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Vibration Analysis of Motor Yacht

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Research article

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Abstract

The comfort on board large motor yachts has become the object of specific attention by most important Classification Societies that issued new rules and regulations for the evaluation of noise and vibration maximum levels. These rules, usually named as "Comfort Class Rules", contain the general criteria for noise and vibration measurements in various yacht areas, and maximum limit values that such measurements should comply with them. In order to improve the comfort level onboard superyachts the University of Genova, in cooperation with one of the most important Italian shipyards, decided to start a comprehensive investigation on the dynamic behaviour of superyacht structures. The first phase of this activity has been represented by a complete review of existing comfort rules of most important Classification Society, a synthesis of which is presented in this work. In the second phase a detailed FEM analysis of a 60 meters superyacht have been carried out in order to investigate the natural frequencies of the main steel deck and of the superstructure light alloy decks. The numerical results have been compared with a first series of experimental data gathered during the vessel construction.

Keywords: Motor Yacht; Vibrations; Fem Modelling; Analysis

Introduction

Yacht designers and builders are continuously looking for new solutions to reduce construction costs and to improve the quality and innovation of their vessels. In the case of superyachts, over 30 meters in length, performances are no more a primary objective and the efforts of technical offices are addressed mostly on other aspects related more to the aesthetic impact of the project and to the on board comfort. From this point of view, vibrations and noise represent most difficult issues to deal with for designers, both in the initial phase of the project, when it is necessary to have preliminary information about the response of the structure not yet defined, and during construction, in case some critical behaviours arise in any part of the structure.

Given the objective difficulty in making any change to the dynamic behaviour of hull structure after construction, it is extremely important to perform FEM predictive analyses to identify the natural frequency of the hull and of local structures, such as decks and bulkheads, and then their response to exciting loads induced by propellers, engines and waves. Vibration problems, in particular, are certainly more critical in the case of steel and light alloy yachts, even if it's well known that also FRP vessels are not free from this kind of problem.

In this respect, most Italian shipyards always devoted great attention to the analysis of the dynamic behaviour of their pleasure ships, both steel and GRP. Recently, a specific research has been started by the Naval Architecture Section of the Department of Electrical, Electronic, Telecommunication Engineering and Naval Architecture (DITEN) of the University of Genova, aimed at deepening the knowledge of all those aspects related to the vibrations of hull structures and the possible actions that can be undertaken to reduce them.

In this work, an initial comparison between the limits imposed by the most important Classification Societies, for the assessment of comfort levels on board yachts, is carried out. Later, a numerical model of a superyacht is created.

Figure 1: A 60-meter superyacht under construction.

Comfort Rules

In the last decade, the sensitivity to the issues related to the human reaction to stresses related to the operations on board of any means of transport is widely increased. In the maritime field, in particular, this sensitivity resulted in an increasing attention to the impact of environmental factors of the ship on human health and comfort, according to the response of crew and passengers. These themes can be assessed by considering the human reaction to the energy and material (pollution) releases of means of transport. Another topic, which is also of growing interest but not subject of the present discussion, affects the response of other species to these emissions (sensitivity of aquatic animals to noise, direct and secondary effects of pollutant emissions on flora, etc.).

Vibrations generated by the energy produced by engines for ship propulsion or crew and passenger on board liveability are elements that interfere, without any doubt, with the presence of people on board and their activities. The consequences of vibrations, in accordance with the relative levels, are relevant for passenger and staff comfort, health and efficiency and for the proper functioning of sensitive apparatus.

The evaluation methodologies of ship vibrations within the mid-bass frequency (from 1 to80-100 Hz) as well as indications on the acceptability of the resulting levels, are delegated within the International Organization for Standardization (ISO) in the standard with the reference number 6954 (afterwards ISO 6954). The first issue of this regulation dates back to 1984, from which the term ISO 6954-1984. The latest review occurred on 2000, so that the code is named ISO 6954-2000.In the following paragraphs, the two versions of the rule ISO 6954 will be compared in order to identify their main advantages and disadvantages.

ISO 6954-1984

The ISO 6954-1984 was born as the basis for vibration measurements on board merchant vessels over 100 meters, from which the overall acceptability of the measured values should be assessed, in relations of technical aspects and the impact on the comfort and the crew performances [1]. The standard sets the procedures for measurement starting from ISO 4867 and ISO 4868 and for the evaluation of the human impact refers to ISO 2631-1. In Figure 2 and Table 1, the admissible range of vibrations from situations commonly accepted on board and measured over time is shown. As it is possible to see in the graph, the limit values have a bi-linear trend with the frequency, decreasing between 1 and 5 Hz and constant between 5 and 100 Hz [2-4].

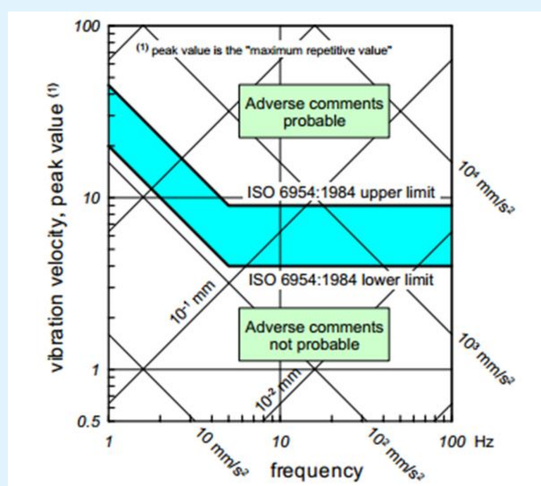

Figure 2: Vibrations limits as proposed by ISO 6954:1984.

ISO 6954:1984	Frequency range	
	1 - 5 [Hz]	**5-100 [Hz]**
Values above which adverse comments are probable	Peak acceleration	Peak velocity
	285 [mm/s²]	9 [mm/s]
Values above which adverse comments are not probable	Peak acceleration	Peak velocity
	126 [mm/s²]	4 [mm/s]

Table 1: Limit vibrations by the ISO 6954:1984.

The Maximum Repetitive Value (MRV) is the value that must be measured and compared preserving the maximum value of the three main directions evaluated separately.

An important critic against ISO 6954:1984is formulated in 2006 ISSC Committee II.2 in relation with the definition of the Maximum Repetitive Value. MRV cannot be easily defined and this could be a source of ambiguity in the vibration level assessment. The situation does not improve in case MRV is deducted from the r.m.s value (Vrms). In fact, the rule does not specify the bandwidth and average period from which the value depends on. The equation to obtain the MRV through the Conversion Factor (CF) is:

$$MRV = C_F \cdot \sqrt{2} \cdot V_{rms} \qquad (1)$$

The Crest Factor is defined as:

$$Crest\ Factor = C_F \cdot \sqrt{2} \quad (2)$$

The value to be assigned to C_F coefficient was (and still is) the subject of a long and lively debate. Nevertheless, in case of uncertainty, a tentative value of 1.8 is suggested. However, the value of 1.0 in the case of the peak value of a single line of a signal frequency spectrum can be considered reasonable. As a matter of fact the C_F corresponding to a purely sinusoidal signal at the frequency of interest is used. When MRV is obtained from a frequency spectrum (FFT), its value varies depending on the acquisition parameters set up, such as block-size and resolution. Even in this case a significant uncertainty of results derives from the assumed settings, not specified by the rules. Finally, the 1984 rules involve the strict stationariness of sources, in relation, as an example, to the rotation conditions of machinery. In fact, since only the higher peak value of the spectrum is maintained, it is necessary that, during the acquisition time, the reference frequency does not drift on adjacent spectral lines, thus providing contiguous lower peaks value than the value of the corresponding source if stationary.

Since 1997 the concept of overall frequency weighted r.m.s. is introduced in order to answer to multi frequency phenomena at Crest Factor below 9 (the typical vibration signals on board are found generally well below this value). The definition of a generic quantity g in terms of overall frequency weighted *r.m.s* in the time domain (T= measurement duration; g_w= weighted quantity) or in spectral terms (W_i is weight function of the i-th band; g_i= value *r.m.s* in the i-th band) is defined by the rules as (ISO 2631-1):

$$g_w = \sqrt{\frac{1}{T} \cdot \int_0^T g_w^2(t)dt} \quad g_w = \sqrt{\sum_i (W_i g_i)^2} \quad (2)$$

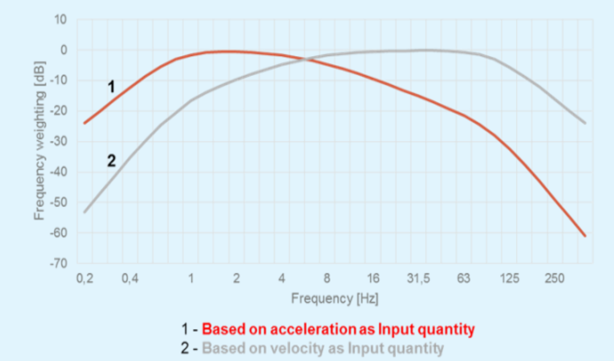

Figure 3: Combined frequency-weighting curves (ISO 6954-2000).

This approach has spread rapidly to report on human exposure aspects (ISO 2631-1) and the habitability of on board spaces (ISO 2631-2) [5]. It is demonstrated that this measure is highly insensitive, i.e. stable, compared to the elements mentioned above regarding the 1984 version. The ambiguity and arbitrariness related to the definition of MRV are removed.

Given the potential provided by current acquisition/analysis systems, the overall acquisition (i.e. integrated on the frequency interval) does not affect the diagnostic capabilities provided by the acquisition in narrowband specified by the previous ISO. With the new acquisition units, it is always possible to capture and save simultaneously the format both in broadband and narrowband. Due to all the reasons listed above, in 2000 the rule ISO 6954 is aligned to this new approach.

ISO 6954-2000

The ISO 6954-2000 governs the measurement and evaluation of vibrations from 1 to 80 [Hz] on board merchant and passenger ships regardless of length, with particular attention to aspects of habitability of space, is divided into three different categories:

– A: passenger cabins;
– B: crew areas;
– C: workspaces.

It provides measurement rules starting with ISO 8041 and refers to ISO 2631-1 for data processing and evaluation of habitability on board [6,7]. In Table 2, the admissible range of vibration in terms of overall frequency-weighted r.m.s. is shown. The maximum value of the three main directions for at least two measurement positions for deck is maintained and only a measurement in the vertical direction is required in the other points.

ISO 6954:2000	Area classification					
	A		B		C	
	Acceleration	Velocity	Acceleration	Velocity	Acceleration	Velocity
Values above which adverse comments are probable	143 mm/s²	4[mm/s]	214[mm/s²]	6[mm/s]	286[mm/s²]	8[mm/s]
Values above which adverse comments are not probable	71,5 [mm/s²]	2[mm/s]	107[mm/s²]	3[mm/s]	143[mm/s²]	4[mm/s]

Table 2: Limit vibrations by the ISO 6954:2000.

In conclusion, in Table 3 the 1984 and 2000 revisions of the standard are compared:

Revision 1984	Revision 2000
Aim	
Title: "Mechanical vibration & shock – Guidelines for the overall evaluation of vibration in merchantships"	Title: "Mechanical vibration – Guidelines for the measurement, reporting&evaluation of vibration with regard to habitability on passenger&merchantships"
– Merchant ships – Length>100 m – Evaluation overall of vibrations	– Merchant and passengerships – Anylength – Habitability of occupied spaces divided into three different categories (A-B-C)
Acquisition / processing of signal	
– Instrumentation: ISO 4867/4868 – Frequency range: 1- 100 [Hz] – Size: acceleration or speed component – Direction: three main components	– Instrumentation: ISO 8041 – Frequency range: 1- 80 [Hz] – Size: acceleration or speed component – Direction: three main components for two points for deck, only vertical for other points.
– Format: Maximum Repetitive Value MRV, or(V_{rms}.$\cdot \sqrt{2} \cdot C_F$)	– Format:Overallfrequency-weightedr.m.s.
Sensitivitycurve and frequency weighting	
– No explicit mention (the trend bi-linear in frequency of the limit values indirectlyintroduces a sensitivity curve)	– Direct reference to curves (ISO 2631-2)
Limit	
Comparison of the processed value with the curves bi-linear reference or the Table 2, defined fields of adverse comment probable and adverse comment not probable.	Comparison of the processed value with the limit in the Table 3, defined fields of adverse comment probable and adverse comment not probable.

Table 3: Comparison between two rules.

The change in the approach suggested by ISO 6954-2000 does not appear motivated by newer or more specifically targeted experimental evidences. It seems rather as the end of a process of transferring assessments from other fields without feedback of specific factors such as expectations of passenger's life on board and the duration of the trip or the peculiar environmental factors. As a matter of fact, the ship-owners and Classification Societies opinion on the introduction of the new ISO is still very cautious. The ISO 6954-1984 is still widely adopted, as shown in Figure 4.

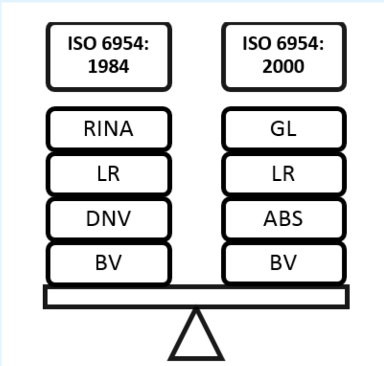

Figure 4: Classification Societies division that ISO is adopted.

Classification Societies

Baker and Mc Sweeneyon 2009, as an example, present a complete analysis of present ABS Rules concerning vibrations and noise published in the 'Guide for the Class Notation Comfort - Yacht' on 2008 [8,9]. Two notational options are considered: COMF(Y), which establishes a level of comfort based on ambient noise and vibration alone and COMF(Y+) which adds slightly more demanding criteria for noise and vibration, and provides additional criteria for the assessment of motion sickness. ABS Yacht Comfort guide, however, have been recently revised in some aspects.

Other Classification Societies produced similar rules for the assessment of comfort levels on board yachts and superyachts. In the following a list of the most important ones are reported:

– Bureau Veritas on 2012, Part E, Sect 5, 'Additional Requirements for Yachts';
– Det Norske Veritas on 2011, Part 6, Chapt. 12, 'Noise and Vibration';
– Germanischer Lloyd on 2003, Part 1, Chapter 16, 'Harmony Class';
– Lloyd's Register on 2011Chapt. 6, 'Passenger and Crew Accommodation Comfort';
– RINA on 2011, Part E, Chapt. 5, 'Comfort on board' [10-14].

Some examples of maximum allowable levels for vibration are shown in Table 4 for American Bureau of Ships, Bureau Veritas, Lloyd's Register and RINA. For what ABS is concerned a synthesis of COM(Y) and COMF(Y+) rules is presented for yachts larger than 45 meters in length [15]. The vibration parameter ν is the spectral peak of structural velocity in mm/s.

Location / C S	Cabins and lounge	Public spaces	Open decks
ABS (Y) Hz	1 - 80	1 - 80	1 - 80
ABS ν [mm/s]	1.25 - 2.0	1.25 - 2.0	1.25 - 2.0
ABS (Y+) Hz	1 - 80	1 - 80	1 - 80
ABS ν [mm/s]	1.0 - 1.5	1.0 - 1.5	1.0 - 1.5
B V Hz	1 - 80	1 - 80	1 - 80
B V ν [mm/s]	1.0 - 3.0	1.0 - 3.0	1.0 - 3.0
L R Hz	1 - 80	1 - 80	1 - 80
L R ν [mm/s]	1.8- 2.5	1.8 - 2.5	1.8 - 2.5
RINA Hz	0 - 100	0 - 100	0 - 100
RINA ν [mm/s]	1.0 - 3.0	1.0 - 3.0	1.0 - 3.0

Table 4: Maximum whole-body vibration according to American Bureau of Ships, Bureau Veritas, Lloyd's Register and RINA Comfort Rules for yachts.

From a structural point of view, vibrations can be assessed at both global and local level. As well known in the first case it's quite impossible to apply any corrective action after the construction and thus it is indispensable to carry out detailed analyses at the design stage.

Although the simplified approach based on variable section beams with concentrated masses remain a valuable tool for the calculation of the first natural frequencies of the hull, the presence of large openings such as aft and side garage doors, considerably complicates the dynamic behaviour of the hull. In this case, only a FEM modelling of the entire structure allows obtaining reliable results. Actually, hull natural frequencies are very low and far from the frequencies of usual exciting forces.

For what local vibrations are concerned, the most critical areas are represented by decks, bulkheads and superstructures. The danger of resonance exists with reference to four main sources of vibrations represented by main engines, electric generators, main propellers and bow thrusters.

Engines and generators are usually mounted on resilient and this significantly reduces their contribution to hull global and local vibrations. On the contrary, main propellers induce high dynamic forces at harmonics corresponding to their blade passing frequencies (BPF); this depends on the number of blades, shaft rate and operating conditions. As stated by Roy et al. on 2011, for large motor yachts (between 50 and 100 m in length) blade-passing frequencies could be between 15-20 Hz at cruise speed condition, up to 30 - 40 Hz at maximum speed [16]. Aft and central parts of the hull are the most exposed areas of the vessel to the propeller exciting forces and their dynamic response mainly depends on the distance from the propellers. The bow thruster propeller should be considered as well; his blade passing frequency BT, much higher than BPF (over 50 Hz) but with minor intensity, can affect mainly the fore part of the vessel.

Several practical approaches are assumed by designers but, as a common general rule, in order to assure that decks do not resonate at any point in the speed range, the design philosophy should be to ensure that the first mode frequency of every deck panel is in excess of BPF. This goal is not so simple to be achieved in case of larger structural components, such as decks and bulkheads, the only possible action in this respect being the increase of the structure stiffness. While a simple plate thickness increase is not advisable because of excessive consequences on hull weight, an accurate selection of deck secondary and primary structure can provide better results.

In case of superstructures, the problem is relatively mitigated by the higher distance from propellers and, consequently, by lower exciting forces; on the other side large spans of unsupported decks lowers their natural frequencies making necessary the addition of cumbersome and anaesthetic pillars. Even so, often it becomes really impossible to keep natural frequencies higher than BPF and the only possibility is to remain below the cruise speed BPF. Even in this case a detailed FEM analysis is the only way to identify critical areas and to look for possible solutions in advance.

Nevertheless, it is plain that every action addressed to the structure vibration reduction lead to an increase of the hull weight; by way of example, it has been estimated that, for a superyacht in the range between 90 and 100 meters, the weight increase to reduce vibrations is more than 100 tons.

The purpose of this work is to verify the dynamic behaviour of the stern area of superstructure decks of a superyacht. In this area the '"upper deck" and the "sun deck" are characterized by a considerable overhang that, in conjunction with the lower aluminium stiffness, may cause annoying vibrations for passengers close to the frequencies of the exciting forces induced by the propellers. The study has been carried out by a general-purpose FEM code in two phases: in the first phase, after a detailed modelling of the stern part of the yacht, the natural frequencies of the superstructure have been determined.

In the second phase, not yet assessed in this paper, a series of frequency analyses should be performed in order to investigate the structure response by varying the intensity of the exiting forces and the structure damping. The calculation results, in terms of vibration velocity of decks, will be compared with the limit values imposed by Classification Society Rules.

Fem Modelling

For this study a three-decks superyacht 60 meters in length, under construction in a famous Italian shipyard, has been taken as case study. The ship has a steel hull and aluminium superstructures longitudinally framed with web frame interval of 1200 mm. In order to perform accurate and reliable FEM calculations a detailed numerical model of the yacht structures has been realized by the multi-purpose code ANSYS version 13.0 [17].

The mesh was created using "SHELL63" elements of the ANSYS library for plating and main reinforcements such as keelsons, floors and girders. For secondary stiffeners, such as longitudinals, simple "BEAM44" elements have been preferred to keep the model dimension as low as possible. The mass and loads of the main deck and the two superstructure decks have been modelled by "SURF154" elements, which are particularly suitable for dynamic analyses.

The hull geometry and structure lay out has been modelled starting from structure drawings kindly made available by the shipyard technical offices. Given the high amount of calculation time needed by dynamic analyses, one of the first aims of this study was to investigate the influence of different schematization approaches in terms of both model dimensions and mesh refinement on the result reliability.

With regard to the first aspect, the objective was to compare the results gathered on a complete model, let's say a numerical model of the whole ship, and the results obtained by studying only a ship portion where vibration problem could be more critical. This second approach, when possible, could involve significant advantages in the calculation time, especially when transient analysis are necessary.

The first phase of the work was then devoted to create a complete model of the 60 meter yacht without taking advantage of any structural symmetry. The hull geometry has been created in accordance to the block assembly scheme (Figure 5) assumed by the Shipyard for construction.

At this point, the second aspect of our initial investigation arose, that is the mesh refinement compatible with reasonable calculation time and result reliability. Many tests have been carried out on a reference model represented by a portion of the aft bottom structure, block FO-02. A good compromise between structure definition and mesh "weight" appeared to be that corresponding to an average panel diagonal around 300 millimetres (Figure 6 and 7).

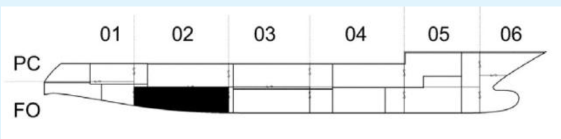

Figure 5: Assembly scheme of hull prefabrication blocks.

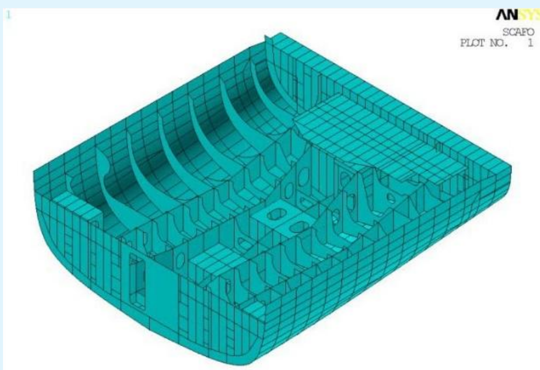

Figure 6: Model geometry of the FO 02 block.

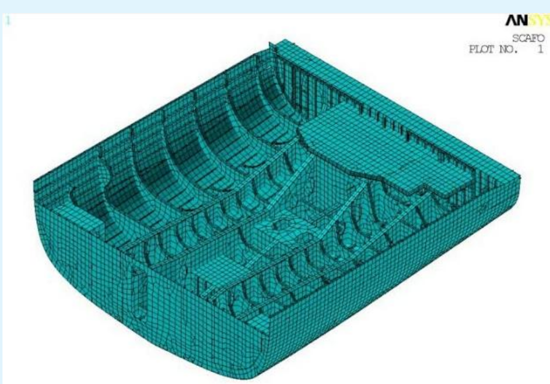

Figure 7: Mesh of the FO 02 block.

The geometry of all the blocks of the hull, in steel, and superstructures, in aluminium light alloy, have been modelled. All blocks have been carefully schematised and, before assembling them in the global model, meshed and tested. A further check has been carried out by assembling the blocks into three parts corresponding to the stern, centre and fore portion of the vessel.

The geometry of the three parts of the vessel are shown in Figure 8, 9 and 10, while in Figure 11 a view of the inside structure is shown by a longitudinal section of the model. After the assemblage of the block geometries all blocks have been meshed and grouped again into three hull portions and, finally, in the global model. In figures 12, 13 and 14 some details of superstructure decks are shown; it should be underlined that, excluding minor details as brackets and barrotts, all other component and details have been included in the schematisation.

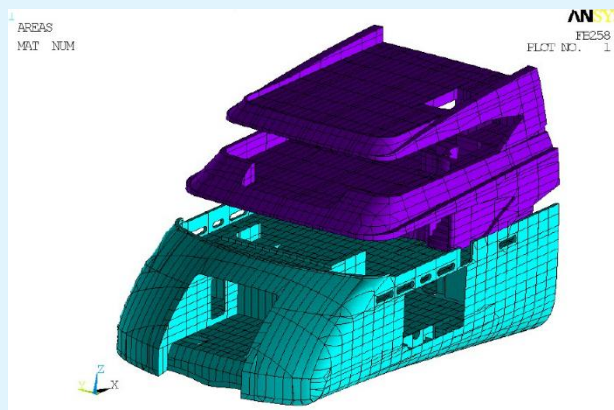

Figure 8: Geometry of the stern part of the vessel.

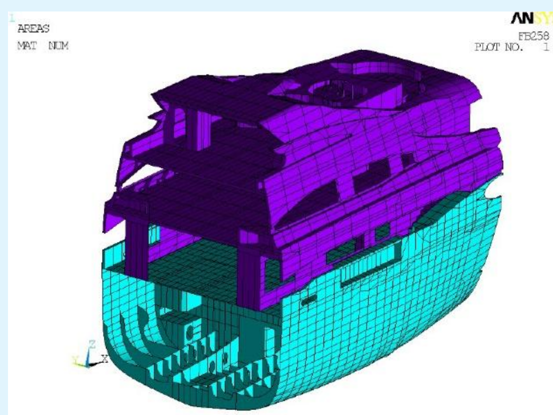

Figure 9: Geometry of the centre part of the vessel.

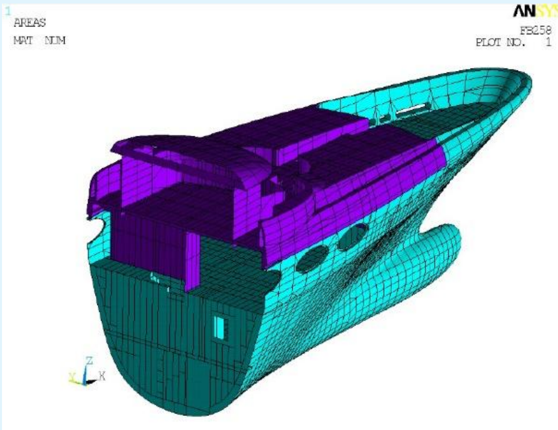

Figure 10: Geometry of the fore part of the vessel.

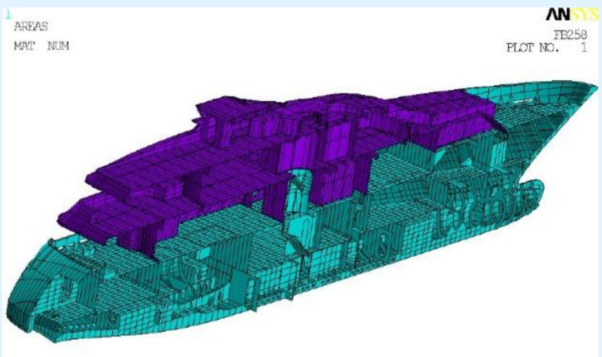

Figure 11: View of the left hand side of the complete model. The light blue part represents the steel hull structures while the dark part represents the aluminium light alloy superstructures.

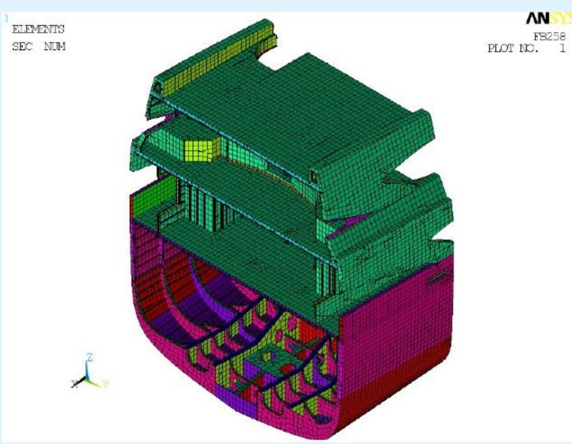

Figure 12: Particular of FO 02 and PC 02 block mesh.

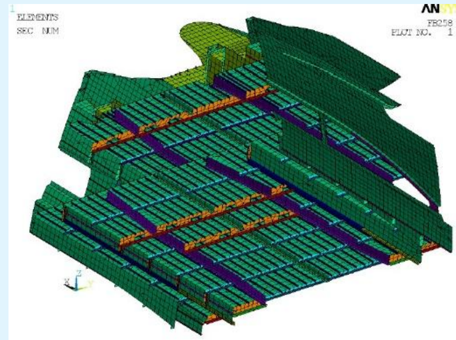

Figure 13: Particular of forward superstructure mesh.

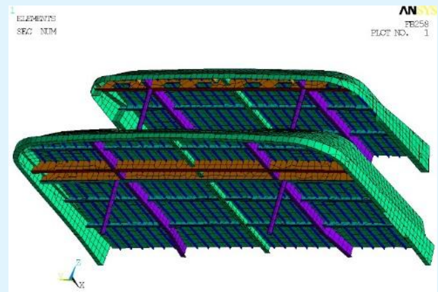

Figure 14: Particular of flying bridges mesh.

In Figures 15, the meshed model of the complete yacht is shown. In total, the numerical models consist of about 263000 nodes and 365000 elements.

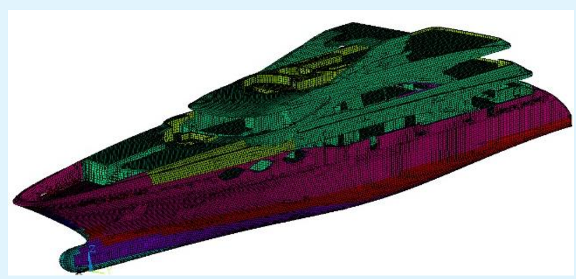

Figure 15: View of the complete mesh.

Calculations have been carried out for different model configurations in order to evaluate which difference each case could imply on the results. The following cases have been investigated:

- Two complete models with different outfitting conditions:

Model A: complete model of hull and superstructures. In this case only steel and aluminium structures have been considered without any kind of outfitting (no outfitting "n.o." case).

Model B: complete model of hull and superstructures including the weight of outfitting and machineries (with outfitting "w.o." case).

- Two partial models (stern part):

Model C: partial model extended from, approximately, midship section to stern. In this model no outfitting components have been included; only steel and an aluminium structure has been modelled.

Model D: same numerical model as case "C" but with outfitting and machinery weights.

- Six single deck numerical models:

Model E: main deck single model without outfitting. The model includes the hull structures from bottom to main deck.

Model F: main deck single model with outfitting.

Model G: upper deck single model without outfitting.

Model H: upper deck single model with outfitting.

Model I: sun deck single model without outfitting.

Model L: upper deck single model with outfitting.

These six last models (single decks) have been used for comparison with the experimental results measured on the real structure under construction in same conditions (only structures). The same comparisons have been carried out on the hull components when only the structure was built before outfitting.

For models with outfitting, on each deck a distributed load has been applied corresponding to the finishing and outfit weight (about 300 N/m2); on the main deck the concentrated load of the tender crane in the garage has been considered as well. The tender garage is located in the stern area below the main deck.
The yacht mass has been increased in way of 80% of displacement to take into consideration the water added mass.

The "C" and "D" numerical models have been constrained in correspondence of section n.17 where the complete hull has been "cut". All nodes located on this section have been completely clamped, thus forcing the structure to behave as a cantilever (Figure 16). Constraints are located far enough from the area of interest; as a consequence the constraint effect on the results of the analysis can be assumed irrelevant. Same solution has been assumed for single deck models; in Figures 17 and 18 the upper and sun deck constraint schemes are shown.

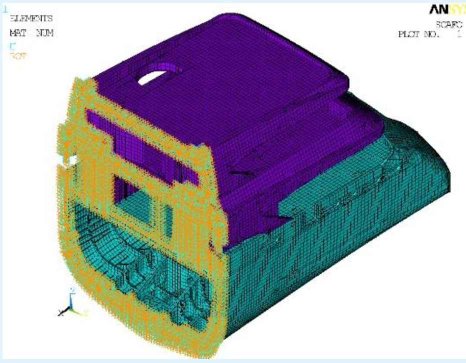

Figure 16: Model "C" and "D" constraint distribution.

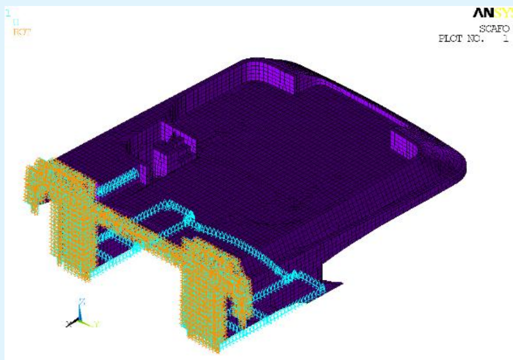

Figure 17: Model "G" and "H" constraint distribution.

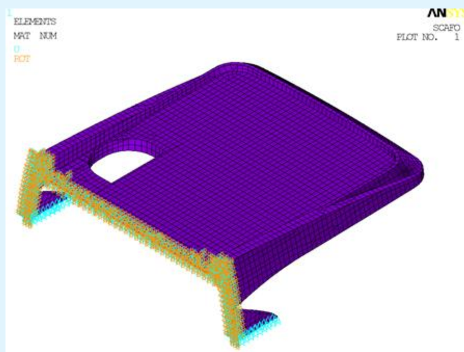

Figure 18: Model "I" and "L" constraint distribution.

Modal Analysis

The first calculation has been carried out on the complete model. The natural frequencies of the whole structure have been determined by modal analysis; the Lanczos mode extraction method (LANB approach in ANSYS code) has been employed. This solver is particularly suitable for large models consisting of shells or a combination of shells and solids.

The first check has been performed on the hull first natural frequency, which resulted to be in accordance with the typical natural frequency of similar yachts built by the same shipyard (see Figure 19). For what local vibrations are concerned, as expected the most significant vibration modes have been individuated on the main deck and on the superstructure decks. The results obtained by FEM calculations have been compared with those measured on the same vessel during construction phases. In Table 5, the most significant natural frequencies, calculated on the two models "A" and "B" and measured on the real structure (assembled and corresponding to the "A" model), are reported.

The displacement plots of the first natural frequencies of the three decks ("B" model) are shown in Figures 20, 21 and 22. These plots can be useful to individuate other local modes taking place at the same frequency. As it is possible to see, while the main and sun deck modes are quite isolate, the upper deck mode frequency implicates many other minor local vibrations.

Item	Calculated Hz		Measured Hz
	"A"	"B"	
Main deck - 1st mode	24.5	12.9	13.5
Upper deck - 1st mode	25	12.6	11.5
Sun deck - 1st mode	14.6	7.6	11.3

Table 5: Calculated and measured natural frequencies of main deck and superstructure decks.

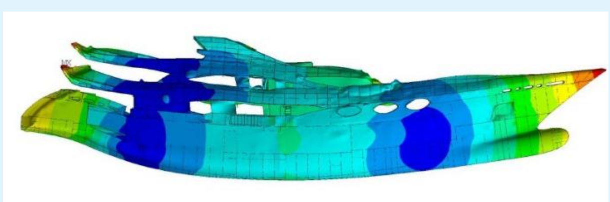

Figure 19: Model "B": first natural hull frequency without added mass (5.87 Hertz).

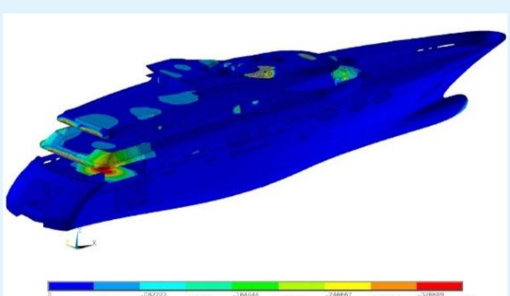

Figure 20: Model "B": main deck first natural frequency (12.9 Hertz).

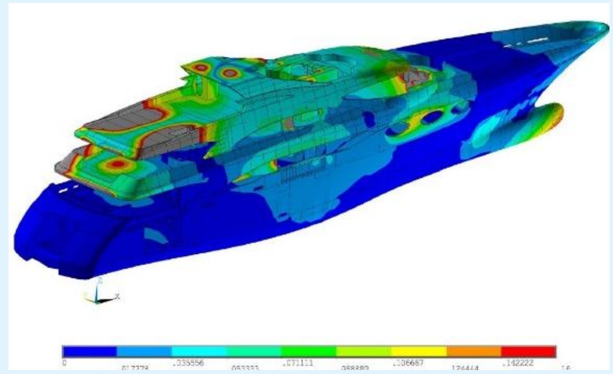

Figure 21: Model "B": upper deck first natural frequency (12.6 Hertz).

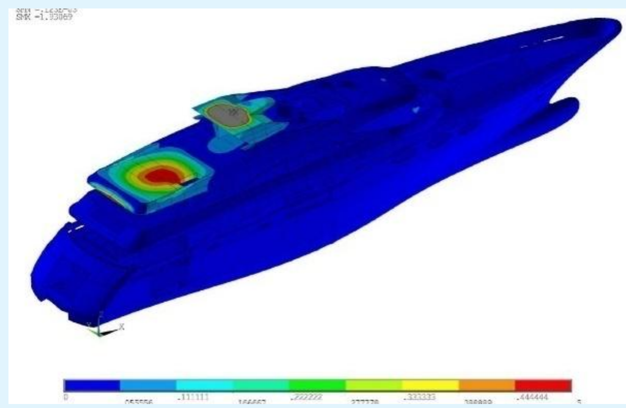

Figure 22: Model "B": sun deck first natural frequency (7.6 Hertz).

At this point, the same calculations carried out for the global model have been repeated on the partial models, the stern part and single decks and for the same conditions (with and without outfitting, "C" and "D" numerical models). The results obtained on the stern model are resumed in Table 6. The displacement plots of the first natural frequencies of the three decks (model "D") are shown in figures 23, 24 and 25. As it is possible to see, the results are very similar to the global model ones.

Item	Calculated Hz		Measured Hz
	"C"	"D"	
Main deck - 1st mode	24.8	13.2	13.5
Upper deck - 1st mode	25.5	12.3	11.5
Sun deck - 1st mode	16.2	8.6	11.3

Table 6: Natural frequencies of main and superstructure decks on the partial "C" and "D" models.

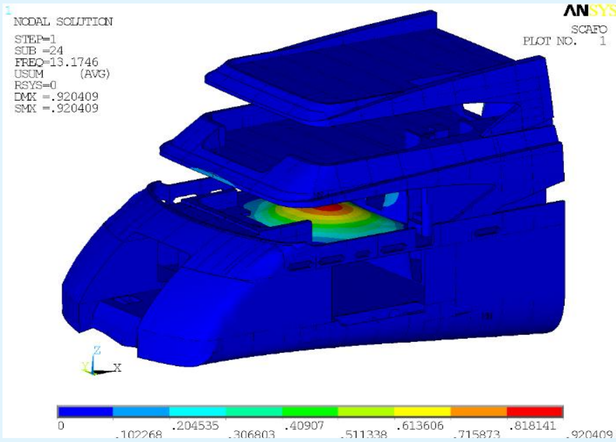

Figure 23: Model "D": main deck first natural frequency (13.2 Hz).

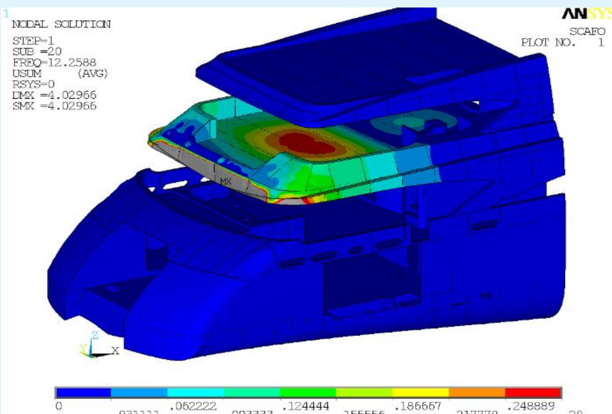

Figure 24: Model "D" - upper deck first natural frequency (12.3 Hz).

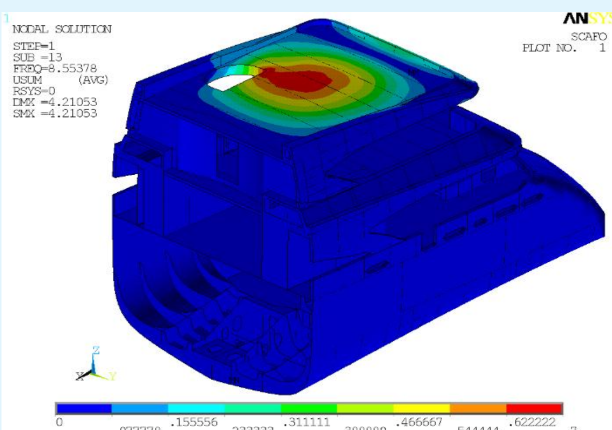

Figure 25: Model "D": sun deck first natural frequency (8.6 Hz).

Finally, the natural frequencies of decks have been calculated on the single deck models with and without outfitting ("E", "F", "G", "H", "I" and "L" numerical models). The gathered results are resumed in the following Table 7. As previously done, the displacement plot of the considered models ("F", "H" and "L" models with outfitting) are shown as well (Figures 23-25).

	"E"	"F"	"G"	"H"	"I"	"L"
Main deck 1st mode	24.1	12.8	-	-	-	-
Upper deck 1st mode	-	-	24.4	12.3	-	-
Sun deck 1st mode	-	-	-	-	17	9

Table 7: Natural frequencies of main and superstructure decks on the partial "C" and "D" models.

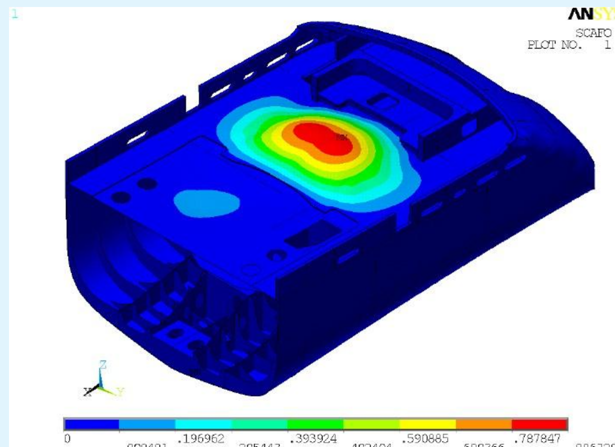

Figure 26: Model "F": main deck first natural frequency (12.8 Hz).

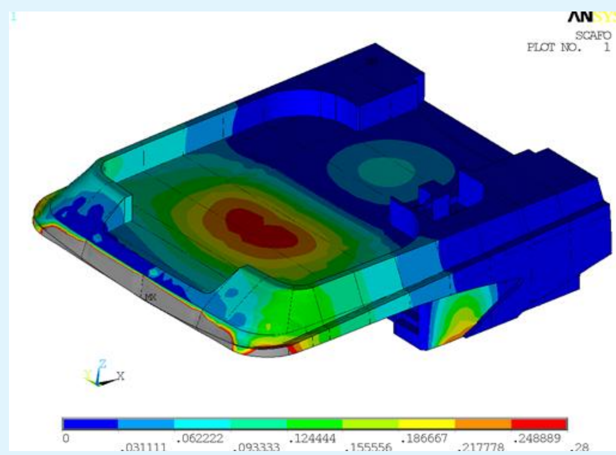

Figure 27: Model "H": upper deck first natural frequency (12.3 Hz).

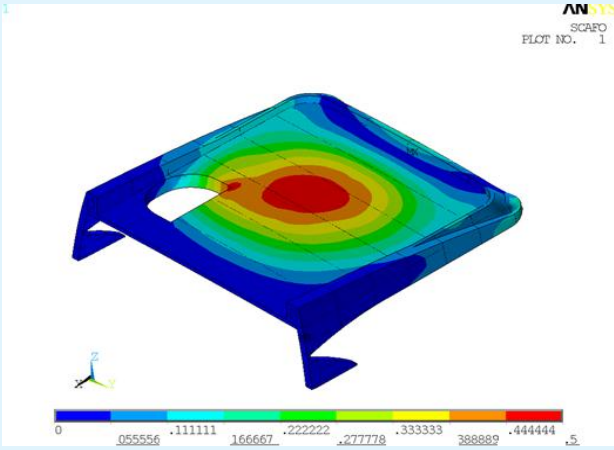

Figure 28: Model "L": sun deck first natural frequency (9.0 Hz).

In the following Table 8, the obtained results for all models are resumed in order to compare them each other. In this case, the name of the models has not been used; mention has been made to the "outfit" or "non outfit" schematisation. From the table analysis is comes out that, if models are properly refined, the results could be reliable also for partial models. This, obviously, could be very useful for time saving sake, especially when only local modes are of interest. For what the matching of FEM results with experimental measurements is concerned good agreement exists for main and upper deck case.

Item	Calculated [Hz]			Measured [Hz]
	Global model	Stern model	Single deck model	
Main deck ("n.o.")	24.5	24.8	24.1	-
Main deck ("w.o.")	12.9	13.2	12.8	13.5
Upper deck ("n.o.")	25	25.5	25.4	-
Upper deck ("w.o.")	12.6	12.3	12.3	11.5
Sun deck ("n.o.")	14.6	16.2	17	-
Sun deck ("w.o.")	7.6	8.6	9	11.3

Table 8: Calculated and measured natural frequencies of single deck.

Only for sun deck, the experimental measurements show a significant difference with the numerical results. At the moment, this is the subject of a specific study in order to ascertain if this disagreement comes from a schematization or measurement error.

Conclusion

The study herein presented is the first part of a research program aimed at thoroughly analysing the structural problems of modern motor yachts. In this phase, attention has been devoted to those aspects having a direct impact on the comfort characteristics of the vessel. At first, a complete overview of recent rules issued by ISO and Classification Societies has been carried out in order to highlight the most important parameters to be considered. Then attention has been focused on hull vibrations, at local level.

The evaluation procedures and limit values suggested by the rules has been applied to a case study, consisting of a 60 meters steel yacht, the dynamic behaviour of which has been constantly monitored during the various construction phases.

Taking advantage of the data availability a detailed FEM model of the case study yacht has been set up with the aim of identifying the natural frequencies of local structures such as bulkheads, main deck and aluminium super-structure decks. The result of this first analysis satisfactorily matches the experimental measurements carried out on board.

As one of the aims of this study was to verify the result reliability obtained by different refinement level models, a number of schematisations have been considered of the same structure, starting from single part models, than passing to a more complete one, representing one half of the yacht (the stern part) and, finally, up to the global model of the yacht. All the models have been analysed with different types of outfitting. As shown in Table 5, if models are properly refined, the results could be reliable both for global and partial models. This, obviously, could be very useful for time saving sake, especially when only local modes are of interest.

At present the research continues in the field of transient analysis in order to verify the effect of exciting propeller forces on the structural dynamic behaviour and, in particular, if the structural response falls within the Rules limits. This is the final test to ascertain the comfort level of the vessel.

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