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The Analysis of Ship's Hull Structure Using Vibration Measurements at Sea

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Abstract

Ship’s vibration represents a constant concern for engineers and constructors. The solutions for each problem come after many trials both at sea and at dry dock. For a good maintenance even after the ship is launched, vibration and noise measurements must be conducted to prevent damages. Ship’s hull vibration is one of the problems encountered onboard. There are different sources which affect hull’s structure, including machinery, propulsion system (propeller and shaft). In this paper we analyze the results of the measurements of training-ship “Mircea” during trials at sea. Our attention was concentrated over the impact of lower frequency vibration to ship’s structure.

Key words: *vibration, hull structure, FFT Analysis, Output-Only Modal Analysis, engine, propeller.*

Theoretical Aspects of Ship Hull Vibration

One of the problems in ship’s vibration is Ship Hull Vibration. Hull vibration has grown dramatically with the increase of ship size and power over the years. As the power-to-weight ratio of ships increased, so did problems with vibrations. Ship Hull Vibration is difficult to analyze because the sources that cause the vibration are interconnected. The major contributors to Ship Hull Vibration are the propeller and ship’s machinery.

When measuring vibration during ship operation, ship’s own vibration sources provide the excitation forces, i.e. various orders of the propeller and engine. In rare cases, the vibration level is caused by slamming-induced impacts or by special flow phenomena [Germanische Lloyd].

For very low frequencies (up to 2 Hz) it can be stated that there are no other excitations than that of the sea [1]. Hull girder’s modes up to 2 Hz may be excited by sea waves, but on the other hand, hull deformations are negligible when compared to both local deformations and ship motion amplitudes.

The propeller creates also problems in Ship Hull Vibration through the propeller design, shaft system and propeller cavitation. Cavitation has a large influence on the static pressure on the hull surface.

The machinery onboard the ship has also an impact on Ship’s Hull Vibration. It creates torsional, axial and lateral vibration patterns. The torsional vibration is due to the variations of gas pressure in the cylinders and due to the varying torque in the crankshaft made by the crankshaft’s mechanism [2]. The axial vibrations come from the arms of the crank, while the lateral vibrations have a less contribution to the vibration.

Onboard a ship, there are numerous sources of vibration and noise. These sources include [3]:

- the prime movers – typically diesel engines
- shaft-line dynamics
- propeller radiated pressures and bearing forces
- air conditioning systems
- manoeuvring devices such as transverse propulsion units
- cargo handling and mooring machinery
- intakes and exhausts
- slamming phenomena

Another source for ship's hull vibration is slamming (after-body slamming and fore-body slamming) which is a shock impact generating supplemental stress in ship's structure.

Propeller induced structural vibration through the thrust bearing is an important source for low frequency shipboard noise. While the propeller delivers thrust to the ship, it also generates oscillatory disturbances. A small variation of thrust is produced when the blades rotate through the non-uniform wake. The frequency of this variation is well known as the blade passing frequency, which is equal to the shaft rotational speed multiplied by the number of blades on the propeller [4].

Local structures close to the propeller, main engine or thrusters might vibrate excessively. To avoid such resonant vibration, it must be ensured that the natural frequency of the structure does not coincide with the respective exciting frequency. Typically, strong local vibration occurs at stiffened panel structures (deck panels, tank walls) or as a coupled vibration of a subsystem and its foundations. The subsystem can be either a structure itself (mast, rudder, shaft bracket etc.) or a machinery and equipment part (gear box, fin stabilizer, generator etc.). The natural frequencies of regular panel structures can be calculated quickly by estimation formulae but for the analysis of coupled vibration phenomenon the finite element method is called for [Germanische Lloyd].

Excessive ship vibration is to be avoided in order to reduce the risk of structural damage on the local structures. Structural damage such as fatigue cracking due to excessive vibration may occur on local structures, including but not limited to engine foundation structures, engine stays, steering gear rooms, tank structures, funnels and radar masts. It should be noted that the structural damage due to the excessive vibration vary according to the local structure detail, actual stress level and local stress concentration and material property of the local structures therefore, the vibration limits for local structures are to be used as a reference to reduce the risk of structural damage due to excessive vibration during the normal operating conditions [5].

Most structural failures during the service life of ships are due to fatigue. The term "fatigue" refers to cyclic loading and the gradual growth of cracks to failure.

A fluctuating stress can initiate microscopic cracks which gradually increase in size until, after a large number of cycles, the cracks have become so large that fracture occurs. Any inter-crystalline slide is accompanied by a material disruption. The first sign of the fatigue is the forming of so-called sliding bands. Under the continuous influence of vibrations, the sliding bands progresses and generate small cracks. Then the cracks extend into the material after the equation below:

$$\frac{dl}{dN} = c\,\rho_r^m\, l_i^n \tag{1}$$

where: l is the length of crack, N the number of cycles of stress, *c,m,n* – constants depending on material properties (in many cases $m = 2$, $n = 1$) and ρ_r is the relative deformation stress.

As the crack extends, the stress in the material becomes so high, that the propagation continues and it produces the disruption of the material through fatigue. Fatigue has a pronounced statistical character because of instable nature of the crack.

For the cases when the amplitude of the stress is variable (random regime), a linear law for disruption summing can be used:

$$D = \sum_i \frac{n_i}{N_i} \tag{2}$$

where n_i is the effective number of cycles for a value of stress which claims a total number of N_i cycles until breaking. Under these circumstances, the disruption occur when D = 1.

Fatigue depends on the magnitude and duration (number of cycles) of loads acting on ship's structure. Generally, for ships, the sources for fatigue are:

- wave induced loads including whipping. Number of bending cycles for ship's life (20 years) is in order of 10^8.
- alternation between loaded and ballasted conditions
- mechanical sources e.g. engine & propeller

In our paper we discuss about the vibrations produced by the main engine and propulsion system.

Experiment and results

In our experiment, we measured the vibrations onboard training-ship "Mircea" during a voyage in the Black Sea. A triaxial accelerometer and a uniaxial accelerometer were used for measurements of the vibrations from ship's machinery, propulsion system and ship's structure. The trials were conducted on a calm sea with small waves.

Vessel "Mircea" has a combined structure, consisting both in metal and wood. Because it's a sailing ship, its structure has more elasticity than a commercial vessel.

The measuring equipment consisted in one triaxial accelerometer type DeltaTron 4506 and one uniaxial accelerometer type 752A 12x, both from Bruel&Kjaer connected to a Machine Diagnostics Toolbox 9727 also from B&K. We used PULSE 12 software for the Fourier analysis and the calculation of the vibration spectra was at the resolution of 0,0625 Hz.

The mathematical algorithm for conventionally used analysis techniques is typically Fourier's analysis. This algorithm is used for the excitation from machinery. On the other hand, propeller most commonly produces time series signatures with significant cyclic perturbations.

The recordings were made in various measuring points. In this experiment, we chose the foundation connections of ship's diesel generators and ship's main engine, and the rudder compartment.

The structure of a ship like training-ship "Mircea" has a particular structural configuration. During sailing, the ship is exposed to vibrations of various degrees. The vibrations limits described above may not be applicable to all of the local structures which have different structural configurations and details. Depending on the vessel configuration, these limits can be less conservative or too conservative.

So we measured in these points the response to vibrations of ship's hull and we analyzed the results in the range 0 – 5 Hz in terms of displacement for rudder compartment, and in terms of velocity for the main engine.

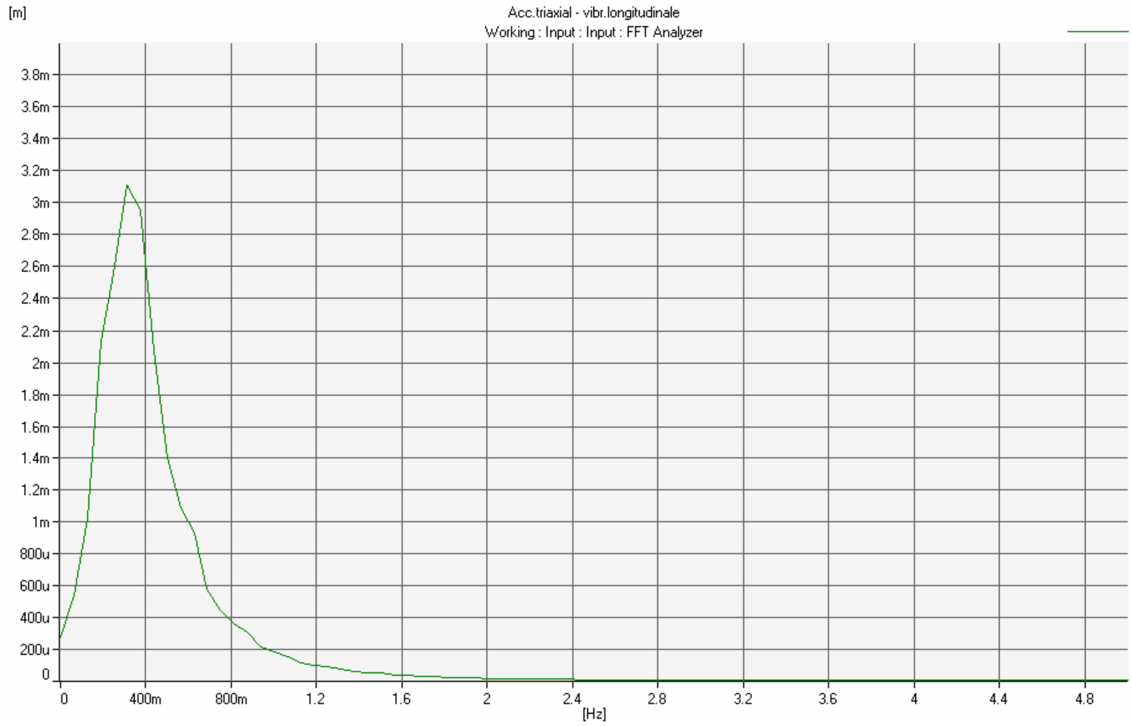

Fig. 1. Rudder compartment, longitudinal vibrations – 3,10 mm at 0,31 Hz

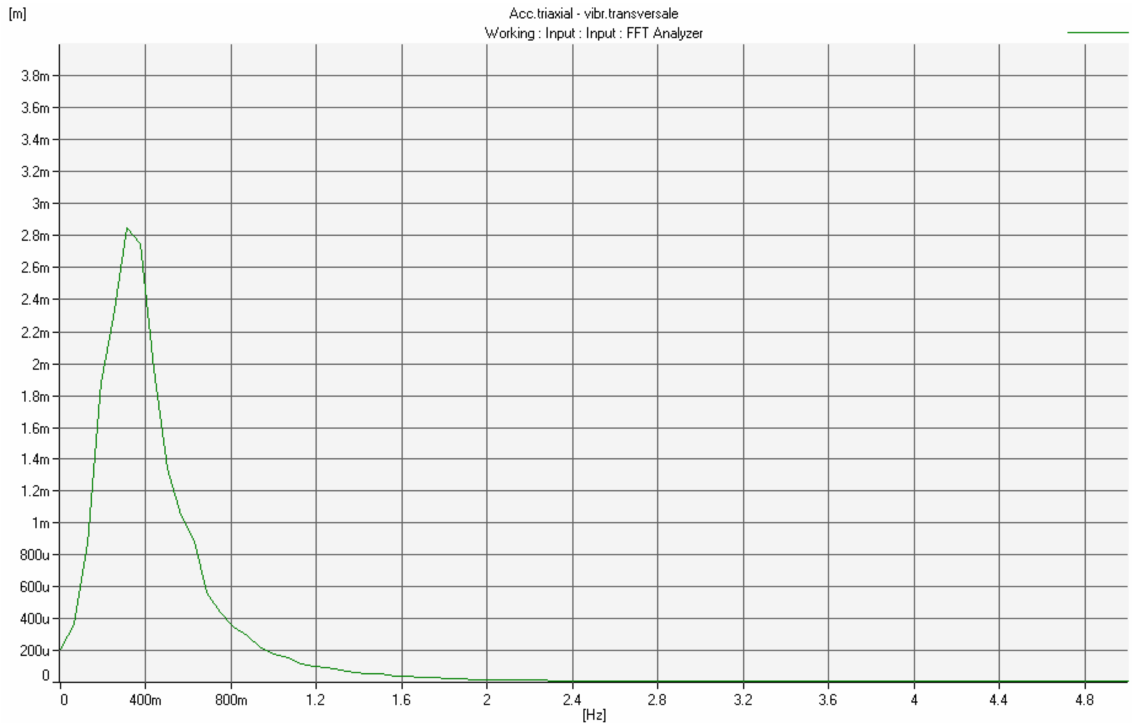

Fig. 2. Rudder compartment, transversal vibrations – 2,84 mm at 0,31 Hz

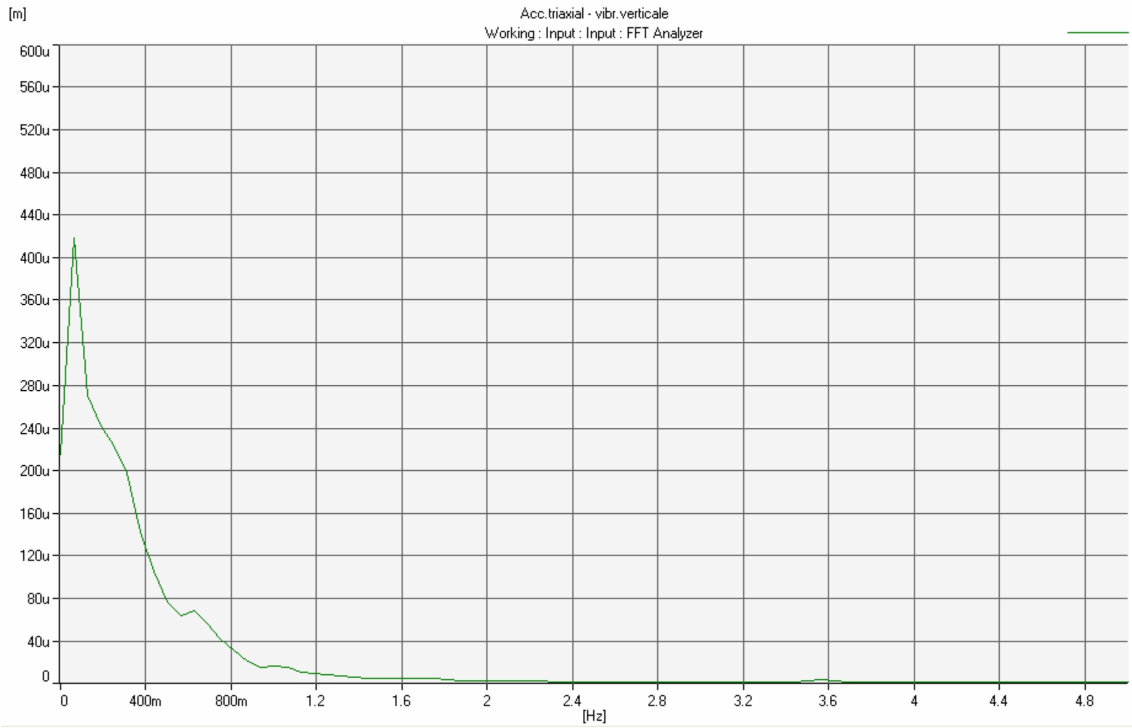

Fig. 3. Rudder compartment, vertical vibrations – 0,417 mm at 0,18 Hz

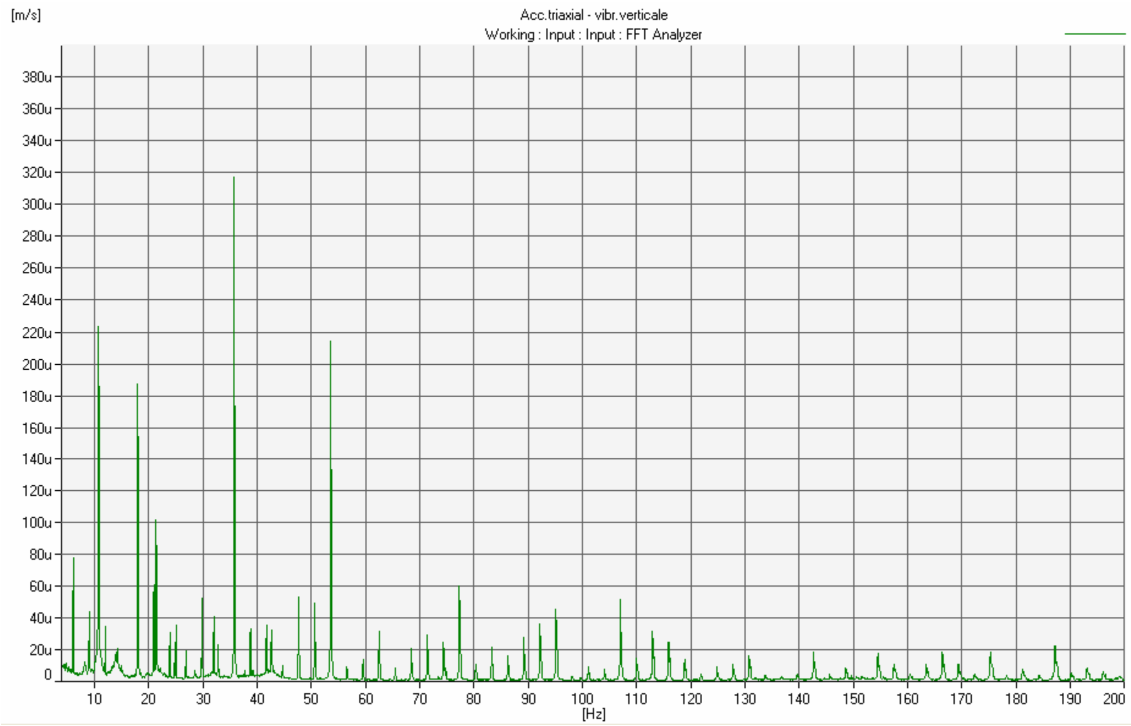

Fig. 4. Vertical vibrations with the triaxial acc. placed on the mountings of the main engine – 0,54 mm/s

Analysis of the results

The results were processed by means of FFT analysis and then through Output-Only Modal Analysis.

We chose to analyze the vibrations in the vertical dimension because the vertical vibrations are better known and more often suggested in the various literatures.

The sources of vibration onboard ships act differently:

- the propeller generates periodic vibrations
- engine and auxiliary machinery generates also periodic vibrations
- effects of the sea conduct to random vibrations

In the next table are presented the frequencies of the main engine and propeller, values obtained after the FFT analysis and according to the formulas from literature.

Table 1 – Values during ship's march on the sea

Frequency [Hz]	Source of vibration		
	Unbalanced moving parts of main engine	Unbalanced moving parts of main engine	Unbalanced moving parts of main engine
Fundamental frequency	5,875	17,625	23,5
1st harmonic	11,75	35,25	47
2nd harmonic	17,625	52,875	70,5
3rd harmonic	23,5	70,5	94
4th harmonic	29,375	88,125	
5th harmonic	35,25	105,75	
6th harmonic	41,125		
7th harmonic	47		
8th harmonic	52,875		
9th harmonic	58,75		
10th harmonic	64,625		
11th harmonic	70,5		
14th harmonic	88,125		
15th harmonic	94		
16th harmonic	99,875		

One can remark that the 3^{rd} harmonic of the main engine coincide with the fundamental frequency of propeller shaft, 7^{th} harmonic with the 1^{st} harmonic of propeller, 11^{th} harmonic with 2^{nd} harmonic of propeller.

It may be expected that if any resonance occurs at all, then these should be found at the same frequencies as the excitation frequencies coming from the engine and the propulsion system.

The values of global vibration do not exceed the values from international standards [5], [7].

The analysis of the vibrations produced in these measuring points can be completed using a relative new technique called Output-Only Modal Analysis. The advantages of identifying modal parameters by performing modal tests using the ambient excitation and measuring only the response of the structure (ambient or operational conditions), made the Output-only modal testing very popular.

The main idea of this method is that the outputs are caused by a broadband excitation of the structure (white noise for example). To obtain the modal parameters, a variety of estimators are used. The techniques involved are Frequency Domain Decomposition (FDD) and Stochastic Subspace Identification (SSI). From the singular value decomposition of the power spectral density matrix (of the output signal) evaluated for each frequency line in the spectrum, the modal parameters are extracted [8].

To estimate the Power Spectral Density (PSD) of a signal and to eliminate the errors, one should have large length of the signal. Only this fact can provide a good estimation of PSD.

There are two basic methods to estimate the power spectral density: the periodogram method and the correlation method. In our analysis, we used the second method.

The correlation method uses the auto or cross correlation between the signals. The correlation function is then Fourier transformed to obtain the corresponding auto or cross spectral densities. Averaging required estimating the PSD is performed in the time domain in this approach. The cross correlation between two discrete signals x(n) and y(n) is given by (x(n) is kept as reference) each with finite block length of N samples,

$$r(n) = \frac{1}{N-n} \sum_{k=0}^{N-(1-n)} \left(y(k)\, x^{*}(n+k) \right) \tag{3}$$

Windowing gives optimum solution to this problem at the expense of less resolution. To cope with it, cross correlation function is multiplied with window and then it is Fourier transformed to find the frequency function. So, the final expression for estimation of cross power spectral density is given by following mathematical equation:

$$S_{yx}(w) = \sum_{n=-M}^{M} \left(r(n)\, w(n)\, e^{-jwn} \right) \tag{4}$$

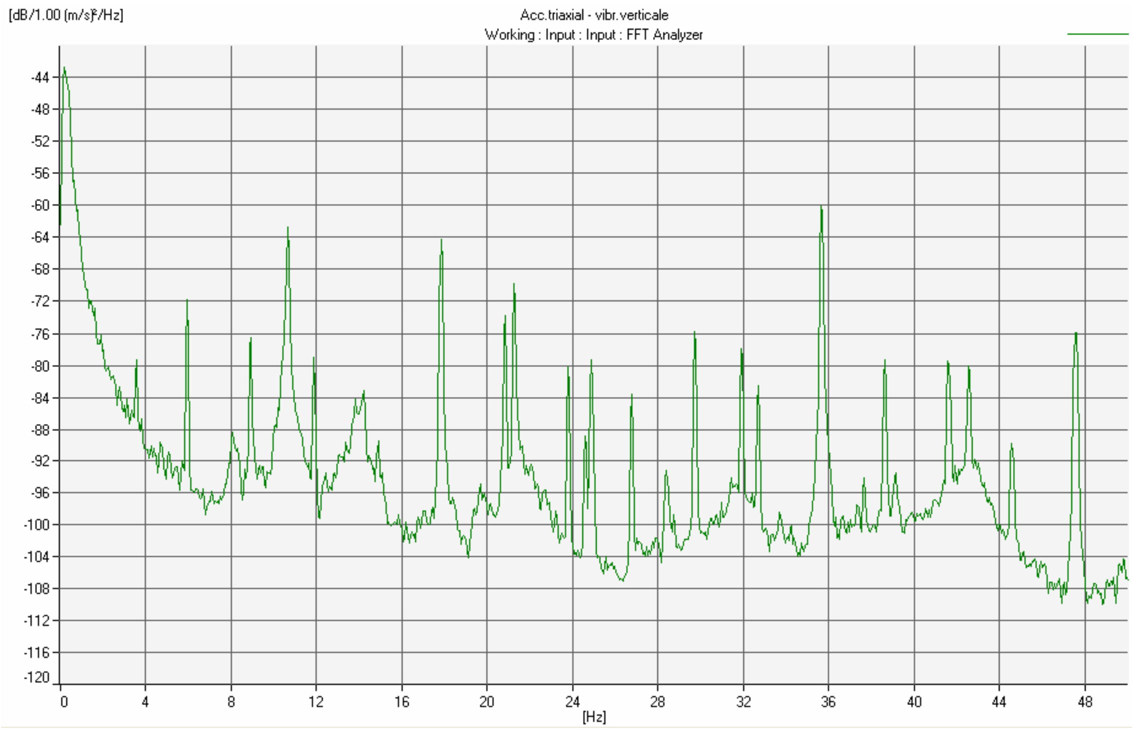

Fig. 5. PSD of vertical vibrations with the triaxial acc. placed on the mountings of the main engine

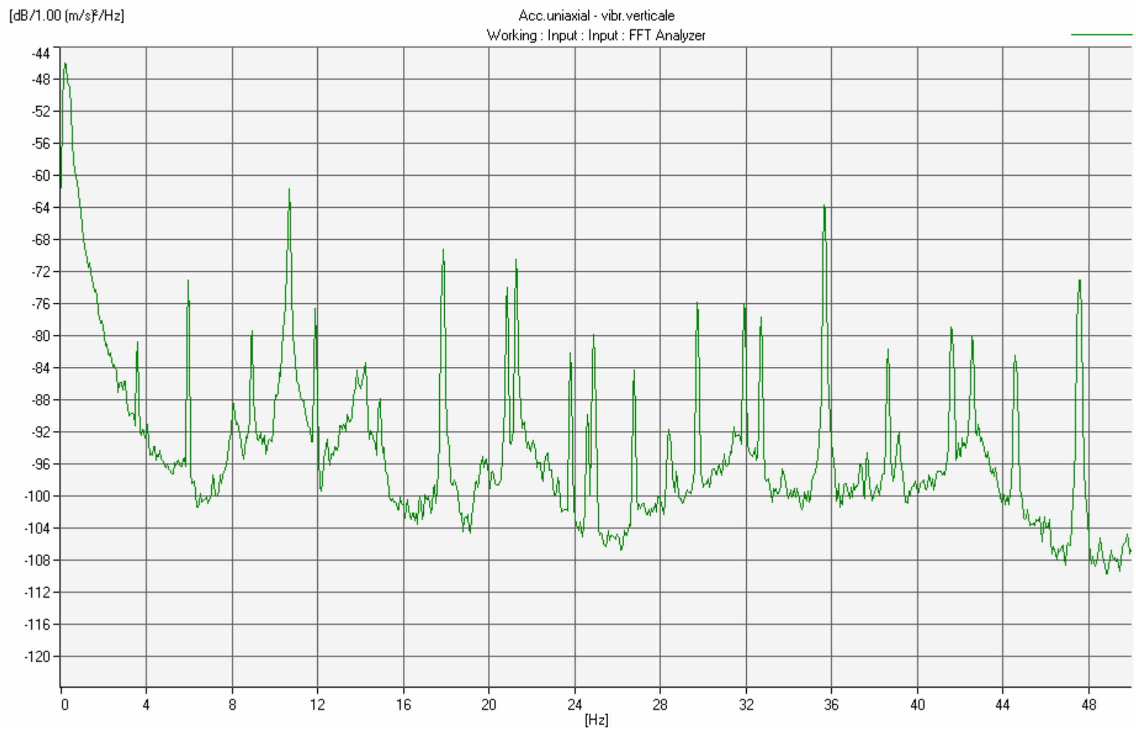

Fig. 6. PSD of vertical vibrations with the uniaxial acc. placed on the mountings of the main engine

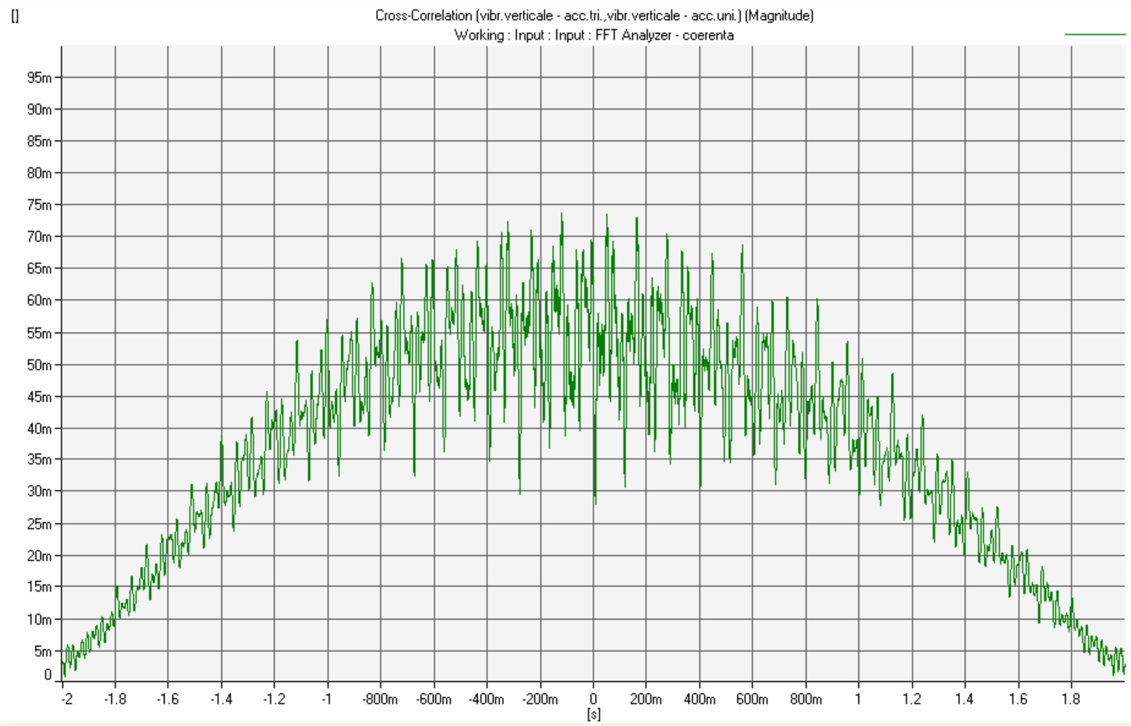

Fig. 7. Cross correlation between the signal from triaxial acc. and the signal from uniaxial acc. placed on the mountings of the main engine (vertical vibrations)

After performing these steps from the Output-only Modal Analysis, the main conclusion is that the power spectrum of the vertical vibrations for the signals in figures 5 and 6 is very much similar. There are very slight differences between the two signals (in terms of magnitude) which can be seen in figure 7.

The frequencies of 0,31 Hz (longitudinal and transversal vibrations) and 0,18 Hz (vertical vibrations) are close to the computed values of natural frequencies of ship's structure – 0,347 Hz and 0,26 Hz.

The limits specified by the American Bureau of Shipping – between 1 and 2 mm – are exceeded in the measurements in rudder compartment. But these values are specified in the frequency range of 1 – 5 Hz. Also, one must note that these limits above may not be applicable to all the local structures with different structural configurations and details. The exceed of the limit imposed by international standards can be explained by the fact that the vibrations are amplified by ship's motion.

In the low frequency range, the only vibrations are the one induced by the sea waves through ship's motion. These vibrations were expected to appear in this range.

The range from 5 to 200 Hz presents no particular peaks. In this frequency range, there were identified the frequencies corresponding to the vibrations from the main engine and propeller. But the amplitudes of these frequencies are small. Some resonances are observed, but with low amplifications.

The Output-Only Modal Analysis proves to be a very useful tool for the analysis of ship's vibrations. The effects of fatigue are especially severe in locations of high stress concentration, and fatigue cracks have sometimes proven to trigger fractures. The resistance of a material to fatigue failure depends, among other factors, on the presence of stress. In welded structures like ships, the fatigue cracks are usually found in details, i.e. hatch corner reinforcements, beam-bracket connections etc. A source of vibration creates also fatigue problems. That's why our attention to the vibrations produced by the main engine and how the vibrations are transmitted to ship's hull structure and how good is the damping. After the analysis, we can conclude that the vibrations are well absorbed and there is no influence to ship's structure.

Conclusions

The values of the vibrations measured in these trials are in the acceptable range mentioned in international regulations. Vibratory behavior of the ship can be considered as very good after these measurements.

However, each measuring point presents distinctive characteristics which must be investigated in future experiments.

In the rudder compartment, the vibrations exceed the 2 mm limit. But we must consider the specific of this compartment which is affected by the shifts of the rudder and by the direct flow of the water under ship's hull.

The range up to 10 Hz contains harmonics which affect the human organism, and from 20 Hz to 20 kHz the ear and the hearing of humans are affected. The range above 10 Hz will be analyzed in future work in order to correlate the vibrations to the noise recorded in each compartment. This will determine which component of the vibration spectra affects the hearing of ship's personnel.

Also, the Output-Only Modal Analysis must be extended to other points where vibrations can cause damages in order to obtain a global and complete status of the stresses affecting the ship.

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Analiza structurii corpului navei prin măsurarea vibrațiilor navei pe mare

Rezumat

In această lucrare sunt analizate vibrațiile corpului navei în urma marşului efectuat în Marea Neagră de către nava-şcoală Mircea. Vibrațiile corpului navei sunt întotdeauna în atenția inginerilor şi contructorilor de nave datorită influenței lor asupra rezistenței corpului navei. Apariția unor valori mari ale amplitudinilor la rezonanță conduce la uzarea prematură a structurii şi la fisuri nedorite. Măsurătorile au fost efectuate asupra motorului principal al navei şi a sistemului de propulsie. Rezultatele au fost analizate folosind analiza FFT şi analiza modală Output-Only.