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A Fully Hydrodynamic Approach to the Motion in Waves of Ships with Free Surface Liquids on Board.

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Background

The work presented in this paper deals with the development of a numerical framework for the simulation of the motions of a ship in a seaway with free-surface liquids on board. The problem as a whole may be split into two different subproblems. The first regards the appropriate simulation of large amplitude coupled ship motions in waves; the latter is related to the simulation of large amplitude liquid sloshing inside the partially filled tank. In the past, as a general rule, linear ship motion computer codes have been matched to algorithms which solve the partial differential equations modelling liquid sloshing *[Dillingham, 1981; Pantazopoulos, 1990; Lee et al., 1994]*. The weakness of these approaches has been well recognised in the ship motion module and specifically in the roll computation, basically because of the lack of an appropriate nonlinear modelling. As a consequence, in the recent past the uncoupled nonlinear roll equation has been adopted by the authors *[Francescutto et al., 1994; Armenio et al., 1995a]* as ship motion module, the coefficients being obtained from experimental tests without liquids on board. This could appear an oversimplified ship motion approach; despite this, the obtained results show a good accuracy in the whole range of wave frequencies investigated. Presently a parameter identification is used, allowing for a wave frequency/amplitude dependence of the roll equation coefficients. On the other hand it is well established that sway may affect liquid sloshing, which, in turn, reflects on the roll motion. In order to take into account the effective coupling among ship motions and liquid sloshing inside a compartment, a fully hydrodynamic approach has been implemented. The only limitation of this approach consists in the potential formulation for the external problem, and in being two-dimensional.

Formulation and numerical scheme

A fluid dynamics approach is here employed, including an external flow problem for the ship motions and an internal flow problem for the liquid sloshing. As regards the internal flows, a constant amount of liquid is considered, i.e. no inflow-outflow is taken into account.
The 2D ship motions are evaluated by means of the NWT (Numerical Wave Tank) method as reported in Contento *[1995 a,b]*. A recent improvement to the algorithm consists in the stretching of the grid on the free surface: this allows to take into account a large number of wavelengths in the tank with a comparable computational effort and with an undoubted benefit in terms of length of the simulation before reflections from the beach and/or from the wavemaker. Liquid sloshing inside the tank (SLOSH module) is simulated solving the Reynolds Averaged Navier-Stokes Equations by means of an improved MAC method recently developed *[Armenio, 1994; Armenio, 1995b]*. The previous computer codes have been extensively validated by means of comparison with experimental data.

Coupling scheme

The main effort in this work regards the coupling between the ship motions and the sloshing loads modules. In principle the problem can be simply summarised as follows:

$$M\,\ddot{\vec{X}}(t) = \vec{F}_{hull}(t) + \vec{F}_{slosh}(t)$$

being M and $\ddot{\vec{X}}(t)$ the generalised mass and acceleration of the floating body respectively, $\vec{F}_{hull}(t)$ the external actions as derived in the NWT module and $\vec{F}_{slosh}(t)$ the sloshing loads as calculated in the SLOSH module. At each time step the incoming exciting waves together with the sloshing induced loads cause the motion of the ship, which, in turn, are given to the SLOSH module for the evaluation of the internal loads at the new time iteration.

Numerical results and conclusions

Numerical computations have been carried out considering a typical ship section whose main dimensions and mechanical characteristics without liquids are in Tab.1:

Ship section			Flooded tank		
breadth	= 1.0	m	breadth	= 0.9	m
length	= 1.0	m	length	= 0.5	m
draft	= 0.35	m	height	= 0.6	m
mass	= 205.43	kg	water depth	= 0.1115	m
KG	= 0.4	m	water mass	= 50.18	kg

Tab. 1: main characteristics of the ship section and of the flooded compartment

Two different mathematical methods have been considered. In the first one, the dynamic sloshing is neglected and the internal water contribution to the motion is accounted only statically (hydrostatic method) as often used in the analysis of damage stability of ships. In the second one, the above described fully hydrodynamic approach is employed.

The first computational test regards the evaluation of the roll natural frequencies of the dynamic system by means of decay tests (starting heel angle=5°). The Fourier analysis of the time records of the roll motion (Fig. 1) evidences large differences between the hydrostatic and the hydrodynamic approach. The former shows the presence of a single fundamental rolling mode at ω_1=2.83 rad/s, whereas the latter shows two different modes, at ω_2 = 2.30 rad/s and ω_3= 5.07 rad/s respectively. It is well known from simplified mathematical models of ship motions with free surface liquids on board and from experimental tests *[Francescutto et al., 1994]* that, due to the dynamic coupling of the two subsystems, a floating body with shipped liquids on board exhibits at least two fundamental roll natural frequencies. They correspond to the in-phase and opposite-of-phase oscillations of the ship with respect to the liquid. In this perspective, the results derived by the hydrodynamic method appear fully consistent with physics.

In Fig. 2 it is shown the forced motion of the ship in waves (ω_{wave} = 2.98 rad/s) as evaluated by means of the hydrostatic and hydrodynamic approaches. For sake of clearness, the vertical displacement of the center of mass of the ship in the hydrostatic case has been shifted. No appreciable differences appear in the sway and heave motion of the ship. On the contrary, both the transient and the steady state rolling look substatially different. In the hydrostatic case, the roll motion hardly reaches a steady state oscillation experiencing strong beatings, whereas in the hydrodynamic case a steady state rolling is finally achieved. In this case the liquid sloshing acts as an additional damping for the ship motions.

In Fig. 3 and 4 the ship motions and the hydrodynamic forces, as evaluated by the two methods near the fundamental resonant frequency (ω_{wave} = 2.10 rad/s), are plotted. Now both sway and roll motions show evident differences. As far as the sway is concerned, the hydrodynamic computation predicts a lower amplitude oscillation with a higher mean drift velocity than in the hydrostatic simulation. These circumstances are easily explained looking at Fig. 4, where the difference in the amplitude of the horizontal sloshing force can be observed. As regards the roll motion, strong differences appear as in the previous computations. Again the sloshing moment acts as a damping with an evident effect on the length of the transient.

The results presented in this abstract are very preliminary. Basically they show a first attempt to pursue a more realistic approach to the evaluation of the dynamic characteristics of a ship in a seaway when free surface liquids are shipped on board. As an example this is the very important case of RoRo ships after garage deck flooding.

References

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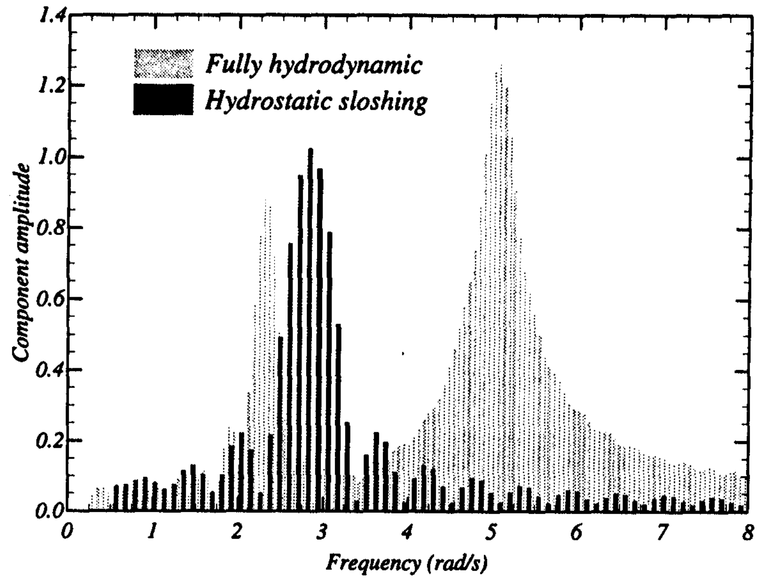

Fig. 1 Fourier analysis of the time record of the roll free decay, using the hydrostatic and the hydrodynamic approaches. (Starting hell angle = 5°, lenght of simulation = 40 s). The analysis is carried out only for evidencing the fundamental roll modes.

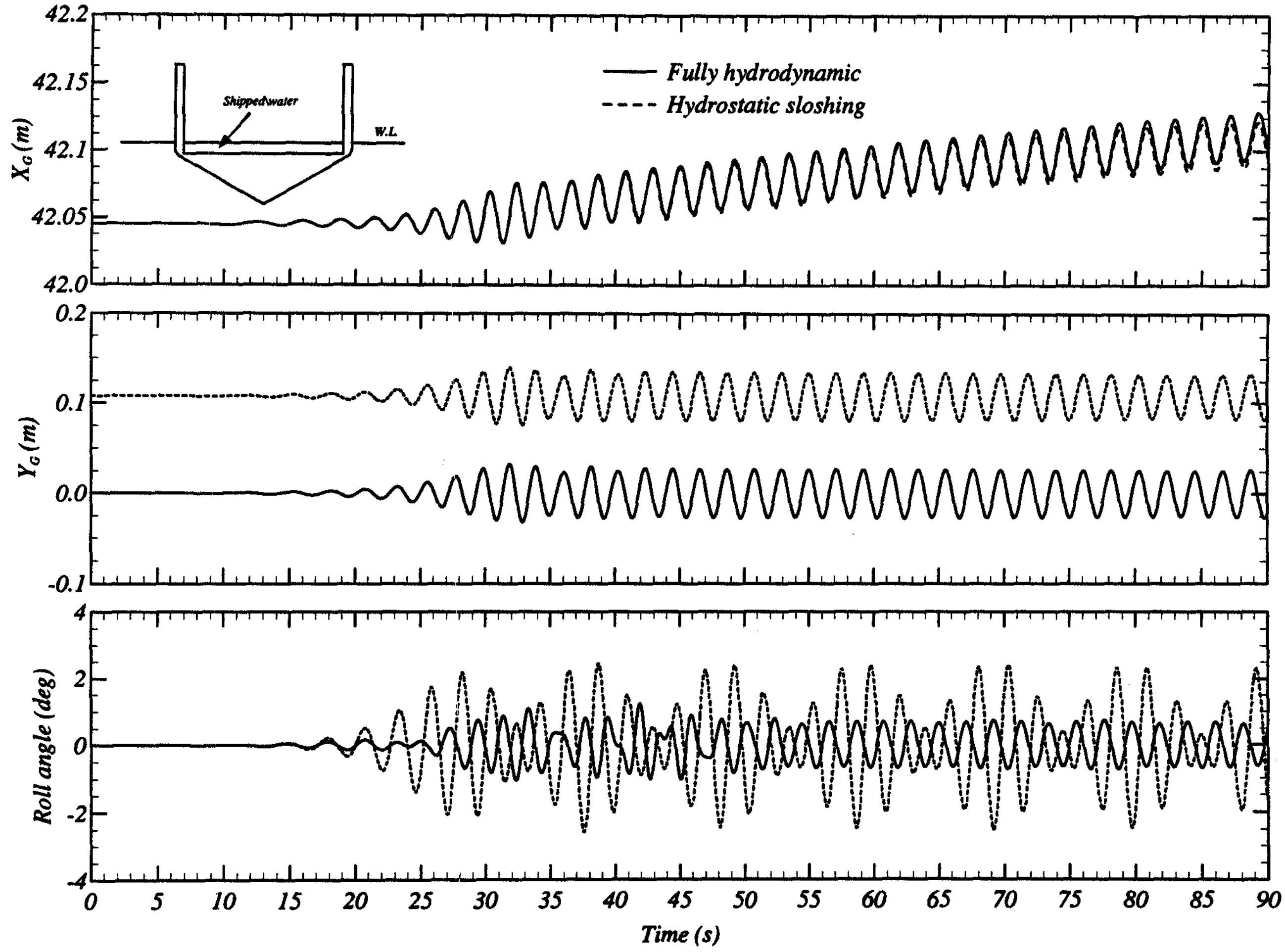

Fig. 2 Comparison between the ship motions derived by the hydrostatic and hydrodynamic methods. (wave frequencies = 2.98 rad/s, wave steepness = 1/130, grid for RANSE computation: stretched 128x128 ; grid for NWT computation: 60 panels on the body, 330 stretched panels on the free surface, 20 panels on the wavemaker and open boundary)

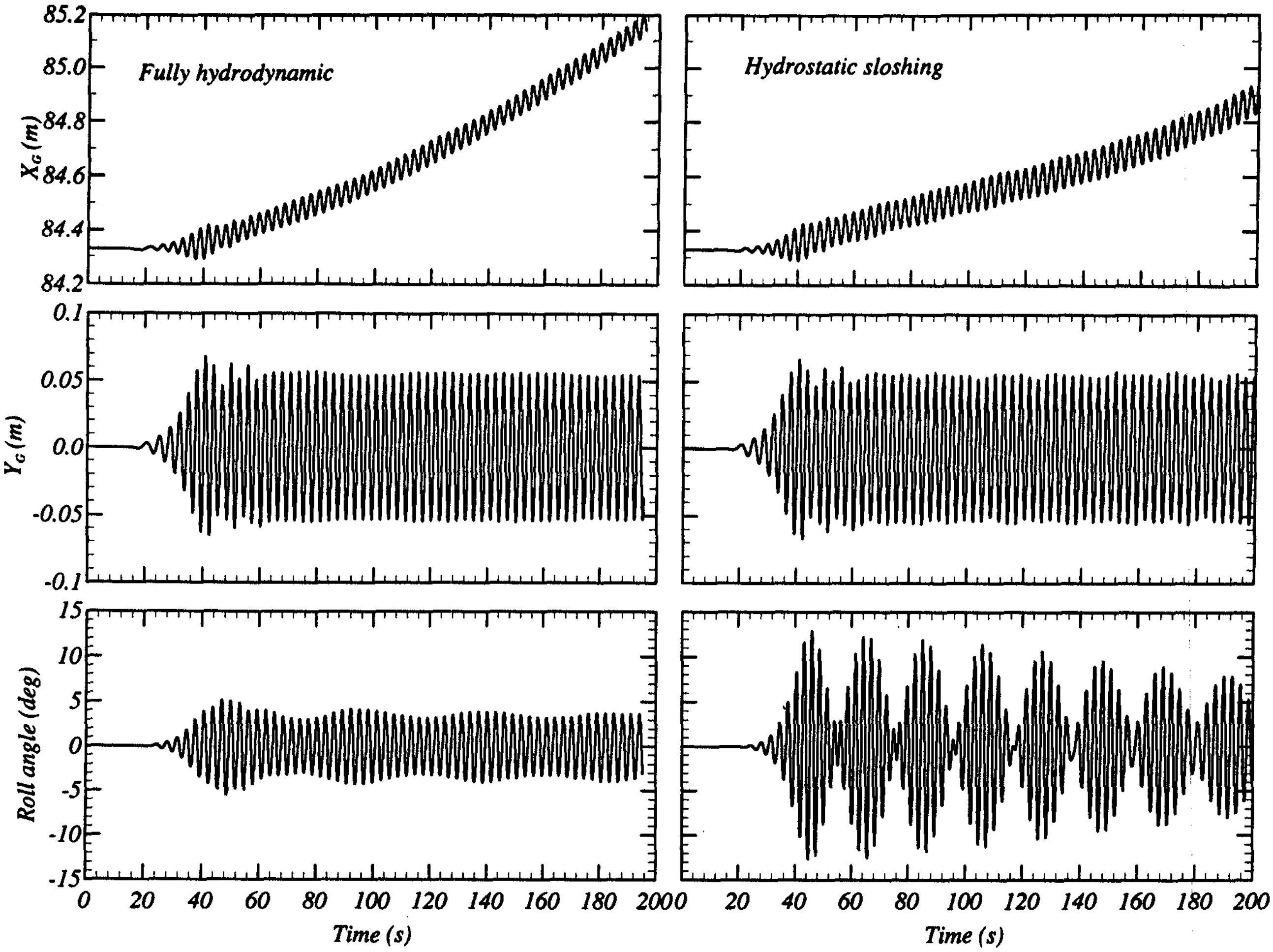

Fig. 3 Comparison between the ship motions derived by the hydrostatic and hydrodynamic methods. (wave frequencies = 2.30 rad/s, wave steepness = 1/130, grid for RANSE computation: stretched 128x128 ; grid for NWT computation: 60 panels on the body, 350 stretched panels on the free surface, 20 panels on the wavemaker and open boundary)

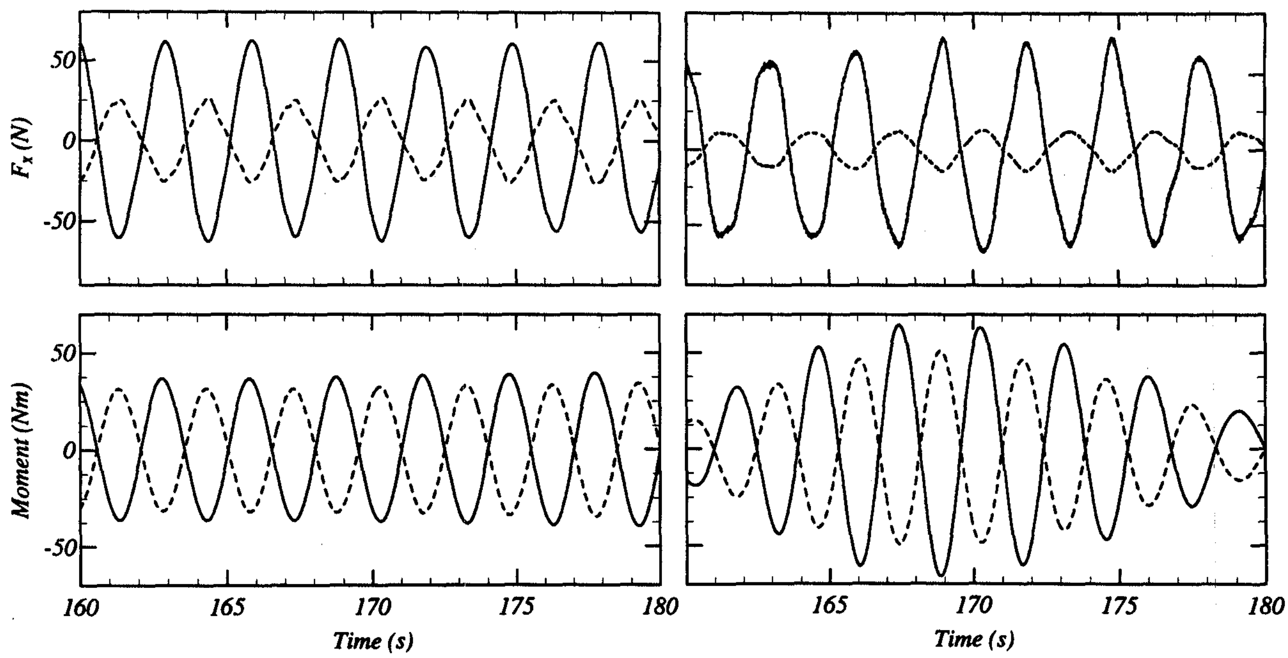

Fig. 4 Comparison between the external hydrodynamic (solid line) and sloshing (dotted line) horizontal force and moment for the simulation of Fig. 3. The results by the hydrodynamic and hydrostatic approaches are on the left and on the right respectively.

DISCUSSION

Tanizawa: You mentioned in your abstract that the coupling scheme can be simply summarised as

$$M\ddot{x}(t) = \vec{F}_{hull}(t) + \vec{F}_{slosh}(t)$$

But for a freely floating body and fluid interaction problem, $\vec{F}_{hull}(t)$ and $\vec{F}_{slosh}(t)$ are a function of $\ddot{x}(t)$. And so the right-hand side of above equation cannot be given explicitly. How do you take $\ddot{x}(t)$ into the computation of $\vec{F}_{hull}(t)$ and $\vec{F}_{slosh}(t)$?

Francescutto et al.: As you mentioned in the question, both $\vec{F}_{hull}(t)$ and $\vec{F}_{slosh}(t)$ in the RHS of the body motion equations depend on the acceleration itself and, due to the nonlinearities involved, the appropriate contribution cannot be given explicitly. As far as the external problem is concerned ($\vec{F}_{hull}(t)$), the problem basically involves the evaluation of $\partial\phi/\partial t$ at a given time t^* in Bernoulli equation. The problem has been solved in the literature in three main ways: numerical differencing, integral equation for $\partial\phi/\partial t$ or the acceleration potential. A simple backward differencing scheme based on the previously calculated values of ϕ means to keep the total derivative $d\phi/dt$ frozen in the intermediate steps of the time stepping scheme (for example by a fourth order Runge Kutta scheme). This may cause the loss of momentum or even the occurence of instabilities. To avoid this, in the present procedure a new scheme has been employed, the terms $d\phi/dt$ and dX/dt, dY/dt needed to obtain $\partial\phi/\partial t$ being evaluated by second-order finite differences consistently with the Runge-Kutta time stepping as follows: at the first substep a backward difference is simply used whereas in the second, third and fourth substeps the derivative is "centered" at time t^* using the values at $t^* - \Delta t_{old}$, t^* and $t^* + \alpha\Delta t_{new}$ ($\alpha = 0.5$ or 1.0) from the preceding intermediate substep. The final derivative is calculated from the four derivatives just obtained using the RK4 scheme for the final jump. Previous checks on mass/energy conservation, presented also at the IWWW'95, show extremely encouraging results even under strong excitation; moreover free decay tests of the heave motion of a half immersed cylinder in calm water corresponding to the experimental measurements of Vugts (1968) have been performed. From the time records of the simulated motion, the added mass and damping at the natural frequency have been derived according to the linear approximation. The obtained values have been compared both with the experimental ones from Vugts in the assigned motion case and with the computations in the assigned motion case. The comparison is shown in IWWW'95 (signed as a star in Fig.1) and can be regarded as satisfactory. As far as the coupling problem is concerned, many computations have been performed at our department in the case of roll equation (ODE) with numerical sloshing, both for inviscid and viscous flow in the tank (Francescutto, Contento, ISOPE'94; Armenio, La Rocca, MB'95). There again the explicit contribution of the force/moment of the sloshing due to the acceleration was not obtainable; so the computations have been performed using a "frozen" sloshing force/moment in the Runge Kutta substeps for the body motion. Despite this, the comparison of the computed roll amplitudes with experimental measurements conducted at our department in a wide frequency range was extremely good.

Huang: In your opinion, does the roll resposnse always have the same frequency as the incident wave? In our simulation of the coupled ship motion and water sloshing on desk, we observed multiple period roll motion, such as the period roll motion is three and a half wave period.

Francescutto et al.: During the numerical simulations, phenomena as those indicated by Mr. Huang, reported also by Lee and Adee (Proc. STAB'94), have never been observed. Also the experimental tests carried out at our department in a regular beam sea in several conditions (water on deck, partially filled compartment, extreme rolling with water on deck up to capsizing) have always been given a roll frequency very close to the wave encounter frequency. The phenomenon above discussed, or ultra- and sub-harmonic oscillations, have been predicted in nonlinear rolling long ago (Cardo Francescutto and Nabergoj, ISP, 1981), but usually their appearance is connected with rolling frequency close to the natural roll frequency and to the exceedance of a treshold for the excitation/damping ratio. Perhaps the above phenomenon is related to some special combinations between the ship rolling and liquid sloshing natural frequencies, or to poor initial stability of the ship in a regular beam sea. It would be nice to know more about the phenomenon observed by Mr. Huang.

Delhommeau: The problem of MAC methods is generally in the computation of forces. What tests have you done to verify the sloshing forces? Why did you use a MAC method instead of a perfect fluid nonlinear method like BEM?

Francescutto et al.: The computer code simulating internal sloshing has been extensively validated. First of all, results have been obtained in very simple cases in which analytical solutions were available. Then experimental tests have been carried out for an isolated tank in roll motion. Numerical results compare well with experimental data in the whole range of filling depths and rolling frequencies examined (Armenio, La Rocca, Proc. MB'95, 1995; Armenio, La Rocca, to appear in Ocean Engineering). We prefer to use a RANSE solver rather than an inviscid model for two main reasons. The first is that internal damping cannot be predicted by using an inviscid model. The second is that practical applications often refer to baffled tanks. In these cases vortex shedding generated by the baffles can affect liquid motion and the induced sloshing loads.