# Whipping Response of a Box Barge in Oblique Seas

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## Introduction

The hydroelastic response of ships moving in waves has been studied for many years to understand and predict the structural failure of a vessel. Accurate prediction of stress is important for failure assessment, both from fatigue and extreme loading. Fatigue loading will cause cracks to form in high stress regions, especially in the deckhouse of naval combatants. Extreme loading can cause a vessel to break apart in heavy seas. This is a complicated problem that requires the combined analysis of the ship structure with the stochastic air-sea environment over the lifetime of a ship.

The goal of the present work is to numerically predict the non-linear response of a vessel in a seaway subject to loading from the wind-wave environment. In particular, the current work is focused on the challenges that arise in large amplitude waves where both global wave bending loads and local impacts are important. The method used in the current work combines a non-linear computational fluid dynamics solver with a modal structural dynamics solver. The two domains are combined in a tightly-coupled time-accurate algorithm. The method is described in Piro and Maki [2] and extended in this work to include rigid body motion in waves.

The motivation for using CFD for the fluid domain is the ability to implicitly capture non-linear free surface topology, specifically, breaking waves that occur during slamming. Furthermore, this method models viscosity and turbulence. Generally viscous damping is small, but can be important, such as in roll.

In this abstract, the method is validated using previous numerical and experimental work by Senjanović et al. [4]. Ongoing work is focused on using the method to analyze whipping response in large amplitude waves.

## Methodology

The starting point of the present work is Piro and Maki [2]. The experimental model has a shear center far from the center of gravity to model a real containership. The finite element model for this case is comprised of Euler beam elements representing the backbone of the experimental model. The difference in position of the center of mass and shear center is achieved by adding stiff bar elements that lower the center of gravity. Finally the hull is represented by transfer shell elements that have no mass or stiffness. The beam and shell elements can be seen in Figure 1.

The displacements and rotations of the beam are linearly applied to shell elements as shown below:

$$\left\{ \begin{array}{c} d_x(x,y,z,t) \\ d_y(x,y,z,t) \\ d_z(x,y,z,t) \end{array} \right\} = \left\{ \begin{array}{c} -\frac{dv(x,t)}{dx}(y-y_b) - \frac{dw(x,t)}{dx}(z-z_b) \\ v(x,t) - \phi(x,t)(z-z_b) \\ w(x,t) + \phi(x,t)(y-y_b) \end{array} \right\}, \tag{1}$$

where $d_x$, $d_y$, and $d_z$ are the shell displacements in the $x$, $y$, and $z$ directions, $v(x,t)$ is the horizontal displacement of the beam, $w(x,t)$ is the vertical displacement of the beam, $\phi(x,t)$ is the torsional rotation of the beam, and $y_b$ and $z_b$ are the $y$ and $z$ locations of the beam. The $x$ axis is along the vessel length and the $z$ axis is defined positive up, opposite the direction of gravity.

The rigid body modes and elastic structure modes are solved separately. The rigid body motion is solved using symplectic integration, which conserves energy and is well suited for long-time integration. The discrete equations for translational motion are:

$$\vec{v}^n = \vec{v}^o + \Delta t \frac{\vec{F}^o}{m}, \tag{2}$$

$$\vec{x}^n = \vec{x}^o + \Delta t \vec{v}^n, \tag{3}$$

where $o$ denotes a value at the old time step, $n$ denotes a value at the new time step, $\vec{x}$ is the position, $\vec{v}$ is the velocity, $\vec{F}$ is the external force, and $m$ is the mass. Underrelaxation and iteration are used to stabilize the equations of motion for vessel. This is necessary when the vessel has relatively large added mass.

An approximate body boundary condition is used on the structural boundary of the fluid domain. The fluid grid moves with the rigid body motion, and the velocity boundary condition contains both the rigid body and structural components. This process is similar to that described in [2]. Using this method, the fluid forces will not contain the hydrostatic stiffness force. Therefore, translational and rotational springs are added to the finite element model to represent the hydrostatic contribution. This procedure avoids the non-robust and expensive mesh deformation process.

Waves are generated using the `waves2Foam` toolkit developed for OpenFOAM by Jacobsen et al. [1]. The toolkit provides inlet wave boundary conditions for the velocity and volume-fraction variable. Also included are "relaxation zones" that help build the waves in the domain upstream of the vessel. These zones are also used to damp the waves downstream.

## Results

In this work the hydroelastic response of a flexible box-barge in oblique seas (heading angle $\beta = 60°$) is studied. The barge is similar to the one studied by Š. Malenica et al. [5], Remy et al. [3], and Senjanović et al. [4], with length $L = 2.445$ m, beam $B = 0.6$ m, draft $T = 0.12$ m, and depth $D = 0.25$ m. Two structures are used, one for the RAO validation study, and one for the slamming study. The elasticity for the RAO comparison is provided by a 1 cm x 1 cm bar located 0.187 m above the waterline. For the slamming study, the size of the bar is increased to 1.8 cm square to be stiffer and more realistically represent a scaled vessel.

**Validation of RAOs** To properly generate the RAOs of rigid and elastic responses of the vessel, the correct wave amplitude must be known. Therefore, simulations are preformed where the wave field is propagated through the domain without the body. A probe is placed where the center of the body will be to generate a time series of wave elevation. The amplitude from this time series is used in the RAO calculation. A focus of ongoing work is to improve the generation, propagation, and damping of these waves.

Response amplitude operators of heave, pitch, roll, vertical bending, horizontal bending, and torsion are compared to the results presented in [4]. Preliminary results to show the capability of the method are shown with four wave periods, 0.8, 1.0, 1.2, and 1.4 s. Figure 2 shows the comparison between the present method and the previous results. The rigid body mode RAOs from the present method compare well with the previous experimental data, as do the RAOs for the bending modes. However, the torsion RAO shows larger differences with the previously published experimental and numerical results. This discrepancy is studied below, and will continue to be the subject of further analysis.

There are several possible explanations for the differences between the current torsion RAO and that of the previously published results. The first and most likely reason is the difference between the dry natural frequencies in the coupled horizontal-bending and torsion modes between the current work and that of Senjanović et al. [4]. The first five frequencies are 4.00, 4.84, 7.28, 9.51, and 12.35 rad/s in the current work while the previously reported values are 5.32, 7.92, 12.7, 15.56, and 21.56 rad/s. These lower frequencies are closer to the forcing frequency, which should yield a larger response. Different modeling strategies are being examined to correct the natural frequencies. Another reason for discrepancy could be the flow between pontoons that is not modeled in the current simulations. This flow should add damping to the system, which would lower responses.

**Non-linear effects and whipping** Ongoing work is focused on understanding the effects of non-linearities and whipping response in large-amplitude waves. Presently simulations are being conducted for a range of wave amplitudes for a single wave length of 1.561 m ($T = 1.0$ s). The analysis will investigate the interaction between the wave bending and slamming responses.

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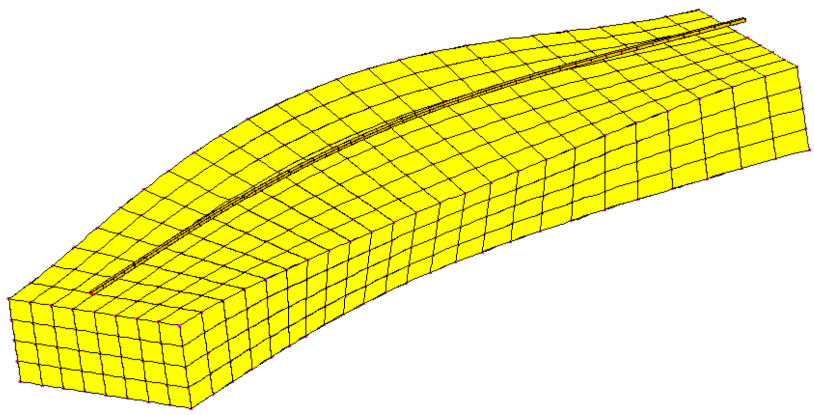

Figure 1: Structural model deformed with first horizontal bending-torsion mode.

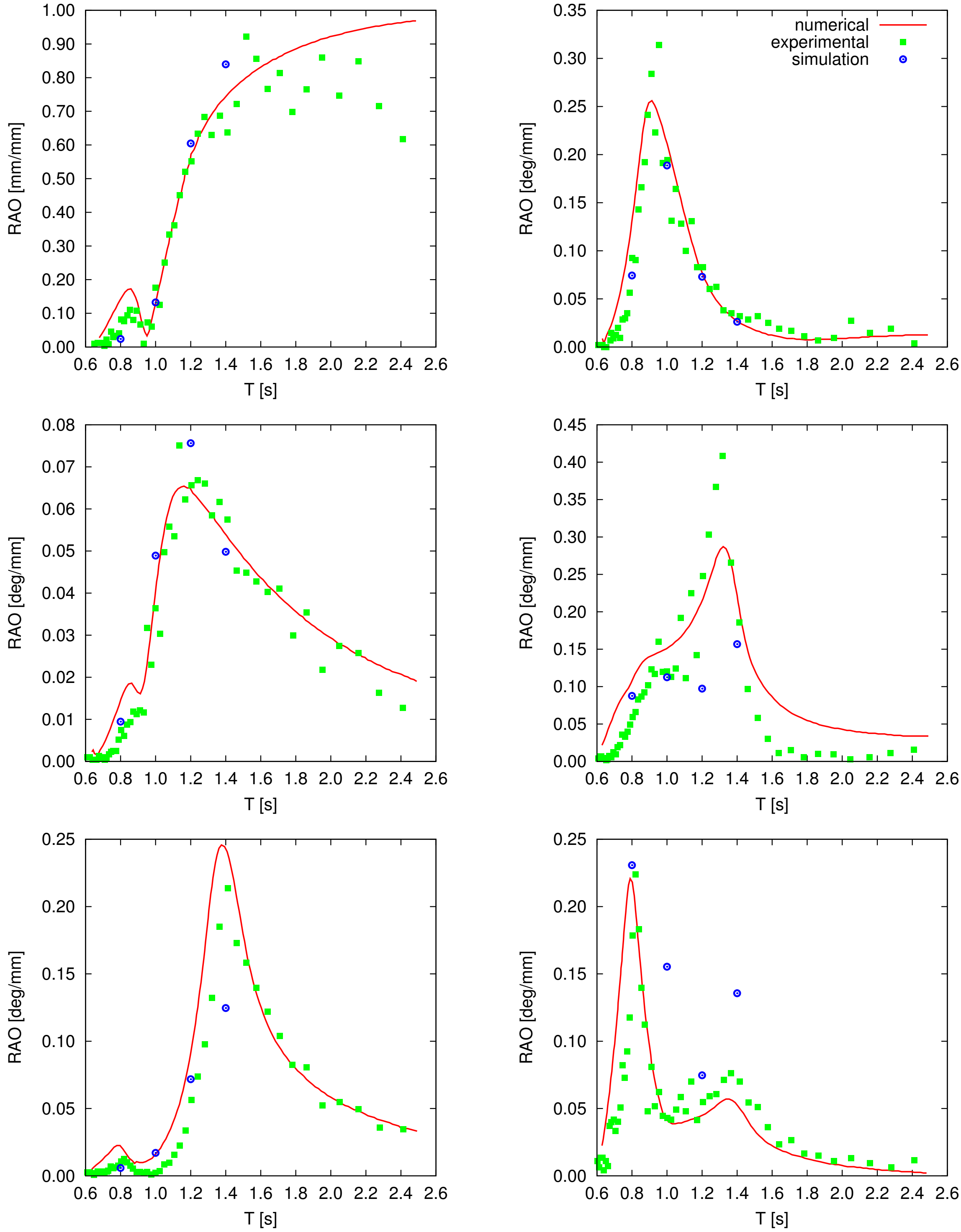


Figure 2: Comparison of RAOs between current simulations (open circles) and Senjanović et al. [4] (solid line and filled squares). The RAOs are of heave (top left), pitch (middle left), roll (bottom left), vertical bending (top right), horizontal bending (middle right), and torsion (bottom right)