Hydrodynamic Interaction Analysis in Marine Accidents

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Abstract

Hydrodynamic interaction effects are critical in congested and confined waters. Increase in speed and size of modern ships makes their consideration in the design of ports, docking stations, navigation channels, an indispensable safety parameter. A brief description of various types of interactions and possible consequences is followed by the main focus of this work, namely moving to stationary ship interaction. A mathematical model amenable to quick computation, including moored ship dynamics is presented. Two cases involving accidents due to such interaction effects are presented along with the author's conclusions from active participation in them.

Keywords

Squat; bank; interaction; shallow; attraction; mooring.

1. Introduction

The utilization of waterways to their limits in our days has rendered the hydrodynamic interaction effects a critical parameter on the channel and port design. Maritime accidents occur primarily in confined and/or congested waters, where hydrodynamic interaction effects dominate. The definition of "interaction" in this study encompasses interaction effects between a ship and the sea/river-bed, a ship and the shore or two ships. The presence of another vessel or shallow water effects will cause coupling and amplification between the interaction effects. The detrimental effects and accident generation from the hydrodynamic interactions have been recognized by Maritime Authorities and related guidance has been issued (MSA 1998). Loss of steerage, collision, down flooding, fire and structural damage are a few examples of what can be instigated on the ship due to hydrodynamic interaction.

The basis of hydrodynamic interactions is the conservation of energy manifested in the form of the well-known Bernoulli equation. Bernoulli equation states that the kinetic and potential energy along a streamline remain constant. Kinetic energy depends on the square of the velocity and potential energy with the depth of the fluid. As a ship moves in the water, there is a region of high pressure at the bow and the stern. The stern high pressure region is of lower magnitude due to frictional loses. Given Bernoulli's theorem, the water displaced by the ship at the bow flows around and under the hull towards the stern and creates a venturi effect under the hull resulting in negative pressure in the mid-ship region.

Immediate consequence of the pressure distribution described above is observed in shallow water, where the restriction between hull and bottom is more pronounced. Venturi effect is more pronounced and the result is depression of the water line in the mid-ship region, which moves with the ship, and a wave-like water rise in the bow and stern. This depression causes a reduction in the under-keel clearance of the vessel and is called "squat". Although the same phenomenon is applicable in deep waters as well, deep water squat is imperceptible. Shallow water is defined as water with depth below two times the draft of the ship.

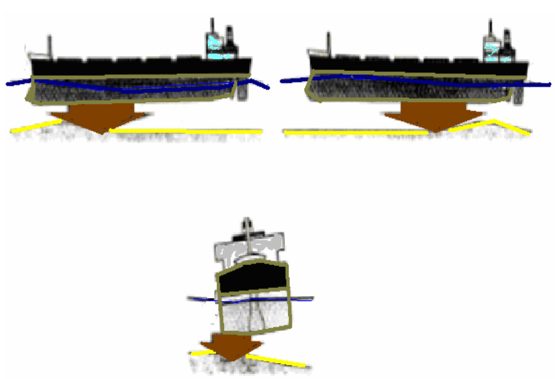

Fig. 1: Squat Induced Trim,Sinkage and Heel

Squat is the thus result of hydrodynamic interaction between ship and bottom. It is not an increase in draft; mean draft remains unaltered. Flow around more "fully bodied" ships is more restricted and it is expected that for these vessels squat will be more pronounced. If the vessel is even-keeled, squat will cause a trim by the bow for fuller vessels (tankers, bulkers). Squat will cause a trim by the stern for finer vessels (container-ships, passenger liners), (Barrass 2006 and 1979). In case the vessel is already trimmed, squat will be further trimmed in the same direction. This is depicted pictorially on Figure 1, where the ship seems to "smell the ground". A dire consequence following squat in vessels with stern trim is the increase of stern trim and the possibility of loss of steerage. This is due to the aft movement of the center of lateral hydrodynamic resistance. If the center of resistance moves aft of the center of gravity of the vessel in shallow water, the lateral resistance force will act against the turning moment of the rudder. The ship will then behave as if running on train tracks, following the bottom, and she will experience loss of steerage. As it is expected, squat depends on the square of the ship speed (Barrass 1979), (Dand & Ferguson 1973). Combined shallow and confined water can yield squat of the order of 5 meters for a VLCC depending on speed. Confined water is defined as a canal with width less than about ten times the beam of the ship. Some headline accidents caused by squat are connected to "Herald of Free Enterprises", "Queen Elizabeth 2" and the "Sea Empress".

Squat is amplified when it is combined with interaction from the shore or "bank effect" and vice versa. In a uniform channel, a vessel traveling in the center of the channel can traverse it with hardly any helm input. Squat might impose sinkage and trim, but otherwise there are no influences from the shore. If the vessel will deviate from the center, a low pressure region will be generated between vessel and bank as depicted in Figure 2, due to accelerated flow; this is a direct consequence of Bernoulli's theorem. The low pressure will manifest into an attraction force on the ship. Furthermore, the asymmetry of the high pressure regions at the bow and stern will result in a yawing moment tending to swing the bow away from the bank (Tuck & Taylor 1970), (Beck 1977), (Cohen & Beck 1983). The ship can be kept parallel to the bank by helm towards the bank as depicted on Figure

2. As the ship speed increases, there is a limit when a long bow wave is generated interacting with the region between ship and bank, generating a high pressure region there. Evidently, in that case the attractive force turns into a rejective one. The speed limit, which dictates the attractive/rejective nature of the sway bank force depends on the water depth and the proximity to the bank, (Yeung & Wooi 1980), (Ch'ng 1991). Similarly to the squat phenomenon, the ship-to-bank interaction increases with decrease of the distance between ship and bank, bank-induced force and moment depend on the square of the ship's speed.

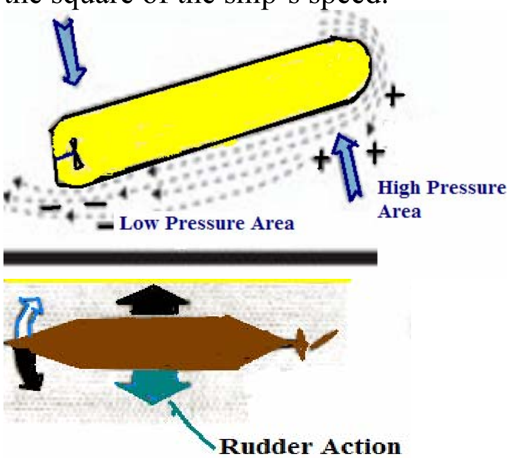

Fig. 2: Bank Effects

A likely position of a ship moving through a canal is the Neutral Steering Line (NSL) of the waterway. This is the sailing line of a vessel along which lateral forces and turning moments due to bank suction are balanced, i.e. no rudder angle input is needed to keep course. In symmetric cross sections the NSL coincides with the centerline of the waterway. In asymmetric sections NSL can be estimated so that it separates the section into two with equal hydraulic radiuses, each of which equals the hydraulic radius of the entire cross section. By definition, hydraulic radius equals the area of the cross section divided by the wetted perimeter. The NSL is always shifted towards the more dredged side/deeper side of the waterway, i.e. towards the dock/dolphin side. The NSL needs to be determined in maritime accidents because that position represents the most stable sailing line for a ship and the most comfortable path for the captain/pilot.

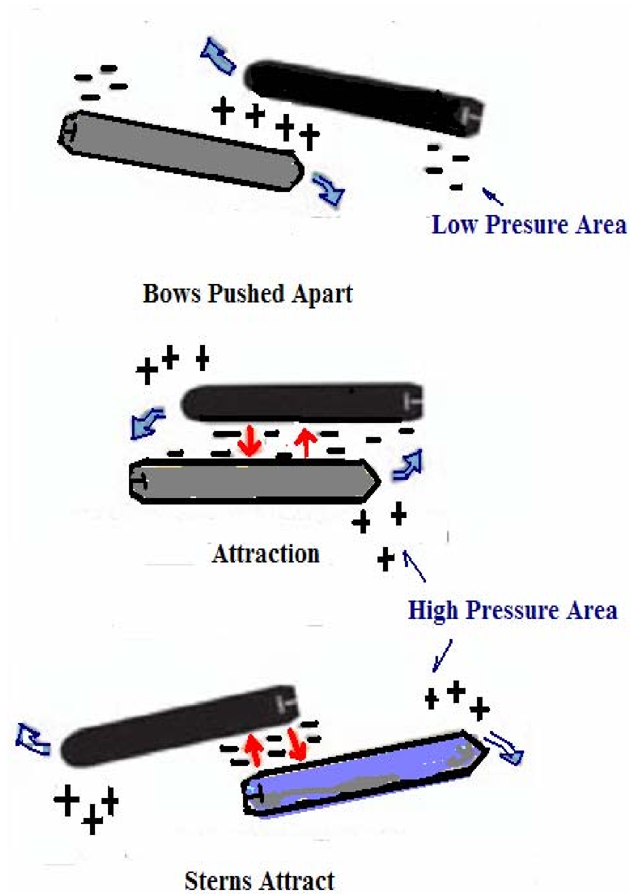

Fig. 3: Interaction in Crossing Ships

Equally important is the interaction between two ships passing each other in close proximity (Tuck & Newman 1974). Hydrodynamic interaction can affect steering and lead to collision. Interaction occurs at any depth but as with the other types of interaction, it is amplified in shallow water. The separation distance between the two ships, besides the speeds squared, is a critical parameter, (Bruce & Fang 1975). Typically, interaction effects are more pronounced if the side shell to side shell distance is less than two times the beam of the wider ship, (Clark 2005), (Yeung 1978). In general the effect on the smaller ship is more pronounced than in the larger one. The well-known "Texas chicken" maneuver performed by the pilots in the Houston channel utilizes hydrodynamic interaction effects or safe passing between ships. The effects on each ship, when crossing each other from opposite directions are shown on Figure 3. In this case at the start and the end of the interaction the two ships are pushed to the starboard. The more critical case of ships overtaking each other is shown on Figure 4. This case is more critical due to the increased maneuver time. In that case, the ships are initially pushed to starboard and finally to port. Phases 2 and 4 present the highest risk of collision since there is both yawing moment and suction force drawing the ships closer to each other.

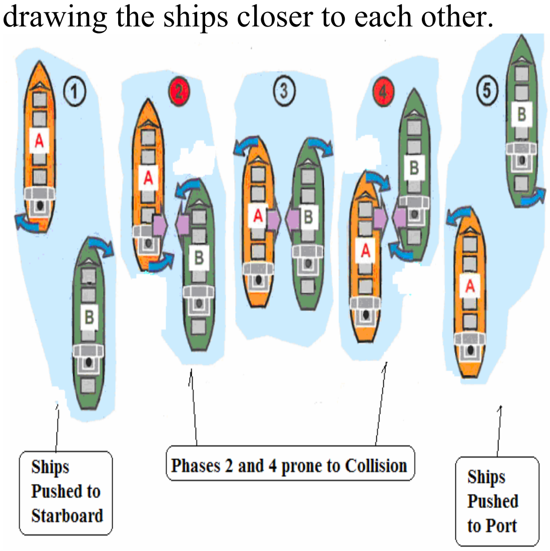

Fig. 4: Interaction in Overtaking

The relative strength of the interaction varies also with the type of hulls involved and their drafts. High block coefficient vessels have in general bluff bows and finer aft ends. Accidents are thus more likely to occur on a smaller vessel like a tugboat at the bow, (Dand 1975). The safest area for a tug is the mid-ship region of a tanker/bulk carrier. It is noted that the attraction force at the bow of a tanker might even lead to the capsize of the tugboat, since it might be manifested to heeling moment as well. Notably, squat is also amplified due to the presence of the other vessel. The amplification is higher at lower ship speeds. Accidents can be avoided if crossing and overtaking takes place at lower speeds and at wider canal sections with the appropriate rudder helm. Of special interest is also the case of the interaction between a moving ship and a stationary one. Accidents on moored ships due to the passing of other vessels will be the focus of the present work.

2. Moving-Stationary Ship Interaction

2.1 Nature of Interaction Forces

Similarly to the case of moving ships, the moored ship starts to feel the effects of the passing ship when the approaching vessel is about two ship lengths away. Typically, the moored vessel will initially feel a repulsive force forward, which will cause it to surge forward and the stern to move away from the dock. The moored vessel is also pushed towards the dock. Evidently, both the spring and the head mooring lines forward will be tried at that time. As the approaching vessels come to about three quarter lengths away from the moored vessel, the surge force becomes attractive and the yawing moment is pulling it aft. When the two vessels face each other the sway force reaches its attractive peak and it dominates the interaction. The sequence

is reversed as the moving ship continues to pass. It is assumed that the speed of the approaching ship is not high enough to generate waves.
Major parameters needed to analyze such type of ship-to-ship interaction are the velocity of the passing ship and the lateral separation distance between the two ships. The velocity of the passing ship can be bounded. A lower bound will be the minimum steerage velocity, whereas an upper bound will be the ship velocity resulting to grounding due to squat. Of course other factors influencing the under-keel clearance of the vessel, such as water density, tides, ship motions, trim and static vessel deflections need to be taken into account as well. Estimation of the possible magnitude of the forces and moment on a moored vessel can be used for the design of a structurally adequate mooring system for the ship and the dock.

2.2 Model of Hydrodynamic Interaction Forces

A simplified and yet accurate enough method is presented, which can be utilized to compute the unsteady sway, surge forces and yaw moments acting on the mooring lines of the stationary ship, (Wang 1975). The force and moment estimation is based on the Lagally theorem (Cummins 1954) and is amenable to quick computation, with the possibility to incorporate shallow water effects. Incorporation of shallow water effects in this case is a "sine qua non" condition, given that such types of accidents typically occur in shallow waters.
The assumption of wave-free surface allows to neglect free surface effects. Then the method of "double-body" flow can be applied. The problem may be considered as that of two double ships with the two slender ship hulls and their images in the free surface. The two ships are separated by a constant lateral distance **n**. Their time varying fore-aft separation is called stagger and it is denoted by ξ as depicted on Fig. 5. It is noted that the equations have been derived assuming that ξ is positive when the moving ship is astern of the moored ship (i.e. shown negative on Fig. 5).

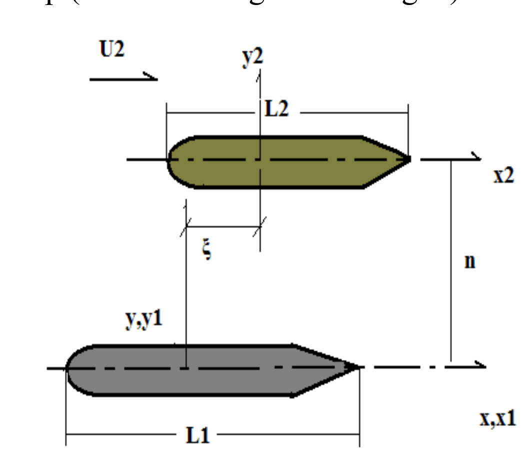

Fig 5: Coordinate Systems

Two body-fixed coordinate systems are defined on Fig. 5. One on the stationary/moored ship 1 (x_1,y_1,z_1) and one on the moving ship 2 (x_2,y_2,z_2). The global (x,y,z) coordinate system is fixed on the moored ship, since our interest is focused on the computation of sway, surge forces and yaw moment on the stationary ship.
The velocity potential φ needs to satisfy the governing Laplace's equation. In addition, the velocity potential φ has to satisfy kinematic boundary conditions on both ships. If n1 and n2 are the unit normal vectors on ship 1 (stationary) and 2 (moving) respectively, then:

$$\frac{\vartheta\phi}{\vartheta n_1} = 0 \qquad (1)$$

$$\frac{\vartheta\phi}{\vartheta n_2} = U_2 \frac{\vartheta x_2}{\vartheta n_2} \qquad (2)$$

U_2 is the speed of the moving ship. It is assumed that the total potential for the flow around the moored ship consists of the potential φ_2 due to the moving ship, which satisfies Eq. (2) and an interaction potential φ_1 , which satisfies Eq. (1). The first potential for a slender body in a uniform axial stream of velocity $-U_2$, relative to the body, can be emulated by a distribution of doublets oriented along the x_2 axis. The doublet strength is given by Munk formula:

$$\mu_2(x_2) = -\frac{1}{2\pi} S_2(x_2) U_2 \qquad (3)$$

Where $S_2(x_2)$ is the wetted cross sectional area of the moving ship. The resulting potential φ_2 expressed on the global system is given by:

$$\phi_2(x,y,z,\xi) = -\frac{U_2}{2\pi} \int_{L_2} \frac{S_2(x_2)(x - x_2 + \xi)dx_2}{[(x - x_2 + \xi)^2 + (y-n)^2 + z^2]^{1..5}} \qquad (4)$$

It is noted that the integration is carried along the length, L_2 of ship 2. The corresponding axial velocity U and cross flow velocity V on ship 1 (y=z=0) are given by the following equations:

$$U(x_1,\xi) = \frac{\vartheta\phi_2}{\vartheta x} = \frac{U_2}{2\pi} \int_{L_2} \frac{S_2'(x_2)(x_2 - x_1 - \xi)dx_2}{[(x_2 - x_1 - \xi)^2 + n^2]^{1..5}} \qquad (5)$$

$$V(x_1,\xi) = \frac{\vartheta\phi_2}{\vartheta y} = \frac{U_2 n}{2\pi} \int_{L_2} \frac{S_2'(x_2)dx_2}{[(x_2 - x_1 - \xi)^2 + n^2]^{1..5}} \qquad (6)$$

The prime in the relations above indicates the gradient of the wetted cross sectional area variation along the length L_2 for the moving ship. Potential $\varphi 1$, satisfying Eq. (1) can be emulated also by a doublet distribution due to ship1 being in uniform streams U and V. The corresponding doublet strengths according to the Munk formula are:

$$\mu_{1x}(x_1,\xi) = \frac{1}{2\pi} S_1(x_1) U(x_1,\xi) \qquad (7)$$

$$\mu_{1y}(x_1,\xi) = \frac{1}{\pi} S_1(x_1) V(x_1,\xi) \qquad (8)$$

Instead of utilization of the unsteady Bernoulli equation to obtain pressure and force distributions, the singularity distributions given above are utilized in conjunction with the unsteady state form of the Lagally theorem, (Cummins 1954) to compute the forces on the moored ship as follows:

$$dX(x_1,\xi) = 2\pi\rho \frac{\vartheta\mu_{1x}}{\vartheta t} dx_1 \qquad (9)$$

$$dY(x_1,\xi) = 2\pi\rho \frac{\vartheta\mu_{1y}}{\vartheta t}]dx_1 \qquad (10)$$

Where ρ is the density of the fluid. The time variance is introduced by the rate of change of momentum due to the singularities distributed on the moored ship. Employment of Eqs. (7) and (8) and integrating along the length L_1 of the moored ship we obtain for the total surge and sway forces:

$$X = \frac{\rho U_2^2}{2\pi} \int_{L_1} S_1'(x_1) \int_{L_2} \frac{S_2'(x_2)(x_2 - x_1 - \xi)dx_2}{[(x_2 - x_1 - \xi)^2 + n^2]^{1..5}} dx_1 \qquad (11)$$

$$Y = \frac{\rho U_2^2 n}{\pi} \int_{L_1} S_1'(x_1) \int_{L_2} \frac{S_2'(x_2)dx_2}{[(x_2 - x_1 - \xi)^2 + n^2]^{1..5}} dx_1 \qquad (12)$$

Similarly for the yaw moment:

$$N = \frac{\rho U_2^2 n}{\pi} \int_{L_1} [S_1'(x_1)x_1 + S_1(x_1)] \int_{L_2} \frac{S_2'(x_2)dx_2}{[(x_2 - x_1 - \xi)^2 + n^2]^{1..5}} dx_1 \qquad (13)$$

2.3 Shallow Water Effects

If the water depth, h, is less than about two times the vessel draft, then shallow water effects need to be considered. The problem is augmented by the addition of two planes at $z = \pm h$, where the kinematic boundary condition as in Eq. (1) needs to be satisfied by the velocity potential. This is achieved by the placement of an infinite series of singularities at $z = \pm 2\nu h$ (where ν = 1,2,…∞). The additional contribution due these image singularities can be accounted for if in the previous formulas the separation distance is replaced by the following relation:

$$n_\nu = (n^2 + 4\nu^2 h^2)^{0.5} \quad (14)$$

The shallow water forces can then be determined via infinite series summation from the deep water terms as defined by Eq. (14) above. The corresponding expressions are:

$$X(\xi, n, h) = \sum_{\nu=-\infty}^{\infty} X(\xi, n_\nu, \infty) \quad \textit{surge} \quad (15)$$

$$Y(\xi, n, h) = \sum_{\nu=-\infty}^{\infty} \frac{Y(\xi.n_\nu, \infty)}{n_\nu} \quad \textit{sway} \quad (16)$$

$$N(\xi, n, h) = n \sum_{\nu=-\infty}^{\infty} \frac{N(\xi.n_\nu, \infty)}{n_\nu} \quad \textit{yaw} \quad (17)$$

The numerical implementation of the mathematical model is straightforward. In case there are no data for the cross sectional areas, parabolic distribution with fore-aft symmetry can be assumed (Wang 1975). The error stemming from this approximation is almost negligible in the computation of the surge force, significant for the sway force and very significant in the estimation of yaw moment (Krishnankutty and Varyani 2004). In general, the cross section distribution of any vessel can be approximated by fitting a least square piece-wise parabolic curve. It is advisable to determine a piece-wise parabola in order to improve accuracy.
Numerical computations in (Wang 1975) with the parabolic cross sectional area and fore-aft symmetry lead to the conclusion that the magnitude of interaction is amplified significantly in s\finite water depth, the amplification being as high as 6 times the deep water values. Further interaction effects increase exponentially as the separation distance between the two ships decreases. The ratio of the passing to the moored ship lengths increases interaction effects as it increases. Evidently, if the moored ship is of smaller size, interaction effects will be more pronounced.

2.4 Moored Ship Dynamics

The critical question is if the mooring system of the stationary vessel is adequate under the action of the unsteady sway, surge and yaw moment from the passing ship. A typical mooring system is depicted in Figure 6.

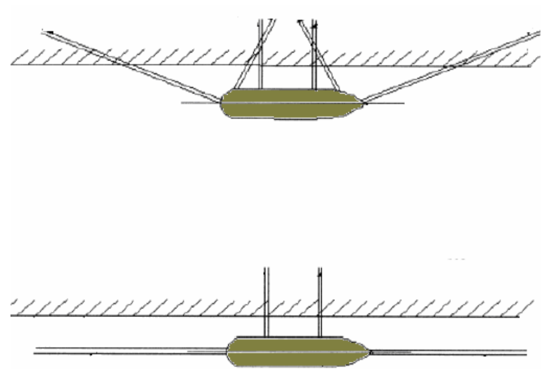

Fig. 6: Mooring System

As it shown above, the mooring system consists of lines, which can resist the surge, sway and yaw movements of the vessel. The fore-to-aft lines are mostly resisting surge and the lateral ones the sway and yaw movements. The later are called breast lines. The dynamics of this typical system are coupled in surge, sway and yaw through the stiffness of the mooring lines. In the coupled dynamics the elastic force on the mooring line is given by:

$$F_M = k\Delta l(x, y, \varphi) \quad (18)$$

Where k is the stiffness of the mooring line and Δl is the elongation of the mooring line. The later is a nonlinear function of the ship movements, namely surge x, sway y and yaw ψ. The second order coupled equations of motion can be easily solved by various numerical techniques, like for example a Runge-Kutta integrator.

The mooring system can be simplified significantly as depicted in the lower part of Fig. 6 where the head and stern lines are taken parallel to the x axis and the breast lines parallel to the y axis. Although this is a significant simplification, valuable conclusions can be drawn on the qualitative merit of the stiffness of the mooring lines. The system will be governed by the following uncoupled second order equations:

$$(m + m_x)\ddot{x} + C_x \dot{x} + K_x x = X(t) \tag{19}$$

$$(m + m_y)\ddot{y} + C_y \dot{y} + K_y y = Y(t) \tag{20}$$

$$(I_z + I_\varphi)\ddot{\psi} + C_\psi \psi + K_\psi \psi = N(t) \tag{21}$$

Where m is the mass of the ship, I_z its mass moment of inertia, m_x, m_y , I_ψ are added mass coefficients, C_x, C_y, C_ψ are damping coefficients and K_x, K_y, K_ψ are the stiffnesses of the mooring lines. The following relations are also valid:

$$C_\psi = \delta^2 C_y \tag{22}$$

$$K_\psi = \delta^2 K_y \tag{23}$$

Where δ is the distance between the aft and the forward breast lines. The lateral mooring forces in these lines are different due to yaw. Numerical results for various stiffnesses have been presented by (Remery 1974) and (Krishnankutty and Varyani 2004). The results prove that the mooring line force can be higher than the interaction force, due to the moored ship dynamics. The results presented indicate amplification up to 50%. As a rule of thumb rigid mooring lines are preferred than soft ones. It is unclear though where is the optimum stiffness, i.e. the one resulting to acceptable excursion of the ship and acceptable mooring line force. The results of uncoupled mooring dynamics follow a resonance type behavior. As the line stiffness decreases the mooring line force increases, but beyond a certain peak it starts to decrease again.

3. Case Studies

3.1 Tanker "Jupiter"

On Sunday, September 16, 1990, the 120-meter tanker "Jupiter" was moored at a petroleum terminal on the Saginaw River in Michigan, discharging a cargo of unleaded gasoline. While the "Jupiter" lay moored the 195-meter long bulk carrier "Buffalo" entered the river en route to discharge a cargo of coal. As the "Buffalo" passed the "Jupiter", the tanker broke away from its berth and its stern swung out into the river, rupturing the discharge hose to the pier and damaging the pipeline on the pier. Gasoline spilled on the pier and onto the deck of the "Jupiter". The electrical cables to two motor-operated valves that closed off the pipelines at the end of the pier were torn apart, causing sparks that ignited the spilled gasoline. Fire spread to the deck of "Jupiter", causing a series of explosions in the cargo tanks that destroyed the entire mid-ship section of the vessel. One crewmember died during the abandonment of the vessel and the "Jupiter" was declared a total loss. Her condition shortly after the accident is depicted on Figs 7 and 8.

Fire was spread because the manifold valve and the ullage covers were open. The burning gasoline spread throughout the mid-ship area and around the open ullage pipes, propagating the fire into the tanks through loosely fitted flame screens. The “Jupiter” was moored with six lines; namely four wire rope cables attached to four winches and two polypropylene lines. The two wire lines numbered 2 and 3 resisted surge and run parallel to the ship center-plane, acting as spring lines. One was forward in the bow (No 2) and the other (No 3) aft in the stern. Forward and aft two polypropylene lines acted essentially as breast lines. The forward line was slightly forward on the ship and the aft polypropylene line was aft. No 1 and No 4 wire cables were also acting as breast lines. Placing both bow lines and both stern lines to single mooring points does not provide enough security for the vessel if one of the mooring devices fails.

Fig 7: Tanker “Jupiter” after Accident

Fig 8: Tanker “Jupiter” after Accident

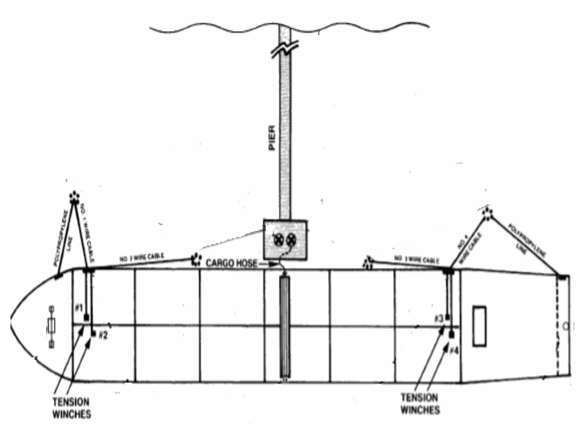

Figure 9: “Jupiter” Mooring

Eyewitness testimony was that initially the “Jupiter” surged forward as is typically the case in hydrodynamic interactions. Further as the “Buffalo”, which is depicted on Fig. 10, passed abeam of “Jupiter”, the tanker surged forward so that the hindmost mooring wire (number 4) tautened. The number four wire was moored the center piling (the “kings-pile”) on the number four cluster of pilings. A typical cluster with a kings-pile is shown in Fig. 11.

Fig. 10: Bulk Carrier ""Buffalo""

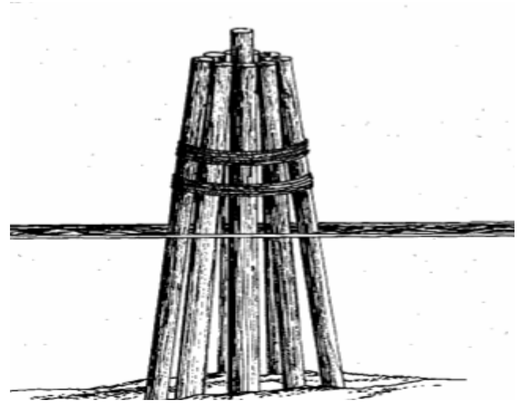

Fig. 11: Pile Cluster with Kings-pile.

When the number four wire grew too taut, the number four kings-pile snapped, releasing both the number four wire and one of the polypropylene lines, leaving the "Jupiter" without any moorings at its stern. The stern swung out away from the dock, stretching the cargo hose until it also broke, spilling its contents (120-300 gallons of gasoline) on the pier and on the deck providing the "fuel" for the subsequent combustion. At the same time, the electric cord for the motor-operated valve, stretching from the wharf to the ship, broke and spewed sparks. The "fire triangle" was complete. The pier's number four kings-pile, which broke was 90-95% rotten. The "Jupiter" had inadequate flame screens. The "Buffalo" was going too fast or too close to the "Jupiter". A cluster of synergistic responsibilities, which led to the accident. Eventually the responsibility was split equally between the tanker, the bulk carrier and the petroleum terminal.

The sequence of events viewed from the hydrodynamic standpoint is depicted on Figure 12 and it consists of six distinct phases. Phase A, where there is no interaction. The moving ship is about two ship lengths away. In phase B, there is a low surge force forward and a low bow out moment. Cables 2 and 4 resist the forward force. In phase C, the maximum surge force with astern direction occurs. A significant bow in (towards the pier) is also exerted on the "Jupiter". The sway force is low. Unfortunately only cable No 3 can provide resistance to the aft surge. Testimony revealed that winch brake was released and presumably paid out the cable to relieve the stress. Evidently in aft movement cables 2 and 4 become slack. Phase D with the two ships abeam, is the phase dominated by the lateral attraction, which reaches its maximum. In phase E, "Jupiter" feels the maximum surge forward force and a significant bow out moment. Only No 2 mooring cable can resist the forward force. Tension from cable No 2 caused the stern to swing away from the pier. Phase F is characterized by low interaction. In this phase a low repulsive sway force will be exerted along with low surge astern and low yaw moment with bow action. As the Buffalo's stern cleared the bow of the "Jupiter", the bow was forced toward the pier. When No 4 mooring cable became taut, the impulsive loading on No 4 pile broke the kings-pile. Interestingly, the failure phase, which left the "Jupiter" to swing unchecked into the river, happened at the last phase of hydrodynamic interaction. This is due to the combined effect of phase E and F on the bow in movement.

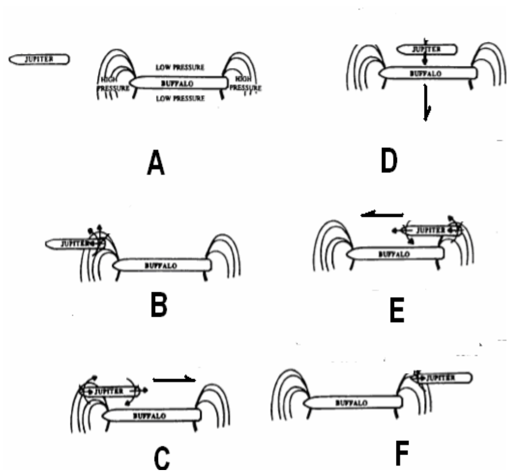

Fig. 12: Hydrodynamic Interaction Phases

As a parting point, the master of the "Buffalo" testified that he was forced to come to a distance of about 18 meters from the "Jupiter". This distance is one beam of the "Buffalo" and is considered a very low separation distance. Amplification effects from shallow water augmented the interaction. The master of "Buffalo" justified his coming close to the "Jupiter", to a wind on his starboard side threatening to ground him on the port shore of the channel. It is interesting to note that even at this low separation distance, he was traversing the channel very close to the center.

3.2 Tanker "Proteus"

A medium sized tanker (M/T "Proteus") was moored at a terminal on the Savannah River just upriver from downtown Savannah, Georgia discharging its cargo. The tanker was oriented with its bow facing upriver. She was secured to the terminal dock with 14 mooring lines. Three of these ropes were used to secure the ship's bow to a dock bollard. Tension in the ropes varied with changes in the tide as well as reduced draft as the vessel off-loaded.

Fig. 13: Damaged Dock Terminal

Crewmembers claimed to have periodically adjusted the tension so that appropriate tension was maintained in each of the lines. A second ship, a container ship, (ship 2) slowly motored upriver at mid-channel outboard of the tanker's starboard side. At the time the container ship passed the tanker, the three bow lines from the tanker pulled the bollard and supporting concrete structure away from the rest of the dock as shown in Fig. 13. A portion of the dock subsequently collapsed and the bollard/concrete structure went to the river bottom just forward of the tanker's port bow still attached to the bow lines. At the time of the incident, a third ship (ship 3) also a container ship, was motoring downriver and was approximately 350 meters ahead of the tanker's bow. The tanker did not incur any damage; the off-loading was temporarily suspended. The position of the three ships is shown schematically in Fig. 14.

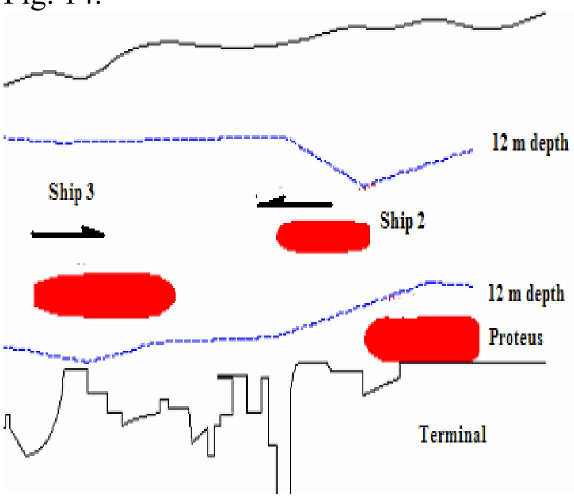

Fig. 14 Ship Position at Accident Time

Notably the tanker had an overall length of 192 meters and a draft of 7 meters at the time of the accident. The container ship had an overall length of 154 meters and a draft at the time of the accident equal to 8 meters. In this case the speed of the container ship was well defined. Its passage was monitored from the security video camera onboard the tanker. It was estimated that the container ship passed at a speed of 4.8 knots. Similarly the third ship passed at a speed of 4.6 knots. Furthermore the separation distance was about 60 meters or three times the beam of the container ship. Given the facts that the speed and the separation distance were low, it was felt that interaction effects might not be significant. Another critical point is that the passing ship was smaller than the moored vessel. It is thus expected that even at shallow water (12 meters depth), the interaction effects would not have been significant. On the other hand the temporal coincidence of the bollard/dock failure at the time of container ship passing was very tempting. It was decided to perform an interaction force calculation and depending on the computed results to proceed with the mooring system dynamic analysis.

The cross sectional area distributions were determined from ships of similar size. Fig. 15 depicts such distribution for "Proteus" (for draft = 7 meters).

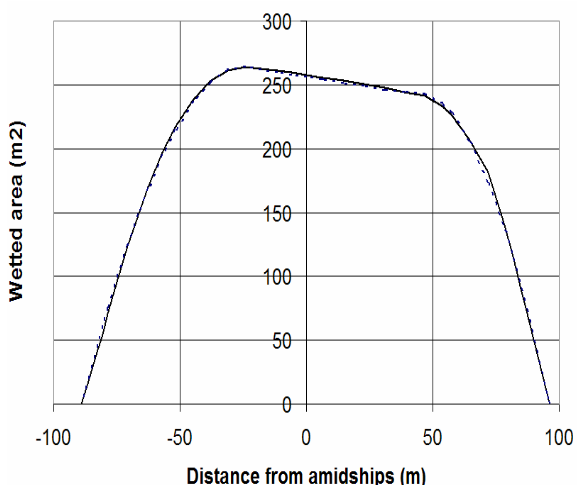

Fig. 15: Wetted Cross Area Distribution

Fig. 15 depicts the parabolic curve fit superimposed on the actual distribution. Evidently the fit is very good. Notably the fit was accomplished in three parts; a linear one (constant slope) representing the parallel mid-body and two parabolic ones for the aft and fore regions. At the ends the wetted cross section is taken equal to zero since in the mathematical model there is not a Kutta condition to account for end effects like for example vortex shedding from the rudder.

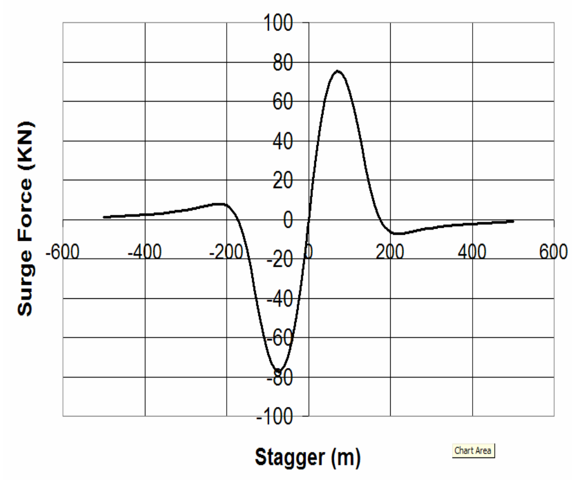

Fig. 16: Surge Force on Tanker

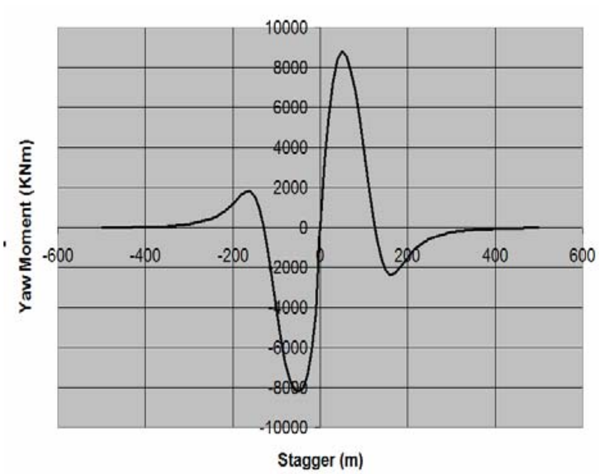

Fig. 17: Yaw Moment on Tanker

Hydrodynamic interaction forces are depicted on Figs 16-18. The resulting forces and moments are relatively very low. It was thus decided that a detailed mooring analysis with input from the interaction study was not needed.

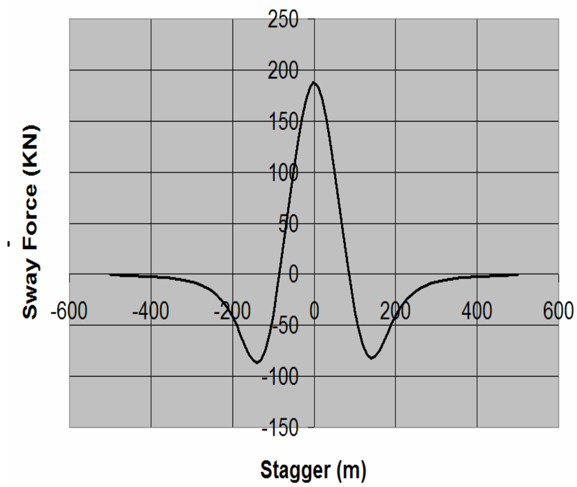

Fig. 18: Sway Force on Tanker

It was thus concluded that there was no single cause of the bollard/dock collapse but it resulted because of the cumulative effects of line pre-tension, draft increase, passing ship, current, and limited capacity of the bollard/dock.

4. Conclusions

Our work refers to a wide array of problems categorized as "hydrodynamic interaction". They are known and have been the primary causal factor in numerous accidents. Focus was concentrated on the particular case of the effects of a moving onto a stationary ship. A mathematical model amenable to quick computation was presented along with two related case studies from the authors' involvement in marine accident investigation. It is our belief that even-though the techniques presented in the current study are some-what approximate, they constitute valuable tools in the study of marine accidents involving interactions. Further refinements such as coupling of the mooring system dynamics in all degrees of freedom, can provide valuable design tools for dock terminals as well as shallow water water-ways.

References

Marine Safety Agency (1998) Guidance Note, “Dangers of Interaction”,

Barrass C.B. (2006). “Ship Stability for Masters and Mates”, Elsevier Publishers

Barrass C.B. (1979). “A Unified Approach to Squat Calculations for Ships”, Bulletin, Permanent International Association of Navigation Congresses, Vol. 1.

Beck R.F (1977). “Forces and Moments on a Ship Moving in a Shallow Channel”, Journal of Ship Research, Vol 21, No 2, pp 107-119.

Bruce M.J., Fang S., (1975). “Passing Ship Effects-From Theory and Experiment”, Offshore Technology Conference, Paper No 2368.

Ch’ng P.W. (1991).“An Investigation into the Influence of Bank Effect on Ship Maneuvering and its Mathematical Modeling for a Shiphandling Simulator”, MSE Thesis, University of New South Wales, Australia.

Clark I.C. (2005). “Ship Dynamics for Mariners”, The Nautical Institute, London .

Cohen S.B., Beck R.F. (1983). “Experimental and Theoretical Hydrodynamic Forces on a Mathematical Model in Confined Waters”, Journal of Ship Research, Vol 27, No 2 , pp 75-89.

Cummins W.E. (1954). “Hydrodynamic Forces and Moments Acting on A slender Body of Revolution Moving Under a Regular Train of Waves”, Report 910, David Taylor Model Basin.

Dand I.W., Ferguson A.M. (1973). “The Squat in Shallow Water”, The Naval Architect, No 4, pp 237-255.

Dand I.W. (1975).“Hydrodynamic Interaction Between a Tug and a Ship”, Safety at Sea International, April.

Krishnankutty P., Varyani K.S. (2004). “Force on the Mooring Lines of a Ship due to the Hydrodynamic Interaction Effects of a Passing Ship”, International Shipbuilding Progress, 51, No 1, pp 33-57.

Marine Safety Agency (1998) Guidance Note, “Dangers of Interaction”.

Remery G.F.M. (1974). “Mooring Forces Induced by Passing Ships”, Offshore Technology Conference, OTC Paper 2066.

Tuck E.O., Taylor P.J. (1970). “Shallow Water Problems in Ship Hydrodynamics “, Proceedings 8th Symposium on Naval Hydrodynamics, Pasadena, California .

Tuck E.O., Newman J.N. (1974). “Hydrodynamic Interactions Between Ships”, Proceedings of the 10th Symposium on Naval Hydrodynamics, ONR, pp 35-70.

Yeung R. W. (1978). “ On the interactions of slender ships in shallow water”, Journal of Fluid Mechanics, vol. 85, pp 143-159.

Yeung R. W., Wooi T.T. (1980). “ Hydrodynamic Interactions of Ships with Fixed Obstacles”, Journal of Ship Research, Vol 24, No 1, pp 50-59.

Wang S. (1975) “Dynamic Effects of Ship Passage on Moored Vessels”, Journal of Waterways, Harbors and Coastal Engineering Division, ASCE, No WW3.