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MODAL IDENTIFICATION OF THE WET MODES OF A MARINE STRUCTURE

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Abstract

In this paper the modal experimental analysis of the bending response of a scaled ship model excited by ambient waves is presented. The physical model, designed to reproduce the global elastic behaviour of a real ship, is made with independent hull portions connected with an elastic supporting beam which the strain gauges are applied to. The identification of the 'wet' modes of the elastic beam is performed by using the Proper Orthogonal Decomposition (POD) on the strain gauges signals, whereas the source of excitation of the structures is 'naturally' provided by the waves generated in the INSEAN linear basin with a wave-maker.

1. INTRODUCTION

The need for estimators of modal parameters has spread only recently in the field of marine structures. This is principally motivated by the fact that ships were built for a long time as quite rigid structures. Nowadays, the design trend especially for innovative ship configurations or simply faster vessels is to reduce the weight in order to increase the payload and/or the cruise speed. As a consequence of this, faster speed regimes imply sea-load spectra shifted toward higher frequencies exciting flexible (with respect to the past) ship structures. The results is the onset of global vibration phenomena (known as whipping and springing) as a consequence of the above mentioned hydrodynamic loads. In this context, operational modal analysis seems to match the request for modal estimators much more than input-output techniques. Ships are generally so large structures that it is quite difficult to provide an adequate and measurable source of excitation. On the other hand, sea loads due to irregular seaways and/or violent impact phenomena of the ship hull on the water surface (slamming), provide the necessary ambient (stochastic) excitation. Indeed, the same considerations also apply to the experimental investigation upon elastic scaled models of ships carried out in towing tank basins. Sea-keeping tests with elastic models can provide an interesting point of view for the hull vibration problem in water, in order to refine coupled fluid-structure models of the ship and to validate the results obtained from numerical simulations. Within the output-only approach, the choice of a particular technique is motivated by the results that can be achieved (mode shape and frequency or even damping), quality of the measured data (number and type of the sensors), and easiness in the application of the technique

In the present paper, the results about the identification of the vertical bending modes of an elastic scaled model of a fast ferry are presented, using two output-only modal identification techniques: the first one is based on the frequency domain decomposition (FDD) and the second one exploits instead the proper orthogonal decomposition (POD). The set of measurements of the system response, separately processed with these techniques, are relative to both strain-gauges and accelerometers. The emphasis in this paper is upon the application of the POD, that demonstrated to be effective in extracting the global mode shapes of the elastic model, leading to a preliminary comparison of its characteristics with those of the FDD technique. An interesting aspect of the present application is related indeed to the fact that the added (fluid) mass varies in time, posing in case of large variations a problem of interpretation of the derived operational modes.

2. OUTPUT-ONLY ANALYSIS

The advantages of identifying the modal parameters by performing modal tests using the unmeasured ambient excitation made the output-only modal testing very popular in recent years. In fact, the test procedure consists only in measuring the response of the system, resulting then in an easier way for characterizing the dynamic behaviour of the structure with respect to the traditional experimental modal analysis, where a measured vibration response is related to a known force excitation. 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. These 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. The main idea of the output–only modal analysis is the assumption that the structure response is generated by a broadband excitation (white noise in the ideal case). Modal parameters are obtained by a variety of estimators, provided by different techniques that can be generally divided into two main groups: (*i*) Frequency Domain Decomposition (FDD), (*ii*) Proper Orthogonal Decomposition (POD) and Stochastic Subspace Identification.

A detailed presentation of both the FDD and POD methods, upon which the results presented in this paper are based, would be quite lengthy and only a short description is given here. In the FDD technique [1], supposing to have a set of displacement measurements of the system at *M* coordinates (number of measure points equal to the number of the considered modes), the *M* x *M* correlation matrix $\mathbf{R}_{\mathbf{xx}}(\tau)$ of the measured signals is first computed and Fourier transformed in order to obtain the power spectral density matrix (PSD) $\mathbf{S}_{\mathbf{xx}}(\omega)$. The singular value decomposition (SVD) of the spectral density matrix is then carried out for each node of the discretized frequency range of interest, providing plots of the singular values with respect to the frequency in order to identify the resonance peaks. When a resonance peak ω_r at a certain frequency is identified the corresponding singular vector $\boldsymbol{\varphi}_i = [\varphi_i(x_1),\ldots,\varphi_i(x_M)]^T$ satisfies the relationship $\boldsymbol{\varphi}_i \mathbf{S}_{\mathbf{xx}}(\omega_r)\boldsymbol{\varphi}_i^H$, providing an approximation of the eigenvector (vibration mode) of the original system. If a singular degree of freedom (SDOF) hypothesis holds in a neighbourhood of the mode frequency, the PSD can be transformed back to the time domain, giving a time history for the corresponding modal amplitude from which the damping can be identified (for instance, using the logarithmic decrement). The idea underlying the POD approach [2],[3],[4] exploits the same concept, *i.e.*, the analysis of the correlation between different vibrating points in order to extract the link between them given by natural modes, in a quite different way. With the same set of measurements considered for the FDD, sampled at *N* time instants, it is possible to construct the *N* x *M* response ensemble matrix as $\mathbf{U} = \left(\mathbf{\Phi}\mathbf{Q}^T\right)^T = \mathbf{Q}\mathbf{\Phi}^T$, where $\mathbf{Q}^T = [\mathbf{q}(t_1),...,\mathbf{q}(t_N)]$ is the modal coordinates ensemble matrix and $\mathbf{\Phi} = [\boldsymbol{\varphi}_1, \boldsymbol{\varphi}_2,...,\boldsymbol{\varphi}_M]$ the matrix of eigenvectors, with $\boldsymbol{\varphi}_i = [\phi_i(x_1),...,\phi_i(x_M)]^T$. It can demonstrated that the response covariance matrix, defined as $\mathbf{R} = (1/N)\ \mathbf{U}^T\mathbf{U}$, for $N \to \infty$ and multi-modal response with distinct natural frequencies, admits as eigenvectors the descretized normal modes $\boldsymbol{\varphi}_i$. The obtained eigenvectors are called proper orthogonal modes (POMs) and the corresponding eigenvalues, called proper orthogonal values (POVs), indicate the amount of energies associated to the POMs and therefore provide an indication of the modal activity in the considered response signals.

Despite the common assumption of using a stochastic source of excitation of the system, several differences between the two methods are apparent. The POD is a plain technique, requiring no user's knowledge about how the system vibrates in order to provide a set of modes orthogonal to each other; on the opposite, in the FDD the user expertise in selecting the resonant frequencies may be effective, especially when multiple mode are present. On the other hand, it is important to remark that in the POD technique the knowledge of the mass distribution may be a critical information to get better results if it is highly variable along the structure, leading in this case to solve the problem $\mathbf{R} = (1/N)\ \mathbf{U}^T\mathbf{U}\,\mathbf{M}$. Moreover, the characterization of natural frequencies and associated damping for a vibration mode is indeed straightforward for the FDD (within the assumption of the system linearity), whereas POD demonstrates to be more versatile when applied to nonlinear systems, where it can be used to produce an optimal (in the energy sense) empirical basis for system reduction.

3. EXPERIMENTAL SET-UP

The experiments [5] were carried out with a scaled model (λ = 1:30, L_{PP} = 4.28*m*, *M*=144*Kg*), of a fast ferry towed by the carriage in the INSEAN basin (220m long), as shown in Fig. 1. The towing tank is equipped with a single-flap wave-maker capable to generate regular and irregular wave patterns that provide the ambient (fluid) excitation during each run. The elastic 'backbone' model is based on the idea of dividing the hull into several (rigid) parts, namely the segments, thus implying that the water pressure distribution is transmitted to the elastic supporting beam only through the legs connecting the 6 segments to it, whereas the bending stiffness is globally reproduced by shaping the 20 constant section elements of the longitudinal beam. The gaps between adjacent segments are made water-tight by using rubber straps. As in standard sea-keeping tests, the physical model is free to heave and to pitch and, moreover, the elastic supporting beam vibrates vertically. In every run the following physical quantities were measured: *(i)* the rigid-body degrees of freedom, *(ii)* the wave height, *(iii)* the beam deformation and *(iv)* accelerations. The strain deformation of the beam is measured in 12 points by using strain gauges glued on the top face of the beam, whereas the accelerations were measured in 5 points along the beam. The horizontal position of the 6 legs, of 12 strain gages and of the balancing masses (vertical lines) can be appreciated in Fig. 2.

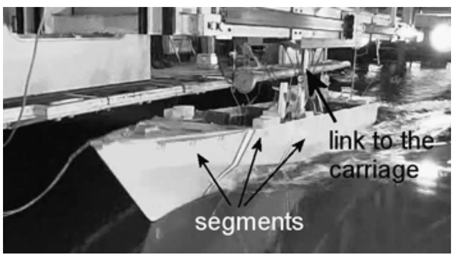

Figure 1. View of the experimental set-up.

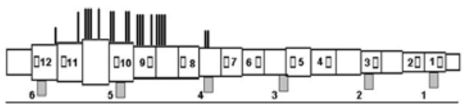

Figure 2. Sketch of the backbone beam.

4. IDENTIFICATION OF THE HULL BENDING MODES

4.1 Elastic response of the coupled systems

The novelty of the present application of the FDD and POD techniques lies in the possibility to identify the so-called wet modes of the ship, *i.e.*, the vertical bending vibration modes of the hull floating water. In this sense, the case under investigation pertains to the broader family of problems relative to the identification of natural modes of coupled fluid-structure systems. In fact, if a numerical model of the structure is usually easy to be derived from the finite element representation of the structure (upon which a modal analysis can be applied in order to extract the dry modes), a greater uncertainty about the theoretical results concerns the estimation of modes (shape, frequency and damping) when the ship is floating or even sailing. In the last case, furthermore the presence of nonlinear behaviour, exhibited in particular circumstances, makes the investigation more interesting, since the ship modal properties may be also affected by the excitation level. Therefore, in order to focus attention on the nature of the observed system, it can be useful to write the equations of the elastic hull floating in water, provided for instance by the finite element representation of the structure, *i.e.*,

$$\mathbf{M}\,\ddot{\mathbf{u}}+\mathbf{C}\,\dot{\mathbf{u}}+\mathbf{K}\,\mathbf{u}=S\left\lfloor \mathbf{M}_f(t)\,\ddot{\mathbf{u}}_r+\mathbf{C}_f(t)\,\dot{\mathbf{u}}_r+\mathbf{K}_f(t)\,\mathbf{u}_r\right\rfloor+\mathbf{f} \qquad (1)$$

where $\mathbf{u} = \{w_1,\theta_1,..., w_P,\theta_P\}^T$ is the vector of generalized nodal displacements, defined at different positions x_k along the elastic beam, $\mathbf{u}_r = \{ w_1\text{-}h_1,0,..., w_{2P+1}\text{-}h_{2P+1},0 \}^T$ is the vector of relative displacements with respect to the wave surface (corresponding in the odd components to the difference **h – u,** with **h** is the vector of local wave elevation), **M**, **C** and **K** denotes structural mass, damping and stiffness matrices, $\mathbf{M}_f$, $\mathbf{C}_f$ and $\mathbf{K}_f$ are the added mass, damping and hydrodynamic stiffness matrices, **f** represents steady force terms (e.g., hydrostatic and gravity forces). The operator *S*, applied over the hydrodynamic force expression, accounts symbolically for the fact that the hull is divided into several (precisely six) segments, that transmit the resulting (*i.e.*, integrated over the segment length) vertical force and moment to the beam through six points (positions of the legs supporting the segments). This model applies also to vibration in air (model hanging on spring) if the coefficient matrices denoted with the subscript *f* are not included. The hydrodynamic terms, due to the interaction between the body and the fluid, depend on time through the vector of relative displacement $\mathbf{u}_r$. Therefore, since the equation of motion introduced above accounts also for rigid-body motions, it yields that, switching to modal coordinates, if large amplitude motion occurs, the system becomes highly nonlinear in the heave and pitch modal coordinates (the hydrodynamic coefficients depend basically upon these variables). Nevertheless, it is worth to note that the system keeps to be linear in the elastic mode amplitudes, since the hydroelastic coupling for the bending modes is low. The variation with time of the added mass, related to the fluid inertia opposing the body motion, may affect the frequency response of the system, as shown in Fig. 3, where the plot shows the dependence of the bending response spectrum with respect to time using the wavelet analysis. This feature is highlighted by the presence of different peaks in the bending response at the cut of the wavelet surface level at 7.4 Hz and 6.5 Hz, in correspondence to the emersion and sinkage of the bow, respectively, during a regular wave test where slamming was present. Indeed, the added mass variation is much more responsible of this frequency shift than the added stiffness.

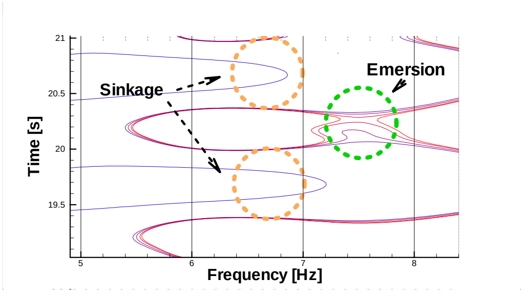

Figure 3. Wavelet analysis of the VBM response at midship.

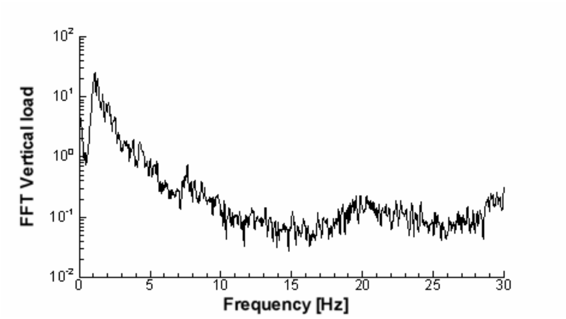

Figure 4. FFT amplitude of the measured force on segment 2 in irregular sea.

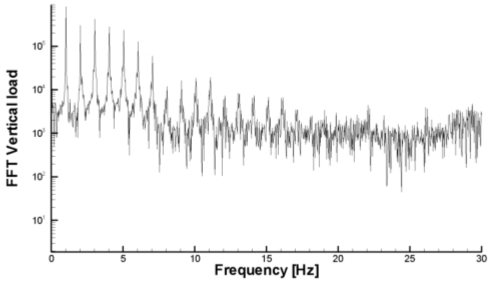

Figure 5. FFT amplitude of the measured force on segment 2 in regular waves.

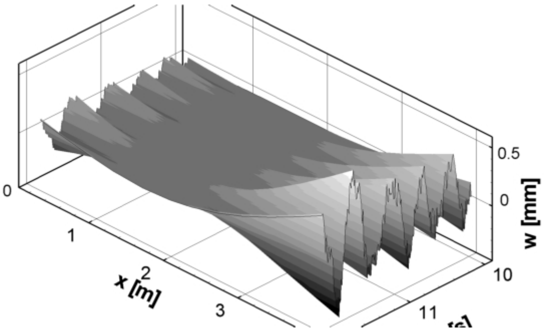

Figure 6. Elastic response of the ship model in a irregular sea test.

4.2 Remarks about the excitation

The experimental tests were performed in both regular (sinusoidal) and irregular (stochastic) head waves for different forward speeds. The irregular sea is represented as a stochastic (Gaussian) process expressed as the sum of harmonic components, with uncorrelated phases and amplitude and frequency of each obtained by the discretization of typical marine spectrum (JONSWAP spectrum in the present case). The random wave excitation under consideration has a frequency range centred on the corresponding significant wave period ($T = 1.37s$ at model scale) and significant wave height $H_{1/3}$=0.066 *m*. The tests were repeated for different forward speeds (0, 0.733, 1.466, 2.82, 3.76 *m/s* at model-scale, corresponding to 0, 10, 20, 30, 40 *kn* at full-scale). It is important to remark that the spectrum of the wave excitation is limited and, in any case, has a frequency content that does not cover even the frequency range of the lowest bending modes. This fact appears even more critical for the regular wave tests, because in this case the wave should be periodic and the main hypothesis upon which output-only methods are based – presence of a stochastic, broad band excitation – seems to fail. However, in both cases, the measured wave loads are capable to excite the entire structure (a logarithmic scale is used in *y*-axis). The explanation of this behaviour is probably due to several physical mechanisms that extend the excitation bands of the hydrodynamic loads, as shown in Fig. 4 and 5. First, if large amplitude motion is present, the nonlinear dependence of the load on the wave elevation is emphasised, thus enlarging the incoming wave spectrum through high-order harmonics, as it can be appreciated especially in Fig. 5. Second, if impact phenomena between the hull bottom and the water surface occur, the time scale of the impact load is very short, exhibiting a band shifted toward higher frequencies (beside the fact that large amplitude motion is surely present to have slamming).

In this case, a wavelet analysis of the spectrum of the measured vertical force acting for instance on the first segment shows the behaviour described above.

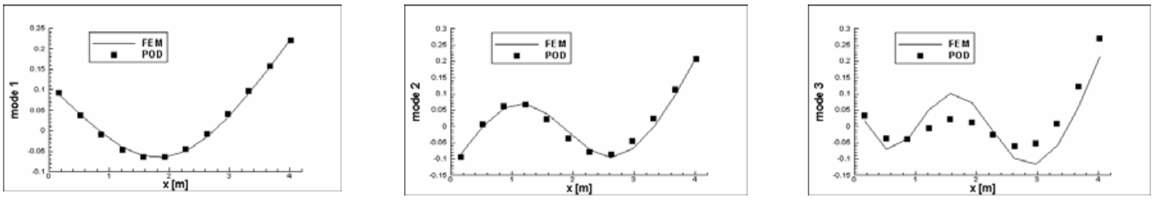

Figure 7. Comparison between FEM and experimental POD modes.

4.3 Mode identification

Whereas data available in terms of accelerations are ready to be processed, if the strain gauge signals are examined they need some pre-processing in order to transform the bending moment distribution into elastic displacement (see Fig. 6) by performing a double spatial integration along the beam (at least the knowledge of the bending stiffness $EI(x)$ distribution is needed, assuming that shear effect are negligible for the slender beam). After that, in order to provide elastic mode coordinates referred directly to the body coordinate system at rest, it is useful to add the linearized rigid-body displacement field; in this way, the intrinsic property of the POMs to be orthogonal to each other determines the nodal line to be coincident to the body *x*-axis. Regarding the overall mass distribution (structural and hydrodynamic masses), needed by the POD, a condensation of the longitudinal mass distribution at the strain-gauge coordinates was properly derived. (a more detail discussion of this aspect can be found in [6]). This hypothesis holds satisfactorily for the irregular wave tests, due to the low amplitude of the rigid model motions.

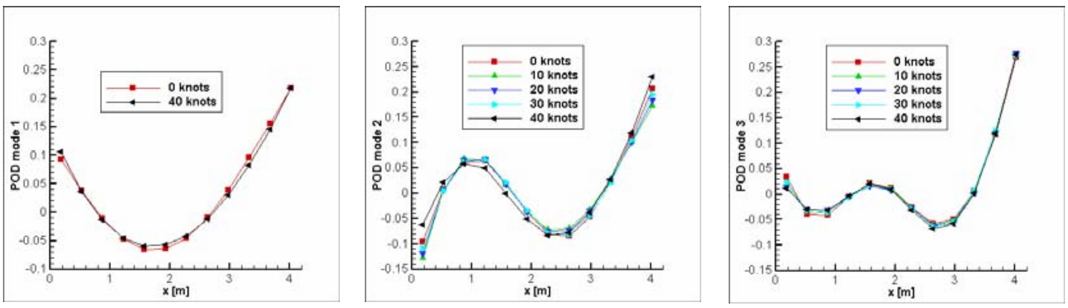

Figure 8. Forward speed dependence of POD modes (irregular sea tests).

In Fig. 7 the comparisons between the experimental POD modes with the FE modes are plotted. The POD modes were identified by analyzing the signals recorded for 20 seconds during a test at zero forward speed and using a sampling frequency of 200 *Hz*. The MAC parameters obtained by these comparisons are: MAC_1=0.997, MAC_2=0.973 and MAC_3=0.707. A satisfactory correspondence is obtained for the first and second POD modes, whereas the global third POD mode appears significantly different from the numerical one. The reasons of this discrepancy could be numerous: corresponding between numerical and experimental model, nonlinear effect due to the fluid-structure coupling, presence of local mode contributes in the response signals, uncertainty on the added mass and therefore on nodal masses values. In Fig. 8 the dependence on the forward speed of the POD modes is presented (for the first POD mode the only two cases of 0 and 40 *kn* are plotted for sake of clarity). This dependence is more evident for the second and third POD modes with respect to the first. The same test condition can be examined also with the five accelerometer signals processed with the FDD technique. Beside the poor spatial resolution, two main point emerge: the dependence of the observed modes on the forward speed (agreeing with those obtained with POD) and the difficulty to detect the 3-node mode (see [7] for more details). Nevertheless, an interesting point is connected to the possibility to identify the modal properties (frequency and damping) associated to the vibration modes, and their dependence on the speed shown in Figs. 9 and 10.

Table 1. Natural frequency and damping of the identified vibration modes.

	'dry' test		'wet' test	
Mode Type	f_n[Hz]	ζ (%)	f_n[Hz]	ζ(%)
2-nodes	10.76	0.40	7.37	1.19
3-nodes	38.67	NA	NA	NA
4-nodes	57.78	1.40	34.76	1.41

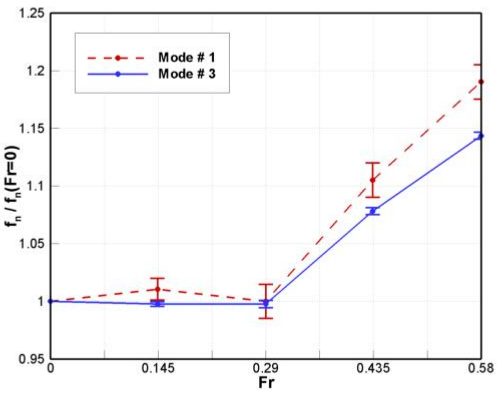

Figure 9. 1st and 3rd frequency mode variation with $V^{(ship)}$.

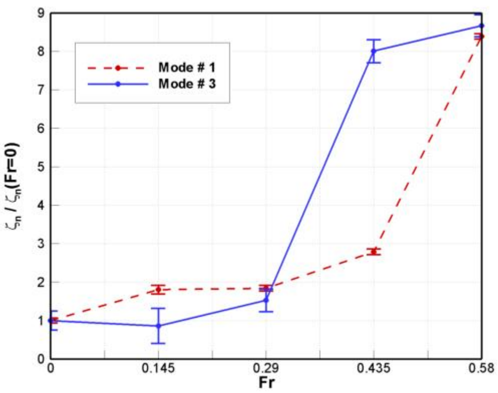

Figure 10. 1st and 3rd damping mode variation with $V^{(ship)}$.

Another interesting aspect connected to the regular wave analysis is the possibility to capture the time dependence of the mode shapes due to the variation of the system coefficients caused by large amplitude motion. Regular wave tests demonstrated to be particularly suitable for this kind of analysis because short time periods can be clearly identified where the variation of the hydrodynamics coefficients is evident and not so large. The analysis was performed in the following conditions: 2.82 m/s of forward speed (30 knots at full-scale), 0.06 m of wave height (3.6 m at full-scale) and non-dimensional wave length $\lambda/L_{PP} = 1.75$; in these test conditions the wave encounter frequency is f_e=0.83 *Hz* (wave encounter period T_e=1.2*s*). Time intervals with a period of 0.3 s in the analyzed signals were chosen in correspondence of the emersion and maximum sinkage of the bow. The sampling frequency in this case was 500 Hz and therefore 150 samples per signal were processed in every time interval. Although the number of samples is limited, the produced results appear reliable and repeatable whatever cycle is analyzed. The analysis of the POMs shape variation with respect to the hull immersion was limited to the first POD mode which appeared to be slightly affected by the projection of the response covariance matrix over the constant mass matrix. The results of this analysis is presented in Fig. 11: the black line is relative to irregular sea test at the same forward speed of 2.82 *m/s* and can be considered as an average condition, the red and green lines are relative respectively to the minimum and maximum relative displacement of the bow (maximum and minimum bow immersion). Each one of these two last curves was averaged on 10 cycles.

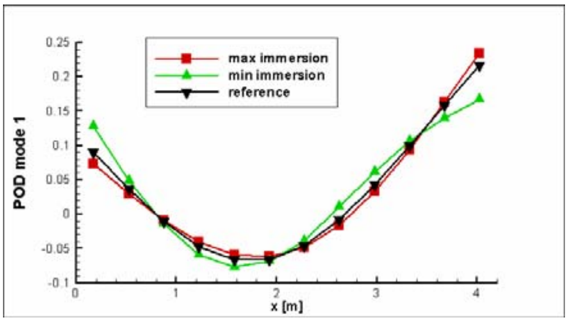

Figure 11. Immersion dependence of 1st POD mode (regular sea test).

4. CONCLUSIONS

In the present paper, the coupled fluid-structure response of the elastic hull was analysed in order to extract the vertical bending modes of the supporting beam. Since the response was excited by the wave loading during towing tank tests, two output-only (POD and FDD) techniques were used and their application was discussed with respect to the frequency content of the excitation and with respect to their capabilities to observe the vibration modes. Since different sets of measurements (strain gauges and accelerometers) were separately considered, some questions still remain to be examined in more detail in order to improve their comparative evaluation.

ACKNOWLEDGEMENTS

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