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Some aspects of whipping response of container ships

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Introduction

Whipping is usually defined as a transient hydroelastic ship structural response due to impulsive loading such as slamming, green water, underwater explosion, etc. Here we concentrate on the slamming induced whipping. Slamming induced whipping is observed both in experiments and in full scale measurements for any kind of ships as far as they encounter heavy seas in which the slamming type of loading is likely to occur. One example of the typical whipping event is shown in figure 1 (taken from [1]). This figure represents the time evolution of the vertical bending moment, following severe slamming event, at the midship of the relatively small ($L_{pp} = 124m$) general cargo/container vessel. As we can see, the whipping contribution to the overall vertical bending moment is not only very important but it also last for a relatively long time due to the low structural damping. One slam event increases multiple extremes in the bending moment which makes the whipping phenomena to be relevant both for extreme and fatigue loading of the ship structure.

However, up to the authors knowledge, the hydroelastic whipping effects are not properly accounted for in the ship design. This is mainly due to the difficulties in the correct modelling of whipping and the needed calculation time. Indeed, in order to calculate the whipping response, one should combine several aspects (seakeeping in large waves at forward speed, slamming, hydroelasticity, etc.) which are difficult to model even independently. In spite of all these difficulties, there is a lot of research on whipping going on nowadays, especially in the context of the ultra large container ships. There are several reasons why the whipping is likely to be more important for the large container ships: high ship speed (close to 30 knots) and large bow flare induce higher slamming loads, large ship size reduce the natural frequencies. In this paper we present the recent numerical developments related to whipping and we apply them to two container ships different in size ($L_{pp} \approx 260m$ and $L_{pp} \approx 360m$). Both extreme loading and fatigue is discussed.

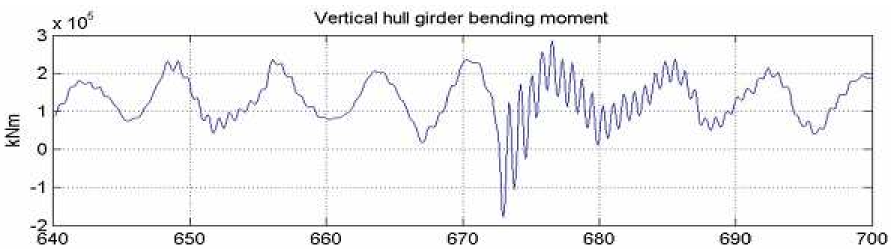

Figure 1: Typical whipping event.

Numerical model

Numerical model that we use is based on the coupling in between the 3D diffraction radiation seakeeping code for hydrodynamic part and the Timoshenko beam model for structural part. The so called modal approach is adopted which means that the ship motions and deformations are represented in a series of 6 rigid body modes and several (5 to 10) dry structural modes. The basics of the theory are presented in [5] and [8] and here below we present just the final dynamic modal equation:

$$([\,\mathbf{m}\,]+[\,\mathbf{A}^{\infty}\,])\{\ddot{\boldsymbol{\xi}}(t)\}+[\,\mathbf{b}\,]\{\dot{\boldsymbol{\xi}}(t)\}+([\,\mathbf{k}\,]+[\,\mathbf{C}\,])\{\boldsymbol{\xi}(t)\}+\int_0^t[\,\mathbf{K}(t-\tau)\,]\{\dot{\boldsymbol{\xi}}(\tau)\}d\tau=\{\boldsymbol{F}(t)\}+\{\boldsymbol{Q}(t)\} \quad (1)$$

where $\boldsymbol{\xi}$ is the vector of modal amplitudes, $[\ \mathbf{m}\ ]$ is the modal mass matrix, $[\ \mathbf{A}^{\infty}\ ]$ is the infinite frequency added mass matrix, $[\ \mathbf{b}\ ]$ is the damping matrix, $[\ \mathbf{k}\ ]$ is the structural stiffness matrix, $[\ \mathbf{C}\ ]$ is the hydrostatic restoring matrix, $[\ \mathbf{K}(t)\ ]$ is the matrix of the hydrodynamic memory functions, $\boldsymbol{F}$ is the vector of the non-impulsive wave loading (linear and nonlinear) and $\boldsymbol{Q}$ is the vector of impulsive loads. The dimension of all the above matrices is $(6 + N_f) \times (6 + N_f)$ where N_f represents the number of dry structural modes. This equation is integrated in time using the 4th order Runge Kutta method.

It is important to note that the linear hydrodynamic coefficients in the above equation are derived from the frequency domain calculations, using the well known method proposed in [3, 7]. The nonlinear part of the loading includes non-impulsive and impulsive parts. The non impulsive part represents the so called Froude Krylov loads which basically corrects the hydrostatics and pressure distribution around the waterline while the impulsive part represents the slamming loads. The slamming part is probably the most difficult part to evaluate numerically. A very robust method is needed and the calculation of the slamming force should not take to much CPU time to be able to preform long time whipping simulations. The only fast and reliable methods for evaluation of the slamming forces which are available today are based on the so called 2D strip approach. The bow is modeled by multiple 2D sections and the slamming force is evaluated at these sections. In this paper we use two different methods for evaluation of the slamming loads at each 2D section: the Generalized Wagner Model (GWM) [9] and the Modified Logvinovich Model (MLM) [4]. The advantage of the second method is much lower CPU time, but the disadvantages is the lower domain of validity. We do not discuss these methods in details here and we refer to [6].
The above described procedure was integrated into a single numerical tool able to perform the long time whipping simulations for any prescribed irregular sea state. These simulations allows for determination of the probability of exceedence of the maxima and for determination of fatigue damage.

Numerical results

As already mentioned, two container ships of different size were chosen for illustration of the overall procedure and for demonstration of the importance of whipping in container ship design. The first ship (S1) has the length between perpendiculars of $360m$ and the second one (S2) $260m$. Only the case of head waves and zero forward speed is investigated which means that the vertical modes only will participate. The first few structural natural frequencies in vertical plane, corresponding natural mode shapes for

Mode	$S1$	$S2$
1	0.432	0.600
2	0.905	1.225
3	1.445	1.950
4	2.008	2.738
5	2.606	3.631

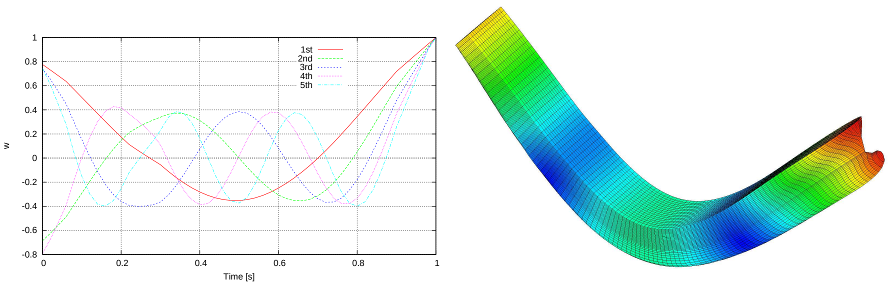

Figure 2: Structural natural frequencies and mode shapes.

S1 and transfer of the first mode onto hydrodynamic mesh, are shown in Figure 2. Only five natural modes are used in the calculations because the tests showed that the higher modes do not participate significantly, and because the mode with the highest frequency determines the stable time step.

The midship bending moment is calculated using different approaches to investigate the effect of whipping. The same wave trains are used for the different calculations for the same sea condition to make comparison more valid. The fist approach is using linear theory only with a rigid ship. The contribution of the non-linear Froude Krylov forces is investigated using a rigid model with the non-linear seakeeping code without slamming. The last approach is a flexible ship using non-linear seakeeping code and applying the slamming loads. For every sea condition six halve hour runs using 250 wave components are glued to obtain enough statistical information.

Example of typical output is presented in Figure 3 where the zoom on the typical whipping event is also shown. As we can see from this figure, the slamming usually occurs in sagging conditions but it last long enough to influence the maximum hogging moment too. These signals are rich of informations and can be used for determination of the maximum expected values as well as for the determination of the fatigue damage by using the rain-flow counting method.

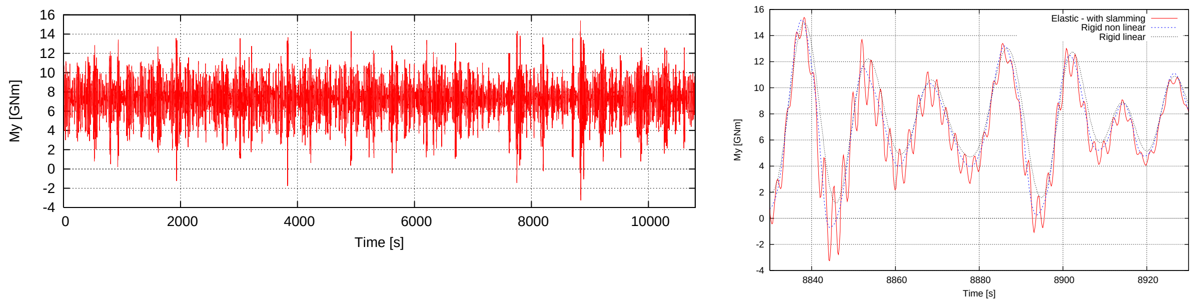

Figure 3: Typical time history of the vertical bending moment and zoom around a whipping event.

The probability of exceedence of the midship bending moment is shown in figure 4 for the different approaches. Even if the relatively mild sea states were chosen, ($H_s = 9m$, $T_z = 11s$ for S1, and $H_s = 10m$, $T_z = 13.95s$ for S2), the influence of whipping is clearly visible. The non linear Froude Krylov forces increase the sagging moment significantly and the whipping response does increase it even more. Figure 5 shows the cycle count of the bending moment using the rainflow method and the calculated fatigue damage of the deck. The whipping increase the fatigue damage significantly by increasing the amplitude of the large cycles.

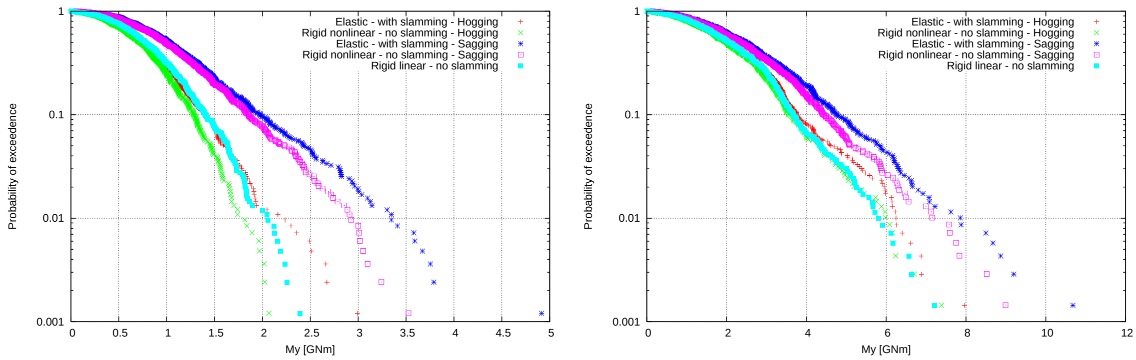

Figure 4: Probability of exceedence of the midship bending moment (S1-left, S2-right).

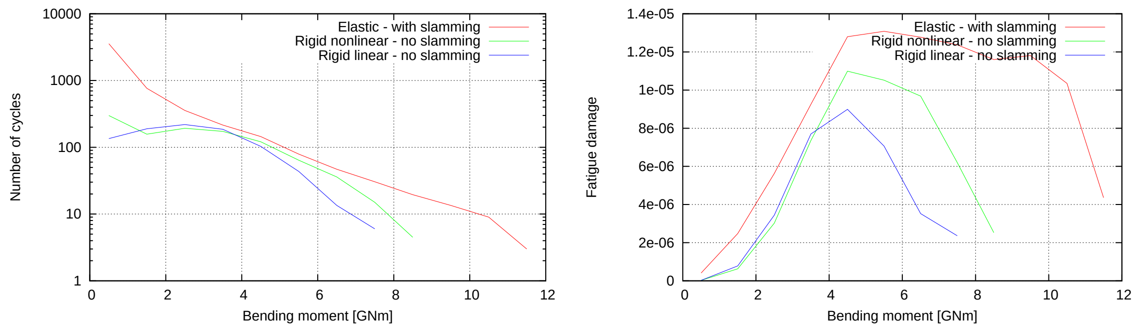

Figure 5: Rainflow count of bending moment and fatigue damage of the deck for S1.

To obtain design values many sea states have to be evaluated. The sagging moment with a probability of 10^{-5} calculated using a Weibull extrapolation and the fatigue damage per hour are shown in figure 6 for a limited number of sea states. In this case the slamming and whipping occurs only in the very severe sea states. When a non zero velocity would be used, the number of cells where whipping occurs will increase.

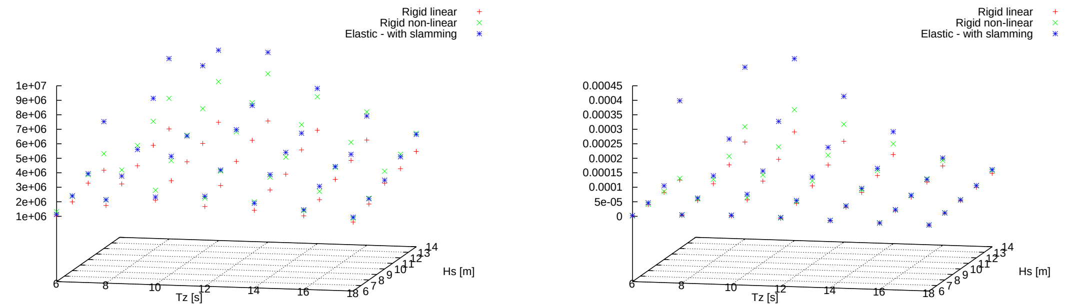

Figure 6: The 10^{-5} probability sagging moment (left) and the fatigue damage per hour (right).

Conclusions

We presented here the numerical method which can be used in ship design for determination of the influence of whipping on wave loadings and ship structural responses. The method was demonstrated on two container ships of different size and in both cases the influence of whipping was found to be important not only for the maximum values but also for fatigue. It should be mentioned that the examples which were chosen are just demonstrative ones, and the general methodology should include more complex set of calculations namely the model should include the effects of forward speed, heading, different loading conditions (full, ballast, etc.) and different sea states. At the same time the sensitivity to some other parameters such as damping, direction of 2D slamming strips, aft slamming, l ength of runs, number of wave components, etc. should be properly investigated. Maybe the most critical point in the analysis is the determination of slamming loads which are extremely difficult to evaluate. In this work we used two different methods GWM and MLM and even if there are some differences in the evaluation of the local forces the final results in terms of maximum values and fatigue seem to be in good agreement. This is important because the MLM method requires much less CPU time. It should also be kept in mind that both methods are limited to the 2D calculations and some 3D correction coefficients need to be employed. How this should be done is not clear yet.

References

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