Ship structure dynamic analysis - effects of made assumptions on computation results

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ABSTRACT

The paper presents identification of main errors which may occur during dynamic analysis of ship's hull and superstructure. The investigation was realized on the basis of calculations of natural and forced vibrations. The analysis was performed for cargo ships with typical propulsion system. Multi-variant computations of ship vibration were carried out with the use of previously identified excitation forces. A computation method for excited vibrations of ship hull and superstructure was developed.

Keywords : hull vibration, superstructure vibration, natural vibration, excitation forces, errors of finite element method.

INTRODUCTION

The presented method of analysis was elaborated for cargo ships (mainly medium size containerships) with the typical propulsion system, i.e. that of a constant pitch propeller directly driven by a low-speed main engine. In the paper an example analysis was performed for a 2700 TEU containership of 207 m length, 35600 t maximum deadweight and 22.7 kn speed, powered by 8 S70 MC-C engine of 24840 kW rated output at 91 rpm. The diameter of its five-blade propeller weighing 42400 kg, was 7.6 m. All drawn conclusions are valid for the containerships of the capacity from 1200 TEU to 4500 TEU. Calculations for above mentioned ships (and also for bulk cargo ships, chemical tankers, Ro-Lo ships, paper carriers etc.) have been performed in Ship Structure Division of Centrum Techniki Okrętowej S.A., Gdańsk.

Patran-Nastran software was used for vibration analysis of ship's hull and superstructure, whereas analysis of excitation forces (dynamic calculations of power transmission system) was made mainly with the use of the author's software.

For the computations two main structural models of 2700 TEU containership were used: the model of the ship in design load condition (Fig.1) and its model for ballast condition (Fig.2) [4,5]. Each of the models contained more than 42000 degrees of freedom.

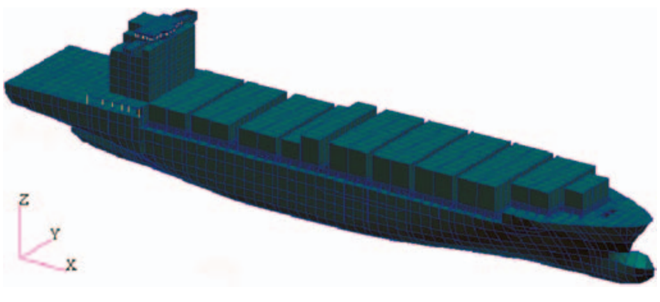

Fig. 1. Model of 2700 TEU containership in design load condition .

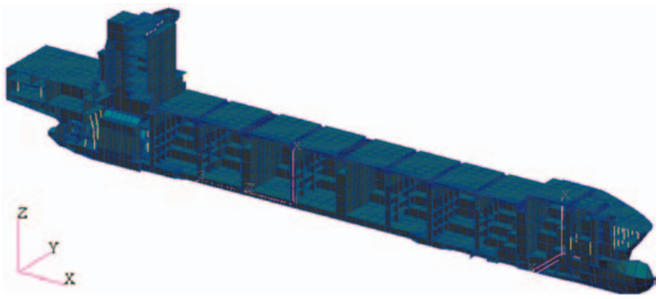

Fig. 2. Model of 2700 TEU containership in ballast load condition .

Unbalanced engine forces and pressure pulses on the hull are specified in the documentation of the installed engine and propeller. They should be applied directly to the ship hull model. Reactions of radial bearings can be calculated from bending vibration of the shaft line, whereas reactions of thrust bearing - from longitudinal vibration of the power transmission system. Vibration of the power transmission system is mainly excited by propeller-induced hydrodynamic forces. There are two possible approaches to computation of ship structure vibrations: the first one consists in accounting for the shaft line when developing the ship hull mathematical model; the other approach consists in finding the bearing reactions by means of a separate mathematical model of power transmission system. The second approach seems to be more appropriate due to possible more accurate modelling of power transmission system (including the crankshaft), particularly when exact stiffness-damping characteristics of bearing lubrication oil film (including their distribution along the bearing) are taken into account. One of the objectives of this paper is to assess whether the first approach is acceptable for computation of excited vibration of ship's hull and superstructure.

ASSUMPTIONS FOR COMPUTATIONS

The first assumption, to be verified during dynamic computation with the use of finite element method, is the way of creating the mass matrix for the mathematical model. A simplified „lumped" matrix where non-zero elements are situated only on the matrix diagonal, or the consistent („coupled") mass matrix in which the mass couplings, i.e. non-zero elements beyond the diagonal, are taken into account, may be applied. Computation results of the first main modes of natural vibrations for the containership in question are shown in Tab.1. Relative error is defined as a difference between frequencies which come from „coupled" and „lumped" calculations, divided by more precise, „coupled" frequency.

Tab. 1. *Natural vibration frequencies of the considered ship, obtained by means of various methods of mass matrix creation.*

Natural vibration modes for the containership in question	Vibration frequency [Hz]		Relative error [%]
	Type of mass matrix		
	„COUPLED"	„LUMPED"	
2-node vertical, for the hull	f1 = 1.5955	f1 = 1.5840	- 0.72
2-node horizontal, for the hull	f2 = 2.8055	f2 = 2.7977	- 0.28
3-node vertical, for the hull	f3 = 2.9253	f3 = 2.8898	- 1.21
4-node vertical, for the hull	f4 = 4.4076	f4 = 4.3026	- 2.38
Torsional, for the hull	f5 = 4.5663	f5 = 4.4524	- 2.49
Transom deck and bow bending	f8 = 5.1435	f6 = 5.1277	- 0.31
5-node vertical, for the hull	f12 = 6.0263	f7 = 5.7951	- 3.84
Longitudinal, for the superstructure	f15 = 6.5917	f8 = 6.5580	- 0.51

The computation time for the first 100 modes of free vibration with application of the „coupled" matrix was 2.5 times longer (80 minutes) than in the case of application of the „lumped" matrix (32 minutes). Moreover, in the „coupled" computation the free vibration frequencies were found only up to the value of 10.6 Hz, whereas in the „lumped" computation 18.0 Hz frequency was also attained. Thus, for the „coupled" computation much more modes of free vibration should be found and the computation time would be then even longer. Negative effects of using the „lumped" matrix for computations are the errors of computed frequencies amounting to 4% and the omitting of some modes of free vibration. However, the omitted modes are of no practical significance (they are mainly isolated vibrations of individual panels of hull plating). The errors of computed frequencies may result in shifting the resonance rotational speed of the propulsion system by no more than 3% downwards. For further computations the „lumped" mass matrices will be used.

For the forced vibration analysis the modal superposition method was applied. In this case the assumption on a number of free vibration modes accounted for should be verified. Taking into account the rotational speed (91 rpm rated value) of the power transmission system and assuming that the 5th harmonic component is the basic one for excitation forces (the number of propeller blades), we may find the maximum frequency of excitation forces equal to 8 Hz. Several variants of forced vibration computations were carried out by taking into account different numbers of free vibration modes. For the analysis the excitation forces generated by non-coupled longitudinal vibrations were used. The computation variants with the following number of free vibration modes were applied :

- ***10 modes***, up to 7 Hz frequency (all the basic modes of free vibrations for ship hull and superstructure)
- ***13 modes***, up to 8 Hz frequency (all the modes below the vibration frequency of excitation forces)
- ***20 modes***, up to 9 Hz frequency (all significant modes of free vibrations for ship hull, superstructure and main engine)
- ***43 modes***, up to 12 Hz frequency (modes with the frequency up to 1.5 times greater than the vibration frequency of excitation forces)
- ***70 modes***, up to 16 Hz frequency (modes with the frequency up to 2.0 times greater than the vibration frequency of excitation forces)
- ***150 modes***, up to 20 Hz frequency (modes with the frequency up to 2.5 times greater than the vibration frequency of excitation forces).

Fig.3. shows examples of vibration velocities at individual points of the hull, depending on the number of free vibrations taken into account. Influence of the number of natural modes on the superstructure vibration resonance curve, is shown in Fig.4. The computation results are presented for the following representative groups of the points (all the points, except the bridge wings, are located in the ship's plane of symmetry) :

MEF – main engine fore cylinder heads
MEB – main engine aft cylinder heads
BOW – hull bow on the main deck
DTR – deck transom edge on the main deck
SBF – fore bottom of the superstructure
SBB – aft bottom of the superstructure
STL – bridge wing (left)
STF – fore top of the superstructure
STB – aft top of the superstructure.

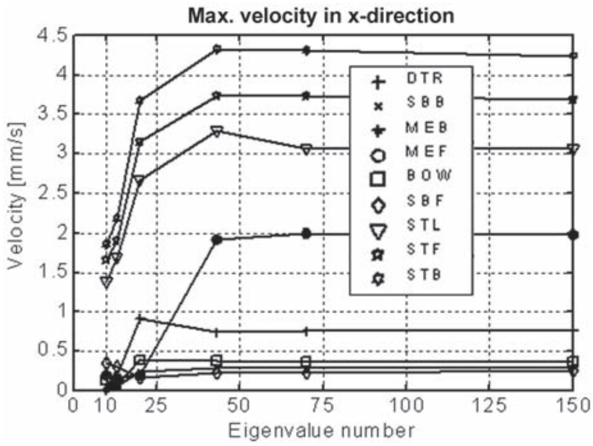

Fig. 3. *Longitudinal vibration velocities at the rated rotational speed of the engine, n = 91 rpm .*

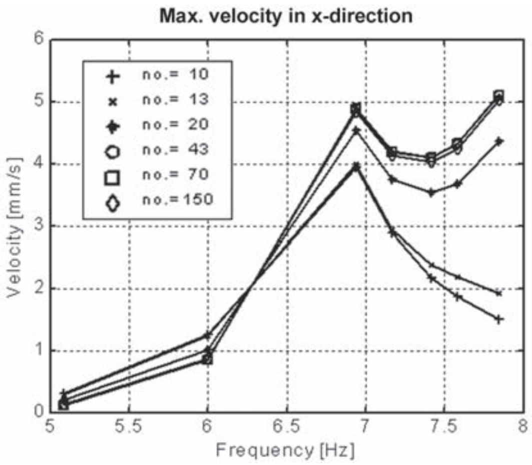

Fig. 4. *Superstructure vibration velocities (at **STB** point) in function of the excitation frequency .*

It may be said that to account for all significant modes of natural vibrations, covering the range of forced vibrations, is insufficient and may be a cause of significant computational errors. It should be noted that even the modes of forced vibration may be incorrect if an insufficient number of forced vibration modes is selected. In application of the modal superposition method the band of free vibration frequencies to be found should be at least two times wider than the excitation frequency range in order to obtain correct values of vibration amplitudes. Forced vibration phases are less vulnerable to the number of found modes of free vibrations – extension of forced vibration range by 50% is sufficient. In the considered case, 70 first modes of free vibrations (up to 16 Hz) are sufficient to be taken into account; hence only these modes are applied to further computations.

NATURAL VIBRATIONS

Fig.5 through 8 show examples of free vibration modes for the ship's hull, superstructure and propulsion system. The effects of changes in loading condition on ship vibration frequency, and the effect of added mass of water wetting the hull surface [1] are presented in Tab.2.

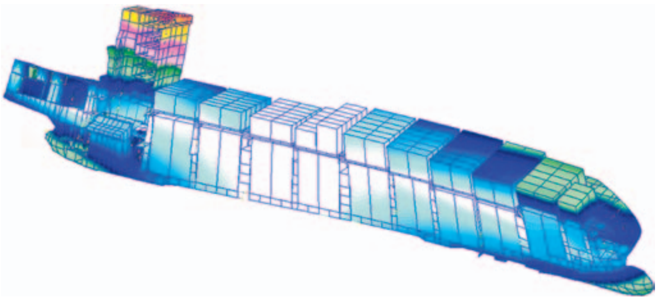

Fig. 5. *Torsional mode of free vibration for the ship's hull at design load condition without added mass of water (4.45 Hz frequency) .*

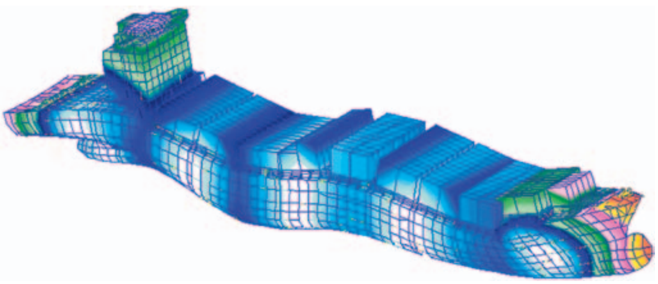

Fig. 6. *5-node mode of free vibration for the ship's hull at design load condition with added mass of water (4.70 Hz frequency) .*

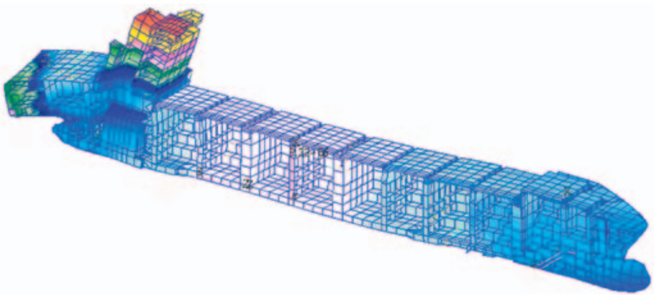

Fig. 7. *Longitudinal mode of free vibrations for the ship's superstructure at ballast condition without added mass of water (7.07 Hz frequency) .*

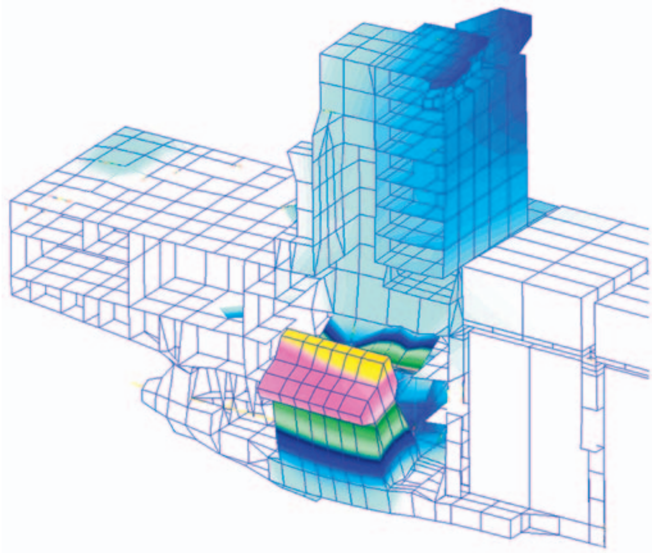

Fig. 8. *Transverse mode of free vibration for the main engine at load condition without added mass of water (6.80 Hz frequency) .*

Tab. 2. *Free vibration frequencies for the ship in question at various load conditions .*

Modes of free vibrations	Design load condition [Hz]		Ballast load without added water mass [Hz]
	without added water mass	with added water mass	
2-node vertical, for the hull	f1 = 1.584	f1 = 1.204	f1 = 1.539
2-node horizontal, for the hull	f2 = 2.798	f3 = 2.301	f2 = 2.587
3-node vertical, for the hull	f3 = 2.890	f2 = 2.255	f3 = 2.972
4-node vertical, for the hull	f4 = 4.303	f4 = 3.436	f7 = 4.496
Torsional, for the hull	f5 = 4.452		f4 = 3.334
Transom deck and bow bending	f6 = 5.128	f6 = 4.404	
5-node vertical, for the hull	f7 = 5.795	f7 = 4.700	f11 = 6.177
Longitudinal, for the superstructure mode I	f8 = 6.558	f10 = 6.315	f9 = 5.856
Longitudinal for the superstructure mode II	f9 = 6.793	f11 = 6.447	f13 = 7.071
6-node vertical, for the hull	f10 = 7.075	f9 = 5.935	f15 = 8.112
Transverse, for the superstructure	f12 = 7.411	f12 = 6.720	f14 = 7.682
Transverse, for the main engine	f20 = 9.229	f17 = 8.693	f12 = 6.795

Changes in ship loading conditions result in changing free vibration sequence. Similar phenomenon happens as the result of taking into account added mass of water wetting the hull surface. Added mass of water reduces values of frequencies. This reduction is greater for the mode of ship hull vibration; the most significant effect of added mass of water may be observed for multi-node modes. The effect of added mass of water on vibration of the superstructure and main engine is not so significant.

Removal of the containers (passing from design load condition to ballast one) does not increase the natural frequency for two first modes of hull vibrations. Increase of ship mass in the design load condition is compensated by increasing the hull rigidity due to the containers. This increase of rigidity has no effect on higher modes of ship's hull natural vibrations. The change of load condition also affects the natural frequency of the main engine. For the ship in full load conditions both the longitudinal modes of natural vibration of the superstructure are much closer to each other in comparison with the case for the ballast load conditions. Such coupling of natural frequencies might be dangerous for the ship in normal service.

EXCITED VIBRATIONS

Comparative analyses were made for the containership in question at design load condition (Fig.1) without accounting for added mass of water wetting the hull. Such assumption was made to shorten time of computation : the computation time in the case of added mass of water amounted to about 6 hours, and only to 25 minutes without accounting for it. It was assumed that the qualitative conclusions maintain unchanged for both the models.

The main problem in defining the propeller-induced pressure excitation field on the transom deck is that only a few standard pressure distribution points are provided by towing tanks. The author's intention was to check the effect of the modelling of accuracy of the pressure field over the transom deck on vibration amplitudes of ship's superstructure, main engine and hull.

Two ways of pressure field modelling were considered: exact and simplified. The pressure value averaged over the entire surface area of the transom deck was used as the simplified field of excitations. In this case only vertical excitation of transom deck and no torsion around the ship axis, appears. Actually the transom deck torsion is caused by unequal amplitudes of pressure generated at starboard and port side of the ship hull. Fig.9 shows differences of forced vibration velocities for two computation variants. The figures show magnitude of possible error in case of insufficiently exact mapping the pressure field on the ship's transom deck.

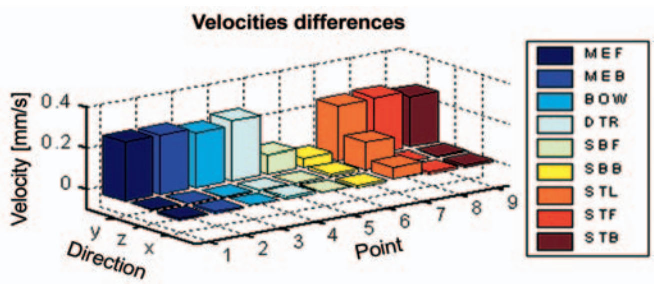

Fig. 9. *Differences of forced vibrations velocities due to exact and simplified pressure distribution over the shell plating above the propeller at its rated rotational speed .*

Application of the simplified pressure field over the transom deck does not involve serious computational errors. Certain difference may be observed only for transverse vibration velocities. The maximum reduction of transverse vibration velocities does not exceed 0.5 mm/s in this case. The differences are negligible for vibrations in longitudinal and vertical directions. Phase errors of forced vibrations are also very small, not exceeding 5°, generally. Accuracy of the pressure field on the stern transom deck is less important than finding total magnitude of excitation.

Non-coupled axial vibrations of the power transmission system generate excitations of ship hull and superstructure through thrust bearing and axial vibration damper [2]. Three kinds of computation were carried out as follows :

- ***Variant 1 (var$_s$)*** – in which exact representation of all excitation forces on the thrust bearing and axial vibration damper was applied
- ***Variant 2 (var$_d$)*** – in which excitation forces on axial vibration damper are omitted
- ***Variant 3 (var$_m$)*** – in which dynamic moment on thrust bearing is omitted.

Fig. 10 shows forced vibration velocities of the superstructure obtained from the above mentioned variants of computation.

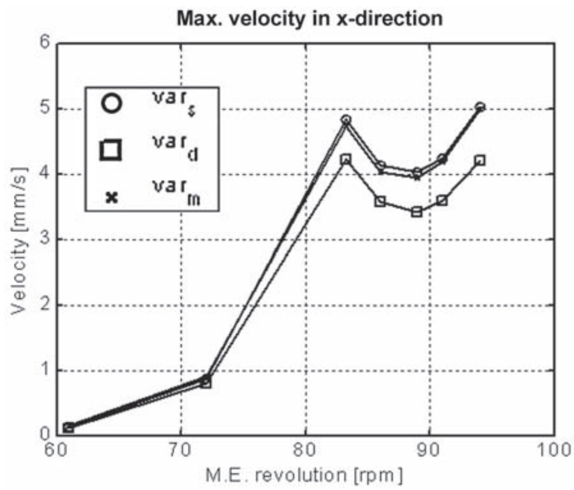

Fig. 10. *Longitudinal vibration velocities of the superstructure (top aft point) forced by uncoupled longitudinal vibrations of the power transmission system .*

The frequently used simplification in which reaction on thrust bearing is assumed to be applied in line with the crankshaft axis, has turned out to be acceptable. Omitting the reaction on longitudinal vibration damper (variant 2) causes a small but noticeable difference in ship structure vibration velocity (below 12%). In the considered case the damper reaction value amounts to ~22% of that on the thrust bearing. If the difference of amplitudes of the two reactions is greater the omission of the damper reaction may be acceptable. The effect of the above mentioned simplifications on phases of forced vibrations of ship's hull and superstructure, is negligible.

Two possible approaches to computation of ship structure vibration were used during the analysis of ship hull and superstructure vibrations forced by transverse hydrodynamic forces. The first approach consists in accounting for the shaft line when developing the ship hull mathematical model, thereafter the transverse hydrodynamic forces are directly applied to the model of ship propeller. The other approach consists in computing the bending vibration by means of a special software (computations are performed for the power transmission system model isolated from ship's hull, with the boundary conditions containing detailed stiffness-damping characteristics). The this way found reactions of radial bearings are applied to the mathematical model of the ship. The first method is faster but less accurate because of problems with proper modelling the shaft line -ship hull interconnection, lack of possibility of representing lubricating oil film and of taking into account the effect of the shaft line setting. In the typical propulsion systems the stern bearing reaction is many times greater than the reactions of the remaining bearings. For this reason it is possible to make the mathematical model of ship hull excitations simpler. Three following variants of computation were carried out :

- ***Variant 1*** – in which dynamic reactions (separate shaft line whirling vibration analysis) are applied to all radial bearings

- ❖ ***Variant 2*** – in which dynamic reaction (separate shaft line whirling vibration analysis) is applied to the stern bearing only
- ❖ ***Variant 3*** – in which hydrodynamic forces and moments are applied to the propeller (an extended model of ship hull and superstructure).

Comparison of vibration velocities of the superstructure and transom deck for all the variants of computation in function of rotational speed of the propulsion system, is shown in Fig. 11 and 12.

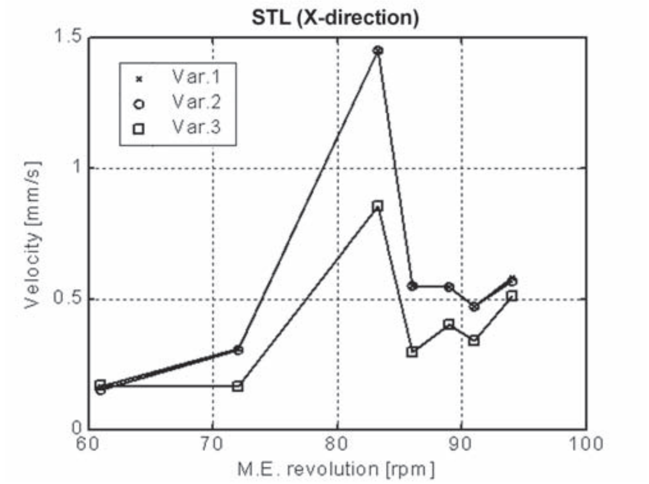

Fig. 11. *Longitudinal vibration velocities for the superstructure (bridge wing) .*

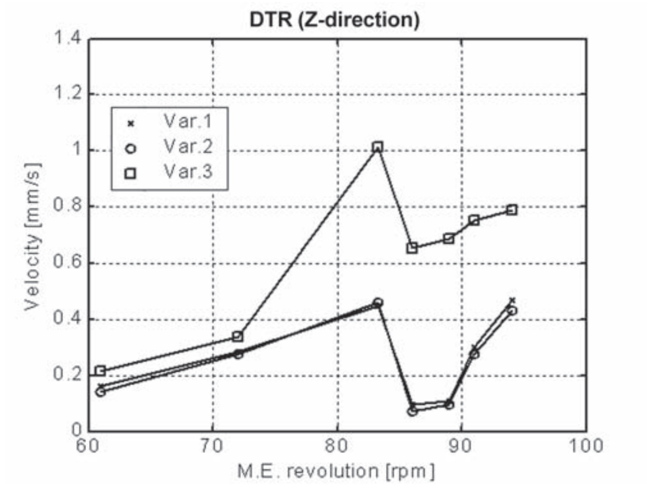

Fig. 12. *Vertical vibration velocities for the transom deck .*

From the performed analyses it may be observed that there are considerable differences between the results of computation variants 1 and 3. The differences are particularly large for the ship hull transom deck. In variant 3 the longitudinal amplitudes are lower, and the vertical ones higher, than those in variant 1. Particularly strong effect of the simplifications of variant 3 may be observed for forced vibration phases. With improperly estimated phases the result of summing the bending vibration forced amplitudes and the amplitudes forced by the remaining excitations, is doubtful. The simplifications in modelling of the power transmission system, used in variant 3, particularly in modelling the boundary conditions, seem to go too far as they cause considerable computational errors. A standard method of computation should include finding the bearing dynamic reactions by means of a special software [3], with accounting for non-linear properties of lubricating oil film and parameters of shaft line setting.

The differences between the results of computation variants 1 and 2 are not large, but noticeable for some reference points (particularly for the transom deck). Application of excitation forces to all radial bearings (variant 1) is not much more labour-consuming than accounting for only stern bearing (variant 2), hence, in the opinion of the author, full spectrum of bearing excitation forces should be used.

CONCLUSIONS

- ❍ When using modal superposition method for forced vibration analysis, the free vibration frequencies should be found within the band at least two times wider than the band of excitation frequencies, in order to make obtaining correct results of vibration amplitude computation possible. It should be noted that the forced vibration modes and resonance curves may be incorrect if an insufficient number of vibration modes is taken into account.
- ❍ Application of the simplified pressure field to the transom deck does not cause serious computational errors. Certain difference may be observed only in the case of transverse vibration velocities. Accuracy of the pressure field on the stern transom deck is less important than finding total (integrated) magnitude of excitation.
- ❍ The frequently used simplification which consists in assuming that reaction on thrust bearing is applied in line with the crankshaft axis, turned out to be acceptable. Omitting the additional moment did not cause any significant change in response of ship's hull, superstructure and even main engine. Omitting the reaction on longitudinal vibration damper causes a small but noticeable difference in ship structure vibration velocity.
- ❍ There are considerable differences between results of basic and simplified computation variants used for determining excitation forces generated by shaft line bending vibrations.
- ❍ Simplification in modelling the power transmission system (when the shaft line model is included to the global ship model), particularly in modelling the boundary conditions, may be a cause of unacceptable calculation errors.
- ❍ A standard method of computation should include finding the dynamic bearing reactions by means of a special software (for shaft line whirling vibration analysis), with taking into account non-linear properties of lubricating oil film and parameters of shaft line setting.

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