Some aspects of hydroelastic issues in the design of ultra large container ships

Malenica Š.[1], Senjanović I.[2], Tomašević S.[2] & Stumpf E.[1]

(1) BUREAU VERITAS - DR, Paris, France (sime.malenica@bureauveritas.com)
(2) Faculty of Mechanical Engineering and Naval Architecture, Zagreb, Croatia

Introduction

Recent trends in increasing the size of container vessels (Figure 1), raise the new hydro-structural issues in ship design both from extreme loading point of view and fatigue point of view. Indeed, due to their extreme size (almost 400 meters in legnth) these ships become much "softer" which means that their hull natural frequencies will be significantly reduced. At the same time ship speed remains relatively high (around 25 knots) so that the risk of hydroelastic resonance between the waves and the structure (springing) is present. In addition, due to the very large bow flare of these ships, the importance of slamming induced vibrations (whipping) is also increased. Finally, local hydroelastic interactions of the bow structure with extreme slamming pressures, also becomes an important issue. When we know that even rigid body seakeeping analysis of the ship sailing in waves is still a challenge from modelling point of view, we can easilly imagine the complexity of the numerical models which need to be put in practice in order to properly evaluate these hydroelastic effects. The present paper assumes that the seakeeping part is known and concentrates on hydroelastic aspects of ship-wave interactions only. It is a continuation of the paper presented at the last Workshop [4].

Global hydro-elastic model of container ship

Unlike the other ship types, the container vessels are more vulnerable to the oblique wave conditions which are usually more complex to model numerically. This is due to their open midship section (Figure 1) which makes the structure much more sensitive to the torsional type of loading. This fact complicates the use of the nonuniform beam model for structural part, because the reduction from 3D to 1D finite element model becomes much more complex due to the warping phenomena and coupling beween the torsional and horizontal modes. However, the advantage of the beam model being its simplicity and low CPU time requirements, it can still be used for pre-design purposes as explained in [4].

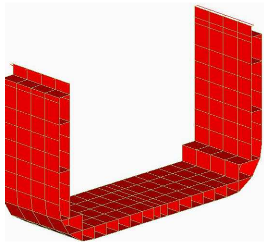

Figure 1: *Ultra large container ship and its typical midship section.*

Linear springing model

Regardless of the structural model (3D or 1D), the most common models for global hydroelastic simulations are based on the so called modal approach. Within this approach the total ship displacement is represented as a sum of different modal displacements:

$$\boldsymbol{H}(x,y,z,t) = \sum_{i=1}^{N} \xi_i(t)\boldsymbol{h}^i(x,y,z) = \sum_{i=1}^{N} \xi_i(t)[h_x^i(x,y,z)\boldsymbol{i} + h_y^i(x,y,z)\boldsymbol{j} + h_z^i(x,y,z)\boldsymbol{k}] \tag{1}$$

where the vector functions $\boldsymbol{h}^i$ denote the modal functions (usually dry structural modes) and ξ_i their amplitudes. This decomposition implies the definition of supplementary radiation potentials with the following body boundary condition:

$$\frac{\partial \varphi_{Rj}}{\partial n} = \boldsymbol{h}^j \boldsymbol{n} \tag{2}$$

After solving the different boundary value problems for the potentials [3], the corresponding forces are calculated and the following modal motion equation written:

$$\left\{-\omega^2([\ \mathbf{m}\ ] + [\ \mathbf{A}\ ]) - i\omega([\ \mathbf{B}\ ] + [\ \mathbf{b}\ ]) + ([\ \mathbf{k}\ ] + [\ \mathbf{C}\ ])\right\}\{\boldsymbol{\xi}\} = \{\boldsymbol{F}^{DI}\} \tag{3}$$

where $[\ \mathbf{m}\ ]$ is the structural mass, $[\ \mathbf{b}\ ]$ is the structural damping, $[\ \mathbf{k}\ ]$ is the structural stiffness, $[\ \mathbf{A}\ ]$ is the hydrodynamic added mass, $[\ \mathbf{B}\ ]$ is the hydrodynamic damping, $[\ \mathbf{C}\ ]$ is the hydrostatic restoring, $\{\xi\}$ are the modal amplitudes and $\{\boldsymbol{F}^{DI}\}$ is the modal hydrodynamic excitation. The solution of this equation gives the modal amplitudes and the (linear) springing problem is formally solved. There are numerous technical difficulties in building up the above equation and herebelow we briefly discuss some of them.

The equation (2) implies the transfer of the structural modal displacements from structural mesh onto hydrodynamic mesh. In the case of simplified beam model [4] this transfer is rather straightforward but in the case of 3D FEM model it involves non-trivial interpolation procedure from one mesh to another. One example of the modal transfer is shown in Figure 2.

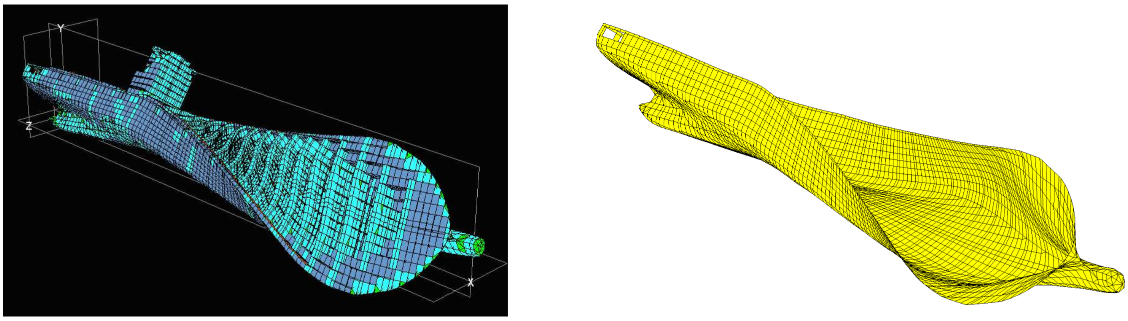

Figure 2: *Structural mode shape and its transfer onto hydrodynamic mesh.*

Validation of the model for coupled horizontal and torsional vibrations

It is clear that the validation of the above described hydroelastic model will be very difficultdue to the fact that the ship motions are usually dominated by the rigid body part so that the ship distortions are small and difficult to measure. A dedicated series of experiments on extremely flexible barge were conducted in ESIM [7] and, in Figure 3 we present few results of comparisons. As we can see the agreement is relatively good and the main tendencies are captured. These results were obtained using the simplified beam model presented in [4], which was built with the goal of obtaining relatively simple tool for predesign hydroelastic analyses when 3D FEM model is not ready.

In order to illustrate the efficiency of this kind of models we present the dry natural frequencies of the container ship from Figure 2.

Mode No.	1(T)	2(VB)	3(T+HB)
1D Beam model	0.859	1.2	1.47
3D FEM	0.861	1.47	1.50

Table 1. *Comparisons of the ship hull natural frequencies (in [Hz]) obtained by 1D and 3D FEM models.*

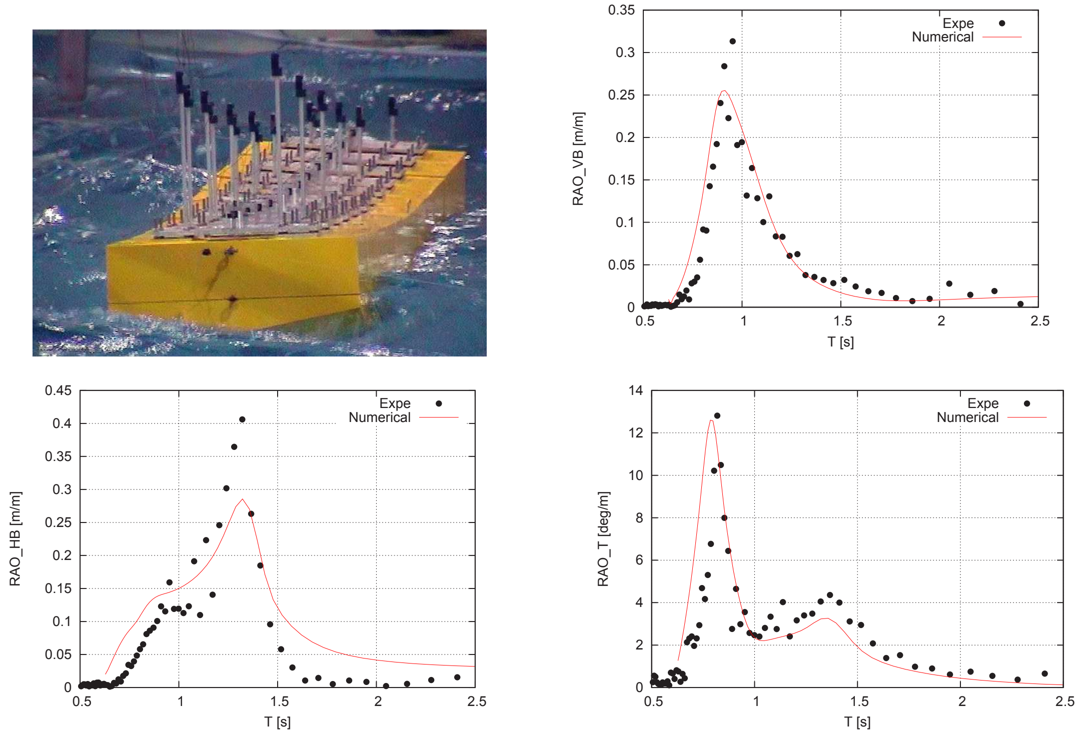

Figure 3: *Experiments on the flexible barge and comparisons of results for the wave incidence of 60 degres (VB - vertical bending, HB - horizontal bending, T - torsion).*

Time domain, slamming and whipping

The whipping is usually defined as the transient hydroelastic ship response induced by slamming. Due to the impulsive character of the loading, simualtions need to be performed in time domain. The method that we use is based on the transfer of the linear frequency domain data into the time domain, using the inverse Fourier transform technique [2], after what the nonlinear impulsive loading is simply added on the right hand side as an additional excitation force $\{\boldsymbol{Q}(t)\}$.

$$([\ \mathbf{m}\ ]+[\ \mathbf{A}^{\infty}\ ])\{\ddot{\boldsymbol{\xi}}(t)\}+([\ \mathbf{k}\ ]+[\ \mathbf{C}\ ])\{\boldsymbol{\xi}(t)\}+[\ \mathbf{b}\ ]\dot{\boldsymbol{\xi}}(t)+\int_0^t [\ \mathbf{K}(t-\tau)\ ]\{\dot{\boldsymbol{\xi}}(\tau)\}d\tau=\{\boldsymbol{F}(t)\}+\{\boldsymbol{Q}(t)\} \quad (4)$$

However, the main problem remains the determination of the slamming forces as they require the solution of highly nonlinear hydrodynamic problem which still remains unsolved for the general 3D case. This fact limits the possibilities of validation to the examples with simple (known) impulsive excitations (hammer tests) or to the so called extinction tests in which an initial elastic deformation is imposed to the model. These extinction tests were also performed within the experimental campaign [7] and some validation results were presented in [3]. Additional comparisons are ongoing and will be presented at the Workshop. Now we briefly present some results concerning the calculation of slamming loads which are the main stumbling block in the analysis of whipping. Indeed, it is fair to say that only 2D slamming methods were sufficiently validated up to now. The correct modelling of the 3D slamming is extremely complex, mainly due to the fact that the wetted body surface is not known in advance but is the part of solution. That is why the so called 3D Generalized von-Karman Model (GvKM) becomes interresting. Indeed, within this model the wetted part of the body is assumed to be the intersection of the body with the undistrubed free surface (there is no wetting correction) which allows for application of the efficient 3D panel methods which can solve the associated von-Karman boundary value problems. One example of the results for a typical ship is shown in Figure 4 where we can see that the numerical results agree reasonably well with experiments in spite of the relatively weak assumptions. However, it will be dangerous to apply this method in every case, and more validation results are necessary.

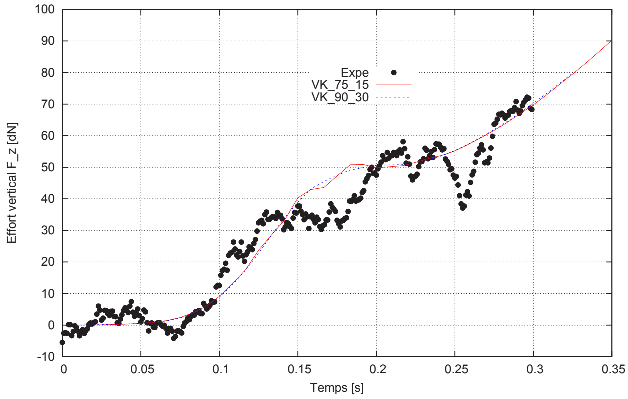

Figure 4: *Slamming force for typical ship bow obtained by 3D Generalized von-Karman method.*

Conclusions

In this abstract, we discussed some aspects of the modelling of the hydroelastic ship responses such as springing and whipping. It is clear that the problem is still open and no fully satisfactory model exists up to now. This is mainly due to the inexistence of the efficient solutions for seakeeping with forward speed and for slamming. However, we have shown that some progress can be made and relatively efficient procedures can be put in practice in order to have an idea of the order of magnitude of hydroelastic effects. We believe that these methods, combined with the experiments, can be used in ship design with reasonable confidence. There is a long way to go before obtaining the fully satisfactory model of springing and whipping.

References

[1] Bishop, R.E.D. & Price, W.G., 1979. : "Hydroelasticity of ships", *Cambridge University Press.*

[2] Cummins W.E., 1962. : "The impulse response function and ship motions.", Schiffstecknik

[3] Malenica Š., Molin B., Remy F. & Senjanović I., 2003. : "Hydroelastic response of a barge to impulsive and non impulsive wave loads.", 3rd. Int. Conf. on Hydroelasticity, Oxford, UK.

[4] Malenica Š., Senjanović I., & Tomašević S. 2006. : "An efficient hydroelastic model for wave induced coupled torsional and horizontal ship vibrations.", 21st IWWWFB, Loughborough, UK.

[5] Newman, J.N., 1994. : "Wave effects on deformable bodies", Applied Ocean Research, Vol. 16, pp.47-59.

[6] Remy F., Molin B. & Ledoux A. 2006. : "Experimental and numerical study of the wave response of a flexible barge.", 4th Int. Conf. on Hydroelasticity, Wuxi, China.

[7] Remy F. & Molin B., 2002. : "Essais GIS-Hydro sur une maquette de barge elastique. Analyse des resultats de mesure.", Technical report ESIM.

[8] Senjanović I. & Grubišić R., 1991. : "Coupled horizontal and torsional vibration of a ship hull with large hatch openings.", Computer and Structures, Vol. 41, No. 2, pp. 213-226.