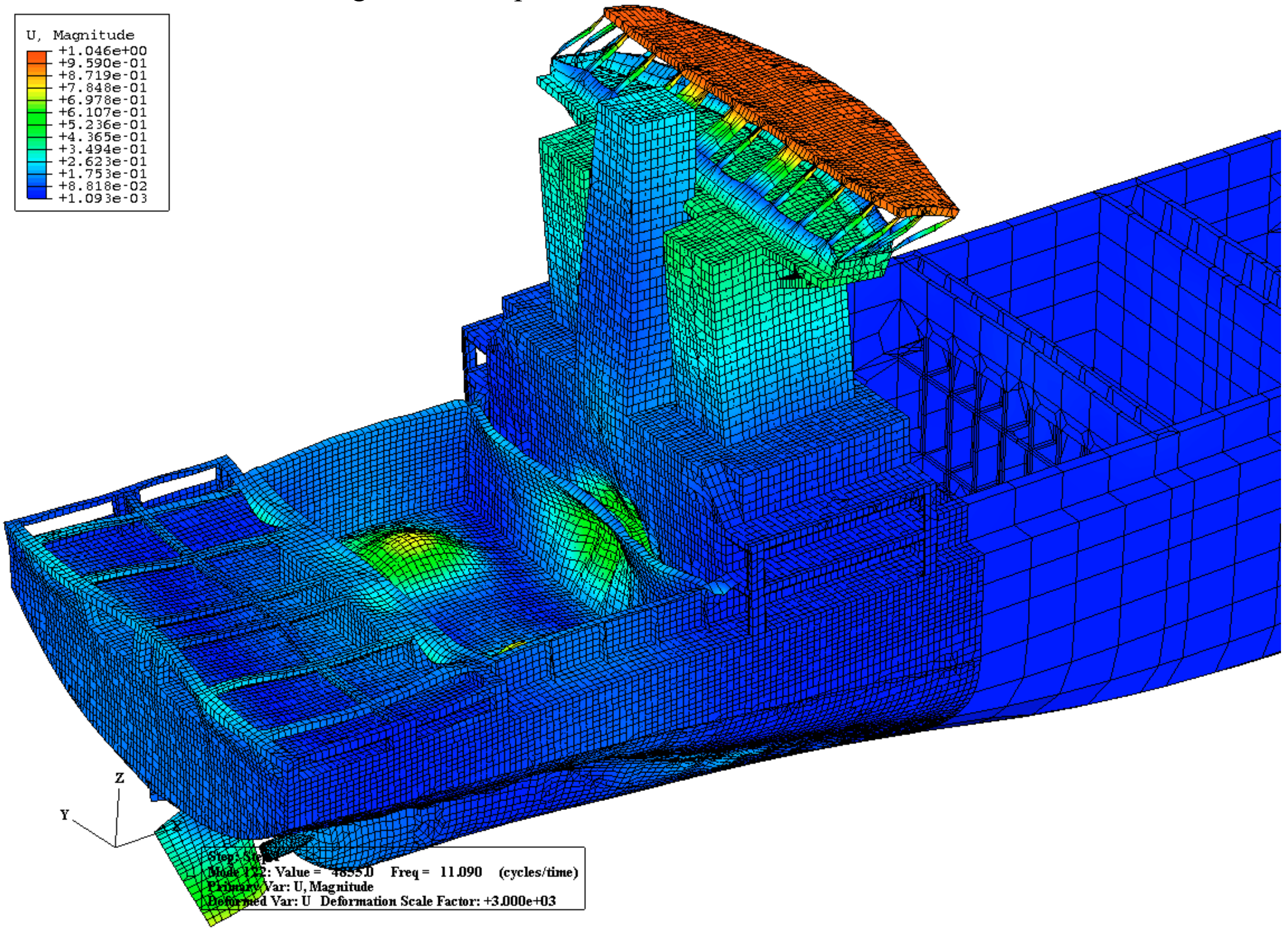

Figure 19 – Superstructure Mode at 11.090 Hz

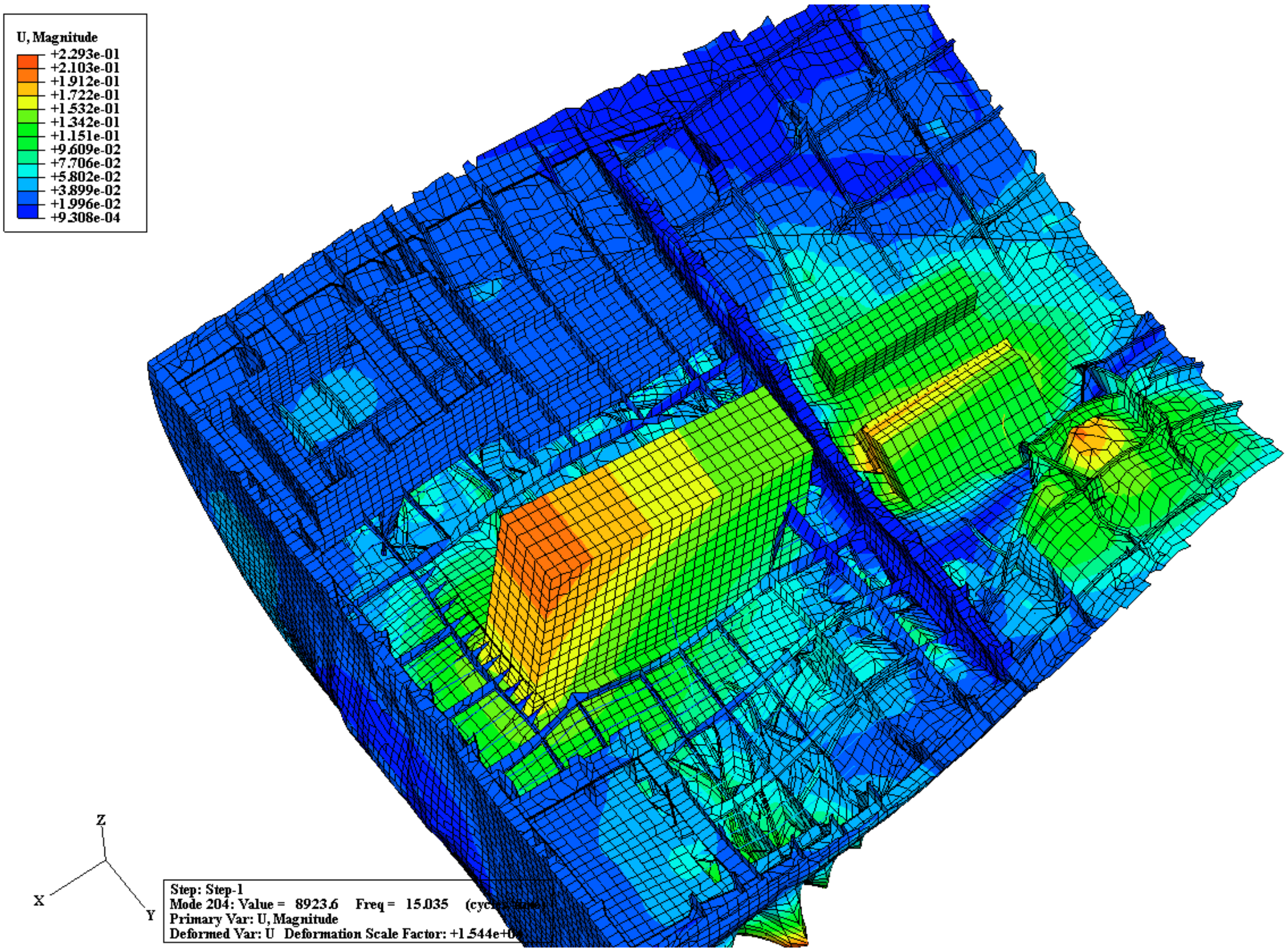

Figure 20 – Engine Room Mode at 15.035 Hz

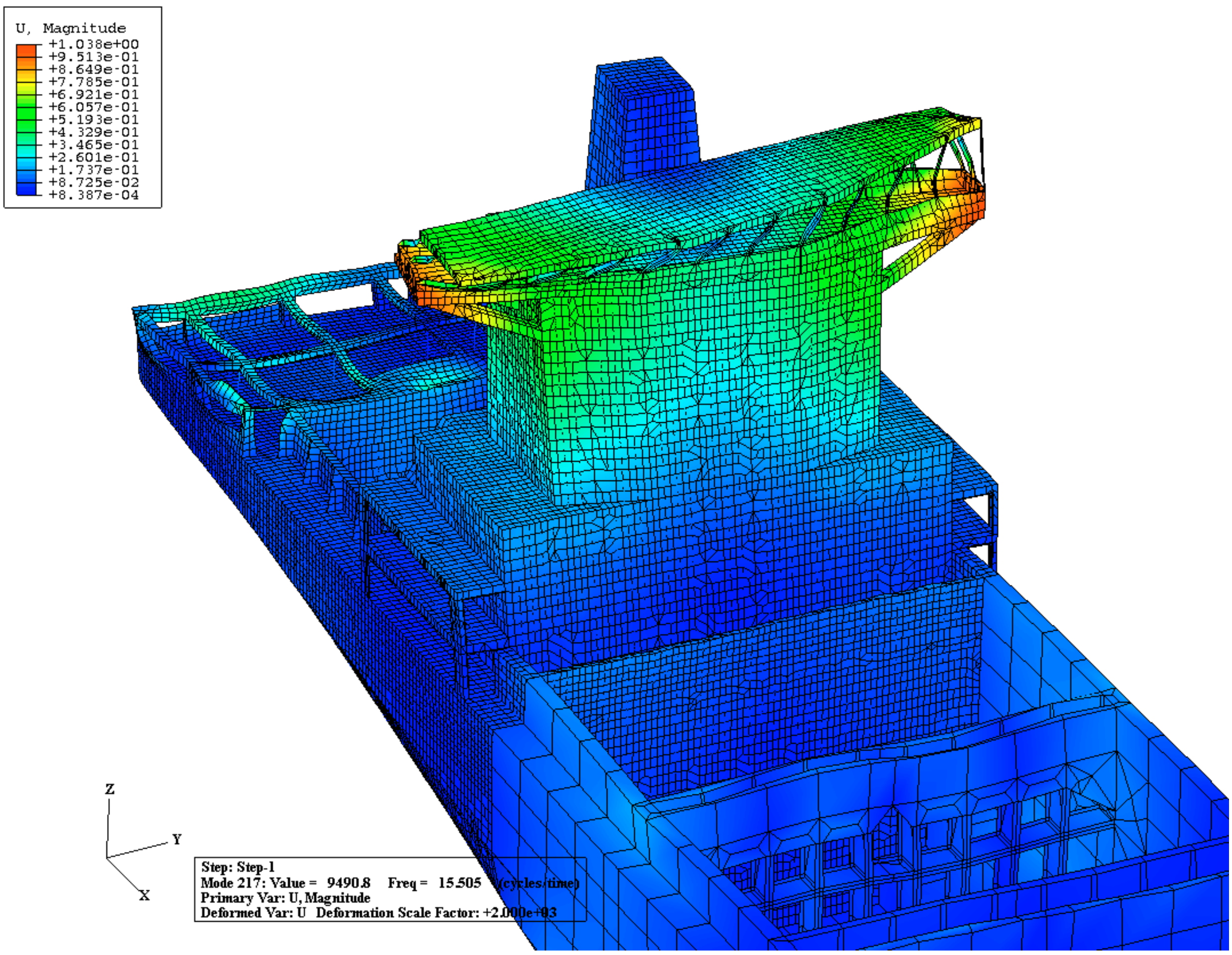

Figure 21 – Superstructure Mode at 15.505 Hz

4.2 Forced Vibration Analysis

Mode-based steady-state dynamic analysis is used in the solutions. In this analysis technique, the response is based on modal superposition techniques; the modes of the system must first be extracted using the eigenfrequency extraction procedure. The modes will include eigenmodes and, if activated in the eigenfrequency extraction step, residual modes. The number of modes extracted must be sufficient to model the dynamic response of the system adequately, which is a matter of judgement on your part.

Theoretically the displacement values obtained from a forced vibration analysis, approach infinite at the resonance frequency without the damping effects. By applying a damping coefficient to the analysis the resulting displacement values could be obtained in more admissible range. However, the damping coefficient of complicated structures such as ships, could not be evaluated precisely.

The overall damping coefficient includes the following effects;

- Material damping,
- Component damping, especially produced by the floor and deck coverings,
- Cargo damping,
- Hydrodynamic damping (water friction, pressure wave generation, surface wave generation),

Material damping can be defined is easily be defined as (0.5 –1.5 %). Component damping mainly depends on floor and deck coverings. Cargo damping mainly depends on the nature of the cargo (container, fluid, bulk, etc). Hydrodynamic damping is generally regarded as negligible in the frequency range of ship vibrations. Damping coefficients used in the analyses are taken from Reference 3.

These nodes which their responses are to be investigated could be seen in Figure 22 below. The results of the forced vibration for some of these numbered nodes could also be seen in Figures 23 to 36. The velocity values are given in mm/s and the frequency values are given in Hz.

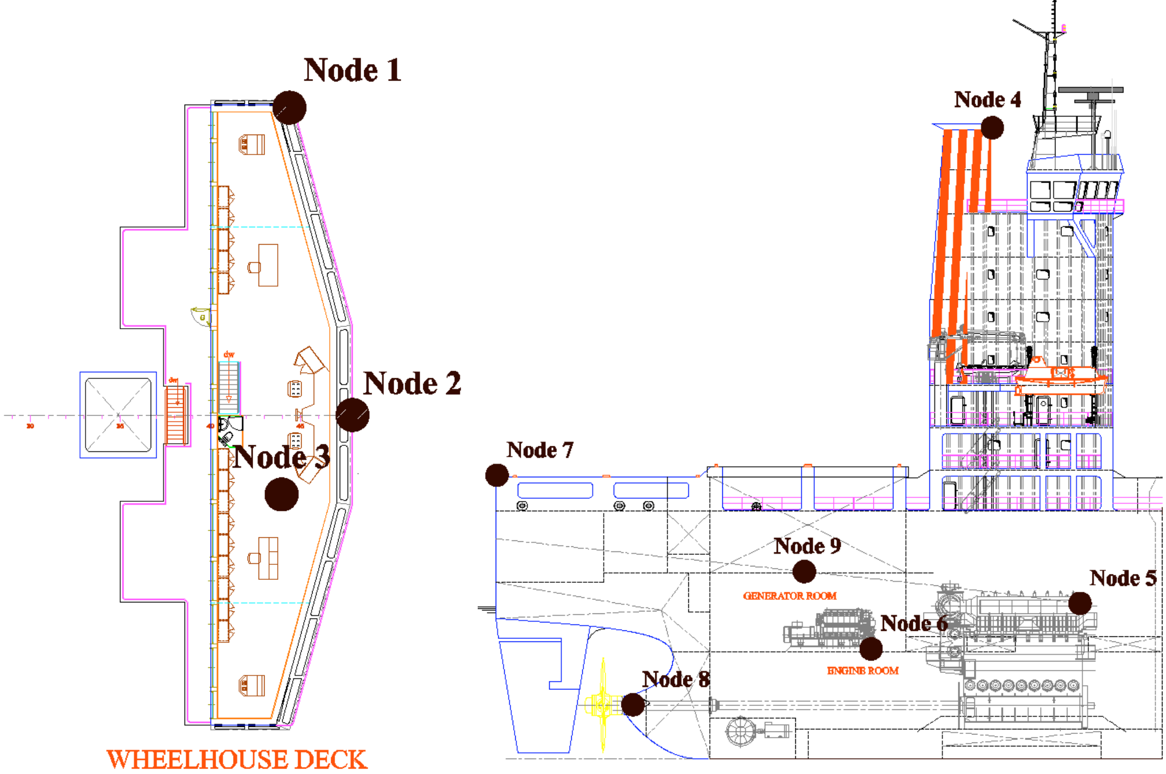

Figure 22 – Nodes for Forced Vibration Response

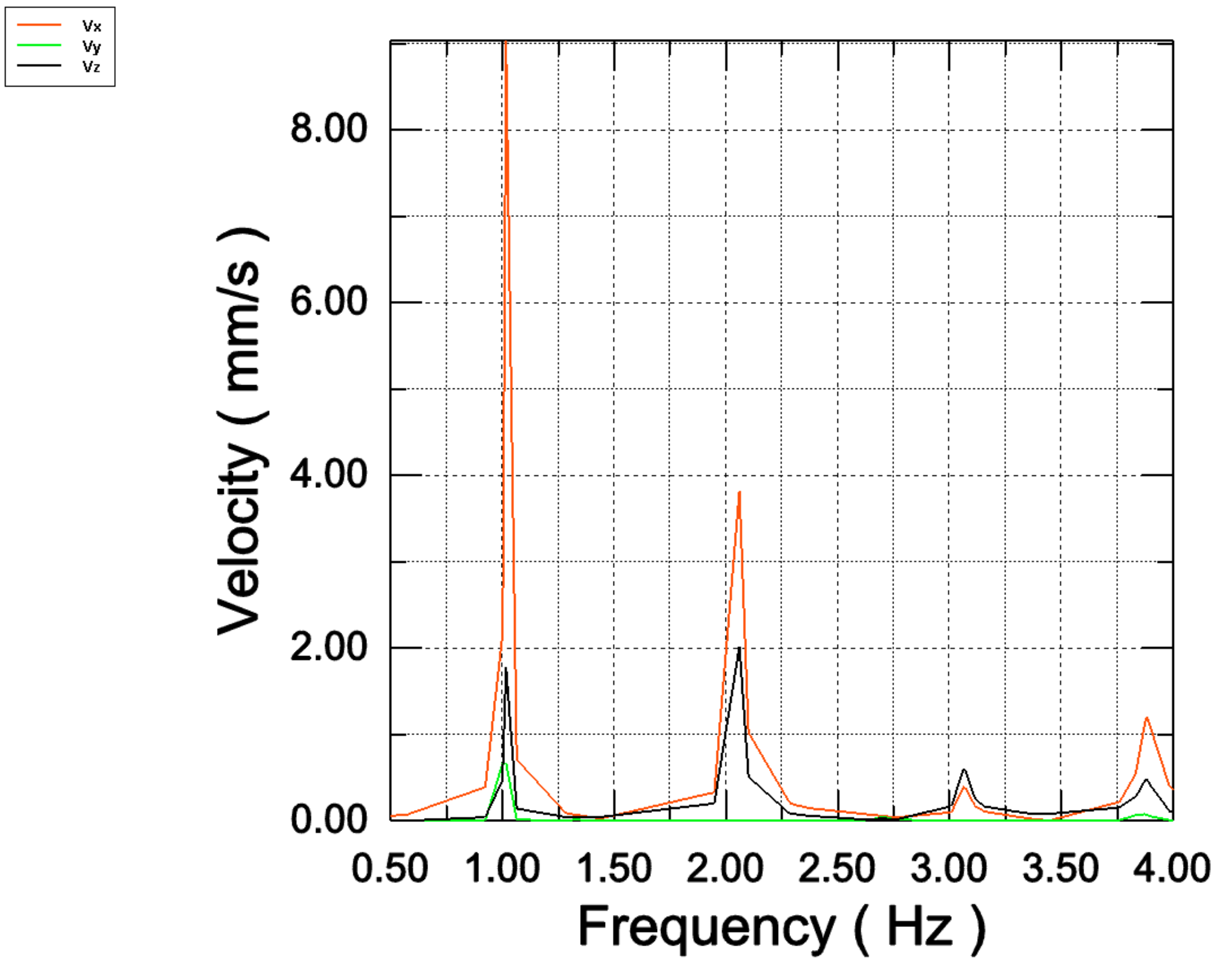

Figure 23 – 1st Order Loading Case (Vertical) - Forced Vibration Response of Node 1

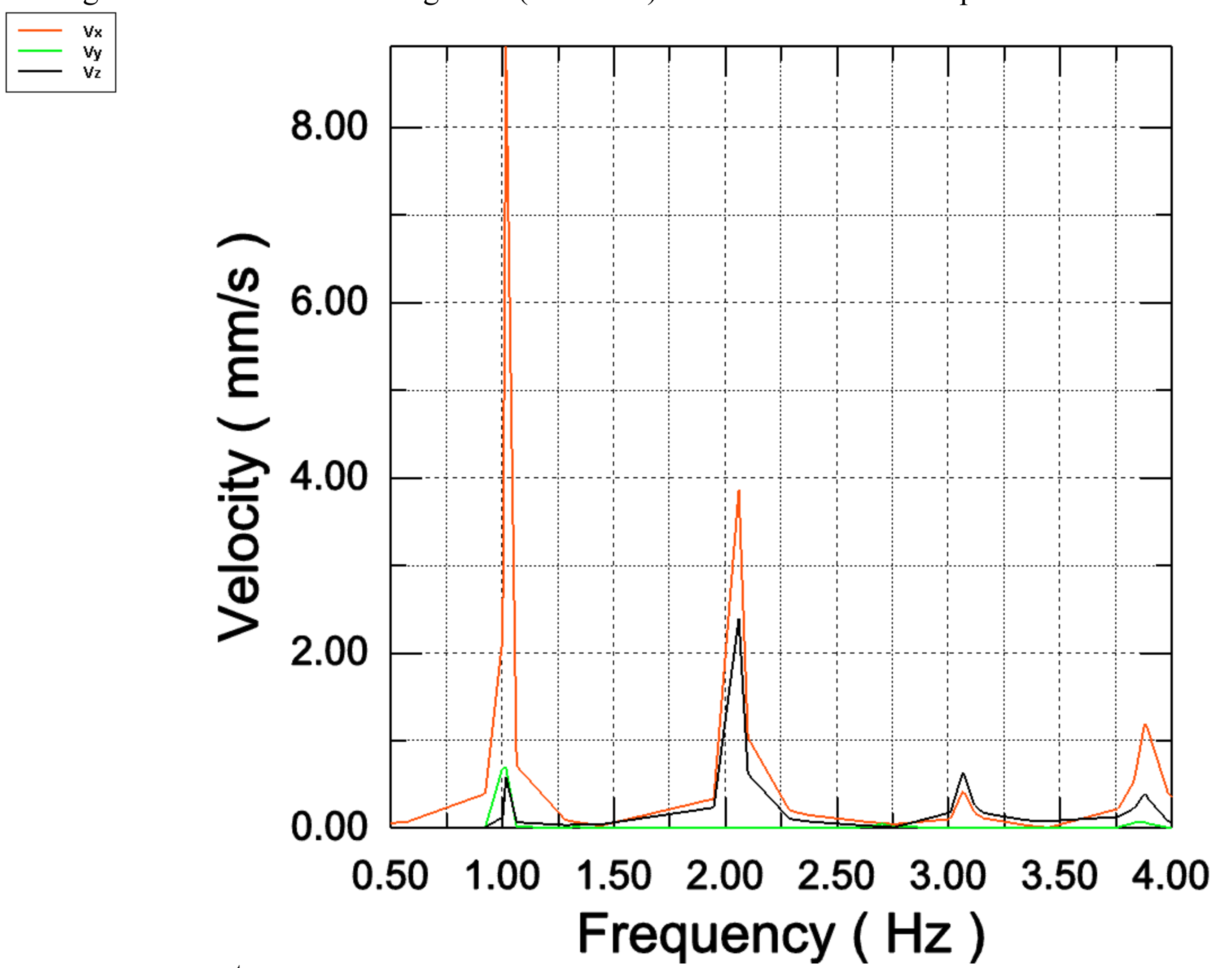

Figure 24 – 1st Order Loading Case (Vertical) - Forced Vibration Response of Node 2

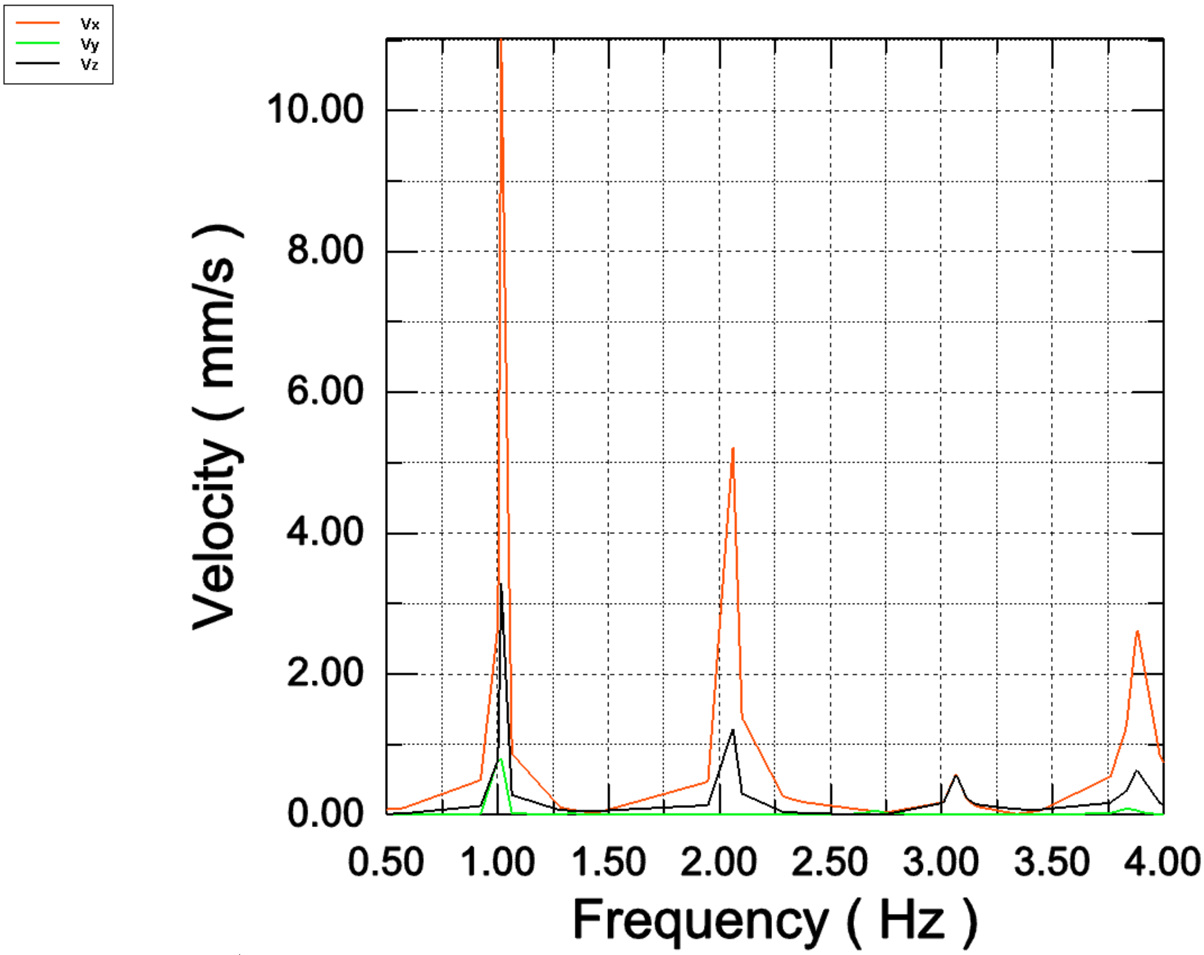

Figure 25 – 1st Order Loading Case (Vertical) - Forced Vibration Response of Node 4

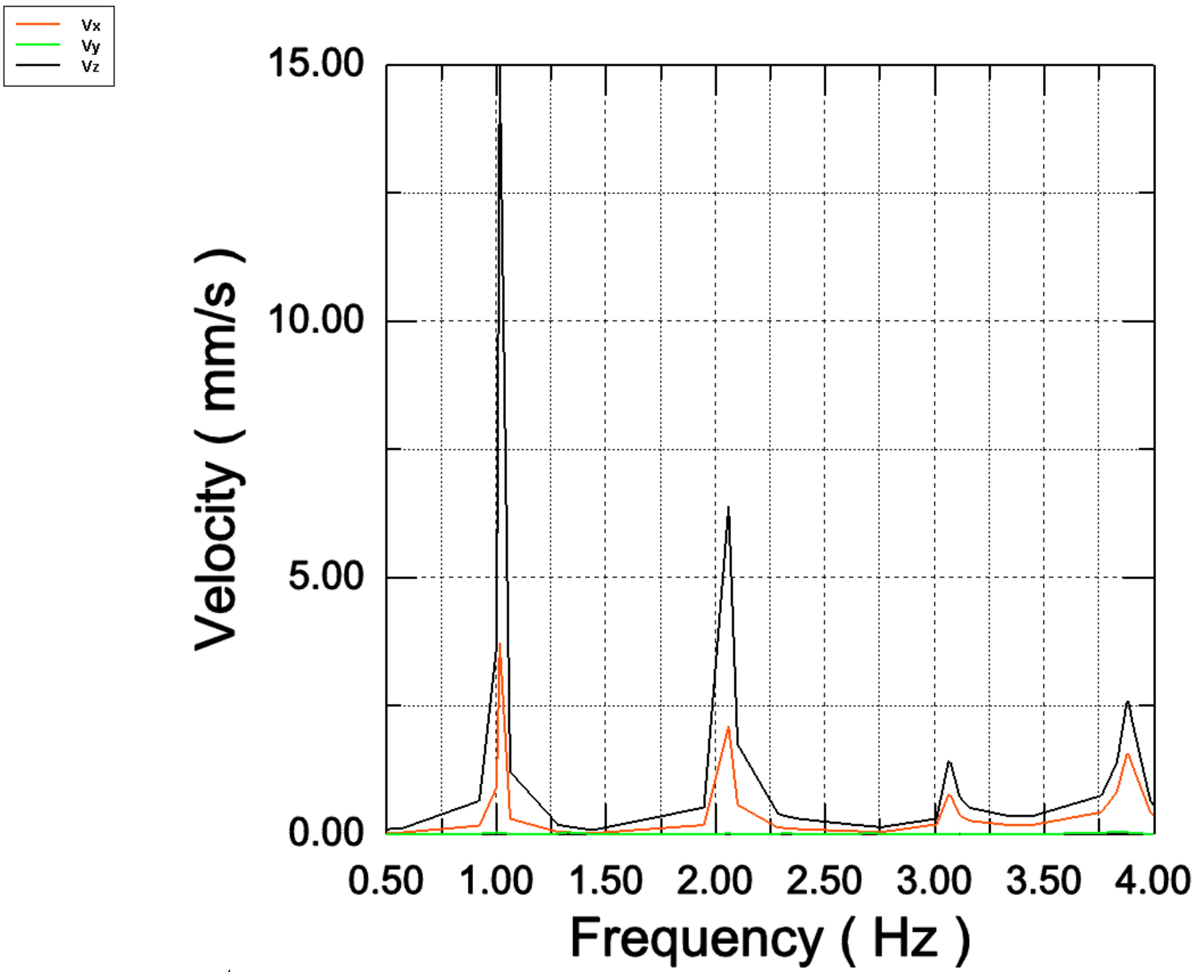

Figure 26 – 1st Order Loading Case (Vertical) - Forced Vibration Response of Node 7

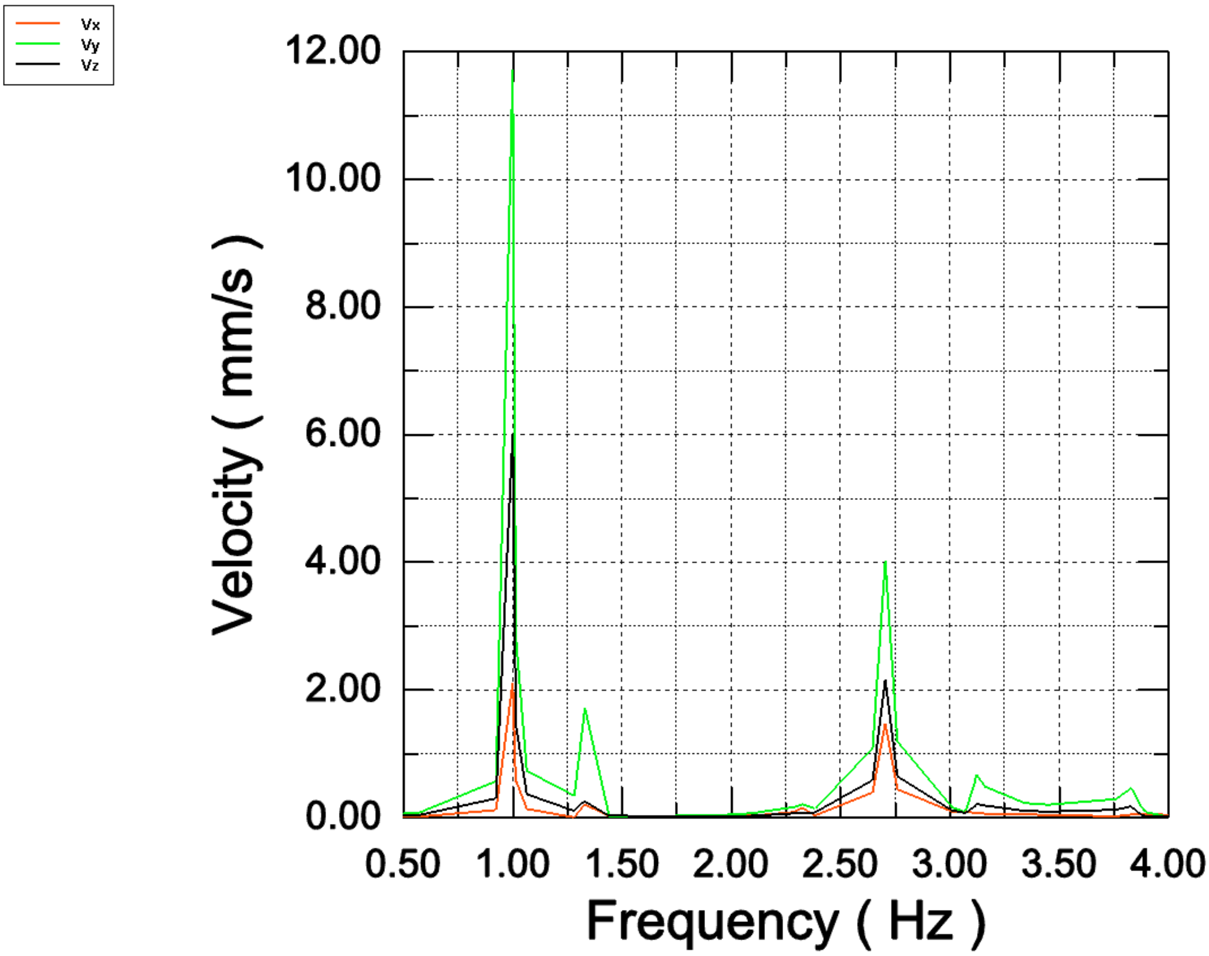

Figure 27 – 1st Order Loading Case (Horizontal) - Forced Vibration Response of Node 1

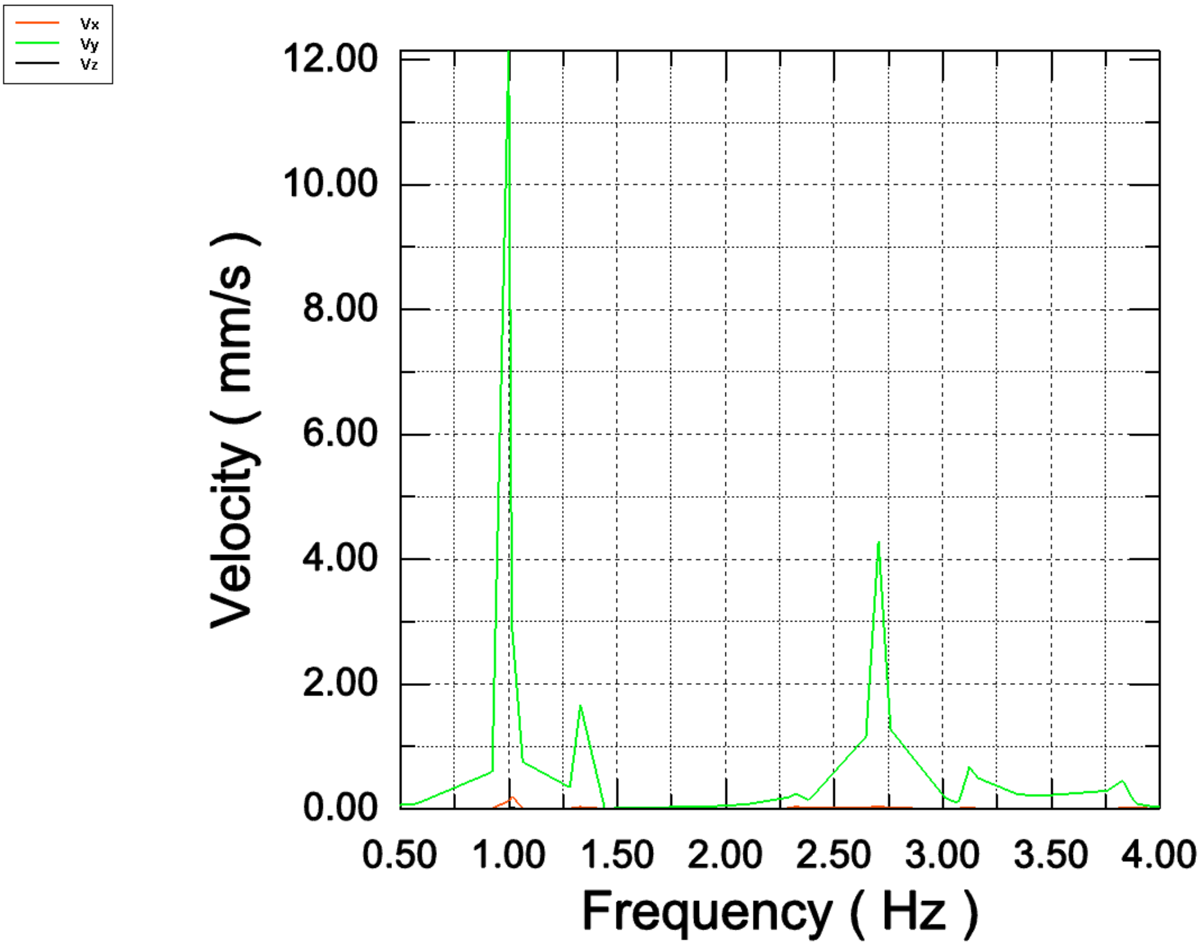

Figure 28 – 1st Order Loading Case (Horizontal) - Forced Vibration Response of Node 2

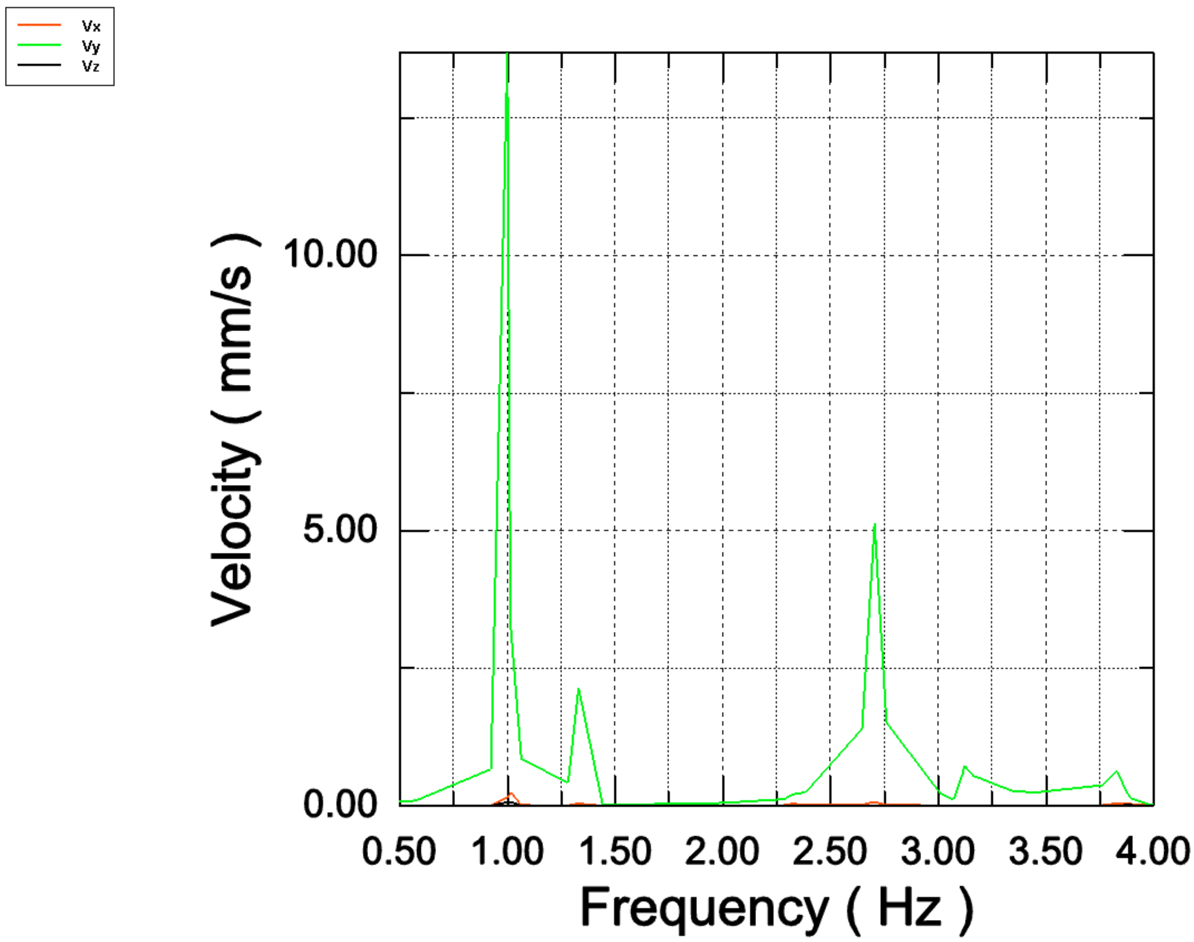

Figure 29 – 1st Order Loading Case (Horizontal) - Forced Vibration Response of Node 4

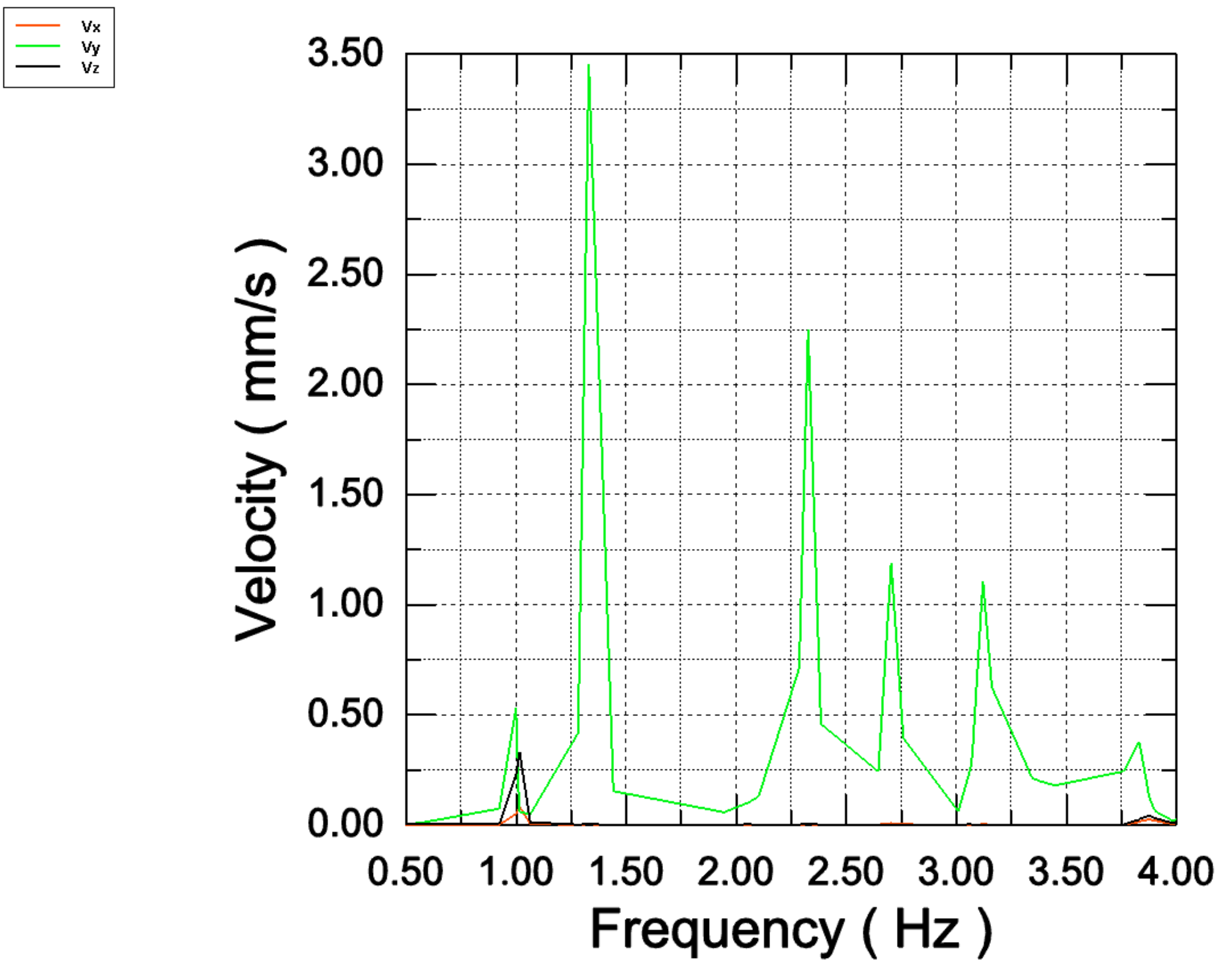

Figure 30 – 1st Order Loading Case (Horizontal) - Forced Vibration Response of Node 7

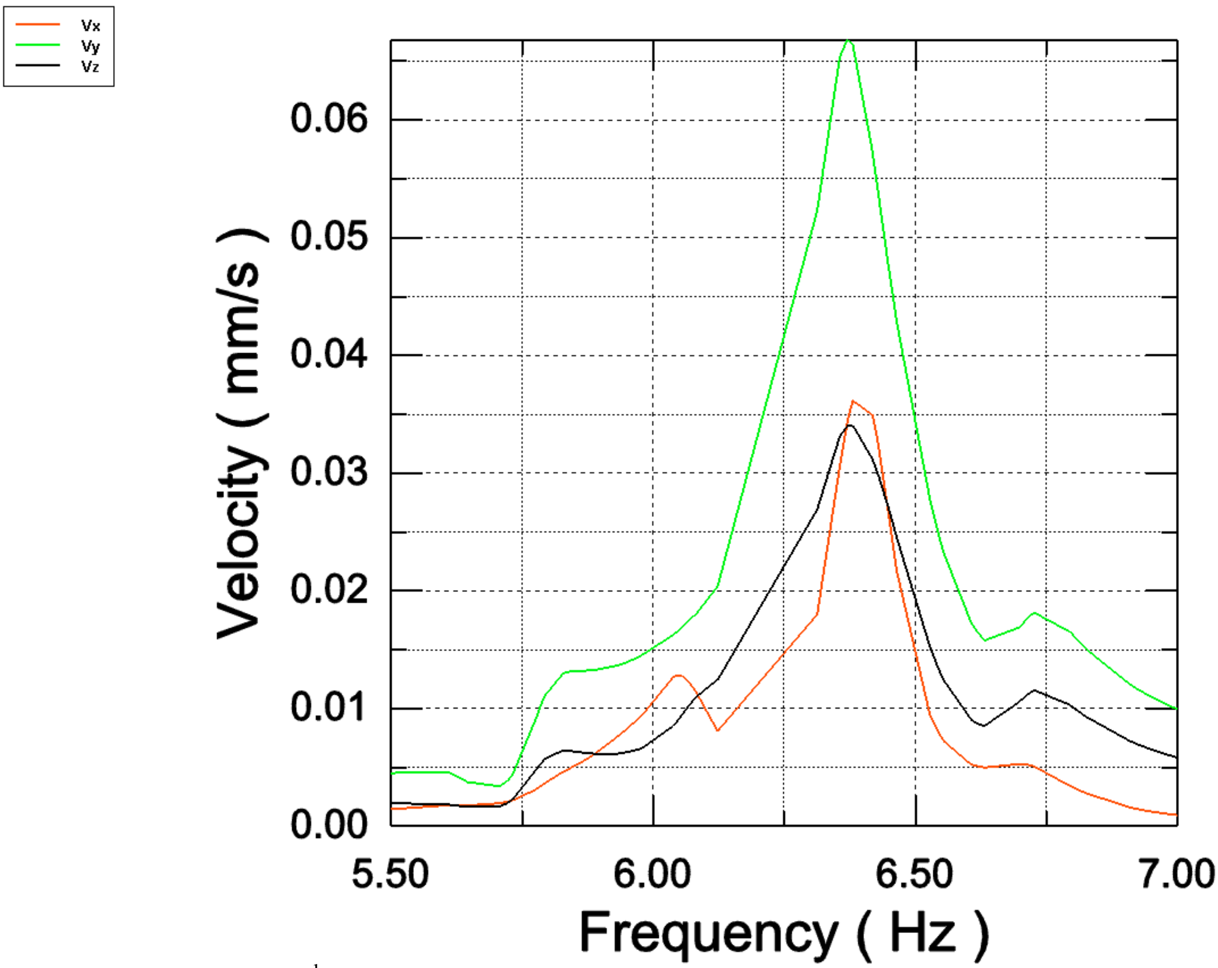

Figure 31 – 3rd Order Loading Case - Forced Vibration Response of Node 1

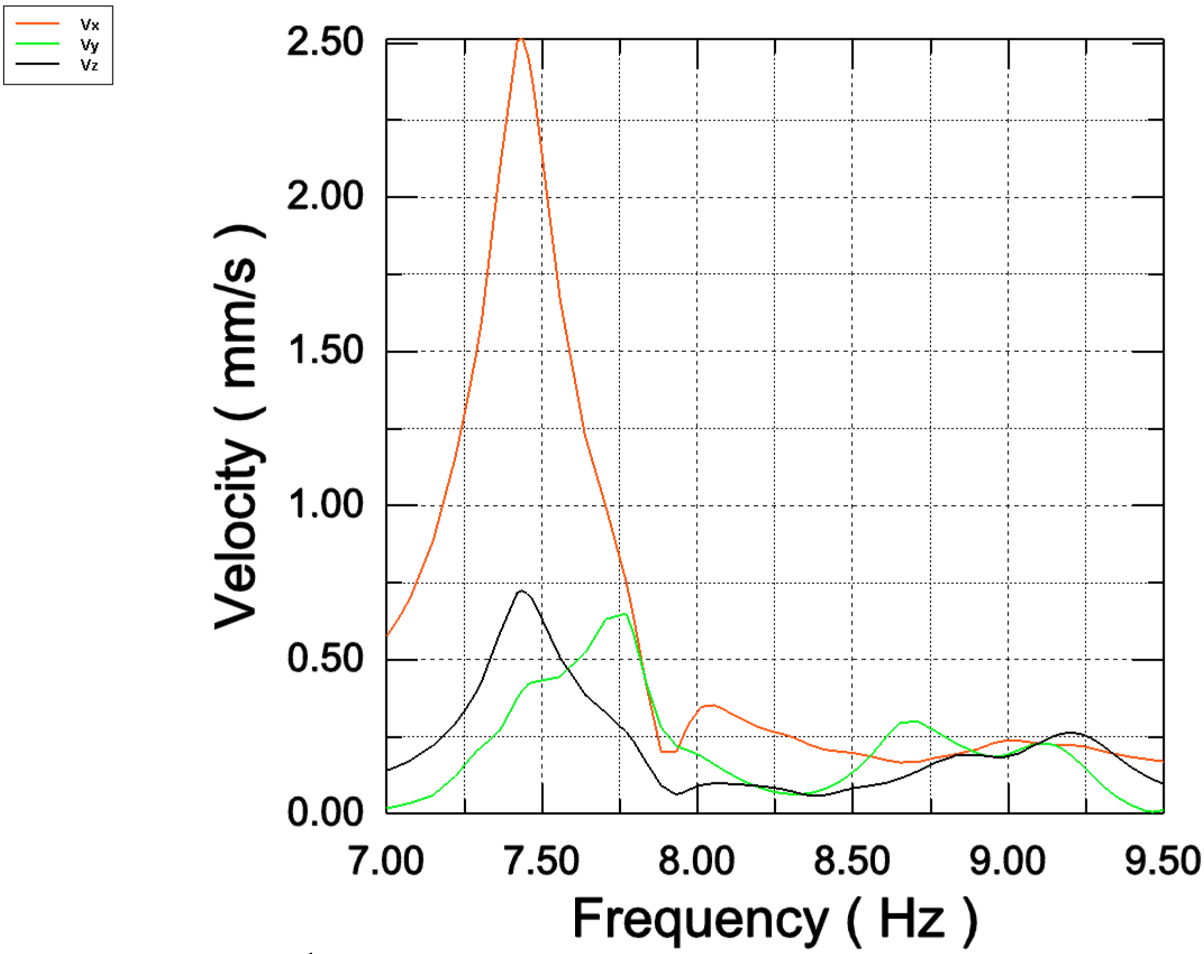

Figure 32 – 4rd Order Loading Case - Forced Vibration Response of Node 2

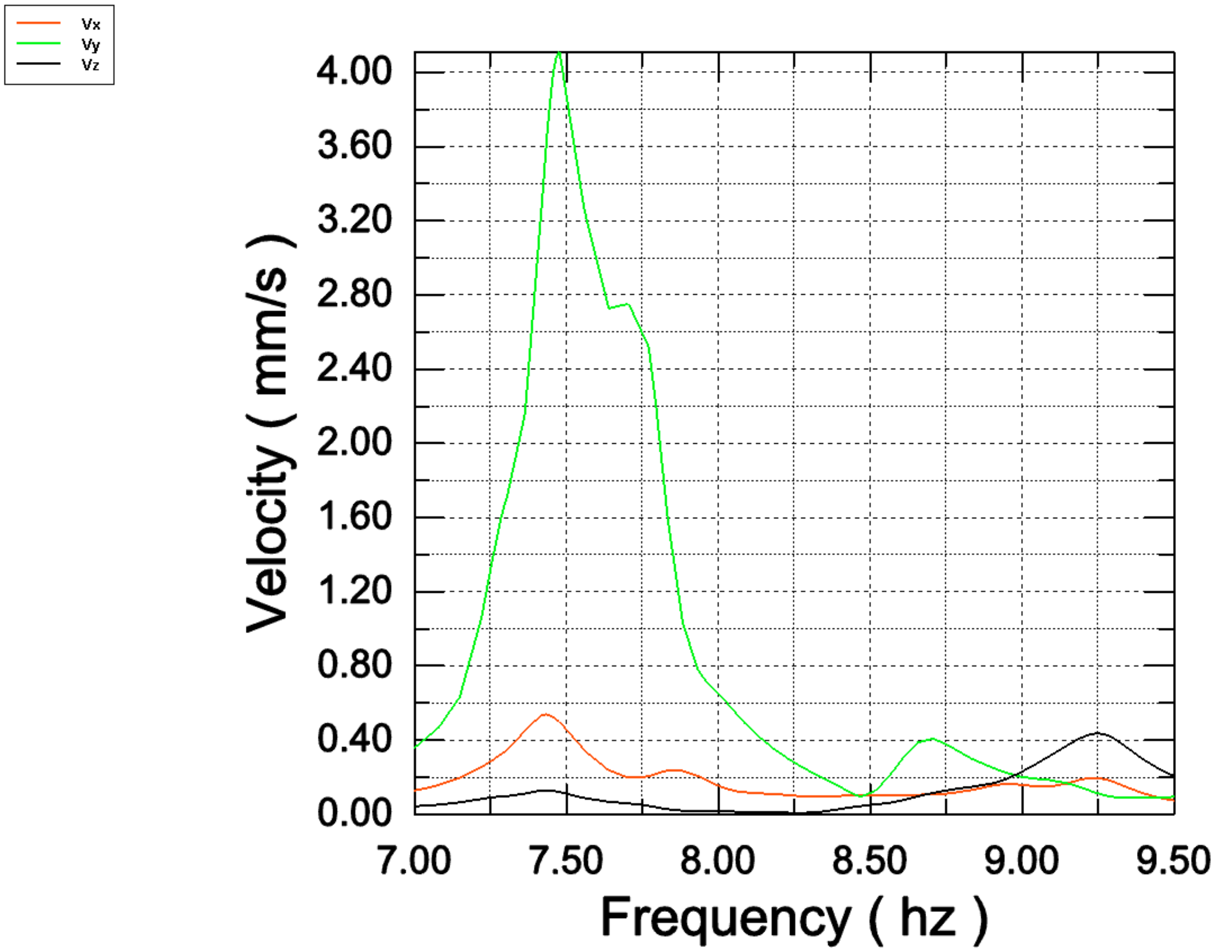

Figure 33 – 4rd Order Loading Case - Forced Vibration Response of Node 4

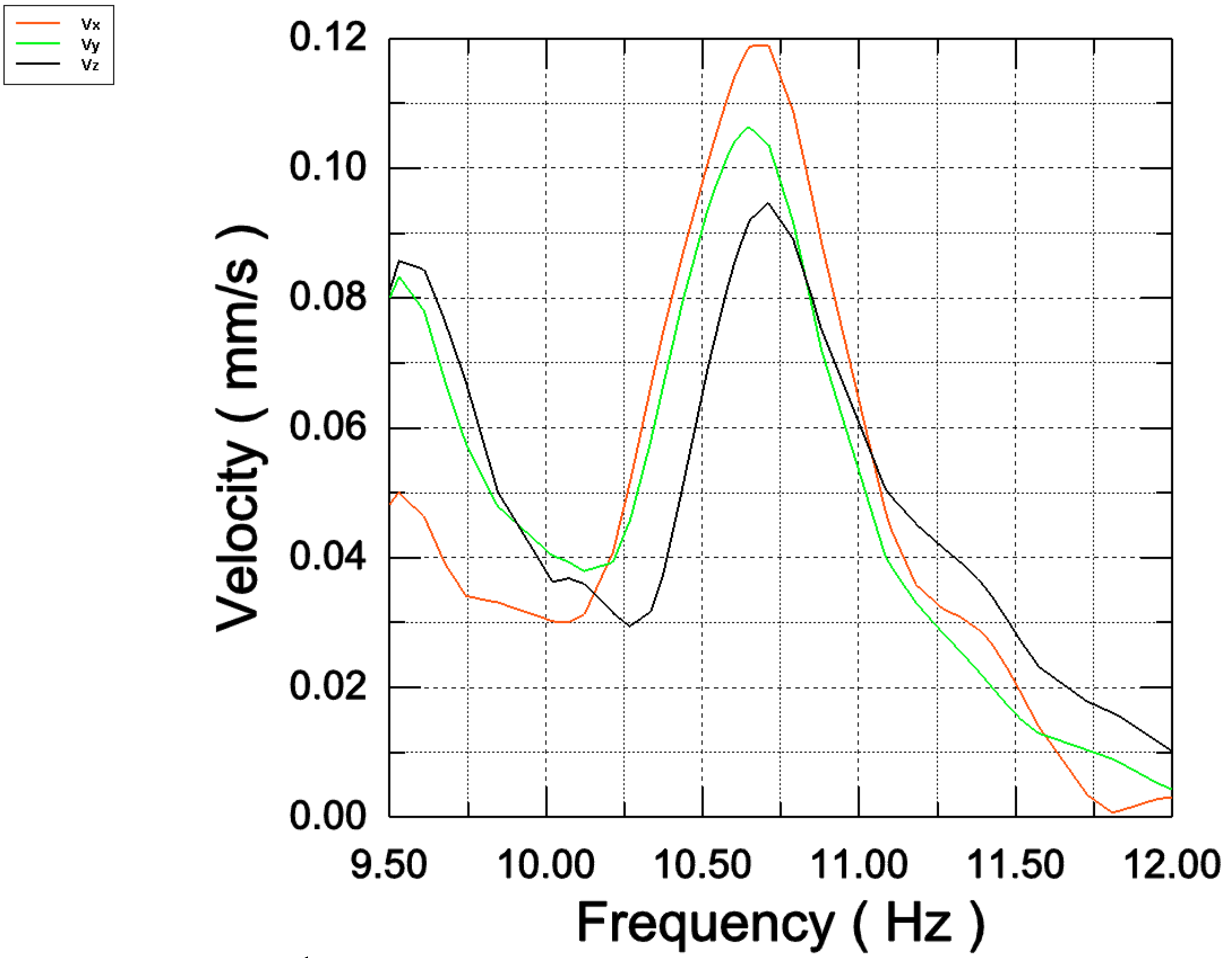

Figure 34 – 5th Order Loading Case - Forced Vibration Response of Node 1

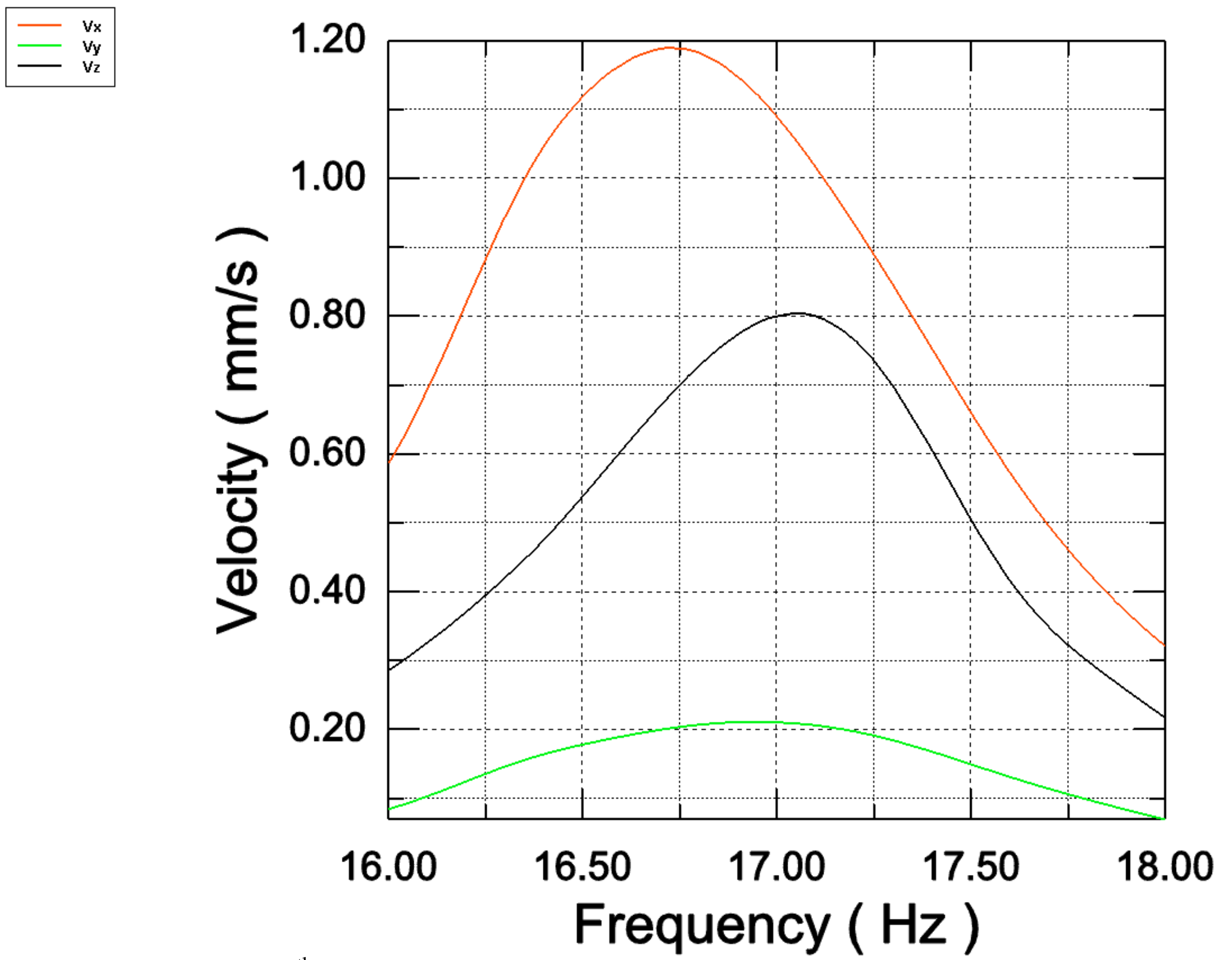

Figure 35 – 8th Order Loading Case - Forced Vibration Response of Node 1

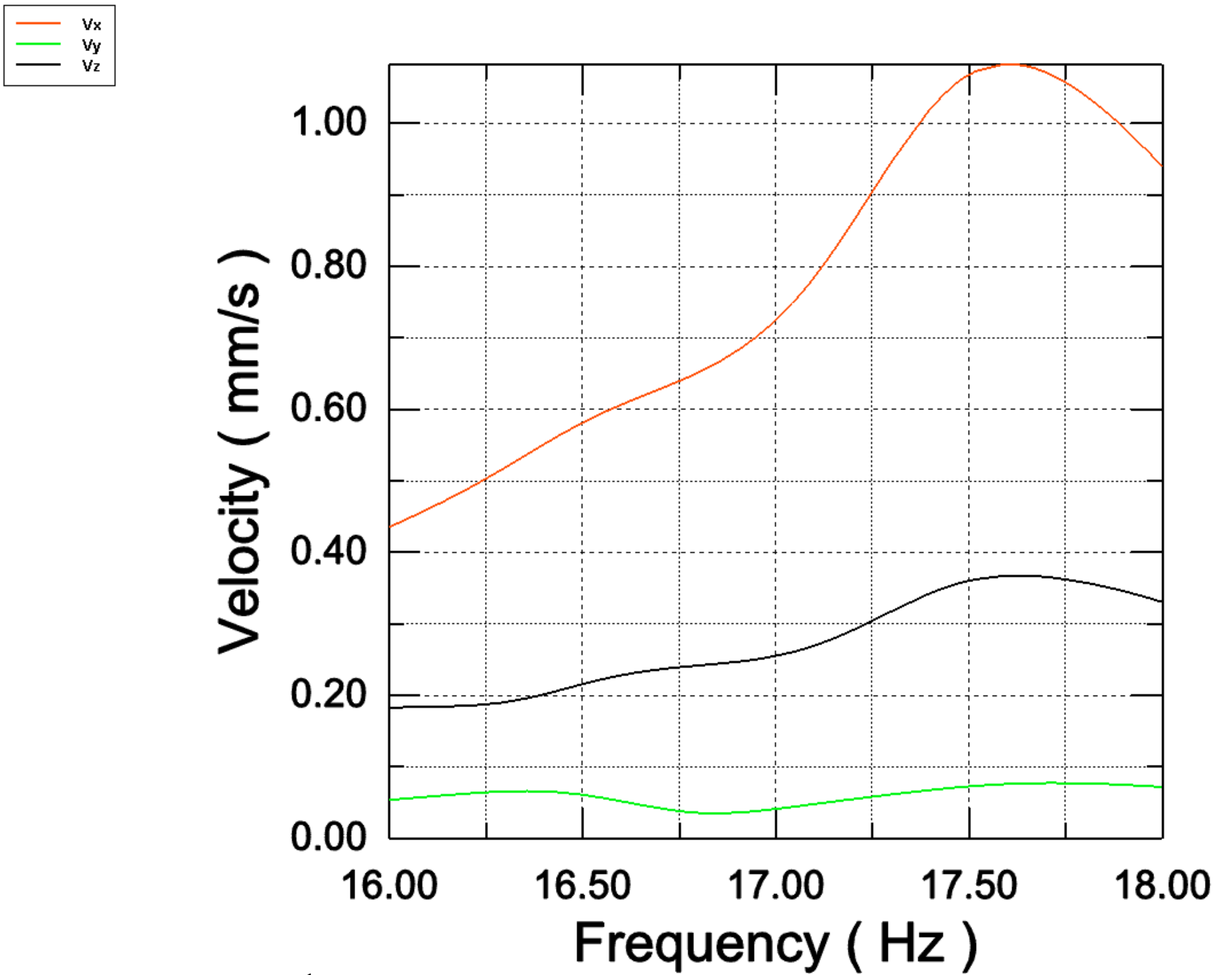

Figure 36 – 8th Order Loading Case - Forced Vibration Response of Node 5

5. CONCLUSIONS

After investigation of the forced vibration analysis results, the maximum velocity values in the frequency range for given loading condition are given in Table III by their locations.

According to ISO 6954 Standard “ Guidelines for overall evaluation of vibration in merchant ships”, vibration levels are classified in Figure 16 with the adverse comments probable and adverse comments not probable regions.

By comparing the velocity results with the ISO 6954 standard (Figure 37), it can be concluded that the vibration levels are in “adverse comments not probable” regions.

Table III - Forced Vibration Analyses Results – Maximum Velocities (mm/s)

	Loading Case 1	Loading Case 2	Loading Case 3	Loading Case 4	Loading Case 5	Loading Case 6
	0<2.116Hz<4	0<2.116Hz<4	5.5<6.35Hz<7	7<8.46Hz<9.5	9.5<10.58Hz<12	16<16.93Hz<18
Node 1	9.041	11.720	0.066	2.163	0.118	1.189
Node 2	8.934	12.142	0.068	2.514	0.087	0.425
Node 3	8.891	11.646	0.066	2.557	0.106	0.719
Node 4	11.032	13.681	0.264	4.111	0.060	0.279
Node 5	2.127	1.123	-	0.835	-	1.083

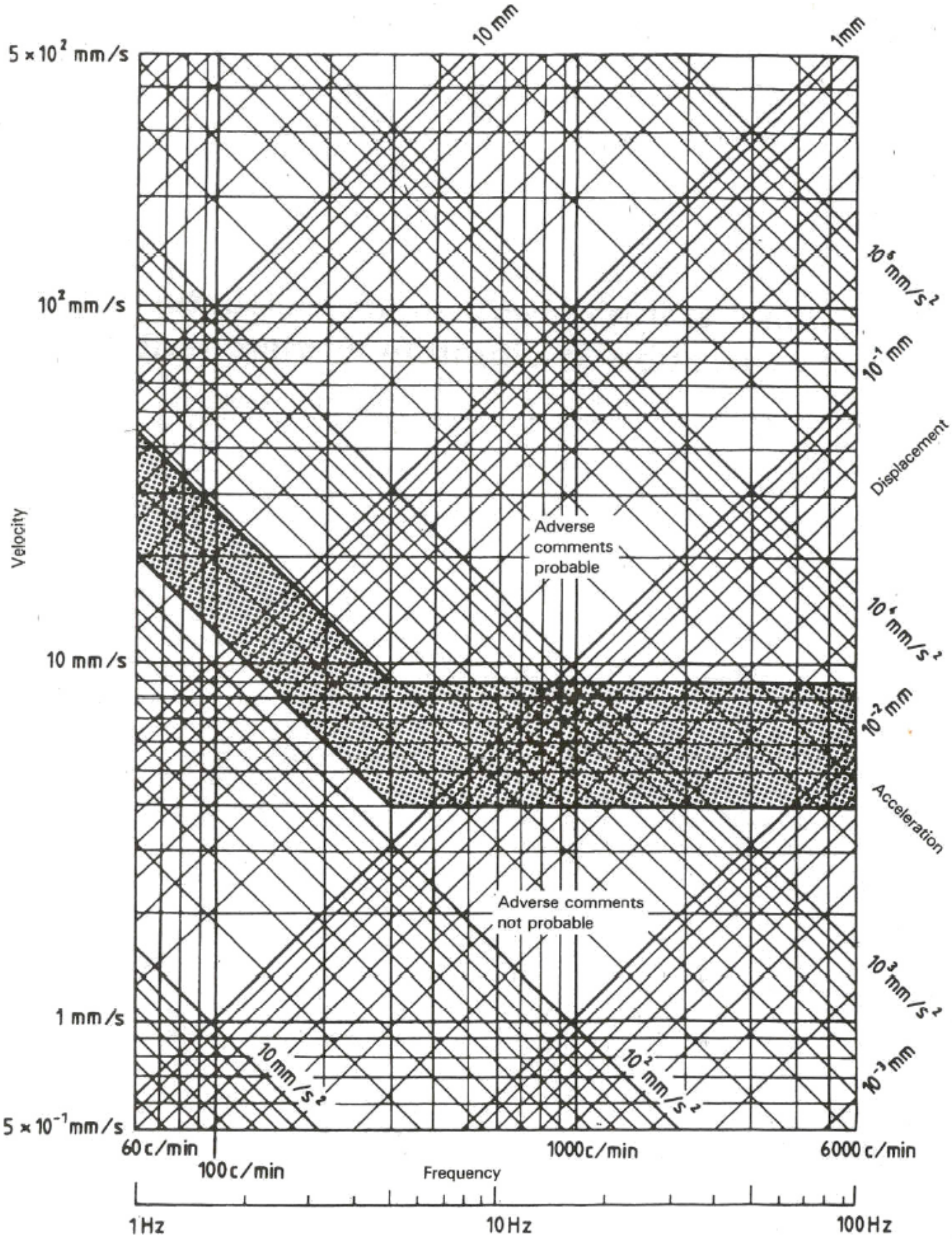

Figure 37 – ISO 6954 Standard

REFERENCES

1- **”** Marine Propellers & Propulsion “ J. S. Carlton Butterworth-Heinemann Ltd. 1994

2- “ Principles Of Naval Architecture “ The Society Of Naval Architects and Marine engineers.

3- “GL- Technology – Ship vibrations” Issue No5 / 2001

4- “ Ship Vibration Design Guide “ Ship Structure Committee SSC-350

5- “ Guidance Notes on Ship Vibration ” April 2006 American Bureau of Shipping

6- “ Prevention of Harmful Vibration in Ships “ July 1983 Det Norske Veritas

7- Abaqus Users’s Manuals