Structure and load identification using wave excitation in sea-keeping tests

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ABSTRACT

The use of segmented models with an elastic backbone in sea-keeping tests has proved in last decades to be a valid tool for the analysis of the structural response of ships in waves, provided that an elastic scaling of the relevant properties is given. The authors have used a segmented model to identify the slamming load and to evaluate the ship-beam response (whipping and springing) and they compared these results with full-scale trials. In this paper, the question that we intend to answer can be considered as an inverse problem with respect to the past investigation: is it possible to identify the structure of the segmented model and also to obtain information about the loads to which it is subjected, just from the analysis of its vibratory response? In other words, we intend to outline the possibility to use a segmented model in a *reverse* way. In the present paper, the answer of the above question regarding structure identification is based on the use of the Output–only analysis technique (which seems quite promising for this kind of investigation), whereas the load identification is based on the slender beam theory.

INTRODUCTION

Given an elastic floating body like a ship or an offshore platform, the prediction of the response due to waves is the classical (sea-keeping) problem that one usually faces. In certain cases, solving the inverse problem may be interesting too, *i.e.*, to reconstruct the wave loading or to identify the body structure by using its deformation field. Since inverse problems are generally ill-posed even if some assumptions (like system linearity) are assumed, it is usually necessary to consider special cases or to accept that only some information can be extracted. Nevertheless, there are still some pseudo-inverse problems that are worth to be considered in the naval engineering practice.

In this paper, the structure identification focuses on the *wet* vibration modes of the thin-walled beam which constitutes the elastic backbone of a segmented model tested in the INSEAN towing tank basin. The wave excitation, given by an irregular sea pattern, is utilized as an unmeasured input to excite the structure. The elastic beam deformation, sensed with accelerometers, is analyzed using a signal processing technique recently developed in structural testing analysis which exploits the ambient excitation, *i.e.*, the so-called Output-only analysis. Thus, the modal properties of the structure are identified and compared with the numerical model of the segmented hull structure. Finite element (FE) modelling of the *dry* ship structure usually achieves a satisfactory prediction of the low frequency vibration modes. When whipping or springing are considered, the closeness between the excitation band and the frequency of the *wet* bending modes of vibration plays an important role. Thus, the vibration mode computation needs also the evaluation of added mass and damping that, when relevant 3D effects due to forward motion are present, motivates some experimental validation. As a final goal, the determination of the *wet* modal properties can provide sufficient data for an updating procedure of the numerical model (mass, damping and stiffness matrices of the structure).

The second issue of this paper concerns the estimation of the hydromechanical load acting on the hull segments on the basis of the signals of the strain-gauges glued on the top face of the backbone beam. This procedure allows to evaluate the loads acting upon the segments if it is not possible to use force transducers or simply for sensor validation when one suspects any transducer malfunction.

The outlined procedure to identify the modal properties and the segment load are here applied to an elastically scaled model of the *MDV3000*, a fast monohull built by *Fincantieri*. This physical model employs the segmented model concept with an elastic backbone which scales the original longitudinal distribution of the bending stiffness. In the present work, the segmented model was towed at several speeds in head sea corresponding to real cruise conditions; both strain gauges (to sense the beam deformation) and several accelerometers (to measure accelerations in different positions) were used. The collected data were processed by using Output-only analysis, providing the low frequency, *wet* bending vibration modes in the longitudinal plane for several Froude numbers. Then, the hydromechanical load is identified by using the strain-gauge measurements and compared with that sensed directly by force-transducers.

IDENTIFICATION TECHNIQUE

The advantages of identifying modal parameters by performing modal tests using the ambient excitation and measuring only the responses of the structure, *i.e.*, ambient or operational conditions, made the Output–only modal testing very popular [1, 2] in recent years. In fact, the test procedure consists only in measuring the response of the system, resulting then an easier way for characterizing the dynamic behavior of the structure with respect to the traditional experimental modal analysis, where a measured vibration response is related to a known force excitation by means of estimators such as H_1 and H_2 [3]. Furthermore, with the output–only approach, it is possible to identify the dynamic properties of the system in real operative conditions where the loading conditions are, in general, unknown or substantially different from those simulated in the modal test. This is also the case of a segmented model, where the use of the hammer or shaker excitation presents some drawbacks.

The developed methods have been also successfully applied to structures where the excitation provided by turbulence, wind, or marine environmental conditions can be used as unknown natural forces [4]. The main idea of the Output–only modal analysis is the assumption that the outputs are caused by a broadband excitation (white noise in the ideal case) of the structure. Modal parameters are obtained by a variety of estimators, whose solution techniques could be generally divided into main groups: Frequency Domain Decomposition (*FDD*), and Stochastic Subspace Identification (*SSI*) [6]. In the *FDD* techniques, the Fast Fourier Transform (*FFT*) is considered to process the output signal. Then the modal parameters are extracted starting from the singular value decomposition of the power spectral density matrix (of the output signal) evaluated for each frequency line in the spectrum [5].

The load identification is achieved in two steps: first, the strain-gauges output is calibrated with simple beam loading (before assembling the segmented model) in order to give directly the longitudinal bending moment intensity; second, after towing tank tests, the *measured* bending moments are processed accordingly to classical beam theory to obtain the applied load. More precisely, the segment forces are obtained by calculating the beam shear force from the bending distribution and then taking the difference of the shear force values between two points sited before and after the leg connection to the beam.

MODEL DESCRIPTION

Several solutions have been proposed to represent the elasticity of monuhulls with physical models (see [8]-[9] for a review of these experimental techniques).

In the present case, the segmented model technique with an elastic backbone (rectangular, hollow and made of an aluminium alloy was built with 20 elements of constant stiffness and shear area) was adopted in order to scale the bending stiffness of the fast ferry Fincantieri *MDV3000* (for more details, refer to [10]).

Each segment is connected to the elastic beam with short legs and the gaps between adjacent segments are made water-tight by using rubber straps. The short legs were built by using steel, whereas the hull segments are of fiber-glass. The materials employed in the model construction were chosen for several technological reasons; among them, the limitation of the total weight was one of the main concerns. Thus, the model-scale was set equal to $\lambda = 1/30$, whereas the number of segments was set to six.

EXPERIMENTAL INVESTIGATION

In the present work, both stochastic sea and regular wave tests were considered: the stochastic sea tests were analyzed to identify the *wet* longitudinal bending modes of vibration by using Output–only analysis (denoted also as *FDD* technique), whereas the regular wave tests provided data to obtain the hydromechanical load both directly (force transducers) and indirectly (strain-gauge measurements).

In order to validate the Output–only analysis, dry-vibration tests were performed as well on the segmented model and the resulting vibration modes were compared with those obtained with the 'impulse excitation' technique (based on the measurement of both input and output via a instrumented hammer). This comparison, aimed to validate the Output–only analysis procedure, was not so obvious because there were multiple 3-node and 4-node vibration modes due to the strong coupling between the beam modes and the modes of the legs connecting the segments to the beam. Thus, the (pure) bending modes of the backbone beam were identified and the frequency and shape of the corresponding vibration modes (determined by using the Output–only analysis, the FRF technique and the FEM results) were successfully compared (for further details, see [11]).

The effect of the ship forward speed upon the modal parameters was investigated by evaluating the response of the segmented hull at different Froude numbers. These tests were carried out in head waves using a Jonswap spectrum characterized at full-scale by a significant wave height $H_{1/3} = 2\ m$, and a period $T_1 = 7.5\ s$. Since the acquisition time depends on the forward speed, then the acquisition settings, used for each test, will depend on the Froude number.The Frequency Domain Decomposition, performed on the power spectral density matrix of the response signals, depicted in Fig. 1, revealed a dynamic behavior of the 'wetted' structure quite similar to the one achieved in dry conditions (in terms of both the type and the number of mode shapes identified). In general, the natural mode frequencies were lower than the corresponding ones achieved considering the structure in dry condition because of the added mass effect. By evaluating the singular value decomposition of the response system, measured at increasing Froude numbers (reported in Figs. 1-3 for $Fn = 0, 0.14, 0.44$)), the estimate of the vibration modes unveiled to be more difficult with respect to the dry-tests, probably due to the 'noisy' fluid-structure interaction with excited structural nonlinearities and damping. Indeed, it seemed that, in absence of forward speed ($F_n = 0$), and for the tests with high Froude numbers ($F_n = 0.44, 0.58$), the *SDOF* hypothesis may still be valid at least for the first and third mode. On the contrary, a fully coupled system was identified at forward speeds corresponding to $F_n = 0.14, 0.29$, affecting the modal identification. The behavior of the multiple 3-node modes at different Froude numbers confirmed the intrinsic coupling of the second bending mode of the beam with the bending mode of the leg-segment structure, as already observed for the dry case. The identification is strongly characterized by unsharpened and low peaks of the corresponding singular values, with respect to the others, with a very weak dependency from the Froude numbers. These 3-node modes, that were previously identified in the frequency range $17 - 41 Hz$ for the dry structure, in the 'wet' condition were found shifted in a narrow, and lower frequency range (as a consequence of the added mass presence). The effect of the different Froude numbers on the modes with 2, and 4 nodes, are furthermore analyzed

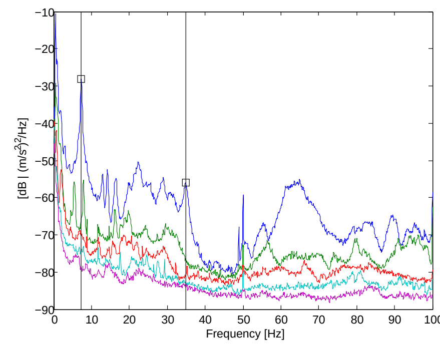

Fig. 1 Singular Value Decomposition at $F_n = 0.00$

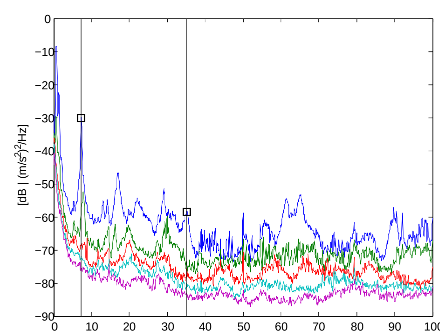

Fig. 2 Singular Value Decomposition at $F_n = 0.14$

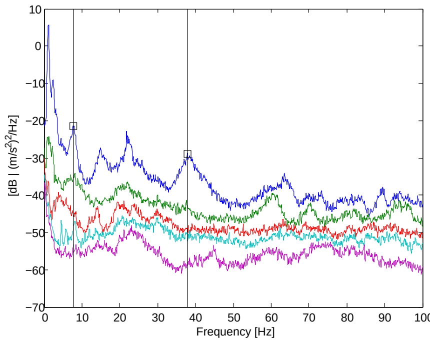

Fig. 3 Singular Value Decomposition at $F_n = 0.44$

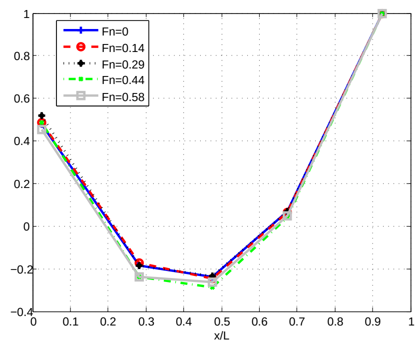

Fig. 4 Modeshape #1 sensitivity to forward speed

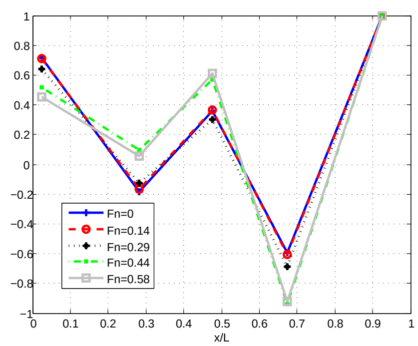

Fig. 5 Modeshape #3 sensitivity to forward speed

in Figs. 4, and 5. Again, the hydroelastic system seemed to experience a change in its dynamic behavior when the F_n crosses the value of 0.44: above this value the mode shapes presented a remarkable difference, with respect to lower Froude numbers, in both frequency $f_n\left(F_n\right)$ and amplitude, being almost constant the positions of the nodes of the eigenvectors. Also the damping $\zeta\left(F_n\right)$, associated to the first and third mode at different Froude numbers, could be estimated through the corresponding free decays by the *FDD* technique. Considering the free decays for the first mode only for $Fn = 0, 0.58$ as reference (Figs. 6 and 7), an increase in the damping characteristics with the forward speed was identified. A global view of the sensitivity of the poles of the dynamic system (*i.e.*, frequency and damping) is finally depicted in Figs. 8 and 9 for both the first and third mode. In Fig. 8 the ratio $f_n\left(F_n\right)/f_n\left(0\right)$ is depicted, where $f_n(0)$ is the frequency of the vibration mode in wet condition with zero forward speed. The damping ratio, $\zeta\left(F_n\right)/\zeta\left(0\right)$, where $\zeta\left(0\right)$ is relative to the $F_n = 0$ condition, is plotted in Fig. 9. As one can see, a remarkable shift in both the natural frequencies and the damping ratios was identified when the Froude number crosses the value of 0.44.

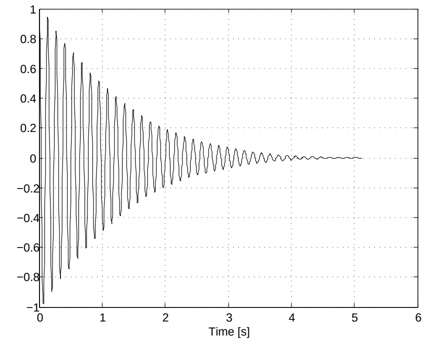

Fig. 6 Free decay at $F_n = 0.00$ for mode #1

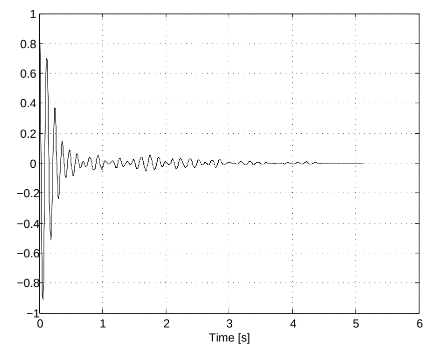

Fig. 7 Free decay at $F_n = 0.58$ for mode #1

A tentative explanation of the experimental results presented above can be based or compared only partially with theories or experiments reported in the past technical literature of which the authors have direct knowledge. Accordingly to [12], in regular waves there should be a reduction of the added mass as long as speed is increased; on the other hand, Riska [13] recorded a reduction of the vibration-mode frequencies due to an added mass increase in still water with growing speed. In the present tests, the frequency data show intermediate trends between those addressed by the cited authors. On the other hand, the damping results seem more dependent on the structure behavior than on fluid-dynamic phenomena.

Regarding the load evaluation, the results for the regular wave tests at $Fn = 0.44$ with $\lambda_{wave}/L_{pp} = 2$ are summarized in Figs. 10-12. For sake of conciseness, only data relative to the second and third segment segment ($No = 1$ segment is the bow). In Fig. 10 and 11 the time histories of the overall vertical load (hydromechanical force plus inertia force) are shown: with the continuous line the load directly measured (with the force transducer located between the beam and the segment leg) is plot, whereas the dashed line represent the load achieve by processing the beam response. It is worth to note that the displayed values are both the difference with respect to the zero initial condition (segmented model with zero-speed and calm water). In Fig. 12, the hydromechanical load for the third segment (dashed line) is obtained by subtracting the segment inertia force; then, this value is compared with the wave load (continuous line) calculated by using recorded motion and relative wave amplitude once geometry, added mass and damping data of the segment are preliminarily known (for this reason, this procedure referred in Fig. 12 as pseudo-numerical). This comparison confirms the validity of the proposed procedure, *i.e.*, the possibility to obtain experimentally the hydromechanical load acting upon the segment from the beam deformation.

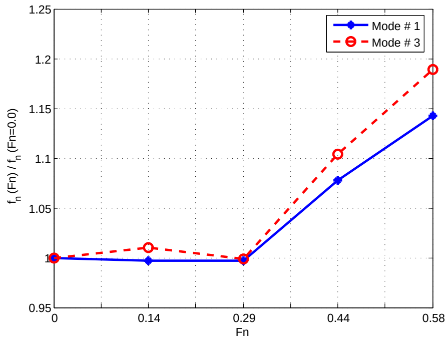

Fig. 8 Natural frequency sensitivities to forward speed

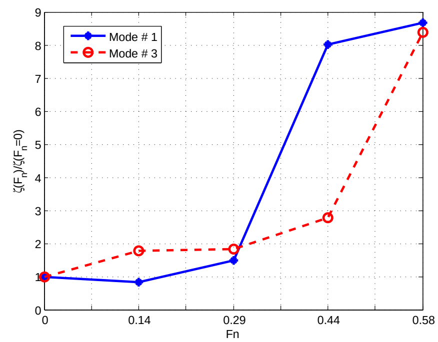

Fig. 9 Damping ratio sensitivities to forward speed

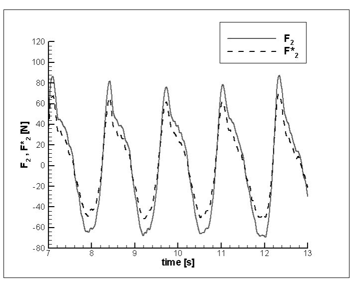

Fig. 10 Comparison between 'strain-gauge' and 'dynamometer' force on the second segment.

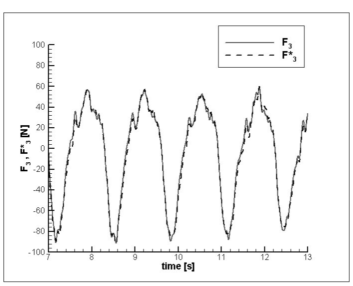

Fig. 11 Comparison between 'strain-gauge' and 'dynamometer' force on the third segment.

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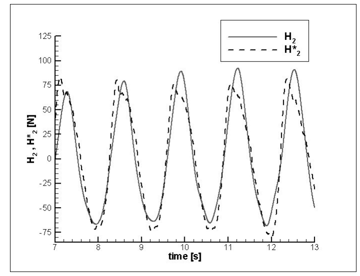

Fig. 12 Comparison between experimental and pseduo-numerical hydromechanical loads.

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