Wave effects on vessels with internal tanks

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The motions of fluid in internal tanks have important effects on the dynamic response of vessels in waves, particularly during loading and unloading operations when the tanks are partially filled. This topic is of special interest for LNG tankers and FPSO vessels. Coupled tank/ship motions have been studied by Kim (2001) and Rognebakke & Faltinsen (2001, 2003), with nonlinear analyses of the interior flow in the tanks, and by Molin *et al* (2002) and Malenica *et al* (2003) who use linear analyses. The tank dynamics are analysed separately from the exterior radiation and diffraction problems. The solution of the coupled equations of motion follows by combining the hydrodynamic forces for the tanks with the vessel's added-mass, damping, and exciting forces.

Recently we have extended the free-surface panel code WAMIT to analyse coupled tank/ship motions, following a more unified approach where the interior wetted surfaces of the tanks are included as an extension of the conventional computational domain defined by the exterior wetted surface of the body. The geometry of the tanks is represented in the same manner as the exterior wetted surface of the hull. In effect all of the tank and hull wetted surfaces form one large global boundary surface. The principal modification required is to impose the condition that the separate fluid domains are independent. This is achieved trivially, by setting equal to zero all elements of the left-hand-side of the linear system for the potential where the source and field points are in different fluid domains. This is equivalent to forming separate sets of linear equations for each domain, and concatenating these into one larger system in a block-diagonal manner. Another obvious modification is to omit the diffraction solution from the domains of the tanks. The exterior free-surface Green function is used for each domain, with vertical shifts of the coordinates corresponding to the free-surface elevation in each tank.

The principal advantage of this approach is that the exterior panel code can be extended to include internal tanks with relatively few modifications. All of the usual hydrodynamic parameters can be evaluated in a similar manner as for vessels without tanks, including the added-mass and damping coefficients, exciting forces, RAO's, fluid pressures and velocities, and the mean second-order drift forces and moments. The geometry of the tanks can be described in the same manner, and generality, as the exterior hull surface. Disadvantages include the larger size of the linear system, which implies some loss of computational efficiency, and the need to re-run the complete interior/exterior analysis in situations where only one or the other is changed, e.g. when the tank depths are modified. Since the entire analysis is linearized, nonlinear sloshing effects are not included.

Computations have been made for various vessels, including the barge model studied by Molin *et al* (2002) where both experimental and computational data are available for comparison. Results are shown here for the spheroid in Figure 1. This vessel has three internal tanks, with the same depth of fluid in each tank. The tank lengths are the same, but the widths and elevations are different. Results are shown in Figure 2 for the first-order motions and drift force in beam waves, for three relative densities of the tank fluid (ρ=0, 0.5, 1.0). The total displacement and waterline plane are fixed as the tank density is varied. The results with zero density are equivalent to the conventional case without internal tanks. All results are normalized by the exterior fluid density, gravity, wave amplitude, and a characteristic length scale of 1m, and plotted vs. the nondimensional wavenumber $Ka = \omega^2 a/g$, where ω is the radian frequency and a=1m is the maximum radius of the spheroid. The vertical center of gravity is in the waterplane, and the radii of gyration are k_x=50cm, and $k_y = k_z$=3m.

Figure 3 shows the six principal added-mass coefficients, which are normalized by the mass of fluid displaced by the hull. The added mass is the sum of the pressure force on the hull and tanks. Thus the coefficients in Figure 3 are linear functions of the tank density, and only the densities 0 and 1 are shown. The damping coefficients are not affected by the tanks.

Most of the added-mass coefficients are singular at the resonant periods corresponding to antisymmetric sloshing modes of the tanks. The surge resonances at Ka=1.184 and 4.686 are the same for all three tanks. There are two resonant periods in sway (Ka=1.653, 2.427) due to the different widths of the tanks. In yaw the first singularity corresponds to the sway mode of the outer tanks (Ka=2.427); the second smaller singularity is associated with the diagonal sloshing mode of the center tank which occurs at Ka=2.922.

In the vicinity of resonance the added-mass coefficients tend to $\pm\infty$. This explains the rapid fluctuations of the RAO's shown in Figure 2. The sway RAO approaches zero at the resonant frequencies, where the added mass is infinite. At slightly higher frequencies, where the negative added mass cancels the body mass, the RAO is large. There is no roll moment induced by the external pressure on the axisymmetric hull, but the tanks induce nonzero roll motions when the density is nonzero.

For heave the tank fluid translates in rigid-body motion, and the RAO is not affected by the tanks. The frequency-dependence of the tank component is an interesting feature of the heave added mass in Figure 3. The velocity potential in each tank for a unit heave velocity is $\phi = (z_t - 1/K)$, where z_t is the local vertical coordinate above the tank free surface and the constant $1/K$ is required to satisfy the free-surface condition. For a tank with constant depth d it follows that the added-mass is equal to $(1 - 1/Kd)$ times the fluid mass. The term $1/Kd$ gives a force proportional to the waterplane area, which is cancelled by the hydrostatic restoring force.

The most surprising results are the sharp reductions in the sway drift force, which coincide with the peak sway response. From momentum conservation, the tanks can only affect the horizontal drift force indirectly, by modifying the motions of the hull, and for this vessel roll has no effect on the drift force. Since the negative peaks of the drift force coincide with positive peaks of the sway RAO, it appears that the reduced drift force is directly related to the increased sway RAO.

References

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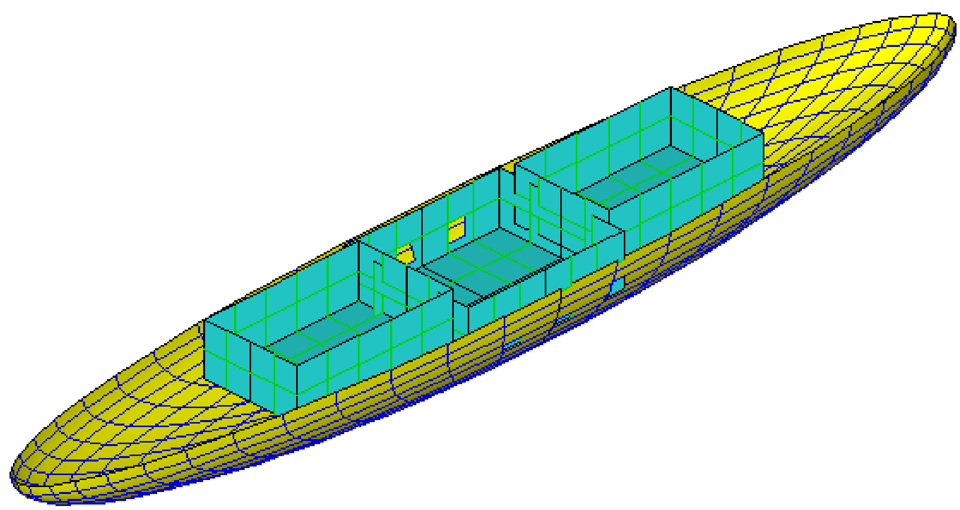

Figure 1: Perspective view of the spheroidal hull. The length is 12m and the midship section is a semi-circle of radius 1m. Each tank is 2m long, and 62.5cm deep. The tank widths are 120cm, 180cm, and 120cm, progressing from x=+3m to x=-3m. The free surfaces are at z= 25cm, 12.5cm, and 25cm above the exterior waterplane.

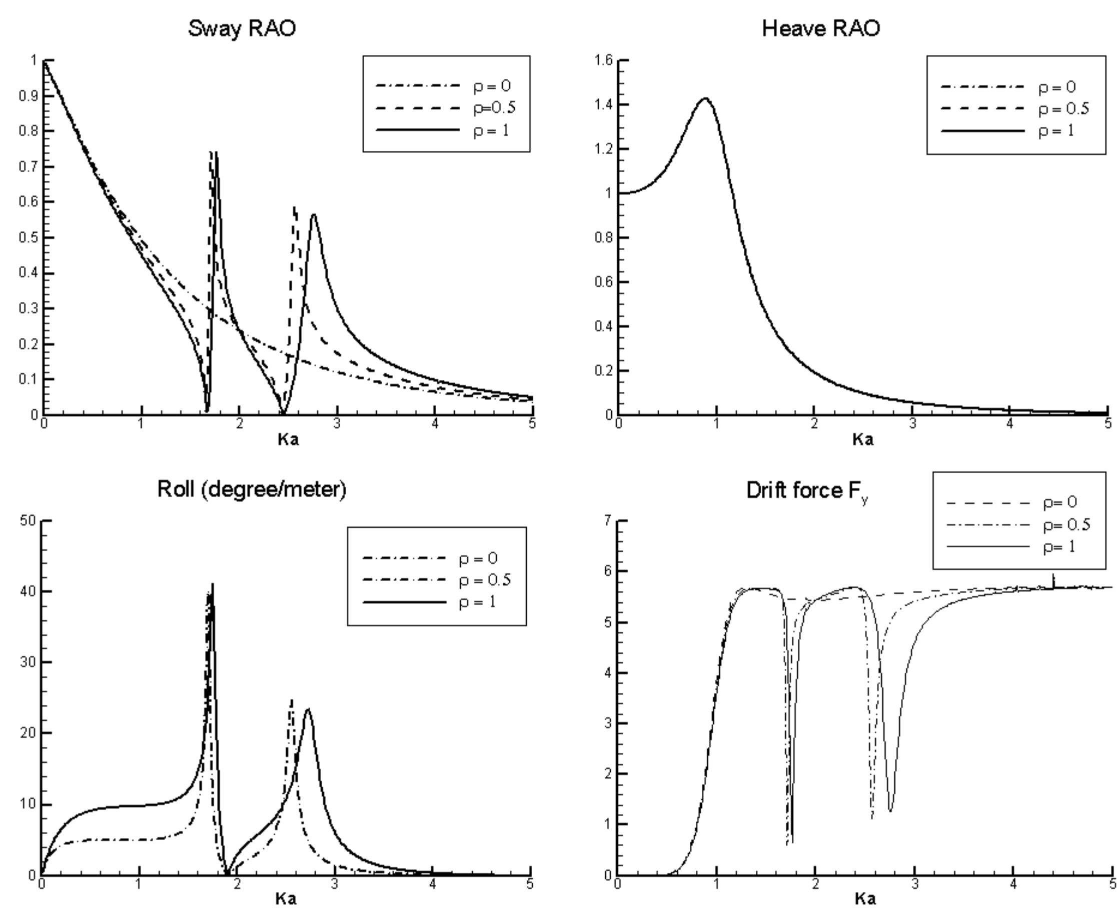

Figure 2: RAO's and drift force for the spheroidal hull in beam waves.

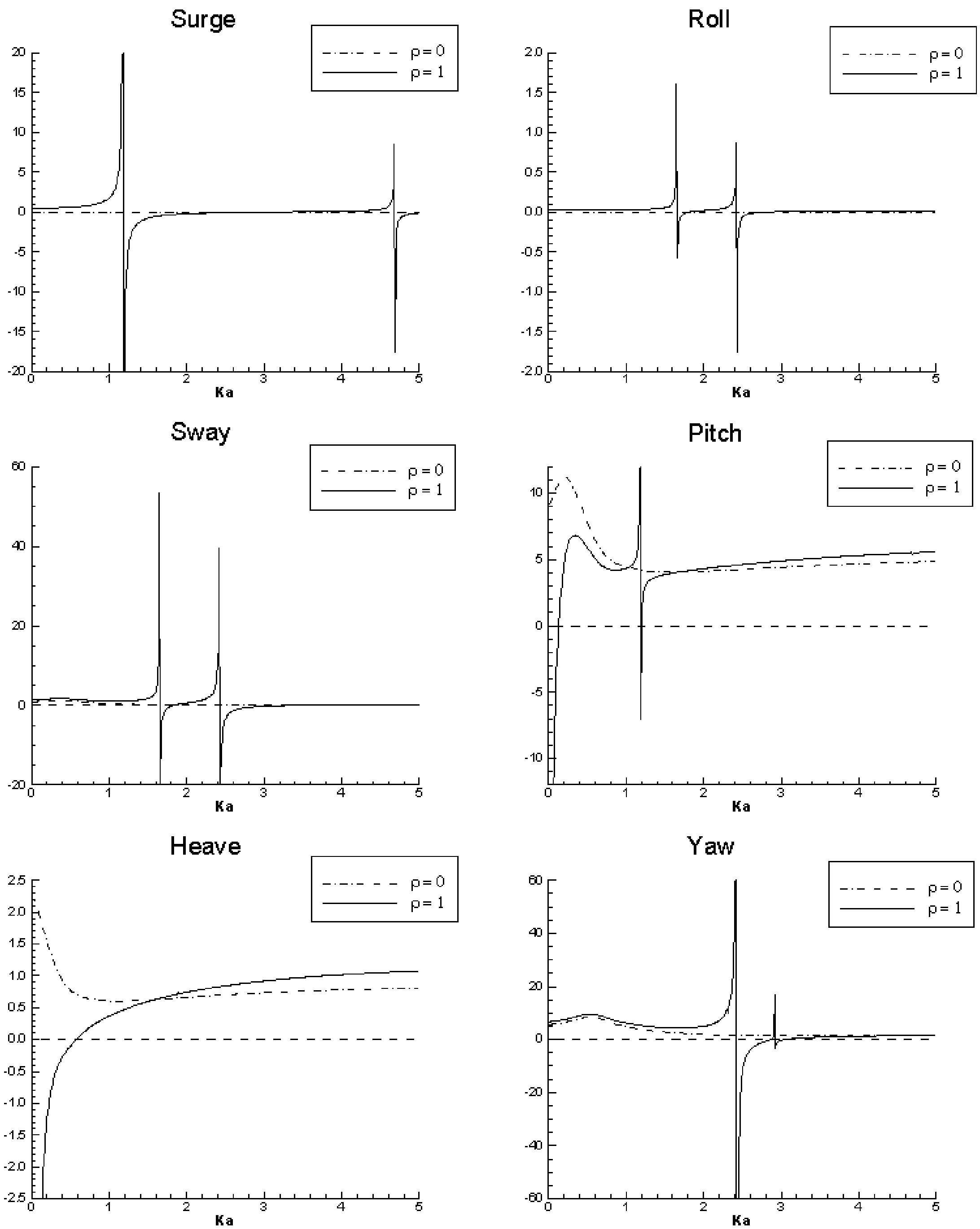

Figure 3: Added-mass coefficients of the spheroidal hull. All coefficients are normalized by the displaced mass and a length of 1m.

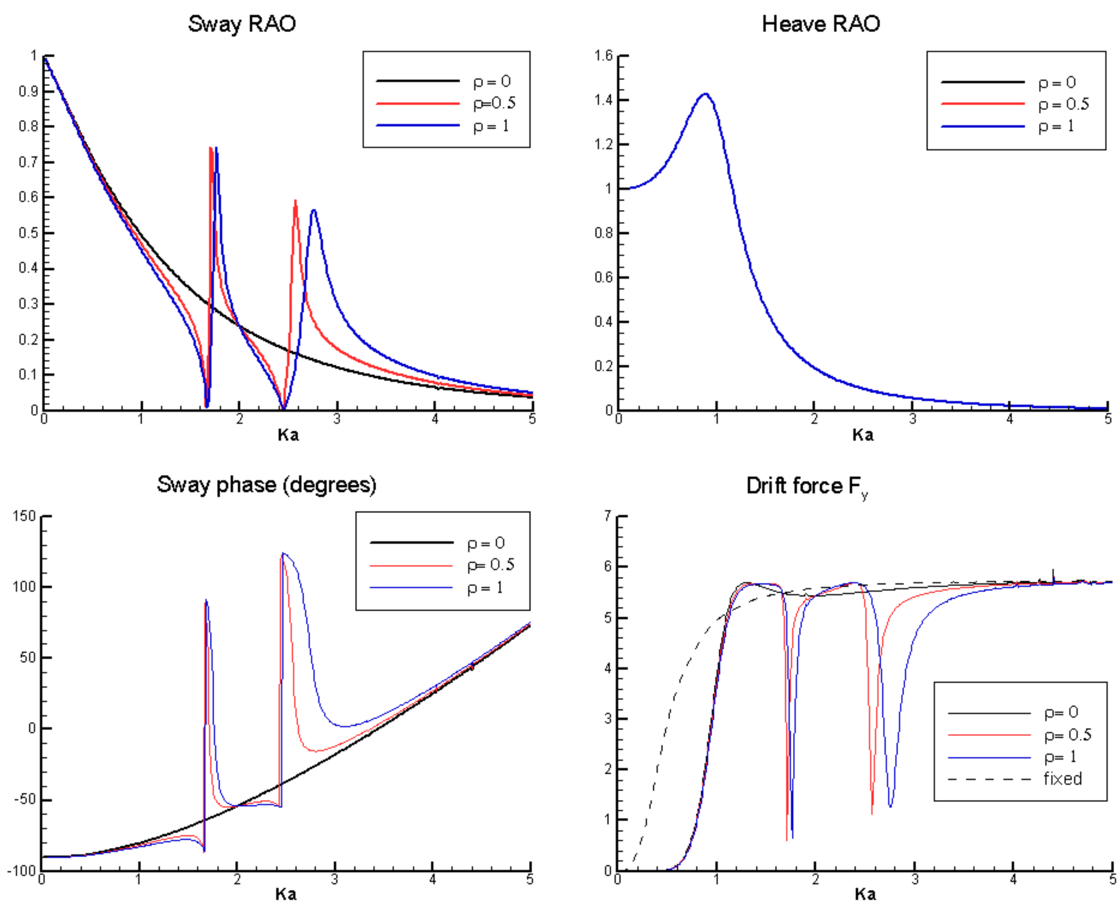

Figure 4: RAO's and drift force for the spheroidal hull in beam waves.