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Ship Entry into a Lock

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Abstract: The problem of ship entry into a very narrow lock is studied with attention to the interaction between the ship's motion and its waves. A one-dimensional unsteady hydraulic narrow-channel model for the flow coupled to the ship's motion in surge, heave and pitch is proposed and numerically implemented. The calculated ship motion is validated by comparison with model experiments carried out in the VBD shallow water tank by Zöllner and Broß (1993). As the ship begins to enter the lock it pushes a mass of water ahead so that a bore is generated. By way of reaction the ship experiences an impulse, decreasing its forward speed and increasing its trim and sinkage enormously. The accurate prediction of such motion is extremely relevant in practice. A movie animation is provided to illustrate this interesting phenomenon.

Introduction

The problem studied here concerns ship entry into a very narrow canal-lock. A typical example in Europe would be an inland ship of length l = 110 m, beam b_m = 11.4 m and draft d_m = 2.5 m moving at a speed of about 5 km/h into a rectangular canal-lock of length L = 200 m, breadth B = 12 m and water depth h = 3 m. We note in passing that this flow problem is analogous to that of a leaky piston pushing into a cylinder filled with a compressible fluid.

Before the ship reaches the lock it moves at a constant speed in a relatively wide lead-in canal. There it is in equilibrium since the net effective propeller thrust equals to the total hull resistance. Because of the low ship speed (F_{nh} < 0.3) the resistance is mainly frictional. As the ship-bow passes the lock-gate, it pushes a mass of water into the lock. Due to a piston effect, the free surface in the lock is at first greatly elevated and the resulting hump of water then slowly runs off through the narrow clearance between the ship and the lock. Owing to the inertia of water, the receding tendency goes on to form a depression in the lock. This process repeats itself becoming weaker and weaker in course of time. As a reaction the ship initially suffers a drastic deceleration, followed by decaying cycles of acceleration and deceleration.

An experimental investigation of this problem on model scale was carried out by Zöllner & Broß (1993) at the VBD in Duisburg. The motion recorded in the model experiments can be exploited to estimate the dominant effect. The average deceleration in the first phase amounts to 0.0025 g. The corresponding braking force is about 20 times larger than the steady frictional resistance according to the ITTC 1957 formula. This highlights the overwhelming importance of inertial effects in this phase.

Mathematical Model

Two coordinate systems are used: an earth-bound system $Oxyz$ and a system $\hat{O}\hat{x}\hat{y}\hat{z}$ moving horizontally with the ship. They are interrelated by

$$x = \hat{x} + x(t) - l/2, \quad y = \hat{y}, \quad z = \hat{z}, \quad t = \hat{t},$$

where l is the ship length and $x(t)$ is the distance covered in the lock (from the lock gate to the ship bow). The sinkage of the ship's center of gravity G is denoted by $s(t)$; the trim, by $q(t)$ (stern up positive). Therefore, its dynamic local cross-sectional area S_d under the undisturbed water level z = 0 can be derived from its static value S_0 as follows:

$$S_d(\hat{x},t) = S_0(\hat{x}) + b(\hat{x})[s(t) + (\hat{x} - \hat{x}_G)q(t)].$$

Here, only the strictly symmetric case is considered, i.e., a symmetric ship entering a symmetric lock along its centerline. Moreover, the problem is simplified to be one-dimensional, i.e., the free surface is $z = \zeta(x,t)$, the mean fluid velocity in x-direction is $u(x,t)$, and the dynamic local canal cross-sectional area filled with water is

$$A(x,t) = A_0(x) - S_d(\hat{x},t) + [B(x) - b(\hat{x})]\zeta(x,t)\,.$$

The flow is, therefore, modeled by the following mass-conservation and momentum equations:

$$\frac{\partial A}{\partial t} + \frac{\partial (uA)}{\partial x} = 0 \tag{1}$$

$$\frac{\partial u}{\partial t} + \alpha_1 \frac{U}{A} u - \alpha_2 \frac{\partial^2 u}{\partial x^2} + u\frac{\partial u}{\partial x} + g\frac{\partial \zeta}{\partial x} = 0 \tag{2}$$

and the pressure is given by

$$p = \rho g(\zeta - z). \tag{3}$$

The empirical α_1-term in Eq. (2) describes a mean local friction on the canal side-walls and the ship-hull surface; it has a dominant effect. The α_2-term represents a small viscous wave damping and may be neglected. Since by virtue of symmetry only three degrees of freedom (surge, heave and pitch) are involved, the relevant hydrodynamic forces and moment acting on the ship are generally expressed as

$$F_x = -\int_{S_w} p n_x dS\,, \quad F_z = -\int_{S_w} p n_z dS\,, \quad M_{\hat{y}G} = -\int_{S_w} p[\hat{z} n_x - (\hat{x} - \hat{x}_G) n_z] dS$$

and specially evaluated as

$$F_x = \rho g \int_{-l/2}^{l/2} \left\{ \zeta \frac{d}{d\hat{x}} \left[S_0(\hat{x}) + b(\hat{x})(\sigma + \theta(\hat{x} - \hat{x}_G)) \right] + \frac{\zeta^2}{2} \frac{db}{d\hat{x}} \right\} d\hat{x}$$

$$F_z = \rho g \int_{-l/2}^{l/2} \left[\zeta + \sigma + \theta(\hat{x} - \hat{x}_G) \right] b(\hat{x})\, d\hat{x}$$

$$M_{\hat{y}G} = -\rho g \int_{-l/2}^{l/2} \left[\zeta + \sigma + \theta(\hat{x} - \hat{x}_G) \right] b(\hat{x})(\hat{x} - \hat{x}_G)\, d\hat{x}.$$

The corresponding equations of motion are

$$m\frac{dU}{dt} = F_x + F_d\,, \qquad m\frac{dW}{dt} = F_z\,, \quad J_{\hat{y}\hat{y}G} \frac{d^2\theta}{dt^2} = M_{\hat{y}G}\,, \tag{4}$$

where

$$U = dx/dt, \quad W = -d\sigma/dt,$$

and F_d is the excess of net effective propeller thrust over the hull frictional resistance. The thrust is assumed to remain constant throughout at its original equilibrium value before the ship reaches the gate. The frictional resistance is updated continuously using the ITTC 1957 formula with an empirical velocity-increase correction similar to Emerson’s (1959).

Introducing static water-plane integrals

$$A_w = \int_{-l/2}^{l/2} b(\hat{x})d\hat{x}\,, \quad M_w = \int_{-l/2}^{l/2} (\hat{x} - \hat{x}_G) b(\hat{x})d\hat{x}\,, \quad I_w = \int_{-l/2}^{l/2} (\hat{x} - \hat{x}_G)^2 b(\hat{x})d\hat{x}$$

and dynamic auxiliary integrals

$$I_{sink}(t) = \int_{-l/2}^{l/2} z\ (x,t) b(\hat{x})d\hat{x}, \qquad I_{trim}(t) = \int_{-l/2}^{l/2} (\hat{x} - \hat{x}_G) z\ (x,t) b(\hat{x})d\hat{x},$$

the last two equations in Eq. (4) can be rendered in a compact form:

$$m(\frac{d^2 s}{dt^2} + a_s \frac{ds}{dt}) + rg\left(A_w s + M_w q + I_{sink}\right) = 0 \tag{5}$$

$$J(\frac{d^2 q}{dt^2} + a_t \frac{dq}{dt}) + rg\left(I_w q + M_w s + I_{trim}\right) = 0 \tag{6}$$

with $J = J_{\hat{y}\hat{y}G}$ for simplicity, and a_s and a_t as empirical damping coefficients for sinkage and trim, respectively.

Numerical Solution and Results

The coupled equations of motion of the fluid and the ship are solved simultaneously by an implicit finite difference method. Gourlay (1999) has given an analytic solution for the steady problem of a ship moving in an infinitely long narrow channel. We check our computer program by comparison with the analytic solution.

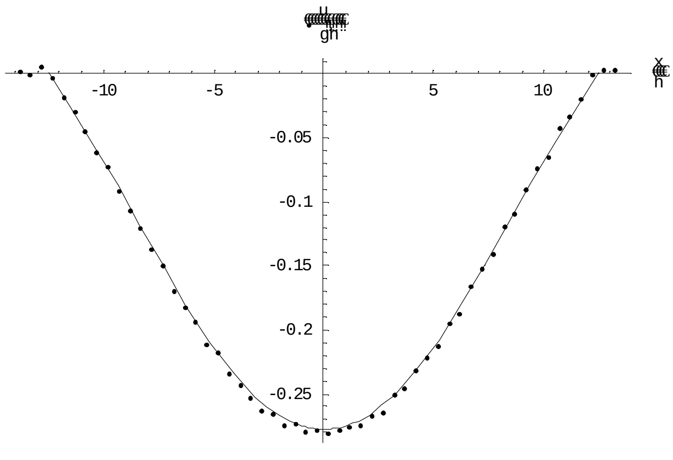

Fig. 1 Numerical and analytical results are represented by the dots and the line, respectively.

Fig. 1 shows a comparison of our numerical results with Gourlay's analytical solution for the velocity distribution along the ship's longitudinal axis for a Wigley hull of length 100 m , beam 11.4 m and draft 3 m at F_{nh} = 0.3 in a channel 12 m broad and 4 m deep. For simplicity, this comparison has been done for a captive hull (no sinkage or trim) and ignoring the empirical factors a_1 and a_2. Evidently, the agreement is satisfactory.

The problem of an inland ship entering a lock was investigated experimentally on model scale by Zöllner and Broß (1993) at VBD. One of their cases was chosen as the first example for our computations. The 1:16 ship model is VBD-M1343 and both horizontal and cross-sectional profiles of the lead-in canal are trapezoidal. The principal dimensions of the ship and lock were already cited in the Introduction. The ship speed is held at F_{nh} = 0.25 (about 5 km/h) as the ship approaches the lock. After the bow passes the lock gate the ship is free to surge.

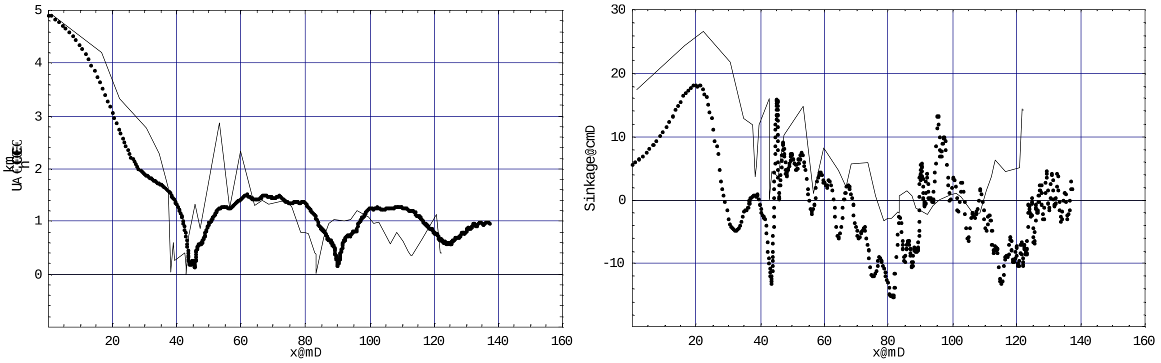

Fig. 2 Numerical and experimental speed and sinkage are represented by the dots and the lines, respectively.

Fig. 2 shows that the numerical computation captures the real unsteady forward motion of the ship with repeated deceleration and acceleration quite well. Even the two near-standstills are simulated with only a slight phase shift. (Note that with a higher initial speed or still larger blockage even afterward motion of the ship could occur temporarily.) On the other hand, the computed sinkage is mostly smaller than the measured one. A probable reason for this discrepancy is that our model does not take into account the propeller effect on the flow, which is further enhanced by the narrow lock. Fig. 3 shows three typical ship wave profiles in the lock at arbitrary instants.

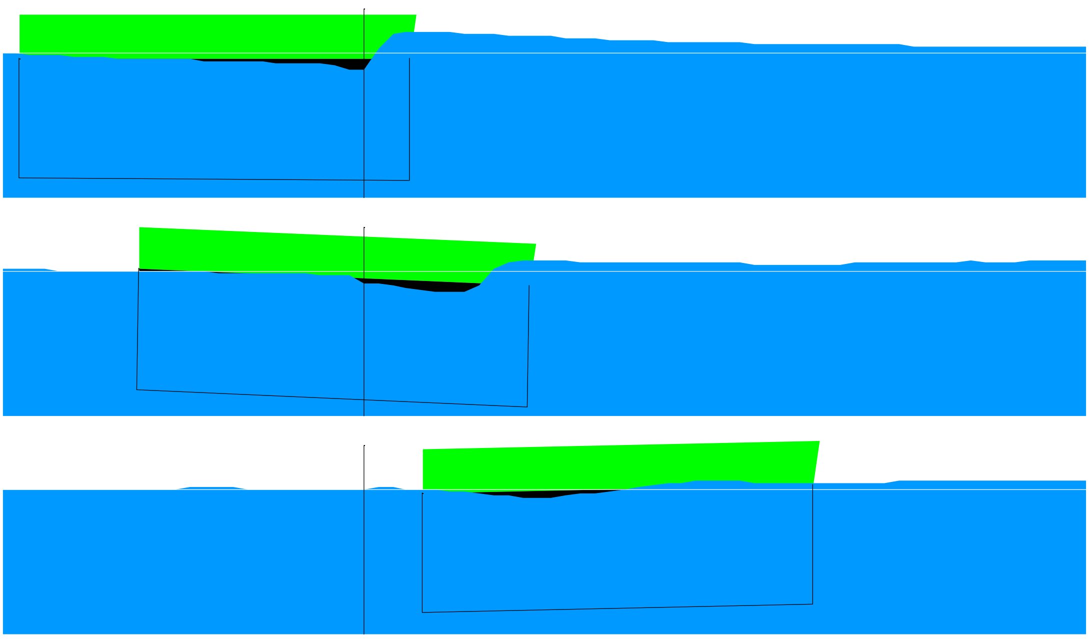

Fig. 3 Calculated ship wave profiles in the lock at three arbitrary instants.

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