Vibration analysis of a 40m ice class motor yacht

Maria Sol Massera

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Supervisor: Prof. Dario Boote, University of Genoa

Reviewer: Prof. Zbigniew Sekulski, ZUT.

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The following document reports the main project developed during the internship performed at Cerri Cantieri Navali in collaboration with the Department of Naval Architecture of the University of Genova. This production will be presented as Master Thesis for the conclusion of the Erasmus Mundus Master in Advanced Ship Design (EMShip).

ABSTRACT

The purpose of this Master Thesis is the study of vibrations of the hull, deck and structure of an Ice Class motor yacht 40 m in length under construction at Cerri Cantieri Navali. This investigation is carried out because of the shipyard's necessity to gather information about the vibration behavior of the yacht, regarding that is not a conventional structure because the requirements of the Classification Society to be approved as an Ice class yacht, it is considerably stiffer than regular semi displacement motor yachts of the same size.
Vibration analysis during the construction process are of great importance, as soon is it possible to detect any probable vibration issue, sooner the changes on structure could be done. Vibration issues are determinant, not only for structural reasons but also when high standards are expected on comfort and wellbeing. If the limits stablished by the standards are not exceeded, the habitability will be good, which means that no health damage will occur.

At the stage of this study, the hull, structure, deck and upper deck are going to be finished. This is a perfect condition to perform measurements and analyze differences between experimental results and FEM analysis, considering that there is no uncertainties regarding damping, and induced errors on input data. The objective of this study is to determine how accurate this FEM model is, to allow further investigations on this project.
Construction plans, technical information and assistance is provided by the shipyard's technical office.

The vibration analysis is developed through the use of the Nastran/ Patran finite element code, the modelling of all the mentioned above was performed to obtain the natural mode and to carry out the dynamic analysis, and after compared with the excitations

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CONTENT

LIST OF TABLES

LIST OF FIGURES

1. INTRODUCTION

The main topic of this document is to present a comparison between the results of FEM model of an Ice class yacht 40 m length, with experimental measurements carried out in the shipyard. This is a useful phase to be employed as first step for future numerical analyses. This was carried performing a FEM numerical model, developed in Patran-Nastran from MSC.

Vibration and noise are problems that a ship can probably face if a study is not performed during the design stage, and nobody wants to find these issues when the yacht is already launched. As sooner a problem is detected, better it is, since is much more costly to find the vibration sources through measurements and solve them once the construction is done.

The need of this research has been highlighted by CERRI CANTIERI NAVALI (From now CCN) Since a 40 m motor yacht is being built in the shipyard and a requirement is to classified this yacht as “Ice Class”, therefore the steel structure of the hull has come out much stiffer than a regular semi displacement motor yacht, resulting in the need to perform a vibration analysis to have information about the behavior of the yacht and predict the possible issues that the yacht could face.

This is very important, not only for structural reasons but also when high standards are expected on comfort and wellbeing. If the limits stablished by the standards are not exceeded, the habitability will be good, which means that no health damage will occur.

At the time of this study, the yacht is under construction, with all steel construction completed. This makes it a perfect condition to perform measurements and compare results with those obtained by FEM analysis.

To carry out this study a finite element model was created, which includes the hull, lower deck and main deck

2. VIBRATIONS IN MOTOR YACHTS

It is of great importance to understand the problem that vibrations represents and what are their influential factors in yachts, even if in this paper not all items are analyzed and developed in detail, being this the first step of a complex analysis.

Vibration is an oscillatory movement; which can be periodic as well as random. In the case of ships this phenomenon is not desirable. A big amount of energy is wasted on it and it might bring structural vibrations and fatigue and noise.

Excluding effects of the seaway which are present mainly in big ships, a vessel is excited mechanically by rotating machinery systems and hydro dynamically by its propeller(s). These excitation sources are essentially periodic, but they are not, in general, simple harmonic, i.e., purely sinusoidal. Because of this, excitations also occur at all multiples of a fundamental exciting frequency associated with each excitation source. The strengths of the various excitations, and their harmonics, are often highly sensitive to the details of design and fabrication. Moderate propeller cavitation, for example, which may be acceptable in all other respects, can produce hull vibratory excitation forces on the order of tens of tons, persistent at frequencies out to several multiples of the blade-rate fundamental.

This motor yacht will be submitted to internal and external dynamic forces during normal operation. These sources of excitations are:

- Waves
- Main engines and auxiliaries engines
- Pressure fluctuations in the stern
- Thrust fluctuation induced by propeller
- Bow thrusters
- Slamming fluid impacts

The vibratory behavior of any complex system can therefore be dealt in terms of the collection of equivalent one-mass systems vibrating simultaneously. For this reason, much insight into the various sensitivities of the vibration of any particular system, whether simple or complex, can be gained by applying a few simple observations from the theory for one-mass systems.

2.1 Equations of motion

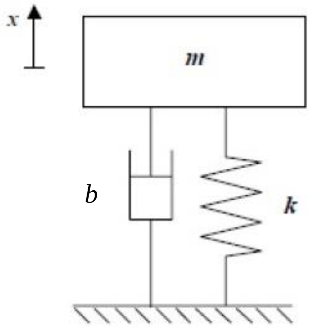


Figure 1- Mass spring damper in free motion

The four basic components of a dynamic system are mass (m), energy dissipation or damping (c), resistance (k), and applied load. As the structure moves in response to an applied load, induced forces are a function of both the applied load and the motion in the individual components. In the case of 1DOF the basic components are only a number, but in the case of multiple DOF problems, m, b and k are represented by matrixes of sizes corresponding to the number of present DOF.

The general dynamic equation representing the vibrations at each instant in time is:

$$M\ddot{x}(t) + B\dot{x}(t) + K\,x(t) = F(t) \qquad \textit{Eq. 1}$$

Where

M = mass matrix

B = Damping matrix

K = Stiffness matrix

x(t) = displacement vector

F (t)= applied load vector

$$\dot{\mathrm{x}} = \frac{\mathrm{dx}}{\mathrm{dt}} = \mathrm{v} = \mathrm{velocity} \qquad \textit{Eq. 2}$$

$$\ddot{x} = \frac{d^2x}{dt^2} = a = \text{acceleration} \qquad \textit{Eq. 3}$$

Typically this is separated as internal forces and external forces, on the left side of equation the internal forces are represented and at right side the external forces are specified. These results in a second order linear differential equation.

The objective of dynamic analysis is to find the solution of this equation. This analysis could be divided in two basic classifications: free vibrations and forced vibrations

2.2 Free vibrations

This phenomenon appears when a mechanical system moves due to an initial input and the system is allowed to vibrate freely. Which means an unforced system (F=0).

Then the basic dynamic equation (considering damping) becomes:

$$M\ddot{x}(t) + B\dot{x}(t) + K\,x(t) = 0 \qquad \textit{Eq. 4}$$

If no damping is considered, then:

$$M\ddot{x}(t) + Kx(t) = 0 \qquad \textit{Eq. 5}$$

If a SDOF is considered, then the ***Eq. 5*** has a solution of the form:

$$x(t) = Asin\omega_n + Bcos\,\omega_n \qquad \textit{Eq. 6}$$

As it can be observed, the response is cyclic in nature, usually termed as circular natural frequency of the structure ω_n. In system with more than 1DOF the subscript indicates a frequency number and the natural frequency is defined by

$$f_n = \frac{\omega_n}{2\pi} = \frac{\sqrt{k/m}}{2\pi} \qquad \textit{Eq. 7}$$

This natural frequency represents a frequency at which a system vibrates when stimulated impulsively from the rest position. For continuous mass and stiffness distributions, the system possesses an infinite number of natural frequencies, even though only a relatively small number are usually of practical significance. On impulsive stimulation from rest, the continuous system will vibrate at all of its natural frequencies, in superposition.

The solution for equation ***Eq. 6*** for an undamped system with a known initial displacement (*x(t=0))* and velocity ($\dot{x}(t=0)$), is represented graphically as a sinusoidal wave whose position in time is determined by its initial displacement and velocity in Figure 2 - 1DOF Undamped free vibrations

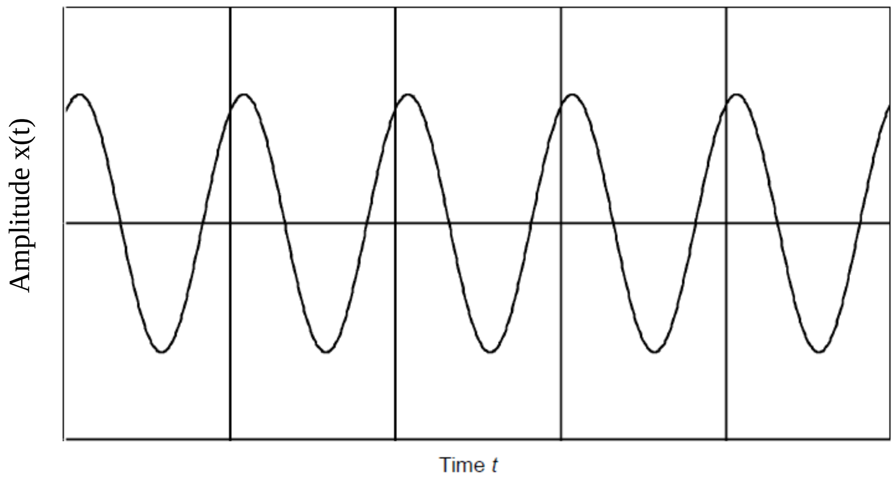


Figure 2 - 1DOF Undamped free vibrations

2.2.1. Damping

Damping happens in a way of dissipation of energy and is finally converted to thermal energy. The hydrodynamic damping is important for ship motions but is considered insignificant in relation to ship vibration. Furthermore the internal damping is the most important for ship vibrations and the most influent in this is the behavior of the material under stress. When the calculations are done the changes in damping due to welding and variations in workmanship should be considered.

Damping matrix is the most complicated one to be defined, since the vibration issues happen as resonances and the responses are inversely proportional to the damping. Vibrations forces generated by engines can be usually well defined, but the ones produced for the pressures in the aft part of the hull due to the propeller are less certain

Several physics contribute to the damping in the ships:

- Material damping

- Component damping (the ones produced by deck and floors by instance)
- Hydrodynamic damping (added mass)
- Mechanical damping (concentrated damping)

Mathematically, the Eq.10 presented before is recalled.

$$M\ddot{x}(t) + B\dot{x}(t) + K\, x(t) = 0$$

For this case the solution is determined by the amount of damping. And three possible cases for positive damping are:

- Critically damped
- Overdamped
- Underdamped

Critical damping occurs when the value of damping equals a term called critical damping ξ, defined as:

$$\xi = 2\sqrt{km} = 2m\, \omega_n \qquad \textit{Eq. 8}$$

In this condition the system recovers the rest condition through an exponential decay curve, without oscillations.

A system is *overdamped* when *b*> ξ and no oscillatory motion occurs as the structure returns to its initial position.

The most frequent condition is underdamped case, when *b* < ξ

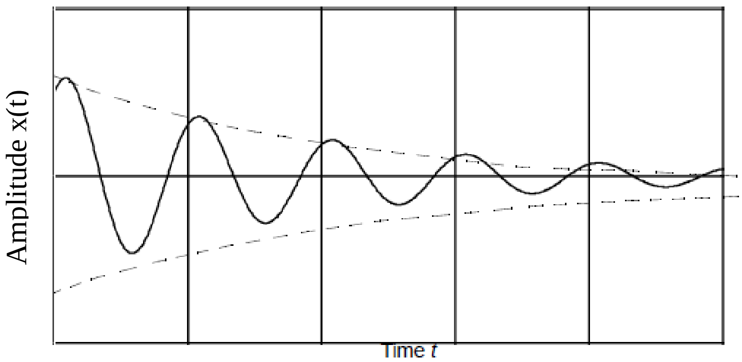


Figure 3 - Free vibration, underdamped case

There is no general damping model to determine its value.

Logarithmic decrement is represented with the following:

$$\frac{x(t)}{x(t+T_a)} = e^{\omega_0 \xi T_a} = e^{\delta} \quad \textit{Eq. 9}$$

$$\delta = \omega_0 . \xi \, T_a \quad \textit{Eq. 10}$$

being ξ the modal damping coefficient.

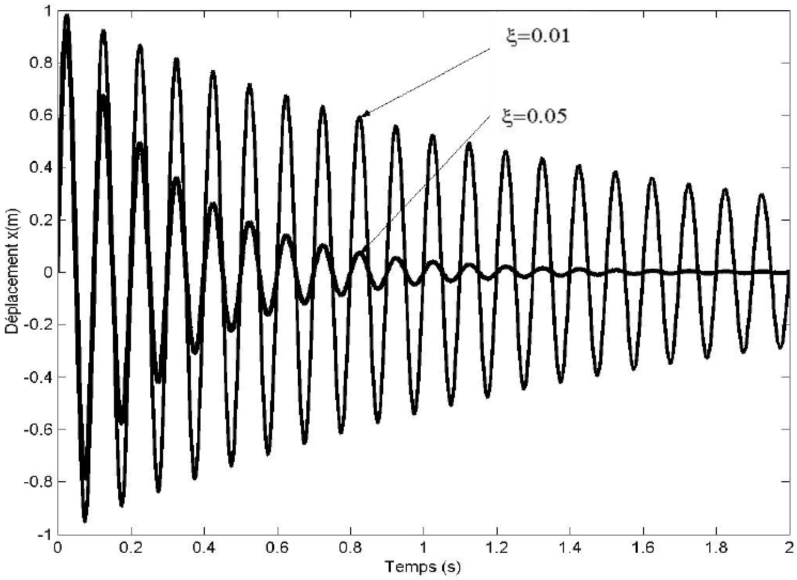


Figure 4 - Damping coefficients

Energy dissipation is weak regarding kinetic and potential energy, but its value controls the displacement amplitude at resonance.

For steel materials:

$$T_a = \frac{2\pi}{\omega_0} \quad => \quad \delta = 2\pi\xi\delta = \omega_0 . \xi \, T_a \quad \textit{Eq. 11}$$

2.2.2. Added mass

This term is used in fluid mechanics and it is the inertia added to the system because the ship is accelerating or decelerating, carrying with her a volume of fluid around her.

A hydrodynamic added mass distribution must be estimated and superimposed on the vessel mass distribution in order to obtain natural frequency estimates with any degree of realism. Estimation of the required added mass distribution to be used in calculating the hull girder vertical modes by way of non-uniform beam analysis is the subject of the next subsection.

Hydrodynamic added mass

Ships are unlike most other vehicles in respect to the substantial inertial effects to which they are subjected by the high density of the fluid in which they operate. The water inertia forces, being proportional to ship surface accelerations, imply an equivalent or effective fluid mass imagined to accelerate along with the ship mass. This effective mass is named “hydrodynamic added mass”.

Hydrodynamic added mass is usually large.

The amount of added mass depends mainly on mode vibration, ship form and water depth. The effect of added mass is smaller for the higher modes of vibration than for the lower ones.

Empirical methods could be used to calculate the added mass and will allow the obtaining of approximate values for added mass which could be compared with the configured added mass in Patran.

Burril or lewis methods could be used to estimate the added mass.

Empirical calculation of added mass

Burril developed in the 1935 a simple estimate to calculate the first vertical mode of vibration of ship, the formula is:

$$Nv = \varphi * \sqrt{\frac{1}{(\Delta + \Delta_1) * L^3 * (1 + Ns)}} \qquad \textit{Eq. 12}$$

Where

Nv= frequency of the first vertical mode of vibration [Hz]

I = Inertial moment of the main section [feet]

Δ = Displacement [tons]

Δ_1 = Added mass [tons]

L = length between perpendiculars

Ns=Shear correct factor

φ = empirical constant

By Lewis, the hydrodynamic added mass per unit length at longitudinal position *x* along the vertically vibrating ship is:

$$m(x) = (\pi/8)\,\rho B^2(x)C(x)J_n \qquad \textit{Eq. 13}$$

Where *ρ = density of water [t/m^3]*

B(x)= section beam [m]

C(x) = section 2D added mass coefficient

J_n= Lewis factor, represents a reduction factor of the 2D to 3D added mass.

The *C(x)* is determined used the Lewis form conformal mapping of ship sections.

The equation *Eq.* ***13*** could be rewritten to make easier the definition of J-Factor.

$$m(x) = m_{2-D}(x)\,J_n \qquad \textit{Eq. 14}$$

$$m_{2-D}(x) = (\pi/8)\,\rho B^2 C \qquad \textit{Eq. 15}$$

and making the integration of this along the hull length, it results:

$$J_n = M/M_{2-D} \qquad \textit{Eq. 16}$$

J_n is the ratio of the total added mass in n-node vibration to the total value assuming two dimensional flow section by section. He assumed that the ratio for a ship was almost equal to a spheroid of the same beam/length ratio. The coefficient Jn valid from the spheroid calculations, can be find in the following Figure 5 - Jn values for added mass

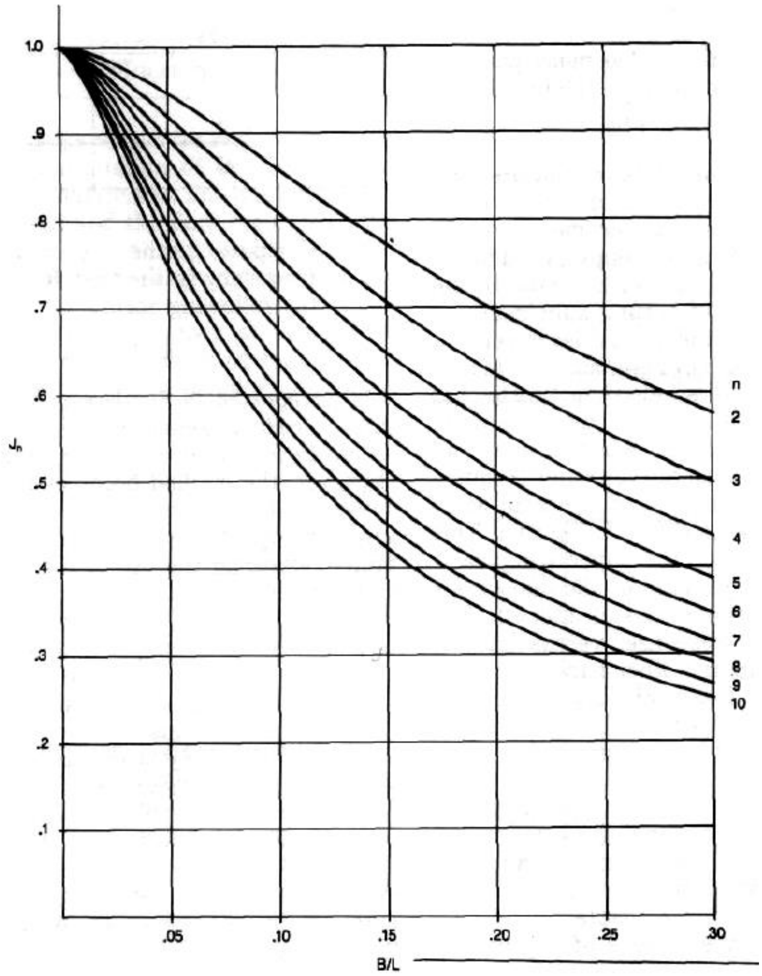


Figure 5 - Jn values for added mass

Any of the previous methods could be used to calculate the added mass, a comparison between these results and the results from FEM model could be a good parameter to know if the results are in a normal range

Estimation of the required added mass distribution for use in calculating the hull girder vertical modes will not be developed in this study, but is subjected to further considerations and next studies.

Added mass in the FEM model

To take into account the added mass, it exists an input function called MFLUID. It is defined as the properties of an incompressible fluid volume for the purpose of generating a virtual mass matrix. This command is very important because it can allow to consider the presence of internal (tanks) or external (added mass), and to consider if it is partially or completely absorbed.

Considering the global vibration analysis, in this case for the added mass MFLUID command was used; the format is:

MFLUID	SID	CID	ZFS	RHO	ELIST1	ELIST2	PLANE1	PLANE2
	RMAX	FMEXACT						

Example:

MFLUID	3	2	15.73	1006	3	4	S	N
		FMEXACT						

Figure 6 - Added mass in Patran

Where:

- SID : set identification number (integer >0) ;
- CID: identification number of rectangular coordinate system used to specify the orientation of the free surface;
- ZFS: Intercept of the free surface of the system referred by CID;
- RHO: Density of the fluid;
- ELIST1 : Identification number of an ELIST with the ID's of two dimensional elements that can be wetted on one side by the fluid;
- ELIST2 : Identification number of an ELIST entry that lists the ID's of two dimensional elements that can be wetted on both sides;
- ▪ PLANE 1 : plane of symmetry, anti-symmetry or not symmetry;
- ▪ PLANE 2: is the plane containing the axes of CID, is plane of symmetry.

In this case the calculation strings to take account of the added mass is:

- mfluid,100,0,1592,1.026E-9,99,,n,n
- elist,99,1,thru,8620

Where it can be noted all the parameters defined earlier.

2.3 Forced vibration

This analysis considers the effect of a time-varying disturbance (load, displacement or velocity) on the response of the system. The disturbance can be a periodic and steady-state input, a transient input, or a random input. The periodic input can be a harmonic or a non-harmonic disturbance, and the forced vibrations can be damped or undamped.

Usually this study is performed after having found the natural modes of the ship. It is possible to simulate the excitations at the frequencies where it is probable to find resonance and like this analyze the behavior of the structure submitted to this excitation.

The type of dynamic loading will determine the mathematical solution approach.

$$M\ddot{x}(t) + kx(t) + B\,\dot{x}(t) = F\sin(\omega t) \qquad \textit{Eq. 17}$$

In this study, only free vibration analysis to obtain natural modes of the yacht were performed, but it is important to consider forced vibrations analysis in further studies of the yacht.

2.4 Natural Modes Analysis

Usually the first step in performing a vibration analysis is determining the natural frequencies and mode shapes of the structure with damping neglected,

Natural Frequency: A natural frequency is a frequency at which a system vibrates when is subjected to a disturbance. The requirement for natural vibration is that the system possesses both mass and stiffness.

Mode: Each different natural frequency of a system defines a mode of system vibration. The modes are ordered numerically upward from the natural frequency with the lowest value.

Mode Shape: A mode shape is a distribution of relative amplitude, or displacement shape, associated with each mode. The deformed shape of the structure at a specific natural frequency of vibration is termed its natural mode of vibration. Some other terms used to describe the natural mode are mode shape, eigenvector and fundamental shape.

The obtained frequencies characterize the behavior of the structure and in this way it is possible to have some information about how the structure will respond under dynamic loading.

Natural frequencies and mode shapes are functions of the structural properties and boundary conditions.

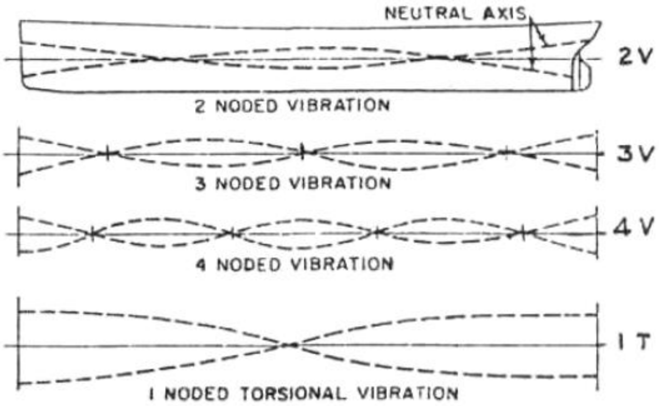


Figure 7 - Ship natural modes

A node is defined as a null point in a distribution of vibratory displacement, or in a mode shape. In general, the number of nodes in a mode shape increases with modal order (natural frequency).

To find natural modes and frequencies, the reduced ***Eq. 18*** needs to be solved. No damping and applied load are considered.

$$M\ddot{x}(t) + Kx(t) = 0 \qquad \textit{Eq. 18}$$

The equation of motion it is possible to rewrite ***Eq. 18*** in the form:

$$x(t) = \emptyset \, . \sin \omega_t \qquad \textit{Eq. 19}$$

Where:

Ø = eigenvector mode shape

ω= circular natural frequency

This is the key in the numerical solution. The harmonic form of the solution means that all the degrees of freedom of the vibrating structure have a synchronized movement.

If differentiation of the assumed harmonic solution is performed and substituted into the equation of Motion, the following is obtained:

$$\omega^2[M]\{\emptyset\} \sin \omega t + [K]\{\emptyset\} \sin \omega t = 0 \qquad \textit{Eq. 20}$$

Simplifying this:

$$([K] - \omega^2[M])\{\emptyset\} = 0 \qquad \textit{Eq. 21}$$

This is the so called Eigenequation, which is a group of homogeneous algebraic equations for the components of the eigenvector and forms the basis for the eigenvalue problem. The basic form of an eigenvalue problem is

$$[A - \lambda I]x = 0 \qquad \textit{Eq. 22}$$

Where

A = square matrix

λ = eigenvalues

I = identity matrix

x = eigenvector

The eigenequation is rewritten in terms of K,ω and M, which are the physical representations of natural frequencies and mode shapes.

There are two possible solution forms for *Eq. 22*

1. $\det\ ([K] - \omega^2[M]) \neq 0$, then the only possible solution is $\{\emptyset\} = 0$

   This is trivial and doesn't provide information from a physical point of view; it represents the "no motion" case.

2. $det([K] - \omega^2[M]) = 0$, a non trivial solution $\{\emptyset\} \neq 0$ is obtained for $([K] - \omega^2[M])\{\emptyset\} = 0$

   The eigenvalue problem is reduced to solve the equation of the form

$$([K] - \omega^2[M]) = 0 \qquad \textit{Eq. 23}$$

The determinant is zero at a set of discrete eigenvalues λ_i or ω_i^2. There is an eigenvector $\{\emptyset_i\}$ which satisfies and corresponds to each eigenvalue. Therefore it can be rewritten as:

$$[K - \omega_i{}^2 M]\{\emptyset_i\} = 0 \qquad i = 1,2,3 \ldots \qquad \textit{Eq. 24}$$

Each eigenvalue and eigenvector defines a free vibration mode of the structure.

The number of possible eigenvalues and eigenvectors is equal to the number of degrees of freedom.

It should be noticed that when a linear elastic structure is vibrating in free of forced motion, its deflected shape at a given time is a linear combination of all of its normal modes as can be seen in the following Figure 8 - Superposition of natural modes

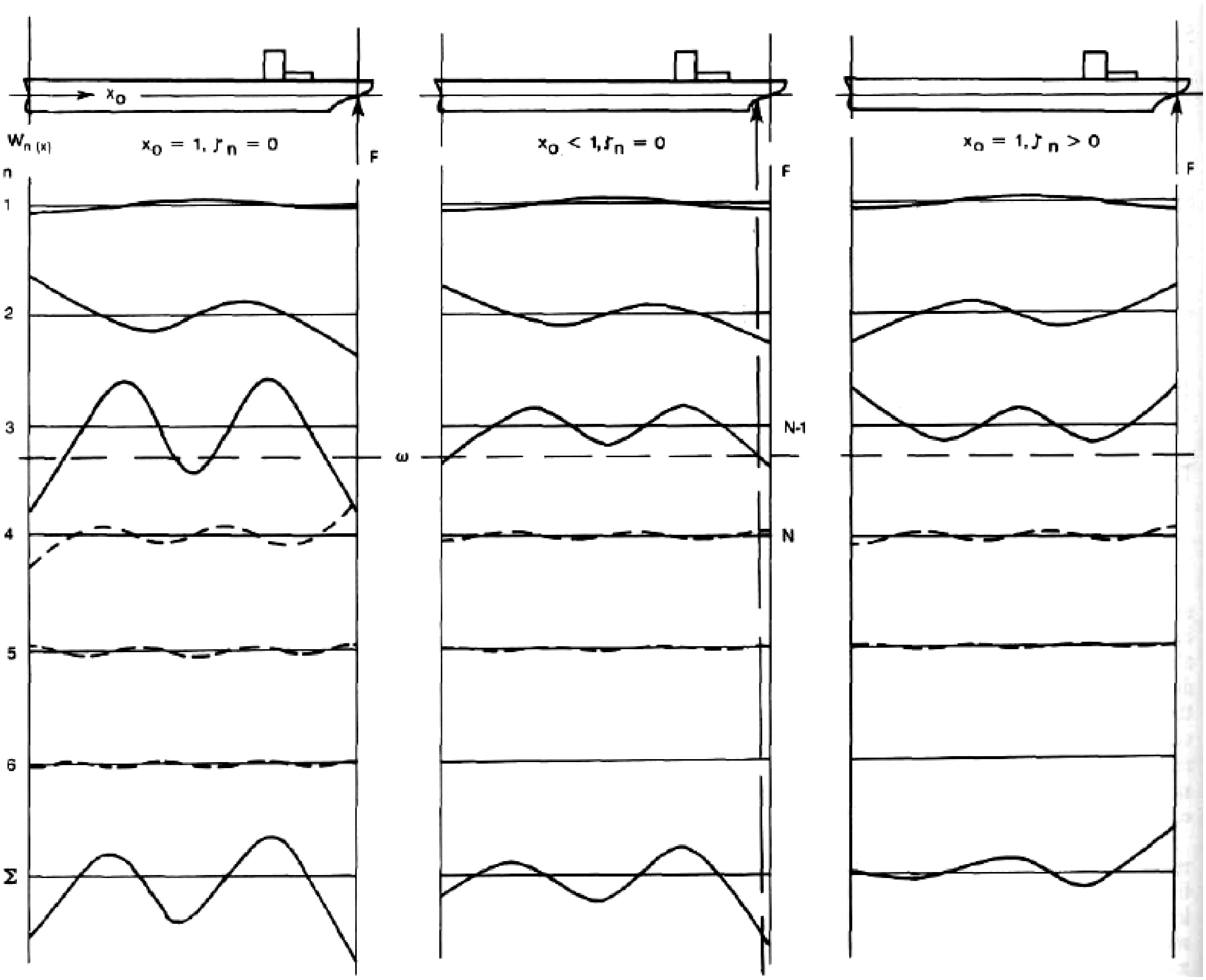


Figure 8 - Superposition of natural modes

For this study the natural vibrations of main deck are studied, the focus is on cinema zone, which will be subjected to the response of the excitation at particular revolutions of the main engines, propellers and auxiliary machinery.

The simplest way to avoid vibrations is preventing the resonance of the excitation source and the natural frequency of the element analyzed. The procedure is successful as long as these two factors could be regarded as independent of environmental conditions, which in the case of ships is frequently unsatisfied, for example different filling states changes the natural frequency in the tanks structures. The design should be selected far from frequency ratio η=1.

A subcritical design must leave all natural frequencies higher than the highest significant excitation frequency. The magnification factor will be dependent on the damping coefficient and also the safety factor.

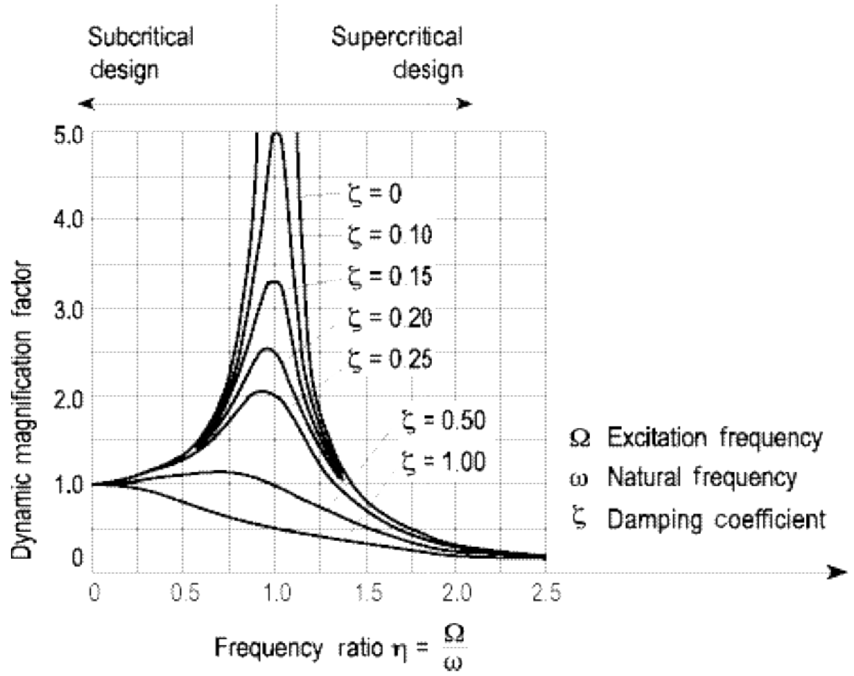


Figure 9- Dynamic magnification for a single DOF system

In the field of vibrations of a ship, three main groups can be distinguished: global hull vibration, vibration of substructures and local vibrations.

In Figure 10, the vibration phenomena relevant in shipbuilding applications are plotted versus frequency. The indicated frequency limits are valid for standard designs and for normal ship types. The transitions between ship motions, ship vibrations and ship acoustics are smooth.

In general, the higher the frequency, the greater the modal density is. As a result, the system response in the higher frequency range is defined by the interaction of more natural modes than at low frequencies.

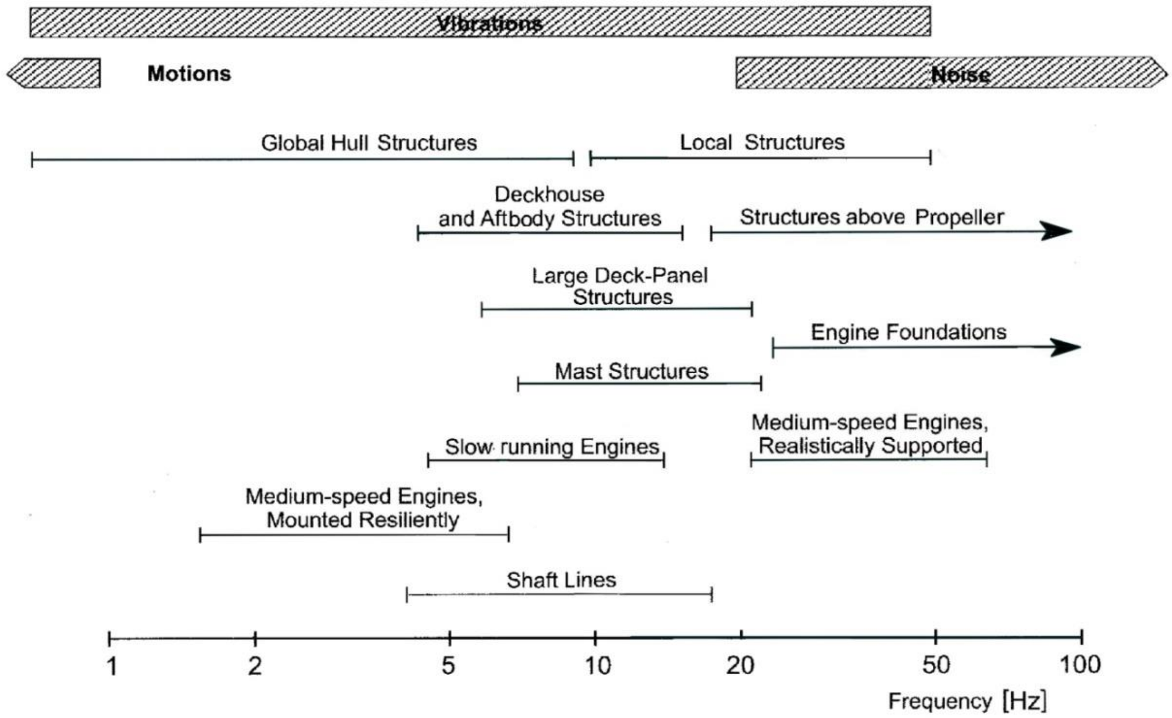
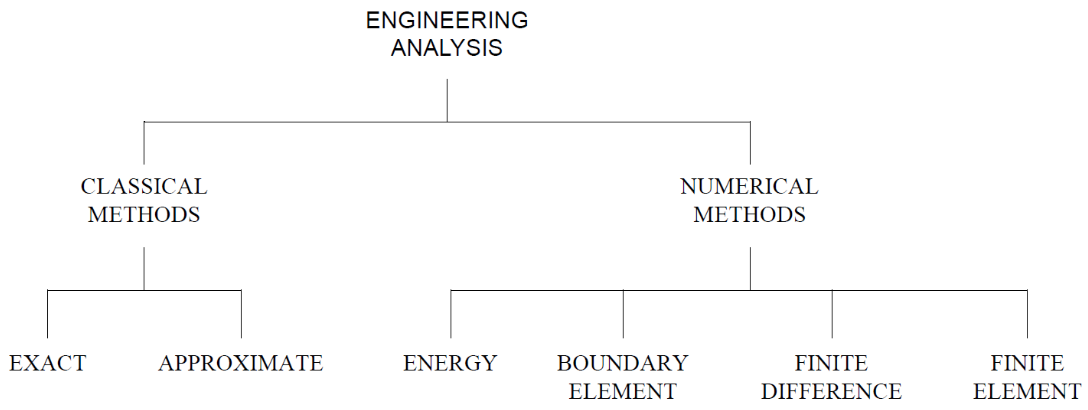


Figure 10- Natural Frequency ranges in shipbuilding

2.5 Finite element method

To solve a mathematical model, it is necessary to discretize the differential equations. Engineering analysis can be divided in two main categories:

1. **The classical methods**; which can result in exact or approximate solutions. Here the problem is solved directly forming governing differential equations based on fundamental principles of physics. Exact solutions are possible only for simplest cases of geometry, boundary conditions and loading. A wider variety of problems can be solved using governing differential equations. These solutions take the form of series expansions that are after truncated to a reasonable degree of convergence. This type of solutions can be used only in simple cases.
2. **Numerical methods.** Here the governing differential equations are discretized; the approximate solution of the boundary value problem obtained by BEM has the distinguishing feature that it is an exact solution of the differential equation in the domain and is parametrized by a finite set of parameters living on the boundary. Finite-difference methods (FDM) are numerical methods for solving differential equations by approximating them with difference equations, in which finite differences approximate the derivatives.



Finite element analysis seeks to approximate the behavior of an arbitrarily shaped structure under general loading and constraint conditions with an assembly of discrete finite elements. Finite elements have a regular geometric shapes and known solutions. The behavior of the structure is obtained by analyzing the collective behavior of the elements.

The first step to solve partial differential equations is to discretize the equations to be able to solve them. In the case of FEM models, the discretization is done through a mesh, which is a geometrical subdivision; they could have different shapes and could be 1d, 2d, or 3d elements. Each node has a certain number of degrees of freedom (DOF). To have an exact solution, the equilibrium and the congruency must be in each internal point of the continuum.

The solutions of the differential equations can be able to give laws of variation of the looked quantities in function of the coordinates in each point.

The fundamental laws of the structural analysis are based on three principles:

- Equilibrium between external forces and internal strength;
- Congruence of strain and deformation;
- Connection between strength and strain due to material behavior.

For real eigenvalue extraction, several methods are provided in MSC Nastran. These are numerical approaches to solve the natural frequencies and mode shapes. The choice for the most suitable method is based on the efficiency of the solution process.

Two groups are distinguished in the extraction of eigenvalue method: Transformation methods and tracking methods. In the first one, the eigenvalue equation is first transformed into a special form from which eigenvalues may easily be extracted. In the tracking method, the eigenvalues are extracted one at a time using an iterative procedure.

For this study, Lanczos Method is used and is briefly described below.

8.1.2 Methods of Computation

The software MSC Nastran has seven different methods to extract the real eigenvalues. All of them are specified for all existing problems and they are numerical approaches to solve natural frequencies and modes shapes.

The methods of eigenvalue extraction belong to one or both of the following two groups:

• Transformation methods;

• Tracking methods.

In the transformation method, the eigenvalue equation is transformed into a special form from which eigenvalues may easily be extracted. In the tracking method, the eigenvalues are extracted one by one using an iterative procedure. Lanczos method is the method to use in this case, which combines the best characteristics of both the tracking and transformation methods.

Lanczos method

The Lanczos method requires that the mass matrix be positive semidefinite and the stiffness be asymmetric. Like the transformation methods, it does not miss roots, but has the efficiency of the tracking methods, because it only makes the calculations necessary to find the roots requested by the user. This method computes accurate eigenvalues and eigenvectors. Unlike the other methods, its performance has been continually enhanced since its introduction giving it an advantage. This method is the preferred method for most medium- to large-sized problems, since it has a performance advantage over other methods.

Also, Lanczos uses Sturm sequence logic to ensure that all modes are found. The Sturm sequence check determines the number of eigenvalues below a trial eigenvalue and then finds all of the eigenvalues below this trial eigenvalue until all modes in the designated range are computed. This process helps to ensure that modes are not missed.

2.6 Excitations

2.6.1. Machinery

Main Diesel Engine

Diesel engine vibratory excitation can be considered as composed of three periodic guide forces and three periodic guide moments acting on the engine foundation. The periodic force component along the axis of the engine is inherently zero, and some other components usually balance to zero depending on particular engine characteristics.

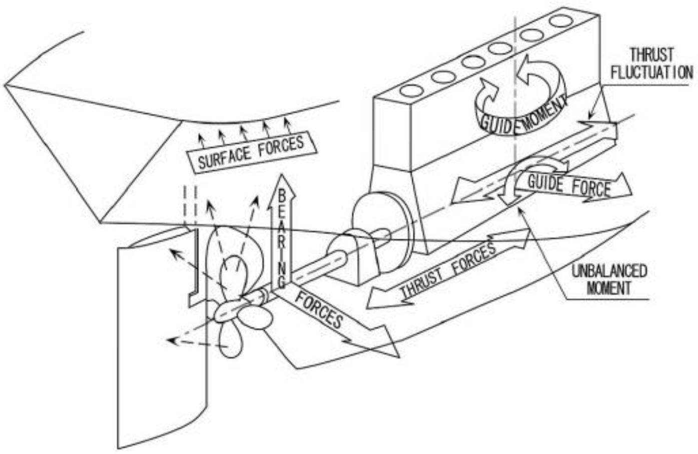


Figure 11 - Forces and Moments induced by main engine and propeller

The vertical force and moment, which are of primary concern with regard to hull vibratory excitation, and the transverse force and moment as well, are due to unbalanced inertial effects. For the engines of more than two cylinders, which is the case of interest with ships, the vertical and transverse inertia force components generally balance to zero at the engine foundation.

This leaves the vertical and transverse moments about which to be concerned. The majority of low speed marine diesels currently in service have 6 cylinders or more. Therefore, the 2nd order vertical moment M_{2v} is generally considered to contribute the most to the hull vibration.

The excitation due to the first harmonics of low speed diesel engines can be at frequencies close to the first natural hull girder frequencies, thus representing a possible cause of a global resonance.

Normally, it is only in case of slow-running main engines that the ship's structure exhibits significant global vibrations caused by internal orders of excitation. Therefore, in computation practice, the engine structure of medium-speed and fast-running machines is not simulated for the purpose of considering excitation forces.

The main engine used in this case will be high speed a diesel engine, operating in the at 2300 RPM, through a gear box (3.542:1) the propulsion goes to the propeller shaft. This will lead a high frequency excitation and should be analyzed since it will be one of the main sources of excitation. Usually in this type of engines the customer ensures that there is no significant imbalance which could affect the global ship vibration, but in the reality there is always some unbalance.

Auxiliary machinery

Diesel engines used as generators, air conditioning, and other auxiliary machines will operate at high speed and therefore the same considerations for main engines are taken into account.

2.6.2 Propeller

From the propeller, excitation forces are transmitted into the ship via the shaft line and in the form of pressure pulses acting on the ship's shell. Whereas shaft line forces are the most significant factor for vibrations of shaft lines, the predominant factor for vibrations of ship structures are pressure fluctuations.

Shaft line forces

The creation of these forces can briefly be described as follows: the relative velocity between the individual profile section of the propeller blade and the water depends on the superposition of the ship's speed and the peripheral velocity at the profile section under consideration. As a simplification, the influence of the wake can be considered as the change in the angle of attack at the profile section, this change being proportional to the inflow-speed variation. The simple computation methods are based on quasi-static considerations that determine thrust and tangential forces at the individual blade directly from the wake induced variation of thrust and moment coefficients throughout one revolution

Pressure fluctuations and thrust fluctuations

Pressure fluctuations acting on the shell are a result of several physical causes:

- Displacement effect (thickness effect) of the rotating propeller. This effect is independent of the wake field, and its contribution to the overall pressure amplitude for the propeller of a merchant ship is about 10 to 30%.

- Portion resulting from or induced by the pressure difference between the back and the face of the blade. This effect, too, occurs independently of the wake field and contributes up to about 10% to the overall pressure amplitude.
- Displacement effect of the fluctuating cavitation layer that typically forms when the propeller blade is moving through the wake peak in the region of the outer radii. For the propeller of a merchant ship, the contribution of this effect to the overall pressure amplitude is approximately 60 to 90%.

Pressure pulses on the shell are also caused by the induction and displacement effect of the propeller tip vortex and the collapse of the individual cavity bubbles.

Considering the above mentioned, it can be concluded that high excitation forces can be obtained only when cavitating propellers are installed.

Pressure pulses caused by a fluctuating cavitation volume V at a point situated at a distance r can be approximate by the following formula:

$$p = \frac{\rho}{4\pi} * \frac{1}{r} * \frac{\partial^2 V}{\partial t^2} \qquad \textit{Eq. 25}$$

Where ρ = density

2.6.3 Sea waves

Considering the results of previous studies it is well known that this factor could influence mainly big ships such as cargo ships, which in full load condition could reach a natural mode for the hull beam around 0.8 Hz.

This is about the encounter frequency that a ship could experiment in some sea states. However this type of vibration could be avoided simply changing the course of the ship and therefore modifying the encounter frequency between the vessel and the wave and makes disappear the resonance frequency.

Springing

Sea waves can result in ship vibrations usually in resonance between the lowest vertical natural frequencies of the hull with wave encounter frequency. This is not a big issue regarding the habitability but it could be relevant for structural fatigue.

Slamming and whipping

These impacts can cause transient global vibrations in the form of whipping as well as local response. In these cases structural strength issues are more relevant than induced vibration problems.

2.7 Vibration criteria and specifications

The design objective of all new ship construction is to meet the criteria or specifications invoked for that project. To control and/or minimize shipboard vibration, it is also necessary to stipulate applicable criteria in specification format. Over the years, a large number of proposal for criteria or guidelines on various aspects of vibration have been put forward. Sometimes such proposal lacks in clearness of concept regarding parameters obtained, range of application and conditions catered for the guidelines.

To define acceptable criteria it is necessary to establish the damages that vibration can create on the yacht. Shipboard vibration is considered excessive when it results in structural damage, damage or malfunction of vital shipboard equipment, or adversely affects the comfort or efficiency of the owner or guests of the boat. Thus, the criteria recommended are based on existing requirements related to:

- Human reaction: the criteria for human reaction throughout the ship remain the same for all areas designated accommodations or working spaces;
- Machinery and equipment malfunction: may occur as a result of the vibration of those structural components to which the equipment is attached or may be due to the sensitivity of the equipment;
- fatigue failure: occur in major ship structures such as the hull girder or bow area in extreme weather conditions.

ISO 6954 (1984) has also been widely used as acceptance criteria for crew habitability and passenger comfort. The criteria are designed to ensure vibration levels below a certain level which crew and passenger do not experience discomfort. ISO 6954 criteria are shown in Figure 97, which can be transformed into the following:

• For each peak response component (in either vertical, transverse, or longitudinal direction), from 1 Hz to 5 Hz, the acceleration is acceptable below 126 mm/sec 2, and adverse comment is probable above 285 mm/sec2

• For each peak response component (in either vertical, transverse, or longitudinal direction), from 5 Hz and above, the velocity is acceptable below 4 mm/sec, and adverse comment is probable above 9 mm/sec.

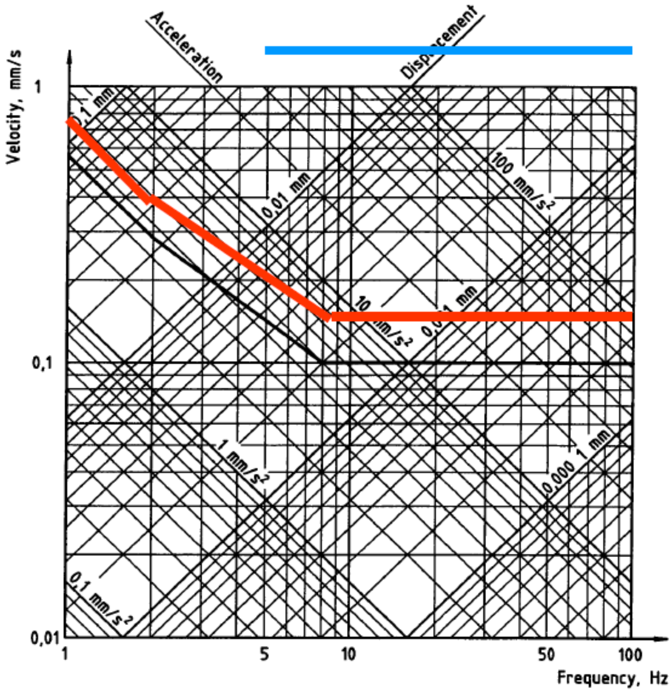


Figure 12 - Human frequency response

To summarize vibration limits in yachts and ships the following table is provided:

Vibration limitation	Underway (mm/s)	Harbor (mm/s)
Avoiding motion sickness and extreme discomfort	7	
Avoidance of vibration induced fatigue	(24hrs) 5	
Ability to sleep comfortably	3	1
Luxury smooth	1	0.25

Table 1- Velocity limitation for yachts

The limits related to discomfort and fatigue are fixed values, independent of the seaway induced motions. The comfort and luxury zones are related to the total sensory environment, including the seaway induced motions, noise, and even to the visual perception of motion.

In the last years the rule was changed, ISO 6954(2000) is the actual guideline for the measurement, reporting and evaluation of vibration with regard to habitability on passenger and merchant ship. The principal difference between these two normative is the results of each measurement; the ISO 6954(2000) impose that the measurements "shall be the overall frequency-weighted r.m.s. value as defined for acceleration in ISO 2631-1:1997, 6.4.2. A similar procedure is applicable for the frequency weighting of velocity spectra".

In this way the normative will become more conservative than the previous one that used the singular component of the velocity. Following graph describes the new normative of evaluation of vibration.

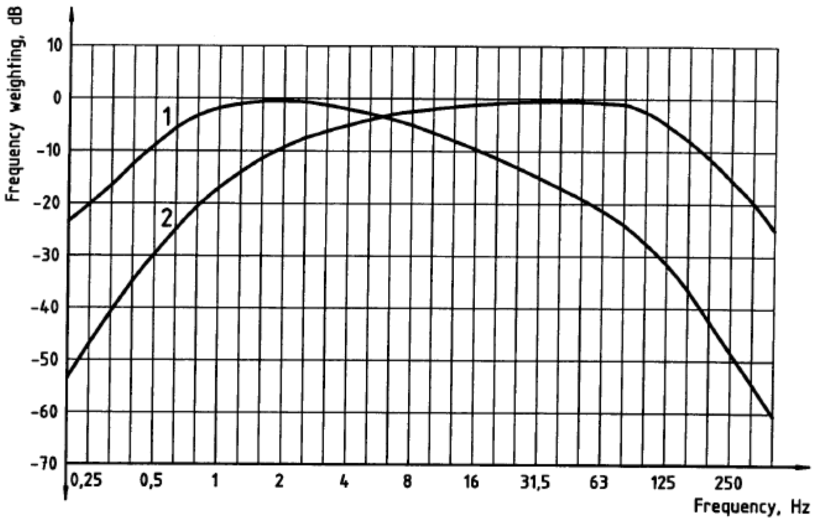


Figure 13 -- Combined frequency-weighting curves. Band limitation

	Acceleration (mm/s^2)		Velocities (mm/s)	
	Minimum	Maximum	Minimum	Maximum
Cabins and lounges	31	63	1	2
Wheelhouse	47	94	1	2
Open decks	63	110	2	3,5

Table 2 - Vibration levels requested by rules

3. CHARACTERISTICS OF THE STUDY

3.1 Motor yacht

The motor yachts represents one of the largest categories in the yacht manufactory; this type of craft can be designed and built in different ways regarding the requirements of clients and the needs of builders. The principal differences among yachts are due to internal and external layout spaces, building methods, performance, materials, propulsion systems, etc.

This concept of "*Fully habitable yacht*" is one of the principal targets of Motor Yacht category: wide superstructure extends for almost the entire length of the vessel and it provided with large windows to guarantee a good natural light. Commonly the crew is living in a different deck onboard, and the guests and owners in different decks.

The analyzed yacht is a particular craft composed by large internal areas distributed in five different decks: Bottom deck, lower deck, main deck, owner's deck and sun deck.

The yacht was designed with big internal and external areas. The bottom deck is where the engine's room is, and living accommodation thought mainly for the crew. In the interior of the lower deck bigger rooms were projected and more luxurious spaces to rely on, where the guests will be housed, the kitchen is also here and a big garage where the tender will be saved was located in the mid part of this deck, this leads a big opening in the side of the vessel. In the aft part of this deck, still in the interior, a big Cinema/ Lounge and small bar was placed. This has two large doors which are connecting this space with a 4 meters exterior deck, so this can be transformed in a big living room. In the interior of the main deck was placed the Captain's cabin, the wheel house, the dining room, the living room. The exterior deck has a big space with a solarium. Owner's deck counts with a big interior where the bedroom is, outside a big solarium to enjoy the exterior. Sun deck is the so called "flybridge" with a large open space.

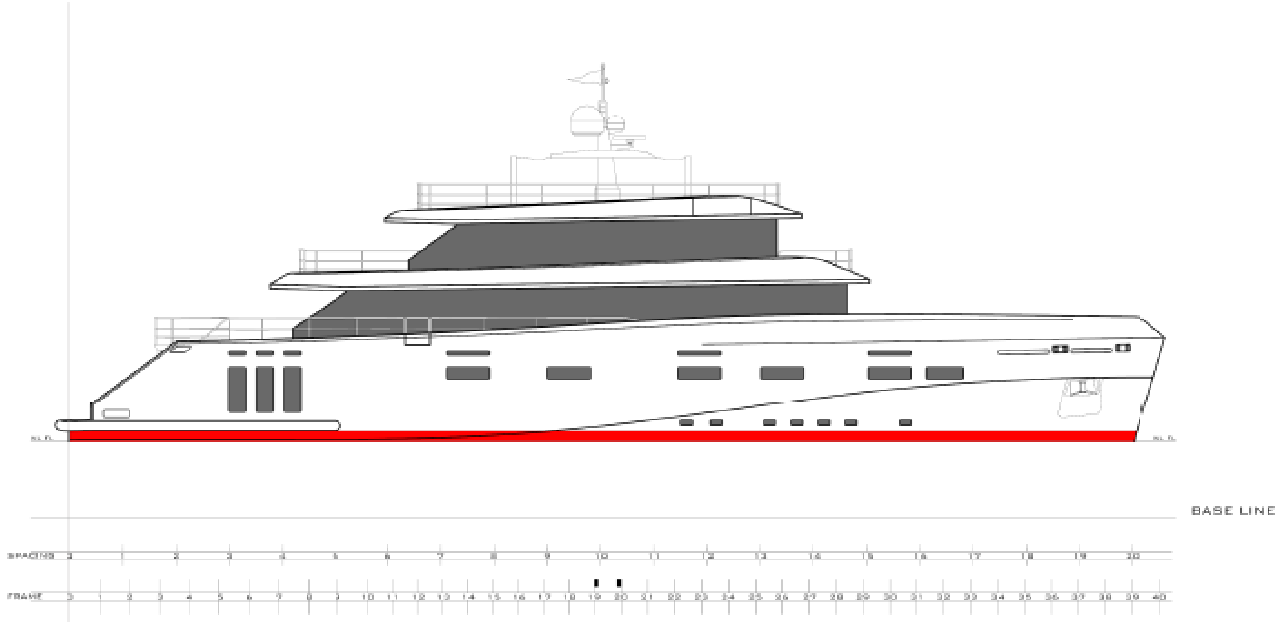


Figure 14 – Profile view of the analyzed yacht

3.2 Main Characteristics

The main characteristics of the yacht under analysis which are useful for this study and to understand the problem are listed below:

Technical Data	Value	Unit
Length Over All - LOA	40.8	m
Length waterline - LWL	39.2	m
Beam max	9.40	m
Draft	2.79	m
Depth	6.3	m
Full load Displacement	438	tons
V max	16	kn

Propulsion Data	**Value**	**Unit**
Engine model CAT C32	x 2	
Power	1081	m
RPM	2300	
Weight	3245	kg
Gearbox	3.542:1	

Propeller Data	**Value**	**Unit**
Diameter	1.4	m
Number of blades	5	

4. FINITE ELEMENT MODELLING

Some basic approximations should be done in the 3D model available, it is needed to find a good compromise between the computational time, the certainty of the results, and the time used to produce the model.

The aim in this case, is the computation of natural frequencies and natural shapes, for a global analysis vibrations, it is intended to perform a comparison between the fem model and vibration measurements on board, to find how precise and accurate is the model, and how trustful the results are.

Thus it is important that it is respected the main structure without altering nothing, small parts which are not relevant for the hull beam structure could be avoided if is not relevant . The principal reinforcements (transverse and longitudinal) and all the principal structure as decks and bulkheads should be modeled in a truthful way as better as possible; these structures influence the frequencies response, so this is the approach to have a proper modelling. The secondary reinforcements as longitudinally decks or longitudinally bottom have been also included. The tertiary reinforcements (grate bars, brackets) have been neglected because they do not influence the results; however, it was take account of them in the weights distribution which it discussed forward.

The subdivision between primary and secondary structures is fundamental for the proper dimensioning of the sections of the various structural elements. To summarize a structural scheme of a ship, the various structural components can be classified in several modes:

- Depending to the shape and to the structural functions, there will be bi dimensional elements (plates) and mono dimensional (beams);
- Depending to the orientation, there will be structural elements longitudinal or transverse;
- Depending to the dimensions, there will be structural elements principal (reinforced) or secondary (commons)

4.1 Geometry in Patran

In Patran it is possible to create the Finite Element Model, but to do this, it is necessary to start from a 3D model of the craft, it would be a very hard task to model the ship in Patran directly. Usually this is not a problem, because the shipyards or design offices have this files if the project is going on.

These files were provided by the shipyard (CCN), and they contain all the tridimensional surfaces or lines that describe both the internal and external part of the hull (including all the secondary and primary reinforcements). The complete model of the motor yacht was created with Microstation, the files created in this software (.DGN) are not compatible with Patran, therefore the files have been exported as .IGES, in this way it has been possible to load them in Patran.

Each file was including a block of the hull, corresponding to construction blocks.

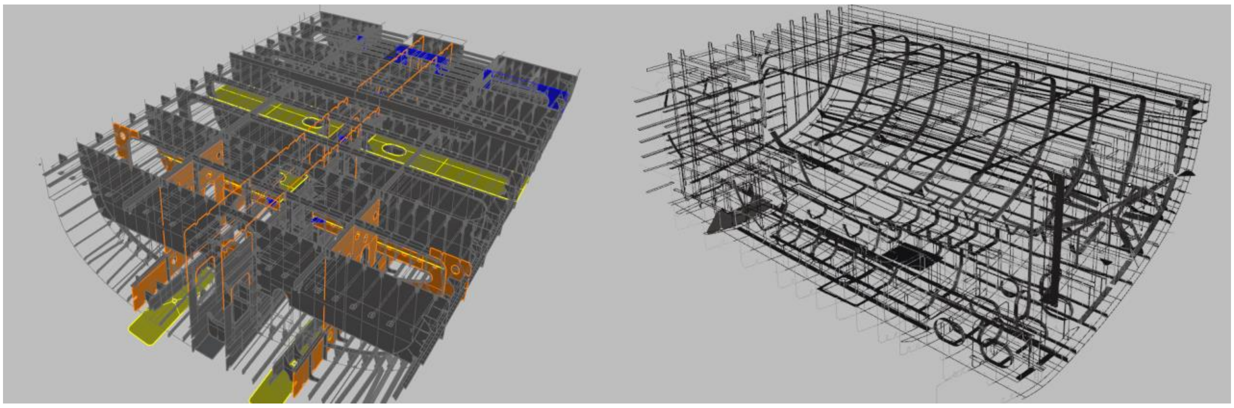


Figure 15 - .IGES Blocks

The original model was composed by lines describing the boundaries of surfaces, polysurfaces considering the thickness of the members, and geometries that are not useful when this is imported in Patran.

Therefore, the first step was to clean these .IGES files to have lighter files only with useful information to create the FEA model.

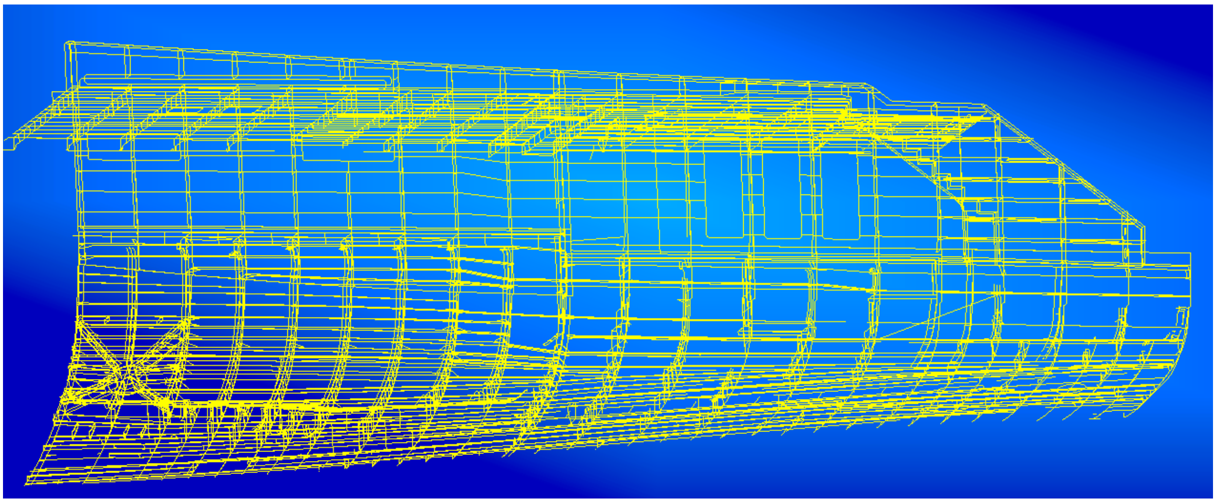


Figure 16 - Geometry in Patran

To create the Finite Elements model, the principal and secondary structure leads the borders of the mesh, because of the necessity to have nodes in the joints. Thus, the mesh can't be automatically created through a complete surface because nodes are placed in wrong places and the correction after takes lot of time. In the following figure the yellow lines represents the used geometry for the mesh generation, it can be noticed that between all of this one element is created.

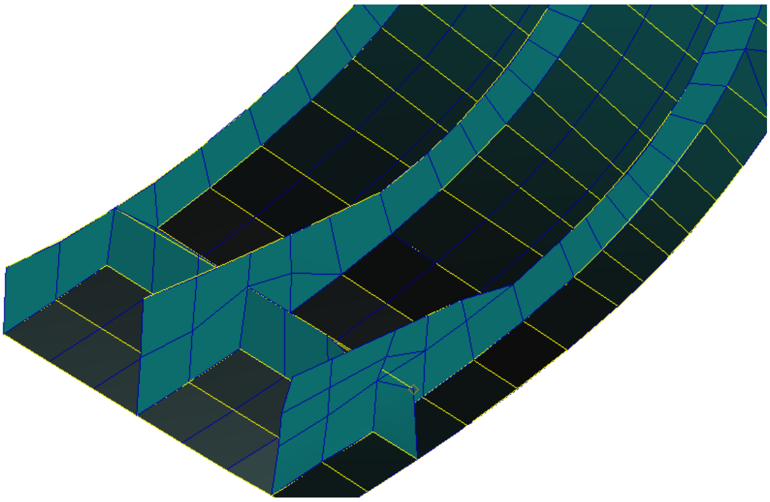


Figure 17 – Geometry and mesh creation

Global coordinate system of the model is right hand Cartesian coordinate system:

- X direction goes along ship's length pointing to bow;
- Y direction goes along ship's breadth pointing to Portside;
- Z direction goes along ship's depth pointing to deck.

Structural model and applied loads in this model are in International System of Units (tn, mm, s)

4.2 Meshing

4.2.1 Shell and structure of the yacht

The discretization of the plating (bottom, sides and decks) is the first step carried out by Patran. It was created with the imported lines of the geometry, creating meshes between 2 curves. When the surfaces were imported in a proper way, the mesh on the surface was possible.

It is possible to see that it was necessary to respect the orientation and position of the stiffeners. Because of this, most of the meshing process was performed "manually". In other words no automatic meshing algorithms were used, except for the "IsoMesh", which allows generating a regular triangular or quad mesh between two curves or surface. In this case it is necessary to have defined the borders where the structure is located.

Shell and primary structure

All plate structures, such as shell of the hull, transverse bulkheads, beams, web frame and longitudinal bulkheads etc. are modelled by CQUAD4 and CTRIA3 shell elements. The model was created as accurate as possible regarding the limitation of mesh size.

Mesh size in longitudinal direction is strong frame spacing, in this case it was chosen mainly to use shell elements with order of magnitude equal to 1/3 the distance between two secondary transversal elements of the ship, and in transverse direction the mesh size is equal to longitudinal spacing.

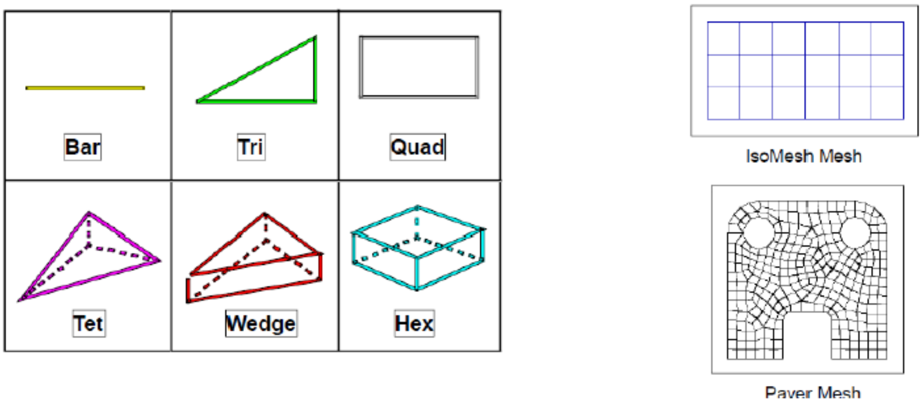


Figure 18- meshing elements (left) and algorithms (right) available in Patran

In the greatest part of the model square elements CQUAD4 were used in order to have a better discretization and therefore more accurate results; in general, almost all the elements have sides comparable with half the distance between two secondary longitudinal elements, this is variable along the hull but the element size is about 300 mm for all the model. In some parts CTRIA3 were used to respect the geometry of the vessel.

CQUAD4 – Quadrilateral plate element: this can represent in plane bending, transverse shear behavior, depending upon data provided. This element is a quadrilateral and connects four grid points.

CTRIA3- Triangular shell element. This is mostly used for mesh transitions, and irregular boundaries, this could exhibit excessive stiffness. Regarding behavior is similar to CQUAD4

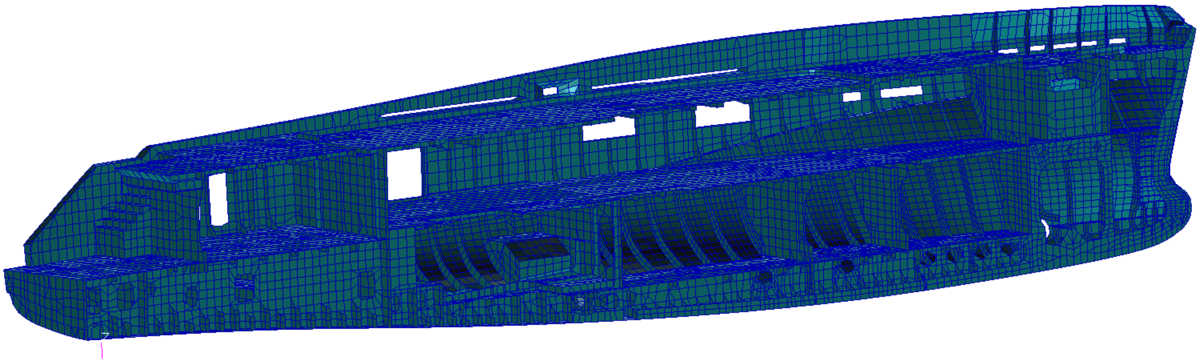


Figure 19 - Shell and primary structure

SECONDARY STRUCTURE – CBAR ELEMENTS

For the secondary structure, it is necessary to create CBAR elements. These elements supports tension and compression, torsion, bending in two perpendicular planes. And shear in two perpendicular planes. The CBAR uses two grid points and can provide stiffness to all six DOF's of each grid point. Its formulation is derived from a classical beam theory (plane sections remains plane) which will be useful after to create the 1D element. It is important to join all the elements and check that they are sharing a same node

Congruent mesh

In many cases small gaps exist between the nodes, therefore it is needed to apply Equivalence in the model, to make collapse these nodes and avoid problems in the future analysis.

Congruence of the mesh: Refers to the condition that every node of adjacent elements should be connected. In other words there should not be a node of an element on the edge of another neighboring element. Otherwise, the program would interpret the elements as belonging to different bodies.

Sufficient skweness: Independently of the solution algorithm used in the analysis, the results obtained depend on the shape of the element. The skewness refers to a relation between the shape of the actual and the optimal cell, where all the edges lengths and corner angles are the same. Presenting poor skewness (or aspect ratio) elements do not necessary stop the program to achieve a solution, but it may lead to erroneous results. Therefore a minimum skewness ratio is advisable.

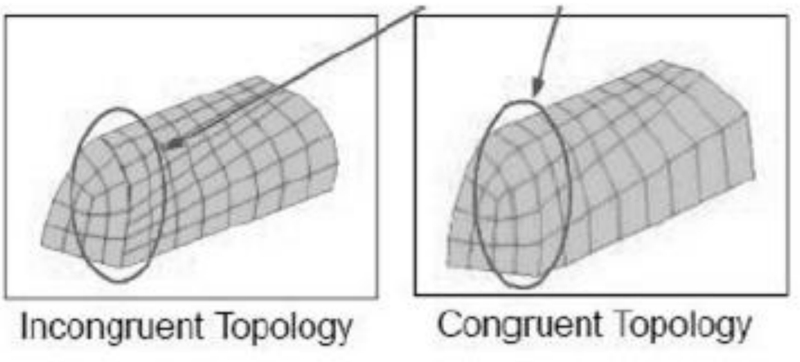


Figure 20 - Incongruent mesh

MSC Patran provides automatic tools to check this and other possible modelling errors such as fixed nodes, nodes without stiffness, intermediate nodes on element edges not connected to the element, trusses or beams crossing shells, double elements, extreme element shapes (element edge aspect ratio and warped elements), incorrect boundary conditions, etc.

4.2.2 Material

When using the MSC Patran computational platform, as a unit system is not imposed, it is necessary to insert the dimensions according to a consistent system of measurements. Different consistent measurement systems are shown in the following table

Quantity	SI	SI(mm)	SI	US Unit(ft)	US Unit(inch)
Length	m	mm	m	ft	in
Force	N	N	kN	lbf	lbf
Mass	kg	$tonne$ $(10^3 kg)$	$tonne$	$slug$	$lbf\ s^2/in$
Time	s	s	s	s	s
Stress	Pa (N/m^2)	MPa (N/mm^2)	kPa	lbf/ft^2	psi (lbf/in^2)
Energy	J	mJ (10^{-3}J)	KJ	$ftlbf$	$inlbf$
Density	kg/m^3	$tonne/mm^3$	$tonne/m^3$	$slug/ft^3$	lbf s^2/in^4

Figure 21 - Units systems

In our case, the analysis was performed using the SI(mm) measurement system. Here, for example, the steel density of (7800 kg/m^3) should be inserted as ($7.8\ x\ 10^9\ tn/mm^3$) .

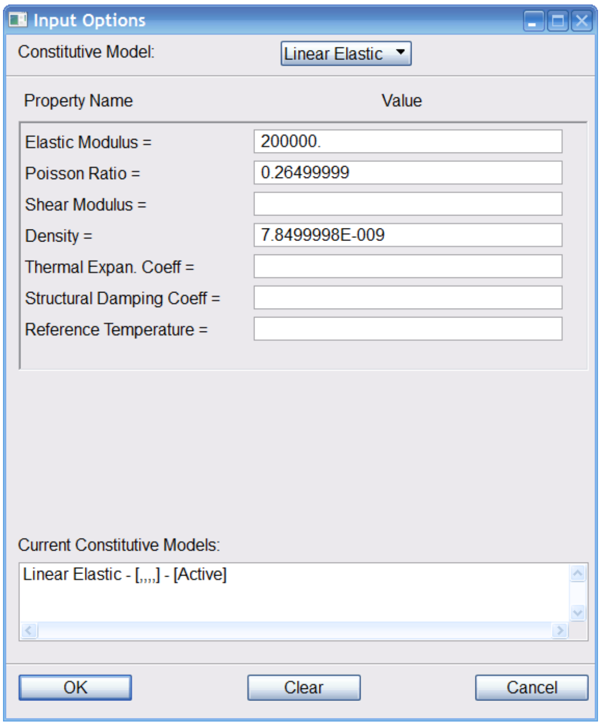
In this case the material used for all the hull, structure of the hull, lower and main deck, is a low carbon steel: ASTM A131 Grade EH36. With this purpose an Isotropic material was created in Patran, so the material property doesn't change with direction of the material.

It is enough for mechanical properties to introduce the Elastic modulus and the Poisson ratio, the density is optional but in this case is of interest because it will give the weight of the model.

The data given by the shipyard corresponding to this material is:

- Minimum yield stress 355 N/mm^2
- Ultimate min tensile strength 490 N/mm^2

The input data in Patran to simulate this material are the following:



4.2.3 Element properties

2D SHELL ELEMENTS.

These types of elements were used for hull shell, bulkheads and all principal members of structure. To create these elements it is necessary to define the material used and the thickness.

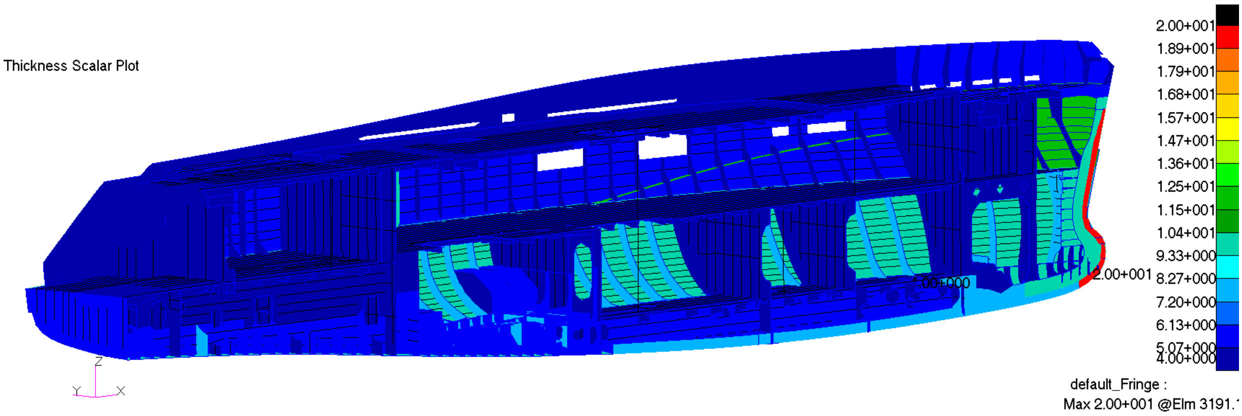


Figure 22 - Shell thickness plot

BEAM ELEMENT PROPERTIES

Beam elements are used for the secondary structural members, such as side longitudinals, deck longitudinals, beams and bulkhead stiffeners.

The creation of this property requires a vector to define the orientation of the cross-section. Several profiles were used in the construction of this yacht, mainly bulbs and L of different sizes. L sections are available in Patran library, but are not the case of bulbs and plates. So the L section was used to obtain the profiles applied for construction. Both bulb sections and simple plates were simulated with L sections because of the lack of these profiles in the software. An example is shown below.

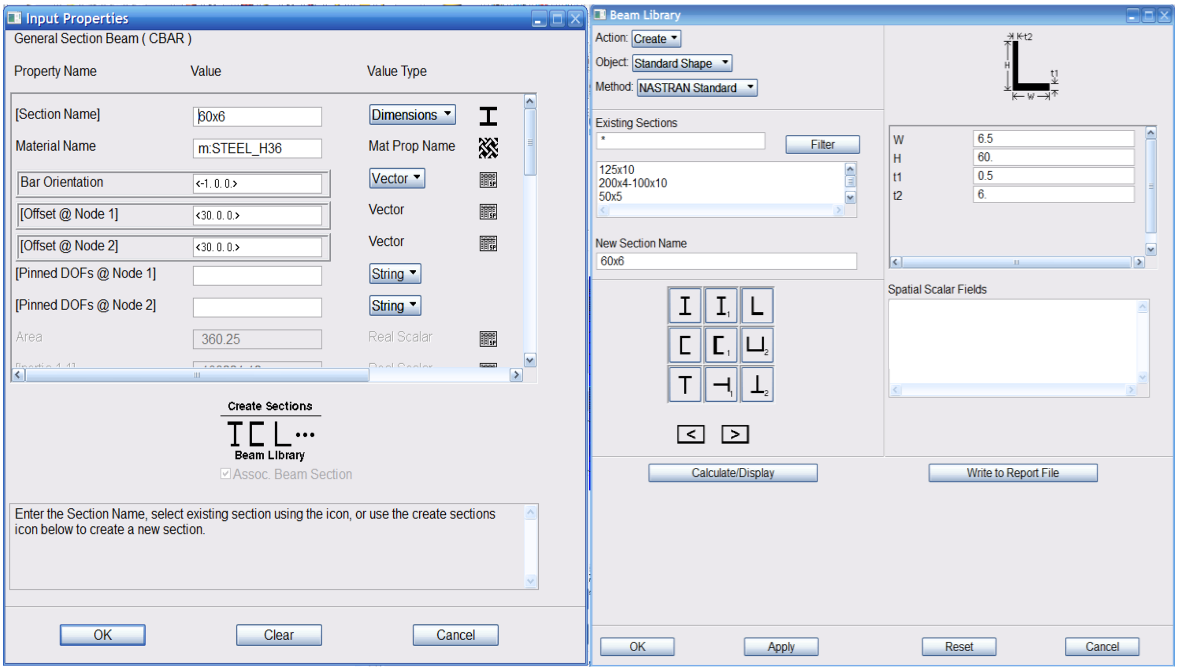
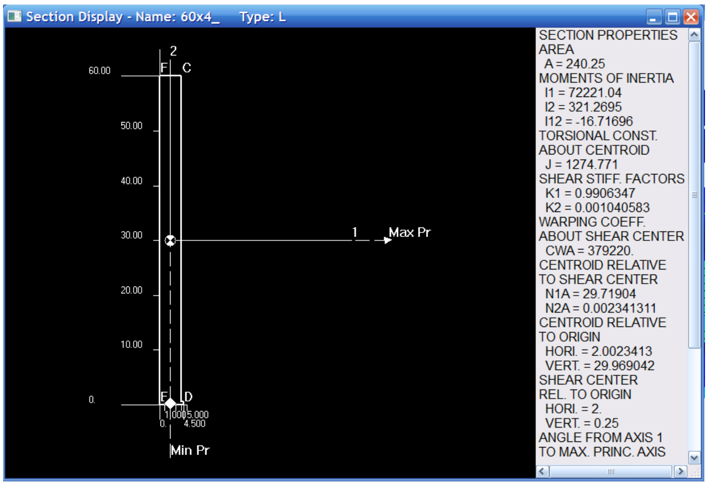


Figure 23 - Beam elements properties

Details of the created profile are given by the software:



Details of used sections are presented in the following table. The bulb profiles were simulated through L sections, due to the lack of bulb profiles in the software.

SHAPE	Nomination	Area (mm^2)	Centroid (mm)	Inertia (mm^3)
BULB	60*4	358	21,8	122296
	80*5	540	31,1	337515
PLATE	125*10	1250	62,5	1628573
	50*5	250	25	52236
	60*4	240	30	72221
	60*6	360	30	108221
	80*8	640	40	341490

Table 3 - Used profiles

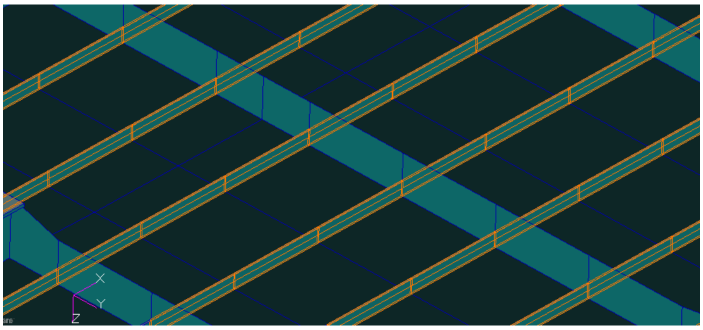


Figure 24 - Beam 1D elements

In Patran library regular beam sections could be found, the L shape was found the most useful. Some calculations were performed in order to obtain the same inertias specified for the used sections in the shipyard for construction.

Comparison of bulb inertias on the model and the ones specified on constructive drawings used in the shipyard are presented below:

		Section (mm^2)	Centroid (mm)	Inertia (mm^4)	SM (mm^3)
FEM BULB	60*4	358	22	122296	5610
REAL BULB	60*4	358	38	122000	3194
FEM BULB	80*5	540	31	337515	10853
REAL BULB	80*5	540	49	338000	6912

Table 4 - Properties of real profiles vs FEM profiles

	Centroid (%)	Inertia (%)	SM (%)
60*4	-27,33	0,12	27,45
80*5	-22,25	-0,07	22,18

Table 5 - Real profiles vs input profiles

Orientation and offset is an important issue, special attention should be given in this point, the offset of the beam should be the subtraction of the height of the beam profile and the centroid height. In this way the beams will be correctly created. Vector to define orientation should be input in such a way that the beam remains approximately perpendicular to the shell.

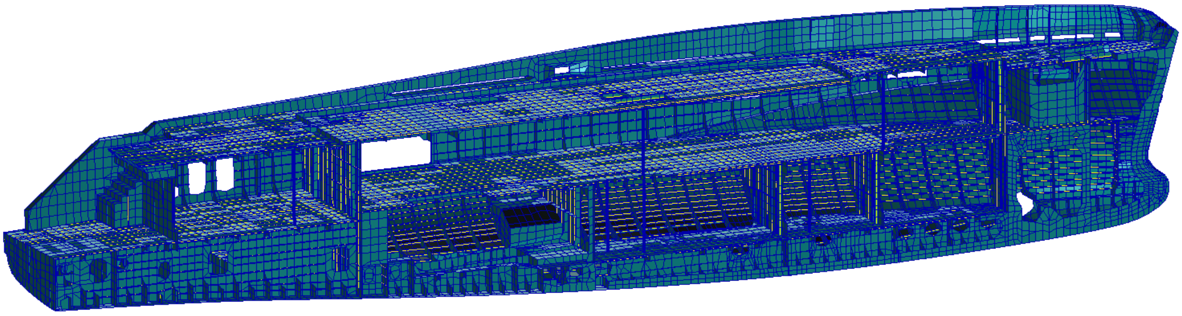


Figure 25 - Hull with primary and secondary structure

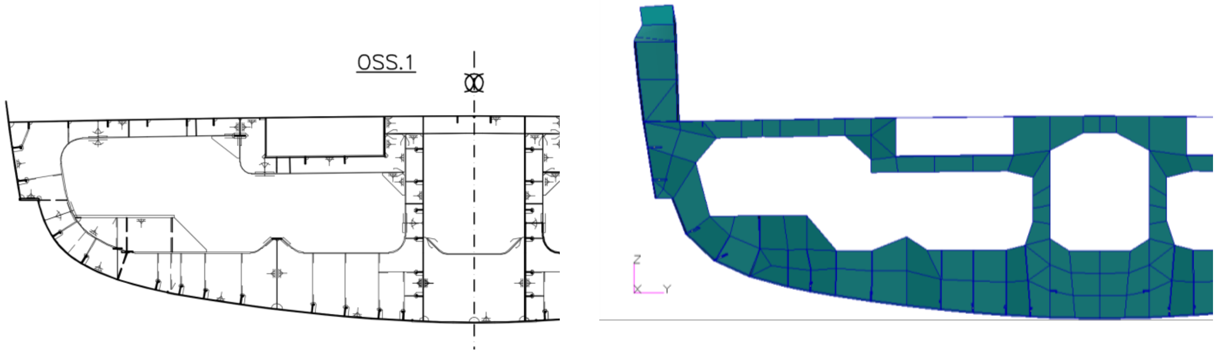


Figure 26 - Constructive drawing vs FEM model

5. MEASUREMENTS PROTOCOL

Usually the measurements are performed when the vessel is already launched and in sailing conditions, in this way the real vibrational response of the yacht is registered and analyzed, all the structural and non-structural masses are involved, and the damping conditions are taken from the real situation. These are parameters that usually are estimated when the FEM model is performed.

In this case the possibility to perform experimental measurements on the ship under construction allows making an accurate comparison of the precision between these and the results obtained by the FEM model analysis, where no damping is considered, no non-structural masses. The real situation is well represented.

Figure 27 - Yacht main deck at the moment of experimental measurements (looking at bow)

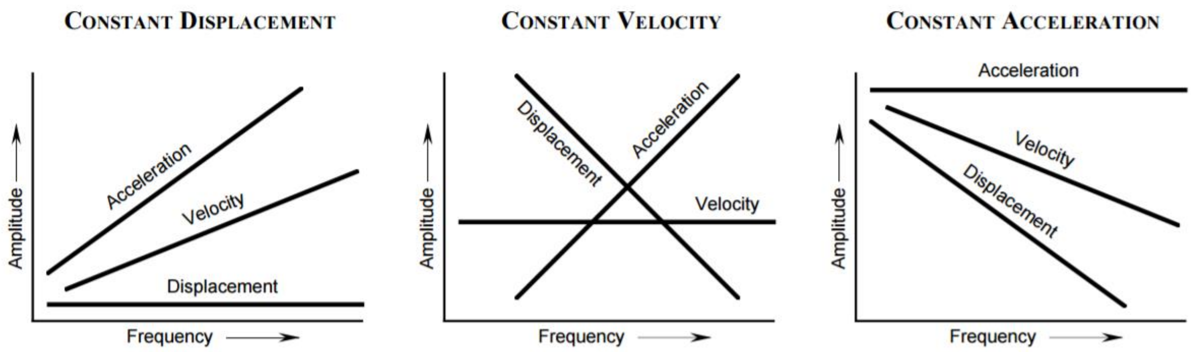
Figure 28 - Yacht lower deck cat the moment of experimental measurements (looking at bow)

Experimental measurements of main deck and lower deck were performed to compare the FEM results with the natural frequency obtained by measurements. This will allow certifying the certainty of the results obtained by the Finite Element Analysis. Measurements were performed with a specialized engineer from RINa.

5.1 Sensors

In vibration monitoring the selection of sensors to use is a critical factor on determining the success of the measurements. Not only the quality of the sensor is critical, but also the type of data that this could catch.

As known, the three parameters representing motion in vibrations are: displacement, velocity and acceleration, these parameter can be measured by different types of motion sensors and are mathematically connected (displacement first derivative is velocity, and velocity first derivative is acceleration). The choice is based on the frequencies of interest involved in each case.



DISPLACEMENT SENSORS: Eddy current probes are non-contact sensors primarily used to measure shaft vibration, shaft/rotor position and clearance. Also referred to as displacement probes, eddy current probes are typically applied on machines utilizing sleeve/journal bearings. They have excellent frequency response with no lower frequency limit and can also be used to provide a trigger input for phase-related measurements.

VELOCITY SENSORS: Velocity sensors are used for low to medium frequency measurements. They are useful for vibration monitoring and balancing operations on rotating machinery. As compared to accelerometers, velocity sensors have lower sensitivity to high frequency vibrations.

ACCELEROMETERS: Piezoelectric accelerometers having a constant signal over a wide frequency range, up to 20 kHz's, for a given mechanical acceleration level, are very useful for all types of vibration measurements. Acceleration integrated to velocity can be used for low frequency measurements. Acceleration signals in the high frequency range added with various signal processing techniques are very useful for bearing and gear measurements.

They are useful for measuring low to very high frequencies and are available in a wide variety of general purpose and application specific designs. The piezoelectric sensor is versatile, reliable

Piezoelectric Sensors Accelerometers operate on the piezoelectric principal: a crystal generates a low voltage or charge when stressed as for example during compression. Motion in the axial direction stresses the crystal due to the inertial force of the mass and produces a signal proportional to acceleration of the mass. This small acceleration signal can be amplified for acceleration measurements or converted (electronically integrated) within the sensor into a velocity or displacement signal. This is commonly referred as the ICP (Integrated Circuit Piezoelectric) type sensor. The piezoelectric velocity sensor is more rugged than a coil and magnet sensor, has a wider frequency range, and can perform accurate phase measurements.

In this study, PCB accelerometers model PCB3530B unidirectional was used

Accelerometers are essential. For structures typical in shipbuilding only the frequency range up to about 100 Hz is of interest from a vibrational point of view. The maximum acceleration values are generally less than 1 G, and in most cases they are distinctly lower.

The signal analyzer includes a calibration setting that allows the voltage signal to be converted back into a measurement of acceleration.

Sensitivity of sensors are calibrated by manufacturers, accurate measurements depends on using the correct sensitivity value for each application. High sensitivity is not appropriate for high acceleration level; too high voltage could saturate the input on the signal analyzer. On the other hand a small sensitivity may produce a signal too weak.

All sensors and measurement hardware is subject to electronic noise. This is due to the sensor cables picking electronic noise from stray signals in the air, noise in the power supply, etc.

Mounting: For low frequency response the mounting technique doesn't affect the results. In this case the sensor was simply placed in the position to measure. In other cases, adhesive mountings, stud mounts or magnetic mounts might be considered.

Figure 29- PCB356A02 triaxial at right and PCB3530B unidirectional at left

Multidirectional and unidirectional sensors are available. In this case the multidirectional device was used.

Calibration

The calibration is done through an automatic system which generates a defined frequency and the sensors are calibrated.

Figure 30 - Calibration device

5.2 Signal Analyzer

The analyzer measures electronic signals with an analog front end that may include special signal conditioning such as sensor power supply, TEDS (transducer electronic data sheets that read the calibration and other information from a chip embedded in the sensor), adjustable voltage gains settings and analog filters. Next, the system converts the analog signal to a digital format via an analog to digital converter (ADC). After the signal is digitized, the system processes it with it a digital signal processor (DSP), which is a mini-computer optimized to do rapid mathematical calculations. The DSP performs all required calculations, including additional filtering, computation of time and frequency measurements, and management of multiple channel signal measurements.

The used analyzer is a four-channel SOUNDBOOK™ system and the acquisitions have been run by the software Sinus SAMURAI™ [19]

Soundbook™, is a universal portable measuring system and works on the basis of Apollo™ platform. The 24-bit A/D converters in combination with the Apollo filter processors provide many channels with high precision and bandwidth.

The basic principle of the software is that virtual measurement instruments provide data for activated measurement channels. Simultaneously with the measurement and analysis, the data provided is stored in synchronous data streams.

5.3 Impact method

This type of measurement is performed to obtain the natural modes of a particular structural component, in this case the main and lower deck. It is usual to perform these types of measurements to check the panels, stiffeners of decks and superstructures before the ship is completed. In case of probable resonances between the main excitation orders (propeller and engine), changes at this stage in the structure are extensively more convenient than further modifications.

It consists in exciting the structure with an impact hammer or vibrator, measuring the frequency response functions between the excitation and many points on the structure, and then using software to visualize the mode shapes. Accelerometers described before measure the vibration levels at several points on the structure and a signal analyzer computes the FRFs (Frequency Response Functions).

An impact hammer is used in this case; the device is a specialized measurement tool that produces short duration vibration levels by striking the structure at some point. The hammer incorporates a sensor (called a load cell) that produces a signal proportional to the force of impact. This enables precise measurement of the excitation force. An impact hammer is often used for modal analysis of structures where use of a mechanical vibrator is not convenient; examples are in the field of very large structures. Different impact tip materials allow tailoring of the frequency content of the impact force. For low frequency measurements, a soft rubber tip concentrates the excitation energy in a narrow frequency range.

In the first trial the excitation device used in tests is an instrumented impulse hammer (PCB Impact Hammer model 086D0) with a medium tip and a weight of 0.32 kg [18]. It was find that the impact to excite the system was very low, so another test was performed using a mass of 110 kg.

5.4 Measurements position

Vibration measurements procedures must be adapted with different configurations and should suite the operational constraints. In this case the measurements were performed in late evening when the workers finish their work day to avoid undesired signals in the measurements.

At the time of measurements, some remaining steel plates were still placed in the main deck of the yacht, which modifies the mass of the measured structure, there is no exact information about this masses.

The position where the experimental measurements were performed is where the cinema will be located in this yacht. This is as space where it is needed to ensure comfort and low vibration and noise level. But is also a compromised zone due to the fact that is located close to main excitations coming from main engine (pressure and thrust fluctuations), and other sources of vibration located at engines room, which could generate some disturbances.

Figure 31 - Sensors position in longitudinal measurement (from left to right S1,S2,S3)

In the picture showed above the position for the three installed sensors could be observed. To the left side of the picture the bow direction of the yacht is located.

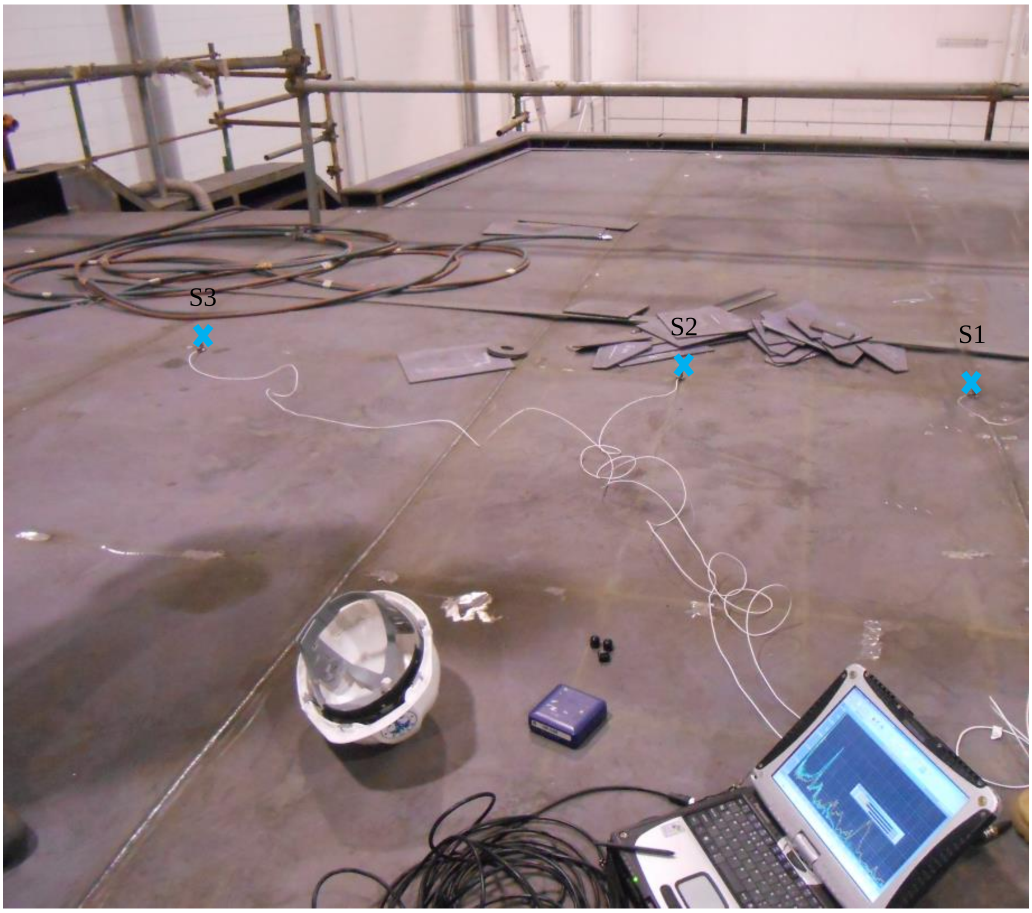


Figure 32 - Sensor position during transversal measurements (looking aft)

6. FINITE ELEMENT ANALYSIS

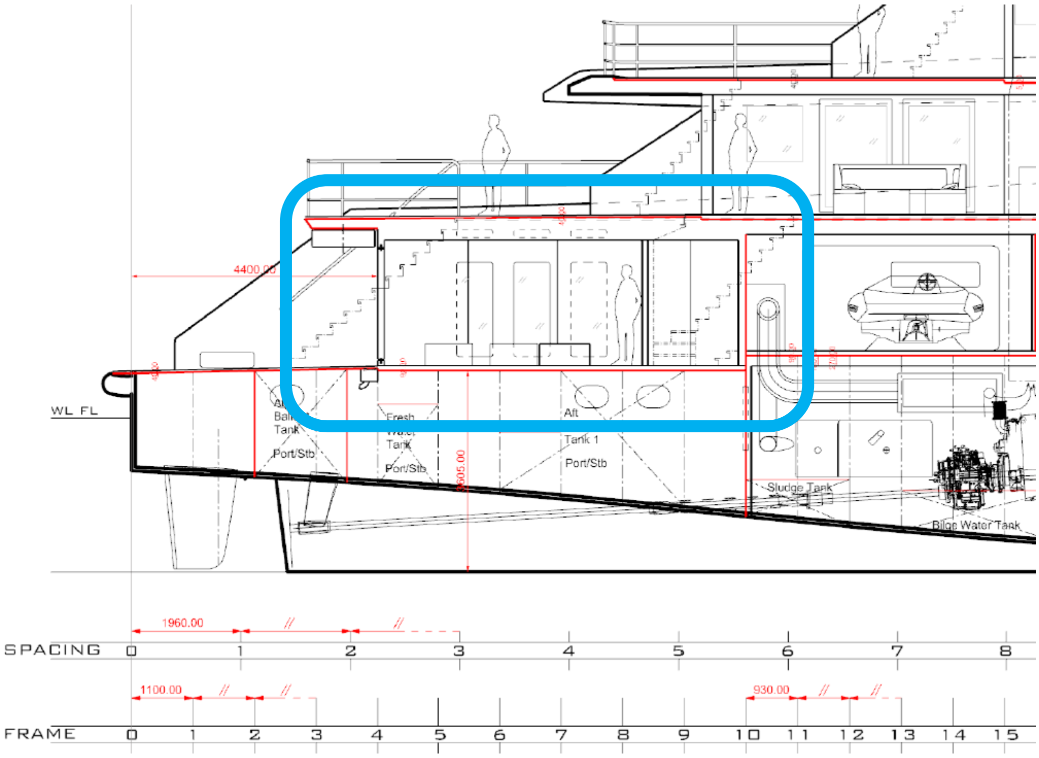


Figure 33 – Cinema lounge location

The focus of this part of the study is to analyze the cinema on the yacht, where later the measurements will be driven allowing a comparison of results. The selected frame to analyze is 7 and longitudinally the stiffener @2450 from CL. These sections will be shown for each case and the measurements after will be driven here.

6.1 Computation of natural frequencies and mode shapes

Once understood the method of computation and chosen the interesting frequency range for the model, it can be started the modal analysis. Before estimating the mode shapes of the model it is important to investigate about the typical values of the frequencies.

The first checking point to know if the results are correct is to detect the first six body nodes (usually the lowest ones); if there are more than six low frequencies, probably the model is incorrect or not all the nodes are perfectly connected. The first six mode shapes describe the six degrees of freedom of the ship in the space, three displacements and three rotational respects to the three principal axis.

The second check should be done on the numerical values of the natural frequencies; for these kinds of craft completely modeled the natural frequencies are higher than 4 Hz, if they are smaller there is an error on the model to solve. Usually elements are not connected, or some illegal geometry was modeled for the elements. These errors are found in the .dbf file or sometimes also in .x06. The understanding of these files is so important as the mesh modeling.

On the generated model, there are about 50.000 elements, with 36000 nodes. The detail is showed below

M O D E L   S U M M A R Y

ENTRY NAME	NUMBER OF ENTRIES
CBAR	4995
CBEAM	10201
CQUAD4	37469
CTRIA3	2494
EIGRL	1
GRID	36111
MAT1	1
PARAM	2
PBAR	1
PBARL	15
PBEAML	41
PSHELL	29

Figure 34 - Model summary

In total 450 modes were run between 0 and 50 Hz, in the following figure the first 15 natural frequencies calculated on the model could be seen:

```
SubcaseId= 1 : SC1:DEFAULT, A3:Mode 1 : Freq. = 1.0045E-5;-MSC.NASTRAN JOB CREATE
SubcaseId= 2 : SC1:DEFAULT, A3:Mode 2 : Freq. = 8.4206E-6;-MSC.NASTRAN JOB CREATE
SubcaseId= 3 : SC1:DEFAULT, A3:Mode 3 : Freq. = 8.1061E-6;-MSC.NASTRAN JOB CREATE
SubcaseId= 4 : SC1:DEFAULT, A3:Mode 4 : Freq. = 4.3927E-6;-MSC.NASTRAN JOB CREATE
SubcaseId= 5 : SC1:DEFAULT, A3:Mode 5 : Freq. = 3.2941E-6;-MSC.NASTRAN JOB CREATE
SubcaseId= 6 : SC1:DEFAULT, A3:Mode 6 : Freq. = 7.174E-6;-MSC.NASTRAN JOB CREATED
SubcaseId= 7 : SC1:DEFAULT, A3:Mode 7 : Freq. = 7.4283;-MSC.NASTRAN JOB CREATED C
SubcaseId= 8 : SC1:DEFAULT, A3:Mode 8 : Freq. = 7.4727;-MSC.NASTRAN JOB CREATED C
SubcaseId= 9 : SC1:DEFAULT, A3:Mode 9 : Freq. = 8.6233;-MSC.NASTRAN JOB CREATED C
SubcaseId= 10 : SC1:DEFAULT, A3:Mode 10 : Freq. = 9.5713;-MSC.NASTRAN JOB CREATED
SubcaseId= 11 : SC1:DEFAULT, A3:Mode 11 : Freq. = 10.285;-MSC.NASTRAN JOB CREATED
SubcaseId= 12 : SC1:DEFAULT, A3:Mode 12 : Freq. = 10.581;-MSC.NASTRAN JOB CREATED
SubcaseId= 13 : SC1:DEFAULT, A3:Mode 13 : Freq. = 12.159;-MSC.NASTRAN JOB CREATED
SubcaseId= 14 : SC1:DEFAULT, A3:Mode 14 : Freq. = 14.447;-MSC.NASTRAN JOB CREATED
SubcaseId= 15 : SC1:DEFAULT, A3:Mode 15 : Freq. = 14.556;-MSC.NASTRAN JOB CREATED
```

Figure 35 - Natural modes

In Patran the user defines the range of frequencies to calculate. This depends on the type of study that the engineer is carrying out. For this case, a value between 0 and 50 Hz was set, because the most noticeable displacements in the zone of the study should be on this range, some computational limitations and computational time were also influent factors on the decision.

In this case the analysis is performed with the complete FEM model created, not only the main deck is analyzed. The system will solve all the possible solutions for the Eigen equation along the entire model, this could be useful or not. It is a work of the engineer to search in the range of interest and discard the frequencies that are not close from the expectations.

In the following figures the most noticeable displacements on the zone of study are showed, latter on a comparative table will be presented.

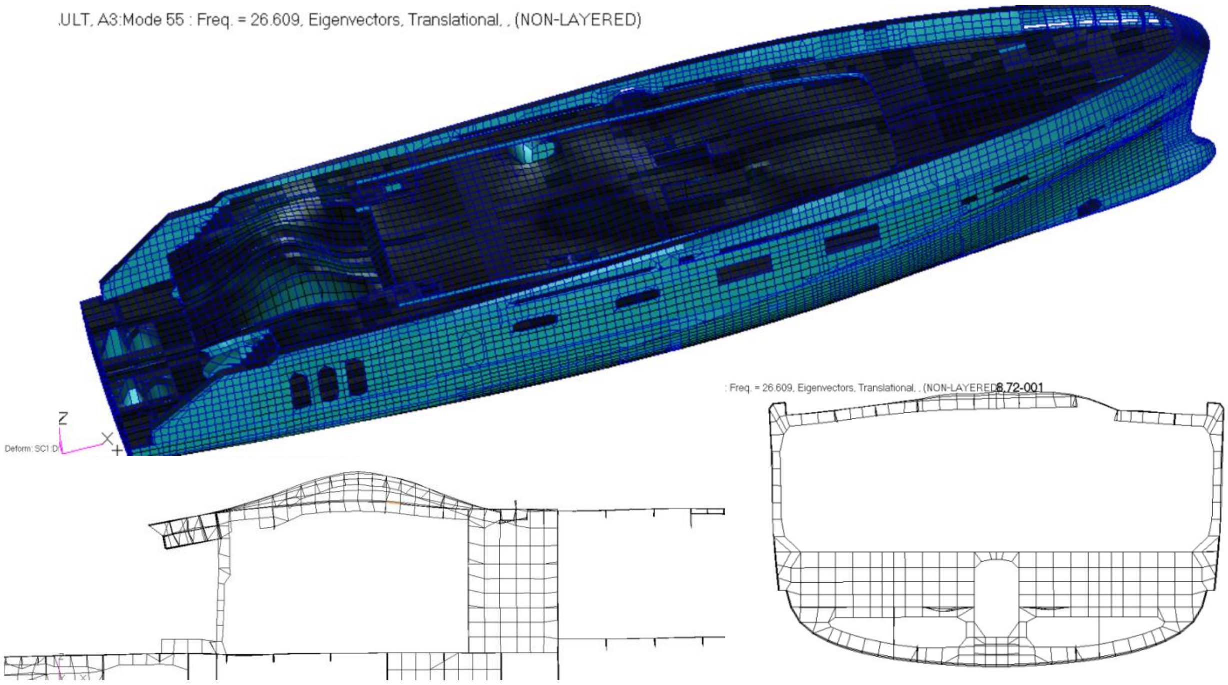


Figure 36 - CINEMA/1st natural mode @26.609Hz

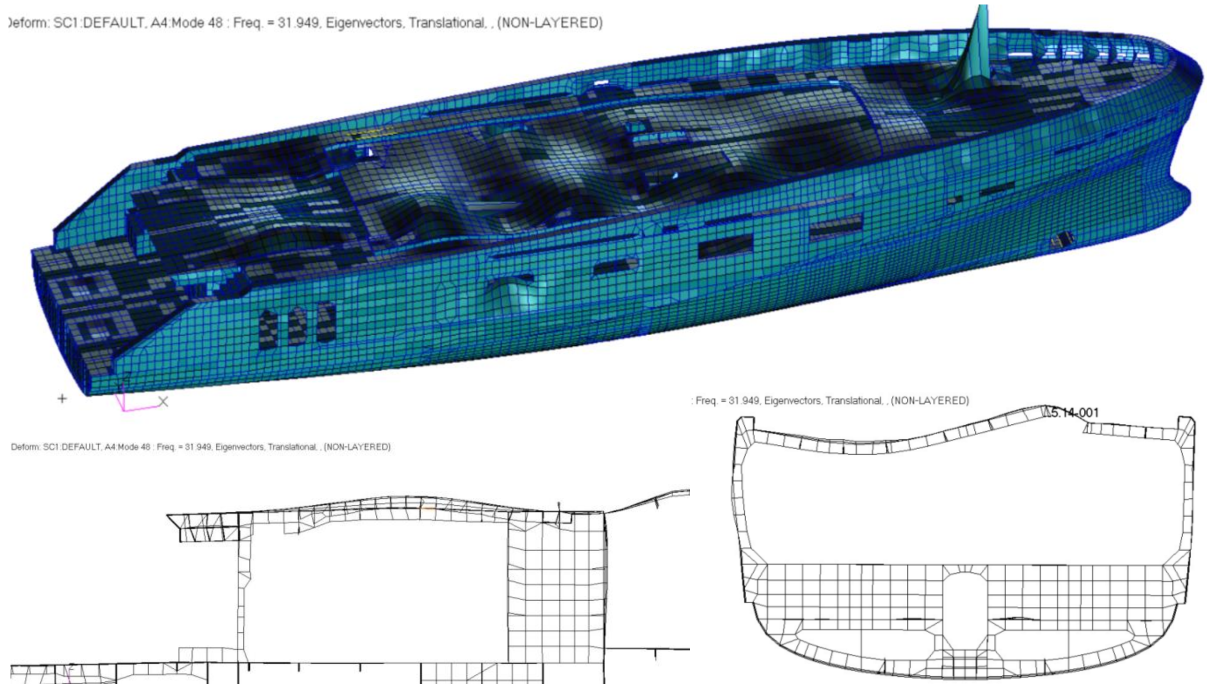


Figure 37- CINEMA/2nd Natural mode@31.949 Hz

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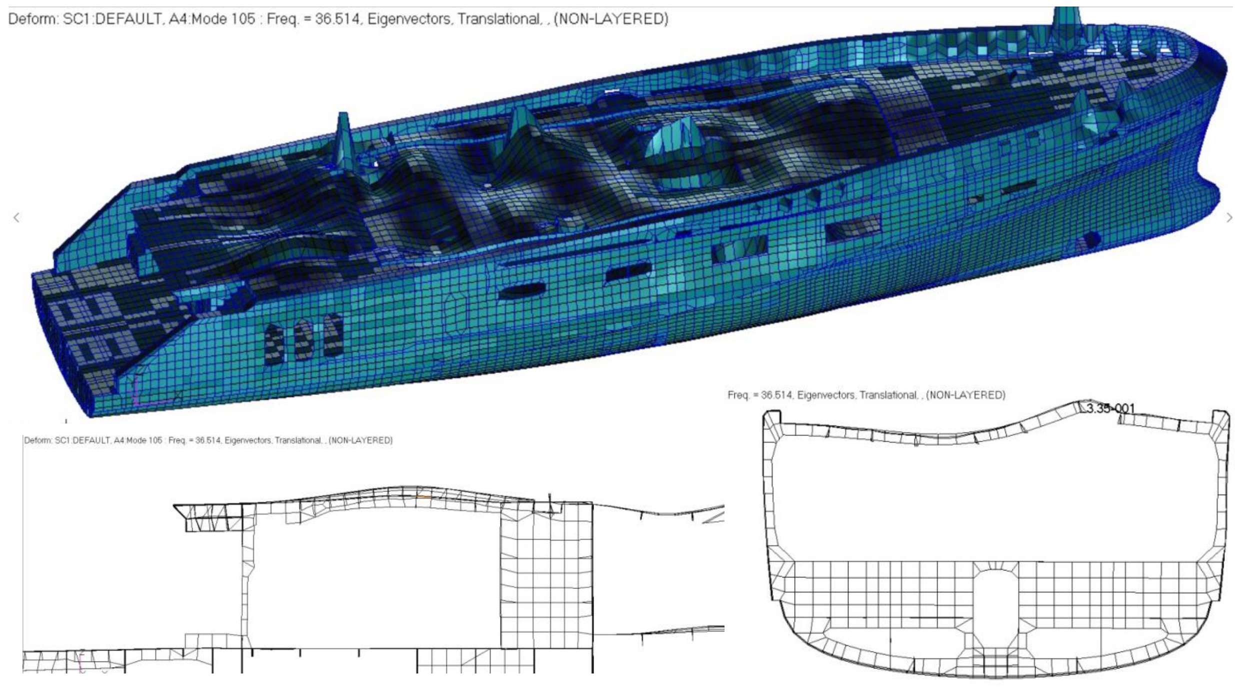

Figure 38 - CINEMA/3rd Natural mode @36.514Hz

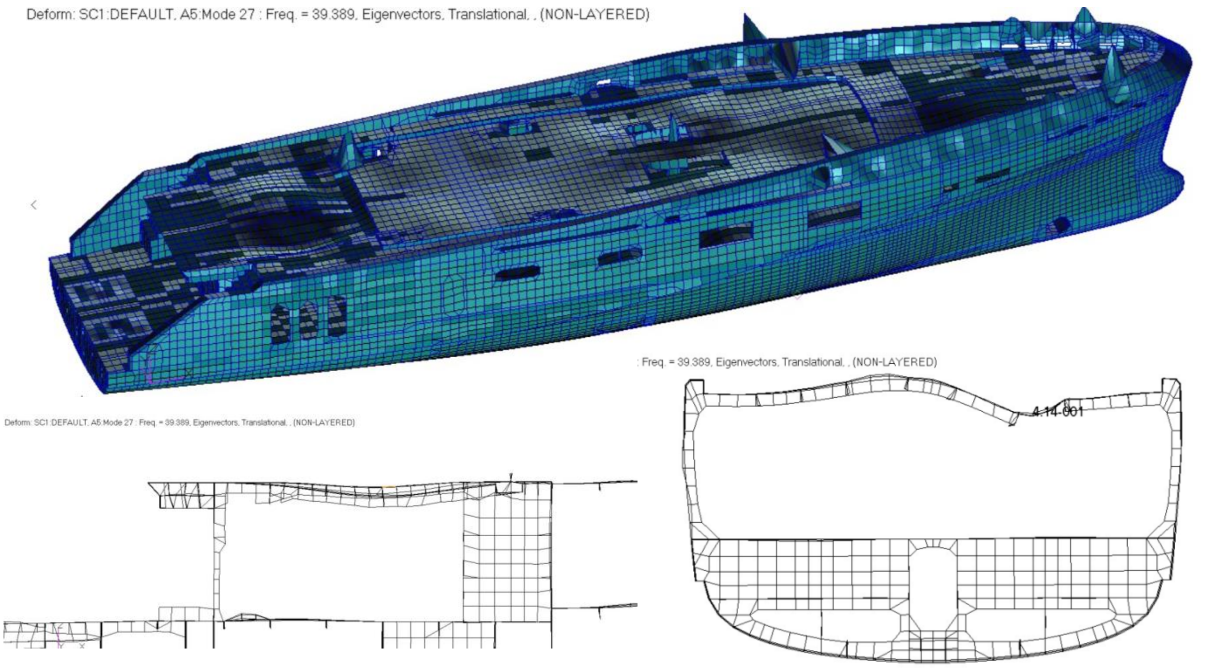

Figure 39 - CINEMA/4rd Natural mode @39.352

In the natural mode found at @26.609 Hz the bigger displacements appears. It could be expected to find this in the measurements as the main response for impact test. The mode shape represents a two nodes shape for both directions, transversal and longitudinal.

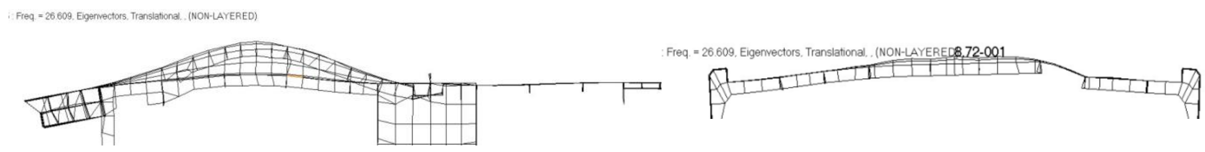

Figure 40 - Two nodes vibration mode – Longitudinal 2450 (left) Frame 7 (right)

At the second natural mode @31.949 Hz, the displacements are not as big as in the first natural mode, the modal shape in this case a 3 node vibration could be observed in the transversal structure.

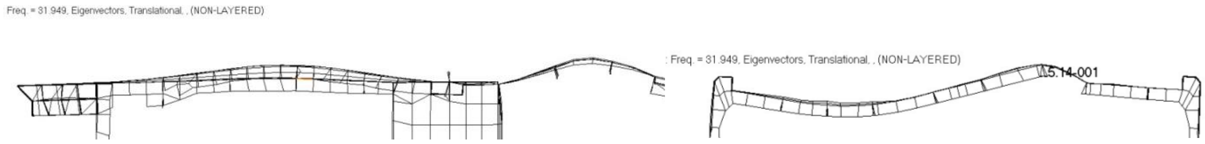

Figure 41 - Three nodes vibration mode - Longitudinal 2450 (left) Frame 7 (right)

Torsional influence is not noticeable in any of these two cases.

From theory it is known that when a linear elastic structure is vibrating in free or forced motion, its deflected shape at a given time is a linear combination of all of its natural modes as it can be seen in the following and this is noticeable in the following 2 natural modes find, where the combination of diverse natural modes existing in the structure such as hull beam, main deck, and others; are revealed.

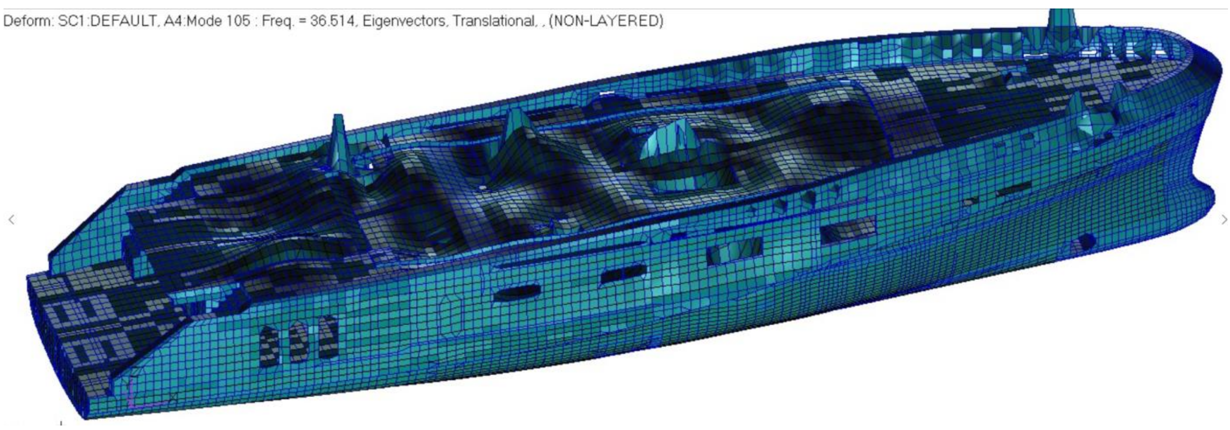

Figure 42 - CINEMA/Natural mode 3

6.2 Boundary conditions: supports

To simulate the precise situation, boundary conditions needs to be applied where the supports of the yacht are located, supporting the yacht. These supports were included after to drive the first FEM analysis.

If some lack of information is happening between the shipyard, and the engineers, or some mistake was done while measurement of supports was being carried out, this study looks to justify also how much the supports influence the results on main deck measurements.

It is known that this will change the natural modes of the entire hull beam, but the quantity of displacement that this is imposing in one point on main deck is not so clear.

Therefore, another analysis was performed, with four supports simulating the real situation. These LBC are considered as simple supports, only in the z direction displacements are not allowed:

- *Translation x* - *Rotation x*
- *Translation y* - *Rotation y*
- ~~*Translation z*~~ - *Rotation z*

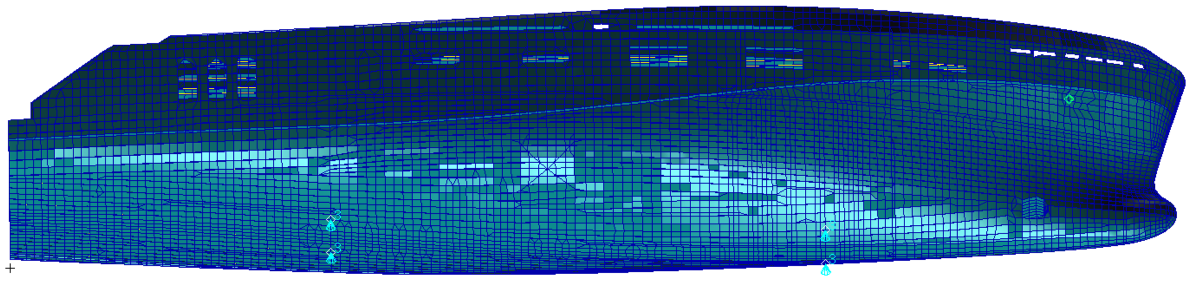

Figure 43 - Supports on the yacht

Two aft supports are located at (11400; + -1450), in mm. remember that the origin is at Baseline, Centerline and aft.

Two of bow supports are located at (28000; + -1400), in mm.

A comparative table is presented below comparing the obtained results with and without supports for the two first natural modes of the cinema.

Natural frequencies FEM no support [Hz]	Natural frequencies FEM with support [Hz]	Error [%]
26,609	26,611	-0,008
31,949	31,826	0,386

Table 6 - FEA Natural frequencies with and without supports

Graphical representation is showed in Figure 38 and Figure 39. It is possible to compare the mode shapes with the ones obtained in the previous case; the observed mod shapes are the same, so the comparison of the obtained natural frequencies is valid.

The observed difference in the results considering the supports is very small. Even though the supports will generate a change on the natural modes of the hull beam, this supports are not representing noticeable changes in the results for deformations on main deck. The hull beam natural frequency with and without supports is shown latter in this section for a better understanding of the problem (see Figure 45 - 2nd cinema Natural mode with supports Figure 46 - Hull beam 1st natural mode @16.659 Hz (without supports)

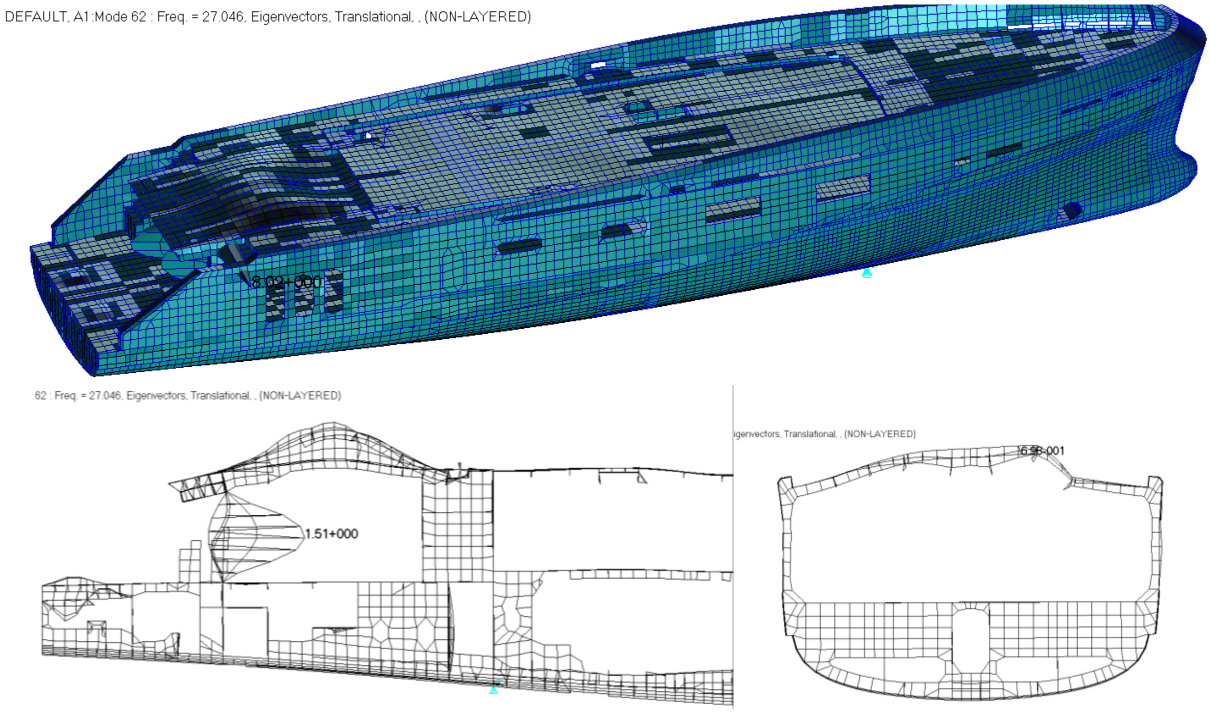

Figure 44 - 1st cinema natural mode with supports

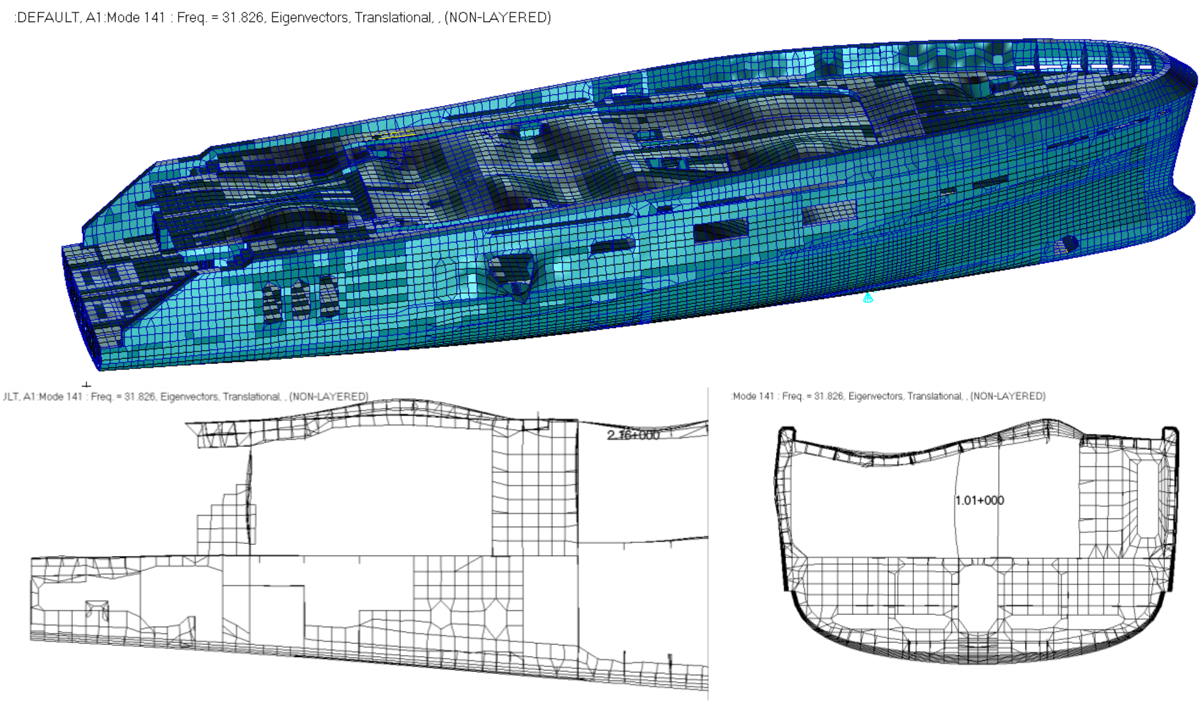

Figure 45 - 2nd cinema Natural mode with supports

It is possible to affirm that the influence of the supports on the natural frequencies on the main deck (cinema zone) is very small and the supports could be even not introduced in FEM model for this analysis.

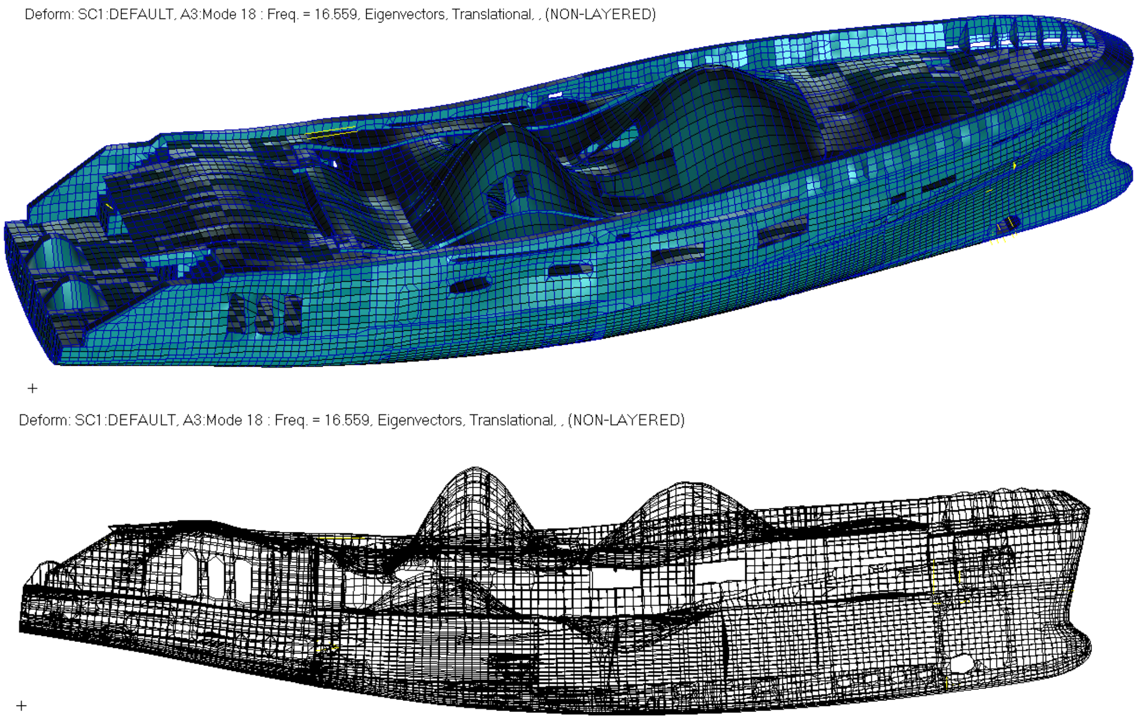

Figure 46 - Hull beam 1st natural mode @16.659 Hz (without supports)

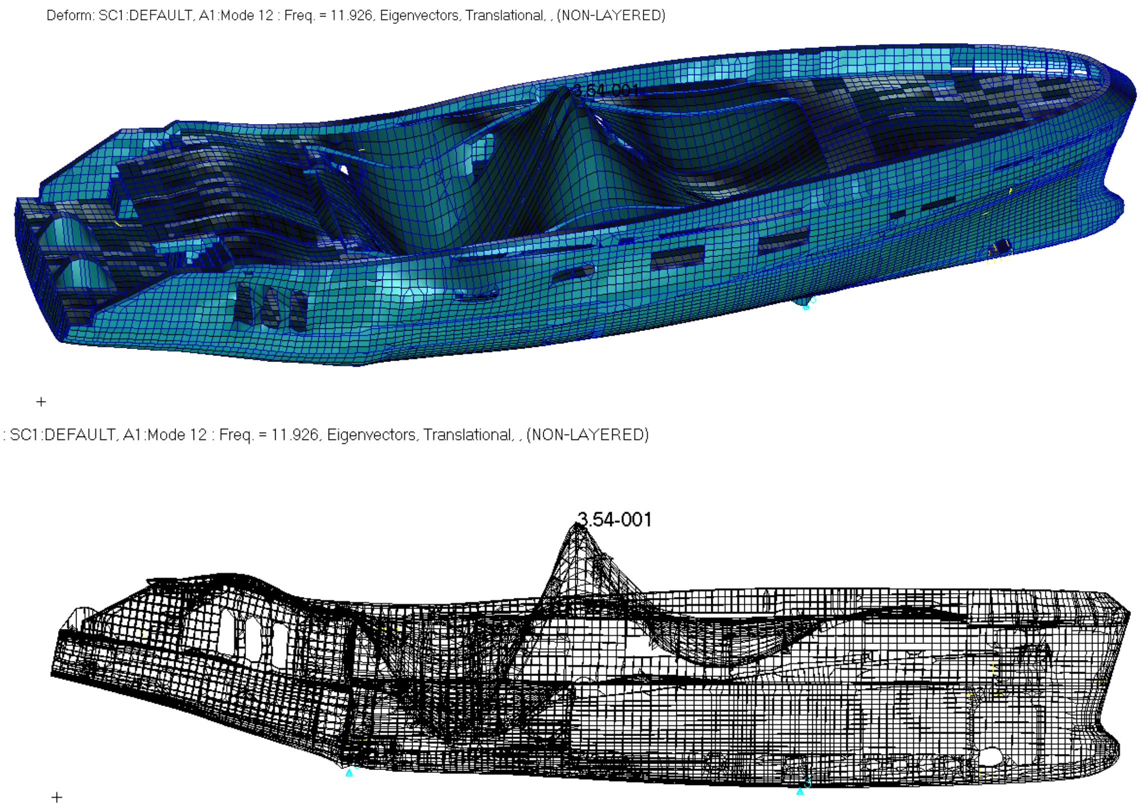

Figure 47 - Hull beam 1st natural mode @11.926 Hz (with supports)

7. MEASUREMENTS

Several measurements were performed to get the most reliable results for each case. The conditions of the yacht at this time were very similar to FEM model. Hull shell, lower deck and main deck with their respective primary and secondary structures are finished, all steel construction.

The yacht is supported in four points which also measures the weight at the time the yacht is being build. At the time of measurements, the corresponding weight is 154tn.

Measurement cases are two:

- M1: Main deck – Transversally Frame 7
- M2: Main deck – Longitudinally between frames 6 and 9

Three uniaxial sensors were used for each measurement oriented in z axis, few trials were done for each measurement position, in this way the results for the same analysis could be confronted and it is much precise for determining the natural frequencies and dismisses the noise that could be present during measurements.

The location of the sensors at the time of measurements is showed in Figure 48 - 3D SENSORS POSITION (*). All the devices are oriented in z direction.

It is possible to observe the natural frequencies at which the structure is responding through displacement graphics, velocity or acceleration. In this case velocity graphs were used because after integration of acceleration (keep in mind that measures are taken from accelerometers) some noises on the signal disappear. On the following section, all the useful measurements are presented and will be discussed later. At least two registrations for each measurement case are presented.

The results are obtained from SAMURAI® software as shown in Figure 50 - Results on SAMURAI.

In the window 1 it is registered the signal of the sensor during a time window; the data is collected until 3 seconds after the recording was initiated, defining in this way the time domain. The duration of each measurement by the software is set in 5 seconds.

Window 2 and 3 shows the Fourier Transform Function (FFT) of velocity and acceleration corresponding to each Frequency. Velocity is obtained through a software integration of the acceleration data obtained by the sensor; a FFT (Kaiser-Bessel) is applied on the recorded data to decompose the original signal in function of time into the frequencies that make it up. The Fourier transform of a function of time itself is a complex-valued function of frequency, whose absolute value represents the amount of that frequency present in the original function.

Window 13, 14 and 15 gives the 1/3 octave information for each sensor which are of interest when noise are studied.

The first case of each measurement case presented was performed with an instrumented impulse hammer (PCB Impact Hammer model 086D0), this reveals more clear results but the amplitudes of the signals are too weak. A second measurement was considered in each case (M1, and M2) with a weight of 110 kg (not calibrated) this shows results with higher amplitudes but more noise because the impact is not as precise as in the case of the use hammer.

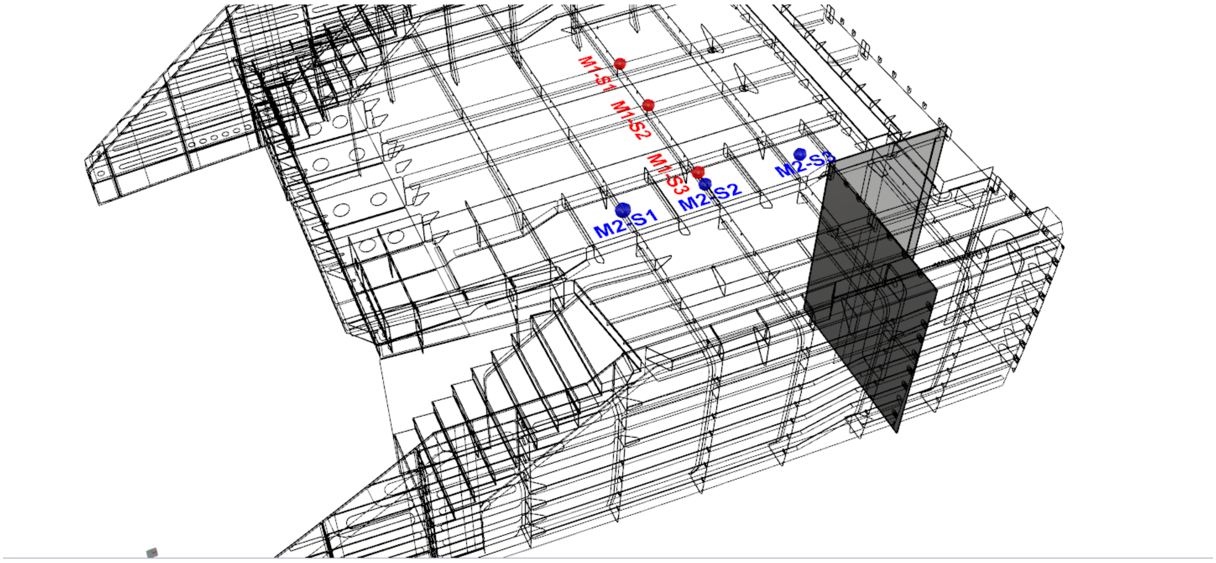

Figure 48 - 3D SENSORS POSITION (*)

(*) M1 is referred to Measurement case 1, transversal measurements.

M2 is referred to Measurement case 2, longitudinal measurements.

S1, S2, S3 refers to sensor number and channel of transmission with the analyzer. Then, for example: M1-S1: Measurement case 1, Sensor 1

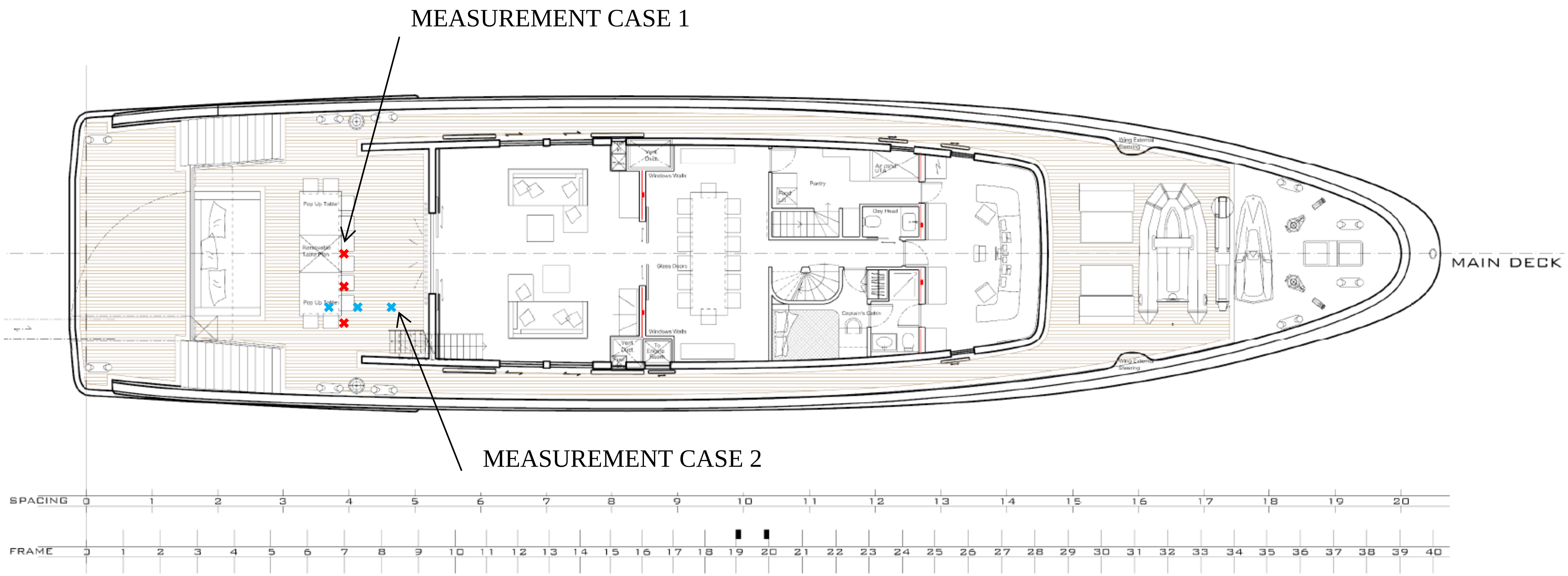

Figure 49 - Position of the sensors for measurements

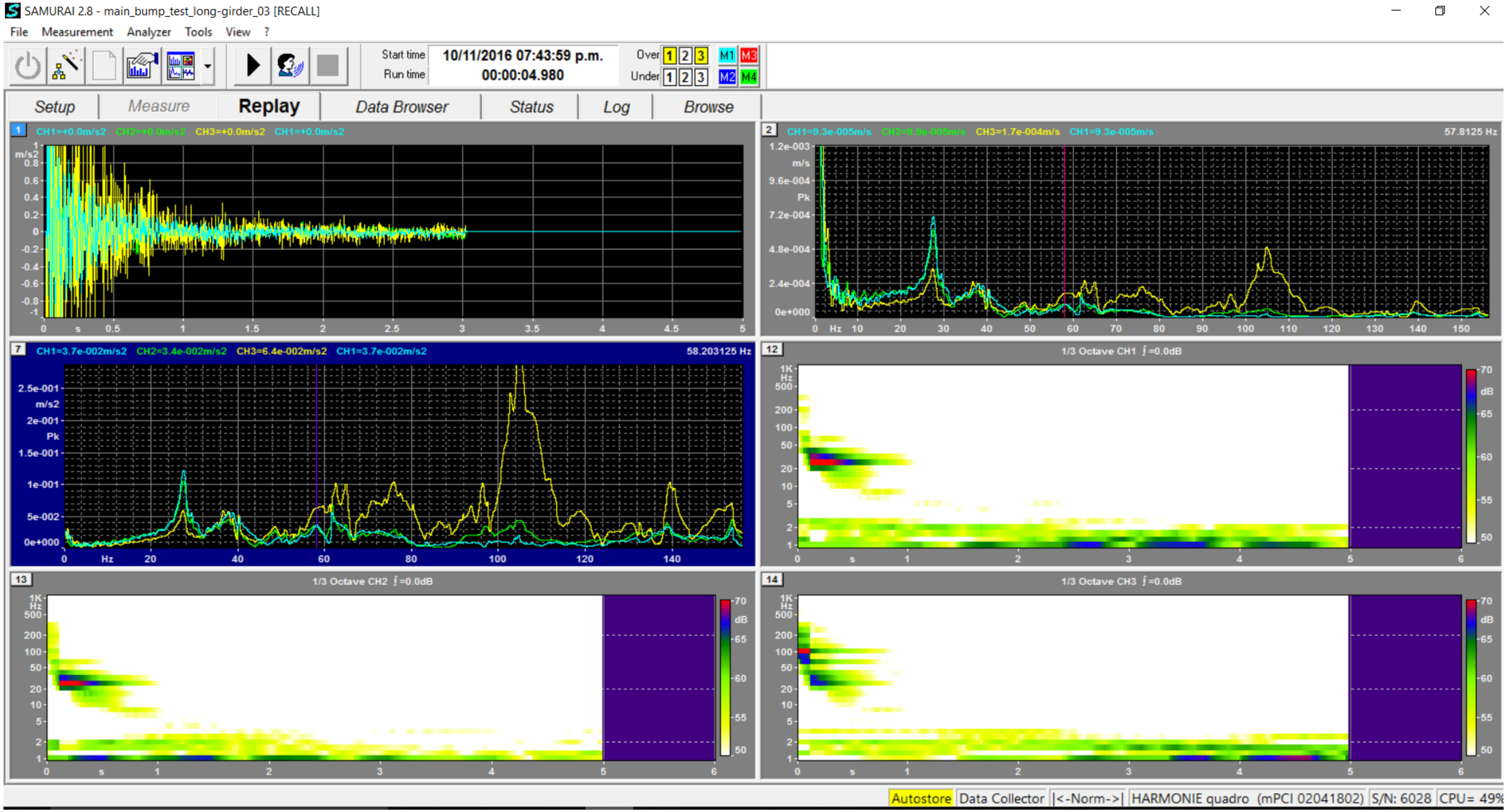

Figure 50 - Results on SAMURAI

7.1 Case 1 measurements – M1

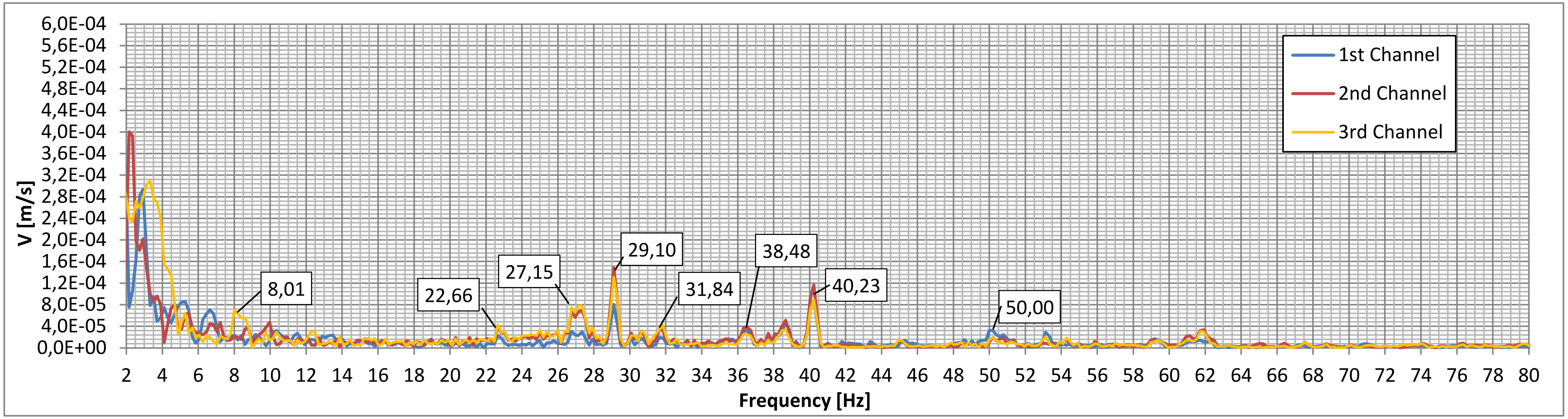

Figure 51 - M1 1st measurement

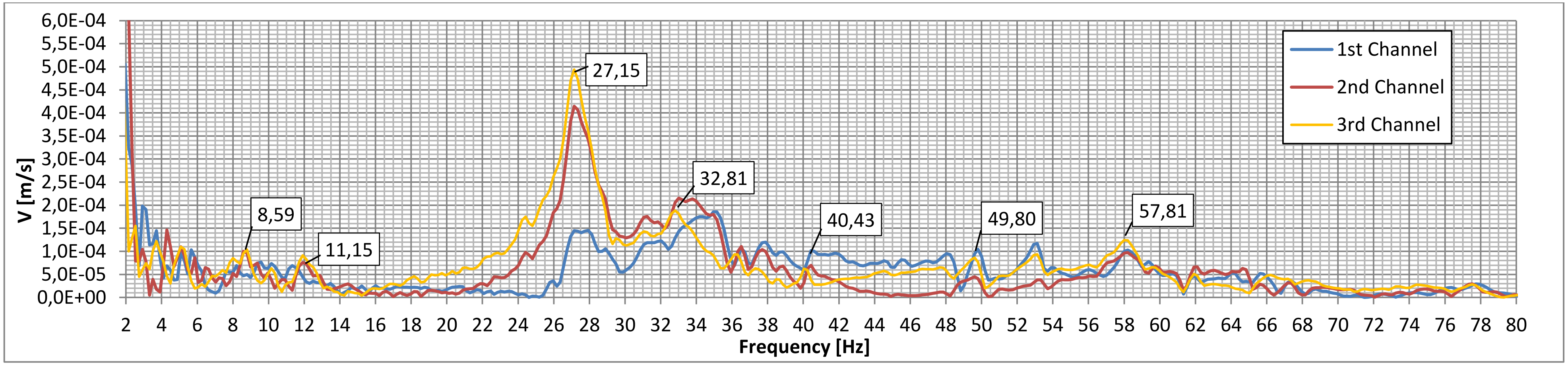

Figure 52 - M1 2nd measurement.

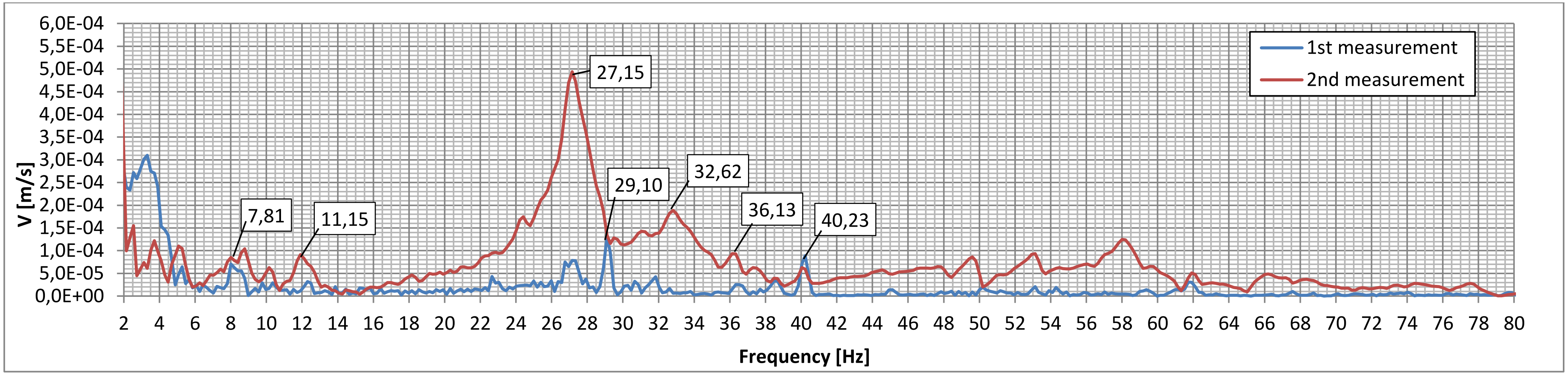

Figure 53- M1 - 1st vs 2nd measurement CH3

From the obtained results in SAMURAI, excel graphics were created after exporting the data from the software for measuring. In this way it is possible to make some comparative graphics which are useful for the understanding and conclusions on this analysis.

Figure 51 - M1 1st measurement and Figure 52 - M1 2nd measurement. the first and second measure of measure case 1 are presented. In the first case it is possible to observe the low amplitudes compared with 2nd measurement, this is due to the fact that for first measurement an instrumented impulse hammer (PCB Impact Hammer model 086D0) was used, this induces much less noise in the signal than the second case that when the impact test was through a free falling weight o 110kg. it was expected to find more Natural modes of the structure, but so far the same modes were excited. Graphically some of them are not perceptible in the first measurement, so the comparison of the two most significant channels for each case is compared in Figure 23.

Here the frequencies where the system seems to have a response are marked in the graph, some of them are observed in the first measurement and some are identified on the second.

For second case of measurements the same graphical procedure was carried, and in the Figure 53- M1 - 1st vs 2nd measurement CH3, both measurement cases are presented in a same graphic, it is possible to observe the frequencies that are excited in both conditions.

A comparative table is presented after the graphical results, where the frequencies corresponding to natural modes are presented.

7.2 Case 2 measurements - M2

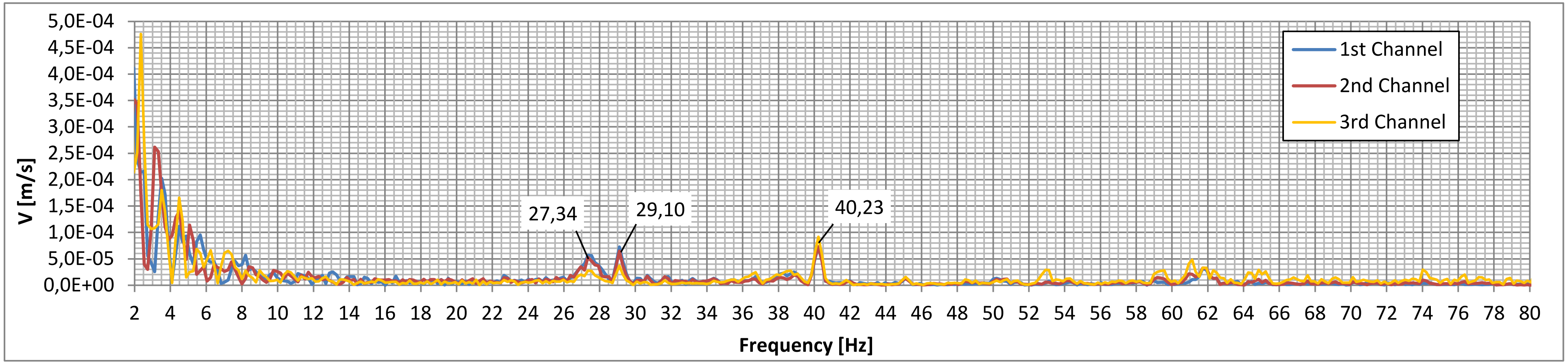

Figure 54 - M2 1st measurement

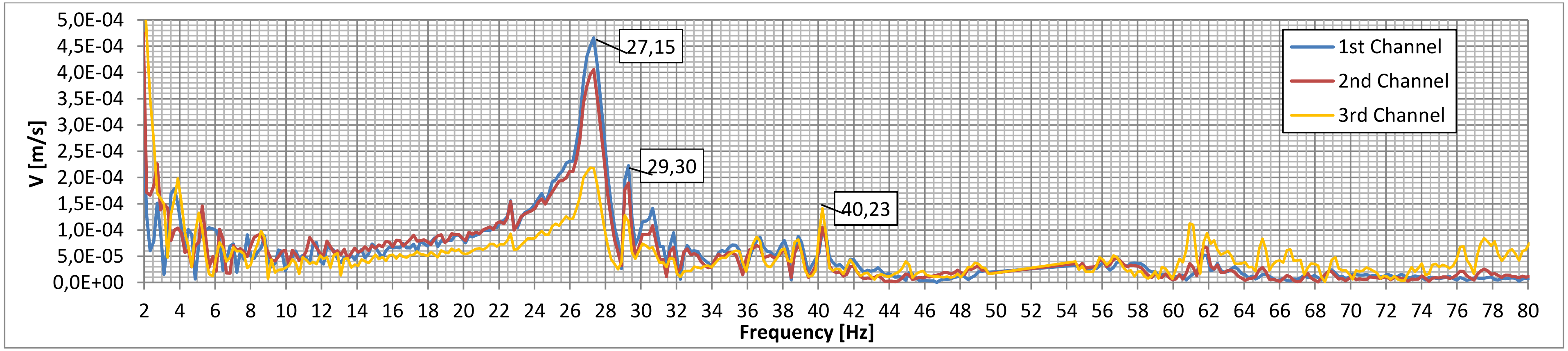

Figure 55 – M2 2nd measurement.

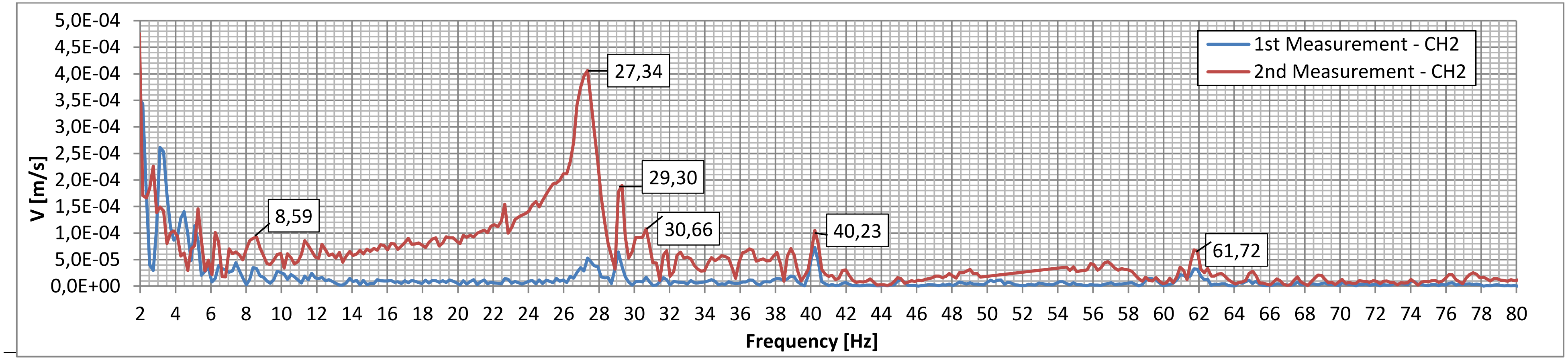

Figure 56 - M2 - 1st vs 2nd measurement CH2

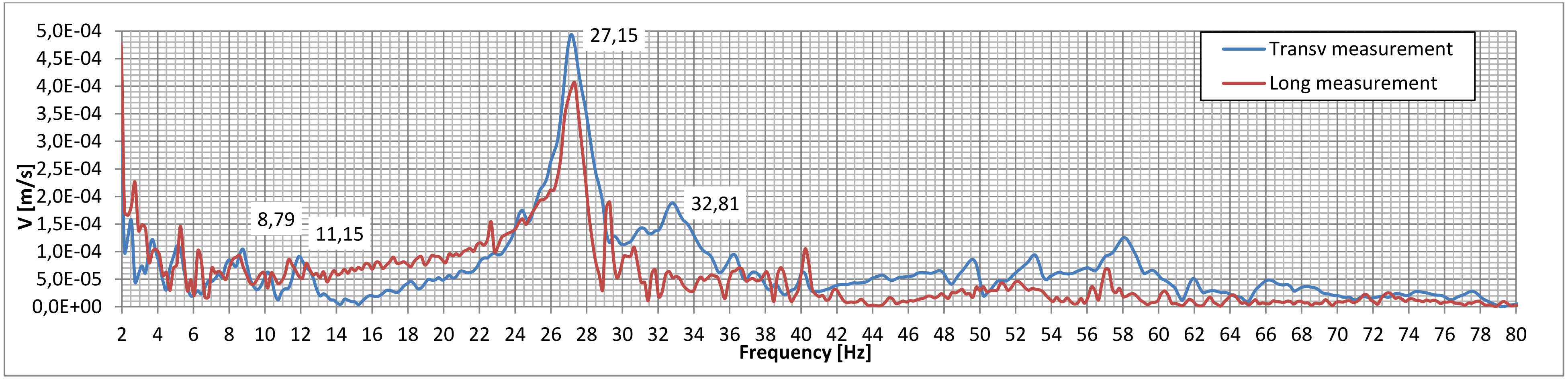

Figure 57 - M2 longitudinal measurement vs transversal measurement

	M1- 1st [Hz]	M1- 2nd [Hz]	M2- 1st [Hz]	M2- 1st [Hz]	M1- mean [Hz]	M2- mean [Hz]
Natural mode 1	8,51	8,59			8,55	0,00
Natural mode 2		11,15			11,15	
Natural mode 3	22,66				22,66	
Natural mode 4	27,15	27,15	27,34	27,15	27,15	27,25
Natural mode 5	29,10		29,10	29,30	29,10	29,20
Natural mode 6	31,84	32,81	31,84	32,81	32,33	32,33
Natural mode 7	38,48		38,48	38,67	38,48	38,58
Natural mode 8	40,23	40,43	40,23	40,43	40,33	40,33
Natural mode 10	50,00	49,80			49,90	

Table 7- Natural modes from experimental measurements

Possible natural modes are expressed in the table presented above. Some of the frequencies were more easily observable in transversal measurements than in longitudinal measurements, moreover, the higher displacements expressed as velocities in the graphs could be find not only in transversal but also longitudinal measurements, as can be seen in Figure 51.

Even if the measurements were performed on a specific zone, and from FEM model it is known the range where the natural frequencies should be present, other natural modes of the ship could be generating displacements.

The three sensors which are taking data from measurements presents small differences in amplitudes as can be observed in Figure 52. This is due to the different positions of the sensors for data collection, if this part is analyzed as a beam case, it is possible to see that sensor 3 is closer from a fixed end and therefore it shows less displacement (half of S1 and S2). In the figure mentioned where the second measurement on longitudinal case is done, it is noticeable the lower response of the 3rd channel (corresponding to sensor 3). Moreover the difference in amplitude should not be the focus of the study, because the most important factor when free analysis is performed is to find the possible resonance frequencies.

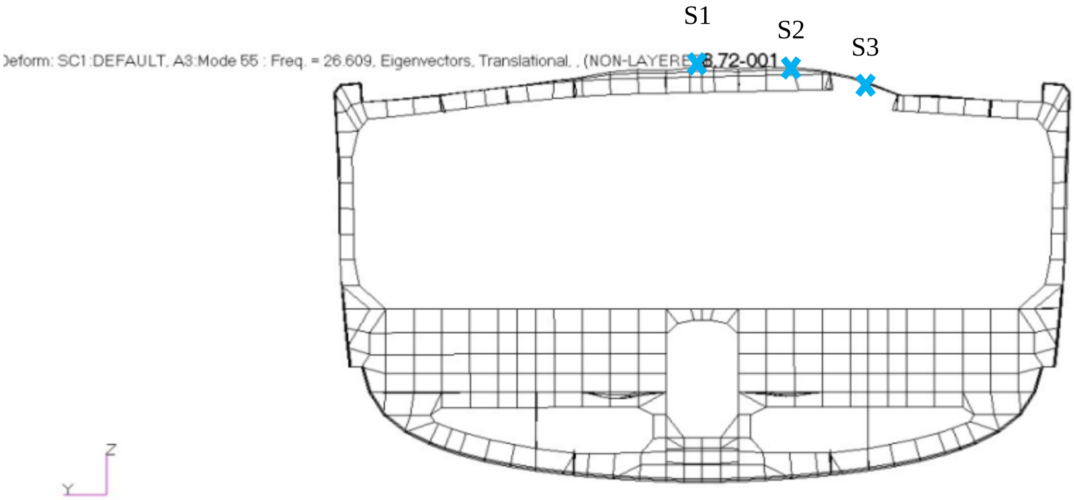

Figure 58 - Sensors/Channels longitudinal position

On the following table a comparison between Natural frequencies found in the measurements and Natural frequencies found by Finite Element Analysis and corresponding errors is presented.

Natural frequencies Measurement [Hz]	Cinema Natural frequencies FEM [Hz]	Error [%]
8,55		
11,15	-	-
22,66	-	-
27,25	26,61	2,38
29,1	-	-
32,33	31,95	1,19
38,52	36,51	5,35
40,33	41,35	-2,5
49,9	-	-

Table 8- FEM Fn vs Measurements Fn

It can be observed that errors are in the range of -2.5% and 5.35%. These differences between the obtained results by FEM modelling and the measurements could be due to:

- Masses onboard at the time of measurement that are not considered in the model
- Induced noise in the impact with the mass of 110Kg (not a hammer)

Regarding the model, let’s remember that when model was done, not all the structure was considered, tertiary structures like brackets and small structures where not considered, this changes the mass distribution of the model, and it is a possible factor introducing error on the Fem model.

The method used to extract the values from the measurements is only graphic, from this source some small variations could come.

8. RESULTS COMPARISON

From Table 8- FEM Fn vs Measurements Fn presented in previous section, it is possible to see that some of the natural frequencies registered on measurements were not found as natural frequencies of the Cinema.

To verify if these results are noise introduced in the system or are representing existing natural modes of main deck which influence the behavior on this part of deck, FEM results are exanimated close of these frequencies (8.55 Hz, 11.91 Hz, 22.66 Hz, 29.1 Hz). Through this process, it should be possible to observe the modes that are exciting the ship at these frequencies:

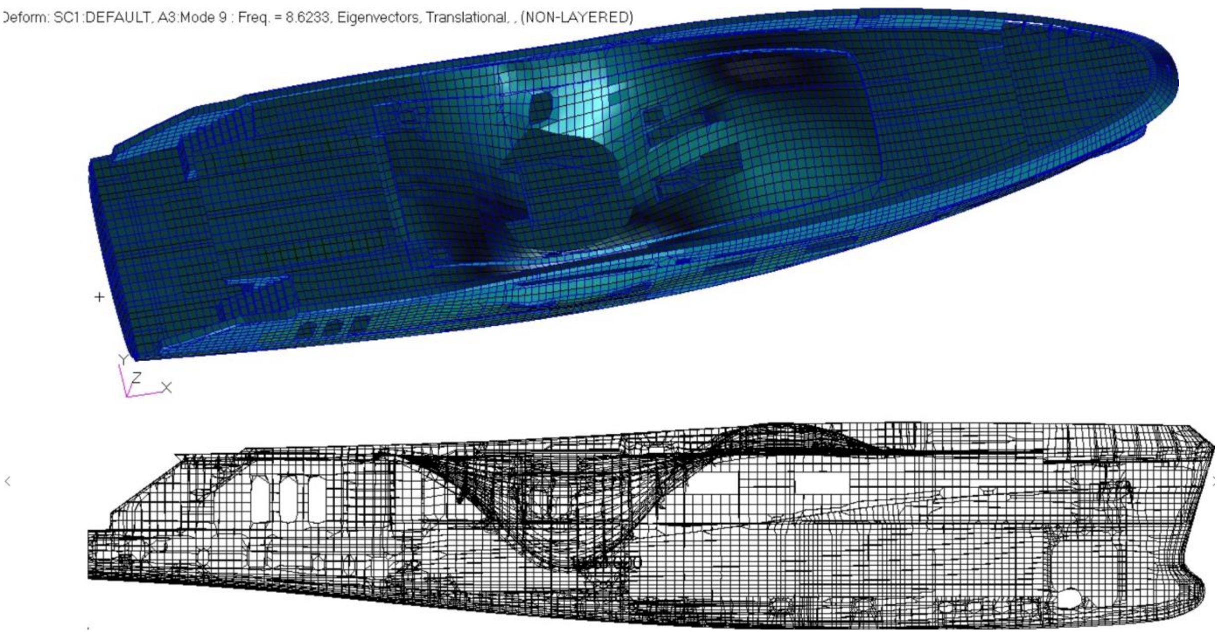

Figure 59 - Natural mode @8.623Hz

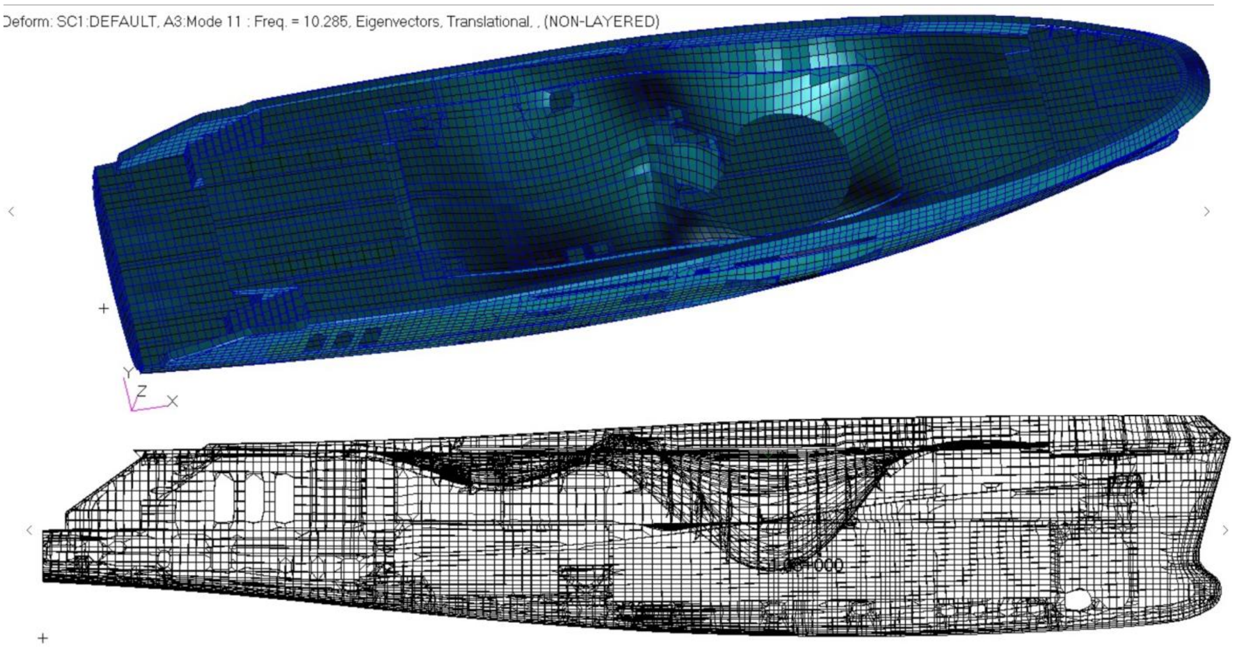

Figure 60 - Natural mode @10.285 Hz

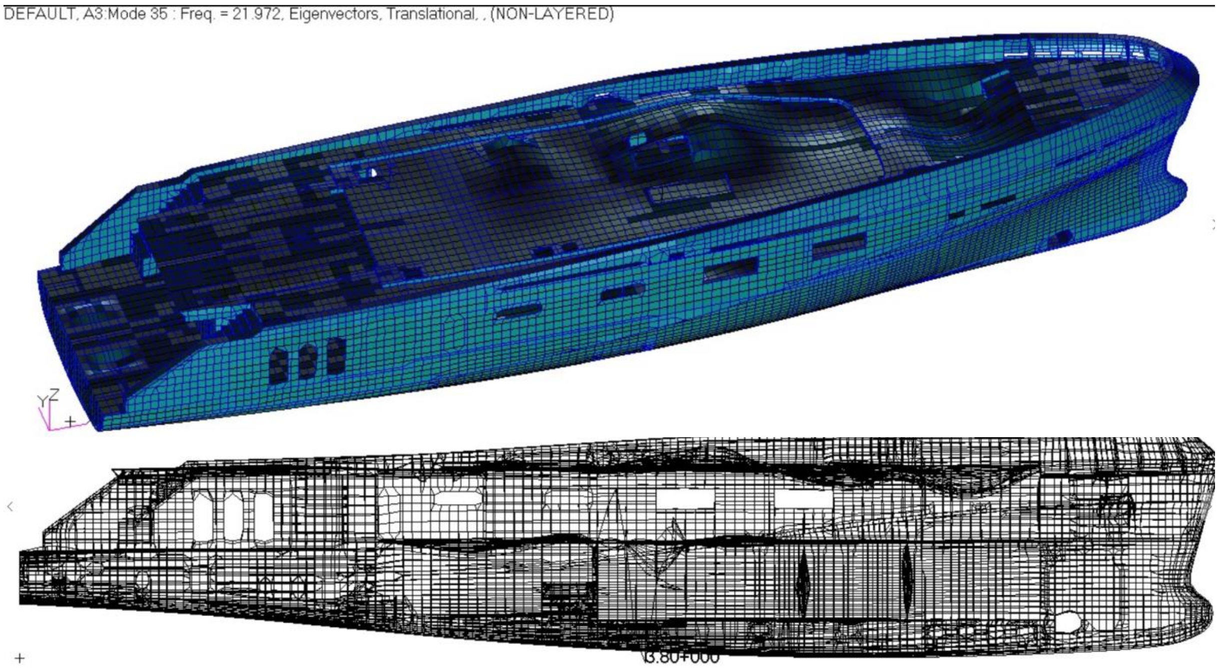

Figure 61 - Natural mode @ 21.972 Hz

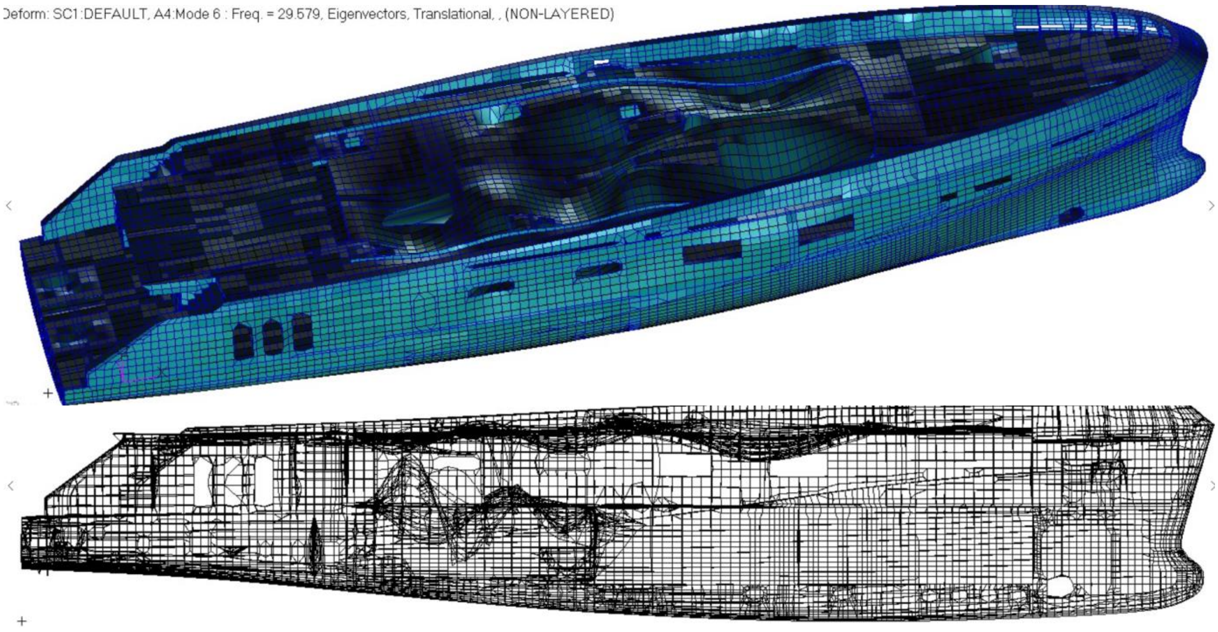

Figure 62 - - Natural mode @ 29.579 Hz

From FEA it is also advisable the presence of natural modes corresponding to the mentioned frequencies which were not found as natural frequencies of the cinema. These displacements are not showing big displacements (velocities or accelerations) in the region where the measurements were performed; moreover, with some attention it is possible to obtain these natural frequencies from the obtained experimental results.

These frequencies are not natural modes of the specific zone considered in this comparison but, natural modes of the main deck are present in these frequencies, which normally influence the behavior of the zone of interest for this part of the study.

At the end of this analysis, it is possible to make a summary of frequencies found in the FEA model, in measurements and compares the respective error.

Natural frequencies Measurement [Hz]	Natural frequencies on cinema FEM [Hz]	Natural frequencies on main deck FEM [Hz]	Error [%]
8,55		**8,623**	**-0,85**
11,15	**-**	**10,285**	**8,071**
22,66	**-**	**21,972**	**3,083**
27,25	**26,61**	**-**	**2,38**
29,1	**-**	**29,579**	**-1,63**
32,33	**31,95**	**-**	**1,19**
38,52	**36,51**	**-**	**5,35**
40,33	**41,35**	**-**	**-2,5**

It is possible to affirm that the global error between the results obtained from measurements and the results obtained by FEM model are always below 8%. This is considering the natural modes coming from all deck. The graphical results obtained on measurements in the range of this frequency are not so accurate and the error could be introduced because of this.

Highlighted results are the ones corresponding to natural modes on cinema, while the others are combinations of natural modes of the main deck influencing the displacement on the cinema. In the natural frequencies corresponding to cinema, the error is about 5%.

The margin of error is acceptable and this model is correct to perform further studies.

9. CONCLUSIONS:

1- Performed FEM model is giving trustful results; the results obtained from Finite Element Analysis are matching with experimental results. It is recommended to use this FEM model for further and future studies regarding vibration analysis of the yacht.

2- The position and computation of the basement support of the yacht in FEM model doesn't show great influence on natural frequency results of the analyzed zone.

3- The higher responses, displacement, velocity and acceleration for the natural frequencies are the ones of the local structure but also global natural frequencies influence the results and can be detected in measurements output.

4- It is of high importance the choice of the position where the measurements must be carried but also the type of source of excitation used for the study. It was observed that for low impacts some natural modes are not excited, moreover the local natural frequencies are still noticeable.

5- Further studies must be performed to simulate the behavior of the ship during navigation. The represented case in this study only represents the yacht in the actual conditions. As next step will be recommendable to include a FEM model of the superstructure and to introduce the added mass and the non-structural masses representing the different load conditions of the yachts, to perform simulations in these conditions and to obtain the natural frequencies of the yacht in real conditions. Giving the tools to compare the natural frequencies with the excitation frequencies coming from main sources of excitation. If resonance frequencies are found, then it will be advisable to perform a dynamic analysis.

10. **Bibliography**

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