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Modular Mathematical Model for a Low-Speed Maneuvering Simulator

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ABSTRACT

This paper presents the mathematical model of the real-time ship simulator for low-speed maneuvering developed by the University of São Paulo, Transpetro and Petrobras, with the technical collaboration of Brazilian Pilots Association (CONAPRA). The software is based on the TPN (Numerical Offshore Tank) numerical code which had several modifications, in order to perform real-time simulations.

After the complete description of the mathematical model, some illustrative results of simulations executed with pilots are exposed.

KEYWORDS

Real Time Simulation, Maneuvering.

INTRODUCTION

Ship maneuvering simulators are used for predicting the navigation safety in restricted areas (ports and channels) and training. The following paragraph, taken from Webster (1992), summarizes the concepts to be studied. "A limited number of simulations using a less-than-perfect simulator, a few select (design) ship types, a few select environmental conditions over extreme ranges characteristic of the local area, and a few pilots with representative local expertise and shiphandling proficiency are sufficient to obtain a useful appraisal of waterway design..."

Some points should be stressed in this statement, which illustrate concisely the benefits and proper way of using simulators. The simulators are never perfect, since mathematical models are simplifications of reality. Therefore, they must always be used in conjunction with the knowledge and experience of local conditions. The use of standards and recommendations (such as PIANC, USACE, ROM, etc.) must also accompany the port studies, corroborating and/or discussing the results of the simulations.

In a recent initiative, Transpetro and Petrobras established a research partnership with the Numerical Offshore Tank Laboratory of the University of São Paulo (TPN-USP) for the development of a maneuvering simulator for port, rivers and offshore operations, with the technical collaboration of Brazilian Pilots Association (CONAPRA). The simulator is named SMH (Portuguese acronym for Maritime and Waterways Simulator) and has two different functions. It is part of the human resources policy of the companies, since the simulator software and the hardware architecture is the basis of the Training Center under construction that will promote the training of their crew. Furthermore, it is an engineering tool that is used for the analysis of several new operations in the Brazilian ports in order to improve their efficiency, due to the increasing oil and gas production and commercialization.

In order to be used as the basis for the Training Center, the simulator was developed in a flexible architecture, and different versions of hardware can be used. For example, it can be configured in a single-station 2-screens computer, that will be used in a class-room for DP and basic maneuvering training. Computer solutions with 3 or 6 visualization screens can also be used for more advanced training. Finally, a full-bridge setup (Figure 1) is also used for an immersive training of maneuvering and procedures.

Figure 1 TPN-USP maneuvering full-mission simulator

The accuracy of the mathematical model is an important requirement for a training simulator, but even more important for an engineering tool. The engineers, pilots and captains must rely on the dynamics of the simulator, since it will be used to give answers to questions related to maneuvering analysis. The simulator can be used to evaluate new channels design, tugboats requirements, environmental window, DP system analyses, or even to define the maximum dimensions of vessels in an approach channel or basin, among others questions.

This paper presents the mathematical model of SMH, adequate for low-speed maneuvering. The software is based on the TPN (Numerical Offshore Tank) numerical code (Nishimoto et al., 2002) which had several modifications, in order to perform real-time simulations. The present simulator applies a Modular Mathematical Model, gathering a large number of models to represent the complete behavior of a ship during the navigation in restricted waters. Other time-domain maneuvering simulators are presented by Ankudinov et al. (1993), Koh et al. (2008), Fossen and Smodeli (2004), among others.

Several Brazilian ports and rivers have already been studied and, in cooperation with pilots and captains, the simulator has been constantly improved. The feedback from the pilots and captains are used to perform a "fine-tuning" of the maneuvering models and environmental conditions in each port. This synergy between the theoretical background and field expertise has proven to be very effective. Some illustrative results of simulations and important results are exposed.

FLOATING BODY DYNAMICS

The mathematical models required to represent the motion of a vessel at low speed are presented in this section. The 6DOF vessel dynamics differential equations are solved using 4rd order Runge-Kutta integration method, considering the interaction with the fluid and the external forces acting on the hull. However, as a matter of simplicity, this paper will only present the equations of motion for the horizontal plane. A complete description of the equations of motions in the simulator is given by Fucatu (2010).

In order to study the ship motions, it is useful to adopt two different coordinate systems as shown in Figure 2. The system *OXYZ* is fixed on the earth (inertial system) and the system *oxyz* is fixed on the ship with the origin on central point of the keel midship section. The center of gravity G is located at the distance x_G ahead from the point *o*, *ox* is the longitudinal axis of the vessel directed to the bow, and *oy* is the transversal axis, pointing to port. The heading of the vessel ψ defines the angle between the *ox* and *OX* axes.

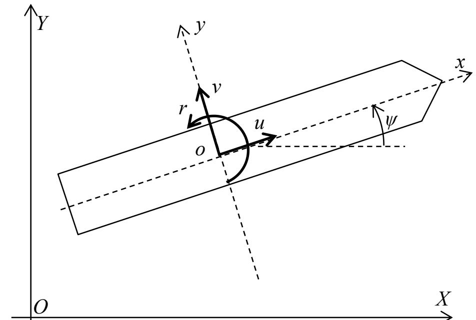

Figure 2 Coordinate system and basic definitions

Applying Newton's law of the motion, the following differential equations describing the relationships between the motion variables, referred to the *oxyz* system, and the external loads are obtained:

$$\begin{aligned} M(\dot{u} - rv - x_G r^2) &= X_{pot} + X_{ext} \\ M(\dot{v} + ru + x_G \dot{r}) &= Y_{pot} + Y_{ext} \\ I_Z \dot{r} + M x_G(\dot{v} + ru) &= N_{pot} + N_{ext} \end{aligned} \qquad (1)$$

where M is the mass and I_z is the yaw moment of inertia of the ship, u and v are, respectively, the surge and sway velocities, r is the yaw angular velocity; the subscript *ext* represents the external loads and the subscript *pot* is related to the potential forces and moment action due to water-hull interaction. The potential parts are evaluated by means of added-mass and potential damping matrixes. For the horizontal motions, the potential damping is negligible, and if the vessel is symmetrical along *ox*, the potential forces are simplified and the equation of motion can be written as:

$$\begin{aligned} &(M + M_{11})\dot{u} - (M + M_{22})vr - (Mx_G + M_{26})r^2 = X_{ext} \\ &(M + M_{22})\dot{v} + (Mx_G + M_{26})\dot{r} + (M + M_{11})ur = Y_{ext} \\ &(I_Z + M_{66})\dot{r} + (Mx_G + M_{26})(\dot{v} + ur) + (M_{22} - M_{11})uv = N_{ext} \end{aligned} \qquad (2)$$

where M_{11} and M_{22} are the ship added masses in the surge and sway directions, respectively, M_{66} is the ship added moment of inertia, M_{26} is coupled sway-yaw added inertia. The last term on the right side of the yaw equation is the Munk's moment.

The external forces and moment may be expressed in terms of different factors:

$$X_{ext} = X_h + X_w + X_{wv} + X_p + X_{tug} \qquad (3)$$

where X_h represents the hydrodynamic non-potential forces, including the current and maneuvering forces, X_w , X_{wv} represent the wind and wave forces, respectively X_p represents the thrusters, propeller and rudder forces and X_{tug} represents the external action of the tug boats, either in contact with the hull or connected by a cable.

HYDRODYNAMIC NON-POTENTIAL FORCES

The non-potential hydrodynamic forces, including current and maneuvering forces, are estimated by the Cross Flow model proposed by Obokata (1987), with modifications for taking into account a non-uniform current field (Fucatu and Nishimoto, 2004).

The model considers a current superficial velocity field $\vec{V}_C(X,Y)$ defined for each point (X,Y) of the water surface. The model is based on the integration of the forces acting on each section of the vessel (Figure 3). The relative speed of the section x meters ahead from the midship of the vessel related to the water ($\vec{V}_{crx}$) is given by (4). The angle ψ_{crx} that defines the relative incidence of the water at each section of the vessel is given by (5). The midship relative velocity and angle are obtained by making x = 0 and are identified by $\vec{V}_{cr}$ and ψ_{cr} respectively.

$$\vec{V}_{crx} = \begin{pmatrix} u_{rx} \\ v_{rx} \end{pmatrix} = \begin{pmatrix} u \\ v + rx \end{pmatrix} - \\ -\vec{V}_C\left(X_O + x.\cos\psi, Y_O + x.\sin\psi\right).\begin{pmatrix} \cos\psi & -\sin\psi \\ \sin\psi & \cos\psi \end{pmatrix} \tag{4}$$

$$\psi_{crx} = \pi + \arctan(v_{rx} / u_{rx}) \tag{5}$$

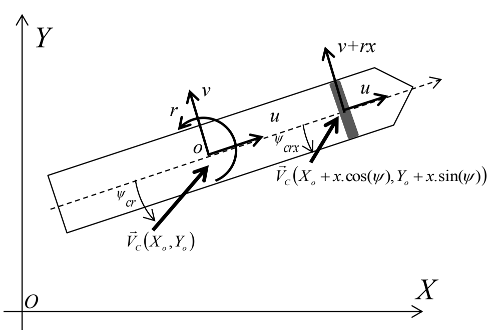

Figure 3 Basic definitions for maneuvering and current forces calculation

The forces X_h and Y_h and the yaw moment N_h are given by the sectional integration (6) along the longitudinal vessel axis, that defines the dependence on vessel yaw speed, as well as a non-uniform current field normally induced by breakwaters near sheltered areas:

$$\begin{aligned} X_h &= \frac{1}{2}\rho T \int_{-L/2}^{L/2} C_x(\psi_{crx}) V_{crx}{}^2 dx \\ Y_h &= \frac{1}{2}\rho T \int_{-L/2}^{L/2} C_y(\psi_{crx}) V_{crx}{}^2 dx \\ N_h &= \frac{1}{2}\rho T \int_{-L/2}^{L/2} \left(C_y(\psi_{crx}) V_{crx}{}^2 - C_y(\psi_{cr}) V_{cr}{}^2\right) x dx \\ &\quad - \frac{1}{2}\rho T L^2 C_z(\psi_{cr}) V_{cr}{}^2 \end{aligned} \tag{6}$$

where ρ is the water density , L is the ship length, T is the draft, C_x and C_y are the current forces coefficients in the *ox* and *oy* directions, respectively, and C_z is the yaw moment coefficient. Those coefficients are obtained by captive towing-tank tests or CFD calculation, for both shallow and deep waters. For tankers with standard hull shapes, the database provided by OCIMF may be used.

With the previous model, the equivalent linear hydrodynamic derivatives can be obtained from the present model. Using the definitions given in ABS (2006), one can show that $Y_r = 0$ and:

$$Y_v = -\frac{T}{L}\frac{dC_y}{d\psi}\bigg|_{\psi=180} \ ; \ N_v = -\frac{T}{L}\frac{dC_z}{d\psi}\bigg|_{\psi=180} \ ; \ N_r = -\frac{T}{12L}\frac{dC_y}{d\psi}\bigg|_{\psi=180}$$

The sway force and yaw moment calculation in (6) assumes that the sectional drag coefficient is constant for the entire hull. This model can be refined if the sectional drag coefficient is estimated or measured (see Simos et al., 2001). In that case, the equivalent linear hydrodynamic derivatives would be different and could represent each vessel more accurately.

It is also possible to define time-changing and oscillatory current profile in the TPN maneuvering simulator.

WIND FORCES

Wind forces acting on the ship hull are modelled using traditional drag force formulation whose coefficients are based on model tests, CFD calculation or OCIMF database curves (OCIMF, 1994). Assuming that there is no spatial variation in speed and direction of wind incident on the vessel, the following relationships for the horizontal loads are assumed:

$$X_w = \frac{1}{2}\rho_a C_{Vx}(\beta_V) A_F V_w^2 \ ; \ Y_w = \frac{1}{2}\rho_a C_{Vy}(\beta_V) A_L V_w^2 \\ N_w = \frac{1}{2}\rho_a C_{Vz}(\beta_V) L A_L V_w^2 \tag{7}$$

where ρ_a is the density of air, C_{Vx}, C_{Vy} and C_{Vz} are dimensionless wind coefficients, A_F and A_L are the emerged frontal and lateral area of the vessel and V_w is the wind relative speed (considering the velocity of the hull). The angle of incidence with respect to the body axis ox is given by β_V. The heave force due to wind is not considered. The application points are the centroids of the wind areas, and their vertical positions will define the heel and trim moments induced by wind.

The simulator allows constant or gusty wind. The wind spectra implemented in the code are Harris, Wills, API and NPD, among others.

WAVE FORCES

The wave forces on the ship hull are evaluated considering separately the second order drift forces (mean and slow varying drift forces) and the first order high frequency forces. The wave forces are obtained by the potential theory, considering zero advance speed for defining the drift coefficients and first-order Hasking Forces. This approximation is valid for low-speed maneuvering. The simulator is able to import the hydrodynamic coefficients from different commercial codes, such as Wamit, Wadam, Hydrostar or AQWA. In the case of shallow waters, the bottom must be taken into account.

Different wave spectrum formulation may be adopted, for both short or long crested waves. It is possible to define either regular waves or irregular sea state, with Pierson-Moskowitz, JONSWAP or Gaussian spectra formulations. Being T_p and $\omega_o = 2\pi / T_p$ the peak period and frequency respectively, the wave spectrum can be written as:

$$S(\omega) = \frac{\alpha_O g^2}{\omega^5}.\exp\left(-\frac{5}{4}\left(\frac{\omega_0}{\omega}\right)^4\right)\gamma^{\exp\left[-(\omega-\omega_0)^2/(2\sigma^2\omega_0^2)\right]} \tag{8}$$

For Pierson-Moskowitz, the parameters must be defined as $\gamma = 1$ and $\alpha_O = 5H_S^2\omega_0^4/16g^2$. For JONSWAP, those parameters are defined accordingly the specific location properties in order to properly define the shape of the spectrum, with:

$$\sigma = \begin{cases} 0.07 & \omega \le \omega_0 \\ 0.09 & \omega > \omega_0 \end{cases};$$

Short crested waves directional spreading is calculated by the cos2s model (Hogben; Cobb, 1986).

The high frequency motion (HF) due to the wave action can be evaluated by two different ways (Pinkster, 1988; Faltinsen, 1990). In the simpler one the HF motion evaluated by the motion RAO is added to the low frequency motion (LF) that is calculated by the integration of the motion equations (without wave excitation).

Alternatively, the wave 1st order forces are applied to the body and all motion components (6 DOF) are obtained dynamically, by solving the equations of motion. In this case, the exciting forces are computed by sub-dividing the sea spectrum in hundreds of components with random uneven frequency ranges and also random phases. Combining (summing up) these components, we define the irregular incident wave. These components combined with the exciting force RAO in each degree of freedom will define the exciting wave forces. These exciting forces are computed only as a function of the incident wave, so they do not double count with other non-potential effects.

The added mass and damping effects are taken into consideration through the convolution of the IRF (impulse response functions) obtained from the frequency domain hydrodynamic coefficients (added mass or potential damping) and the past motions of the ship. In our code this IRF functions were computed under the assumption of an impulsive velocity applied to the floating bodies. We convolve this IRF with the past velocities of the body. Only oscillatory motions will give rise to added mass and damping forces, corresponding to an energy balance between the body motions and the waves radiated due to these ship oscillations. They are function only of the past motions of the ship, and properly include the effect of the waves radiated due to these past oscillations (Oortmerssen, 1976).

This later approach, although more computational time consuming, provides more accurate results, since the body motions are calculated including all external forces, and not simply imposing HF motions based on the motion RAOs. This is important when the vessel is subjected to HF external forces other than waves, such as hawser forces.

The Aranha and Fernandes (1995) approximation to the quadratic transfer functions is applied to the calculation of the slow drift wave forces. Wave-drift damping effects (current-wave interaction) are also considered, following the formulation presented by Aranha (1994).

For bi-modal sea states, both wave components (local sea and swell) are independently evaluated then added for defining the total wave forces.

For port simulations, the effect of the breakwaters must be considered, since it changes the wave pattern into the sheltered area. A spatial map of wave height and direction may be defined as an input to the simulator. The simulator then calculates an average wave height and direction for the vessel instantaneous position, and these wave properties are used for obtaining the forces at that time. This is a simplification, since the spatial variation of the wave along the hull is not considered.

The simulator can also import a large database of hydrodynamic coefficients obtained by the potential software, for different vessel and the breakwater relative positions. This approach is adopted for more accurate simulations or offshore multiple-body analysis (Queiroz Filho and Tannuri, 2009).

THRUSTER, PROPELLER AND RUDDER FORCES

A four quadrant model for the fixed or controllable pitch propellers is adopted, including an extra ad-hoc correction for the yaw motion induced by the reverse action of the main propeller (paddle effect). The dynamics of the propeller axis is modeled as a second order inertia system, considering the thrust and torque (K_t and K_q) coefficients and inlet water speed. The motor overload situation is simulated, causing the reduction of the rotation of the motor. Tunnel and azimuth thrusters may be also considered, taking into account the losses due to transverse water speed. The detailed description of those models is presented in (Queiroz Filho et al., 2014).

A physical-based model is used for the calculation of the rudder forces. Drag and lift forces are given by the dimensionless coefficients C_L and C_D:

$$F_D(\beta_r) = 0.5\rho A_r C_D(\alpha_E)V_r^2 \ ; \ F_L(\beta_r) = 0.5\rho A_r C_L(\alpha_E)V_r^2 \quad (9)$$

where F_D and F_L are the drag and lift forces respectively, C_D and C_L the dimensionless coefficients, α_E is the effective rudder angle and V_r the relative velocity of the fluid onto the rudder (Figure 4). The rudder coefficients must be provided in the [0^o,360^o] range.

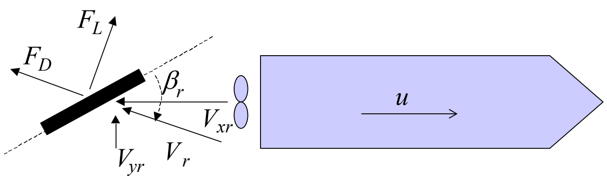

Figure 4 Definitions for the modeling of the rudder

The flow downstream speed of the propeller V_x is calculated using an argument of conservation of momentum, adapted from Molland and Turnock (2007):

$$V_{xr} = V_i + \left(0.5 + \frac{0.5}{1+0.15\,X/D_P}\right)\left(\sqrt{V_i^2 + \frac{8T_{prop}}{\rho\pi D_P^2}} - sign(V_i).V_i\right) \quad (10)$$

where V_i is the inlet longitudinal speed water in the propeller, D_P is the main propeller diameter, X is the longitudinal distance between the main propeller and the rudder axis. For positive advance speed, $V_i = u(1-w)$, being w the wake factor induced by the hull. For negative advance speed $u < 0$, it is assumed no hull influence in the flow that reaches the rudder by the opposite side ($V_i = u$).

When the main propeller operates with reverse rotation, no influence of the propeller in the rudder speed is assumed. Figure 5 shows the four-quadrant model adopted for the rudder-propeller action.

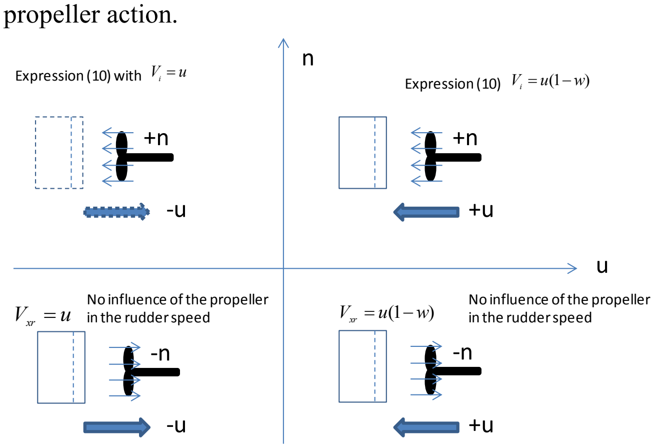

Figure 5 Four-quadrant model for rudder-propeller action

The lateral velocity (V_{yr}) is the water-rudder relative lateral speed, considering the hull rotation, drift and flow straightening. The contribution of the hull to the flow straightening is modeled as a reduction β in the lateral velocity (V_{yr}) compared to the geometric velocity (V_{yr}=κV_y). The value of κ is adjustable and for negative advance speed is taken as 1. The flow straightening induced by the propeller is naturally obtained from the expression (10). Since the longitudinal velocity of the water is increased by the propeller, the geometric drift angle at the rudder (β_r - see Figure 6) is larger than the actual "hydrodynamic" drift angle α_0.

The total flow straightening factor α_0/β_r obtained with the present formulation depends on the advance coefficient and geometric drift angle, and the results are in agreement with the experimental data presented by Molland and Turnock (2007). Typically, the flow straightening factor α_0/β_r is around 0.3-0.5.

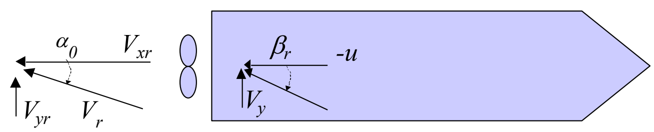

Figure 6 Flow straightening definitions

TUGBOATS

Tugboats may be simulated either as individual floating vessels, connected to the main vessel by cables or direct contact (push and pull), as shown in the Figure 7, or as external forces, depending on the requirements of analysis. In the first case, the tug boats are modeled as ships, including all the dynamic and hydrodynamic effects considered in the simulator code. Two smaller simulators are used to control them in a multiplayer simulation section.

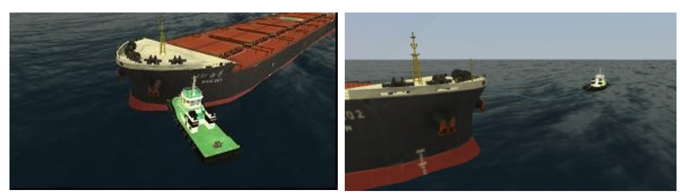

Figure 7 (left) Tugboat pushing the vessel by direct contact; (right) Tugboat connected to the vessel by a cable

In the case of external forces, a control station is used for commanding the tug action. Although in this method the dynamics of the tug is not fully simulated, important effects of the tug action are modeled. The time to change the force intensity and direction is considered using the recommendations from PIANC (1997) and Hensem (2003). The thrust losses due to the wash effect is modeled by a reduction factor (normally about 50%) when a push-pull tugboat is pushing the vessel and the water jet is directed toward the vessel hull. The thruster used by the tugboat to keep its own position is considered as a reduction factor of about 10% of the total thrust. This value is conservative since it was calculated for an offshore tugboat in non-sheltered waters using the data from Silva (2012).

MOORING LINES AND FENDERS

The SMH has model for mooring lines (using catenary formulation) and fenders (considering its non-linear restoration curve) simulation.

The catenary equations can consider lines with multiple segments, each of them with different materials and mechanical properties. This is adequate for modeling steel lines with polyester tails for example. A description of the catenary formulation used in the simulator, as well as the mooring line damping mathematical model, are presented in Nishimoto et al., (1999).

The mathematical model of the fender is extremely complex, since they are elements that provide different types of forces to the vessel and the point of contact may change.

The normal force is due to the elastic restoration properties of the fender, and is modeled by a look-up table relating the restoring force and the compression, normally provided by the suppliers. The contact point is calculated by a geometric formulation that evolves the hull geometry, vessel position and orientation and fender contact plane.

The friction between the hull and the fender also plays an important role in a berthing simulation. This force is modeled as a static friction, using typical dynamic and static friction coefficients for the materials of the fender in contact with steel. The direction of the force is defined by the contact plane and is opposite to the tendency of motion of the contact point projected in this plane.

This tendency of motion can be calculated in the simulator since the fender is not perfectly rigid. It has a flexional stiffness, as indicated in the Figure 8. This flexional force is proportional to the lateral displacement of the fender. If the static friction is larger than the flexional force, than the point of contact is not changed. Otherwise, the vessel will slip on the fender and the contact point changes. The restoration look-up table, static and dynamic friction coefficients and flexional stiffness of the fender are input parameters to the simulator.

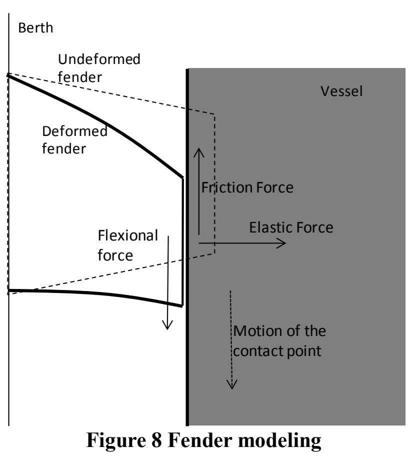

Figure 8 Fender modeling

INTERACTION WITH BANKS AND BOTTOM

The effects of obstacles, such as banks or passing ships, are evaluated by the real-time calculation of the distance between the vessel hull and the external obstacles. Simplified tabulated suction forces and yaw moments are used in order to provide the adequate physical effect expected by the pilots (Souza Junior et al., 2009; Lewis, 2008). More accurate calculation of those effects is still being implemented, by the integration with a potential Rankine-based time-domain code under development (Watai et al., 2013). The squatting is obtained by the formulation provided by ICORELS (1980).

The shallow water effect on the vessel dynamics are considered in several aspects of the mathematical modeling. The potential hydrodynamic properties (added masses and potential damping) and forces (first order and drift coefficients) are obtained by commercial codes considering the bottom in the mesh grid.

The non-potential forces are obtained by C_x, C_y and C_z coefficients. It is desirable that the towing tank or CFD tests used for defining those coefficients are done with the bottom already included. If this is not the case, some ad-hoc corrections are performed in order to account for the shallow water effects.

Ankudinov et al. (1990) presented the correction factor for the hydrodynamic derivatives Y_v and N_v valid for H / T> 1.1 approximately (Figure 9). As already mentioned, Y_v and N_v are directly related to the slope of the curves of C_y and C_z for the incidence of 180°. These correction factors are used for the angles in the intervals [-40^o; 40^o] and [140^o; 220^o]. For incidence angles higher than 40°, the hydrodynamic forces are primarily due to cross-flow. OCIMF (1994) presented current coefficients for loaded tankers with several underkeel clearances and large incidence angles. So, the variation obtained for $C_y(90^o)$ and $C_z(45^o)$ were used as correction factors (Figure 10) for the angles in the intervals [220^o;320^o] and [40^o ; 140^o]. A smooth transition function is applied to guarantee that the curves are continuous after the application of the correction factors.

The C_x coefficient is only corrected in the range [140^o; 220^o], using the model of shallow water resistance to advance presented in Lewis (1998).

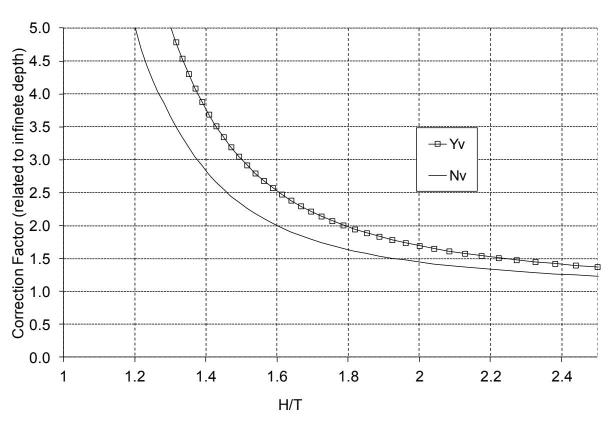

Figure 9 Shallow water correction factors for Y_v and N_v (adapted from Ankudinov et al., 1990)

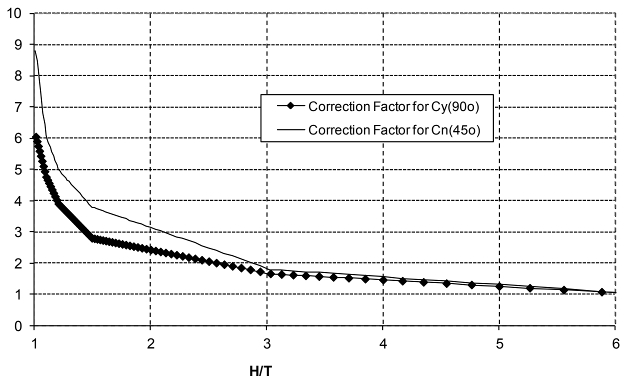

Figure 10 Shallow water correction factors for cross flow effects

DYNAMIC POSITIONING AND AUTO-PILOT

Three main classes of algorithms used in commercial DP systems are also implemented in the simulator. A low-pass filter, called wave-filter, is employed to separate high-frequency components (excited by waves) from measured signals. Such decomposition must be performed because the DP system must only control low-frequency motion, since high-frequency motion would require enormous power to be attenuated and could cause extra wear in propellers. In the present simulator, Kalman Filter is used. Furthermore, an optimization algorithm, called thrust allocation, must be used to distribute control forces among thrusters. It guarantees minimum power consumption to generate the required total forces and moment, positioning the vessel. At last, a control algorithm uses the filtered motion measurements to calculate such required forces and moment. Normally, a wind feedforward control is also included, enabling to estimate wind load action on the vessel (based on wind sensor measurements) and to compensate it by means of propellers. A detailed description of DP algorithms included in the numerical simulator can be found in Queiroz et al., (2014). This paper also presents a validation using the results from a commercial DP System.

The autopilot is designed to control the rudder angle in order to guarantee that the vessel keeps a predefined heading angle or follow a desired trajectory, even in the presence of external disturbances. The complete description of the autopilot implemented in the simulator is given by Yoshimatsu and Moreira (2013).

In order to control the heading angle, a proportional integral derivative (PID) controller was developed. This controller is designed based on the Nomoto's 1st order model, which has one degree of freedom. The control parameters were adjusted by several classes of vessels (From Handymax to Suezmax) using pole-placement technique fine-tuned by simulations, in ballasted and loaded conditions. An automatic interpolation procedure is developed in order to be easily applied to other sizes of vessels.

The Line of Sight (LOS) method is also implemented to path following (Asharafiuon et al., 2010). In this case, the heading is updated in order to guarantee that the vessel follows the required trajectory, even with a drift angle is required to counteract the environmental forces.

APPLICATIONS

The simulator helps in performing various maneuverings in extreme conditions, with different types of ship and in damaged conditions and/or command failures. Accordingly, the envelope of the ship's course is obtained in extreme and operating conditions that, in reality, occur seldom during the year. In this way, it is possible to dimension more adequately the channels, maneuvering basins and escape routes.

The simulator can also be used to verify the impact of civil works, such as the construction of new berths and breakwaters. Very often the maneuvering simulation should be associated with a hydraulic and wave diffraction study in order to predict the alteration of the field of waves and current as a result of such works.

Dredging studies relate to the same context. Deepening a channel does not directly reflect on increasing the draft permitted for navigation. A study should be undertaken with the help of simulators, inasmuch as the maneuverability of the ship with this new draft is altered, as well as other physical phenomena that define the maximum draft, such as squatting, wave motion and current in the channel.

Non-conventional operations, such as ship-to-ship berthing and maneuvering of hulls of future FPSO platforms (without propulsion), can also be examined beforehand using simulation. Moreover, when the dimensions of channels and access bends are very close to or less than obtained by standards or recommendations, it is essential to perform simulations in order to check the risks associated with extreme operating conditions, and possibly to define environmental windows for operation.

The analysis of the results of simulation can also be used to define the navigation signals and lights design (location and type of buoys) and contingency plans. The number, layout and bollard-pull of tugboats to guarantee the safe positioning of ships can also be appraised using simulators.

Several Brazilian ports and rivers have already been studied with the aid of SMH in cooperation with pilots and captains. The feedback from the pilots and captains are used to perform a "fine-tuning" of the maneuvering models and environmental conditions in each port. This synergy between the theoretical background and field expertise has proven to be very effective in order to improve the simulator accuracy.

Two illustrative results of simulations and important results are exposed.

CASE 1 – MANEUVERING ON A DREDGED CHANNEL

A new tanker terminal has been built in the Suape Port (Pernambuco State, Brazil) and is able to receive Suezmax tankers. The approach channel and turning basin have been designed using the PIANC (1997) recommendations. The vessel used to perform the analysis is a 175.000DWT Suezmax, in the maximum draft allowed in the port (Table 1).

Table 1 - Tanker main properties

	Partially Full
Length Overall LOA (m)	285
Beam (m)	49.5
Depth (m)	24
Draft (m)	15.4
Displacement (ton)	184.000
Wind Lateral Area (m^2)	3294
Engine Power	18.881kW @90rpm

The wind and current coefficients are obtained from OCIMF (1994), considering a 1.2 depth/draft (H/T) relation. In the present model, a constant depth is assumed.

Beforehand, the mathematical model is used to predict the results of crash-stop and turning circle maneuvers. The parameters of the paddle-effect model are adjusted in order to obtain acceptable results, with maximum difference between sea-trial and simulation about 20%. The Figure 11 shows the comparison for crash-stop maneuver.

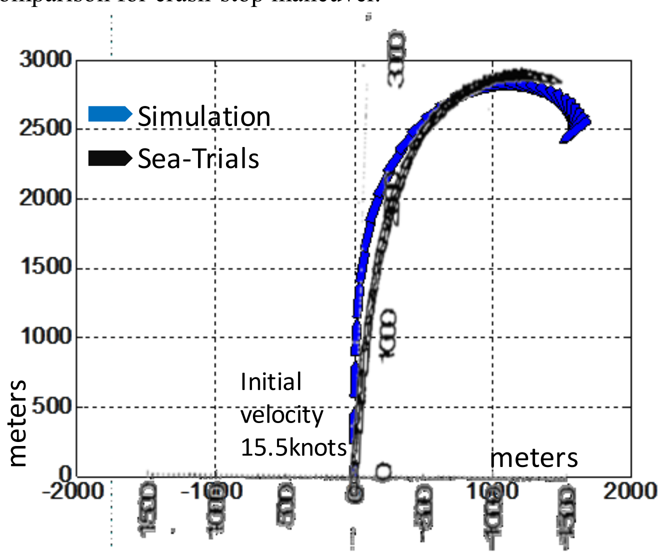

Figure 11 Crash-stop maneuver

The major concern of the pilots related to the dredged channel was the final part of it, when the vessel speed should be reduced to enter the sheltered area (see Figure 12).

Strong environmental conditions coming from SE reaches the approach channel perpendicularly. The vessel must navigate at the minimum speed of 5.5 to 6 knots in order to avoid lateral drift and to be kept inside the channel. At the final mile of the channel, the propeller must be reversed to reduce the speed to approximately 2 knots, and safely connect the tugs in the sheltered area. During this phase, the paddle effect imposes a large yaw rotation to starboard. The only way to avoid this rotation is to impose a forward order to the propeller and to use the rudder, what undesirably increases the speed.

A large number of real-time simulations were then carried out in order to properly define the width and length of the final part of the channel, as indicated in the Figure 12.

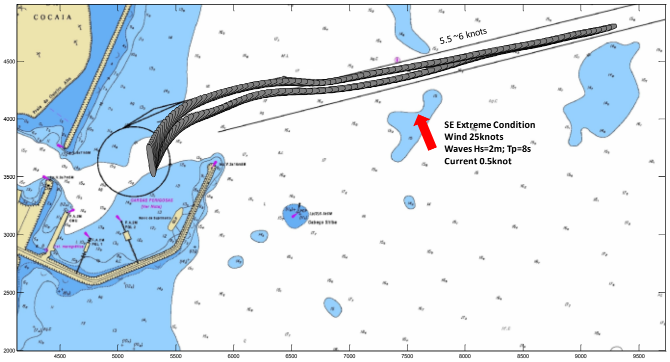

Figure 12 Simulations for SE condition

CASE 2 – MANEUVER OF A VLCC HULL

A new FPSO will be built using the hull of a VLCC, and a maneuver has been planned to take the hull from the anchoring location in Guanabara Bay (Rio de Janeiro state) to the yard, that is located very close to a large bridge that crosses the bay. This is a special maneuver that will require 2 pilots and demands special attention, since the vessel has no propeller and maneuvering equipments. The simulations were carried out to define the best environmental window to execute the operation and to verify the number and bollard pull of the tugboats. The main properties of the hull are presented in the Table 2

Table 2 - VLCC hull main properties

	Lightship
Length Overall LOA (m)	322
Beam (m)	56
Depth (m)	29
Draft (m)	5
Displacement (ton)	65.000
Wind lateral area (m2)	7728

The wind and current coefficients are obtained from OCIMF (1994), considering a 1.5 depth/draft (H/T) relation.

A large number of real-time simulations were carried considering typical morning wind conditions, with flood and ebb currents. The Figure 13 shows three different simulations, indicating that the risk of collision with the bridge pillars must be carefully evaluated. In the upper figure, a mistake in the timing to order the tugboats caused a strong delay for stopping the rate of turn of the vessel. The results indicated the necessity of 6 tugboats with 50ton and defined that the maneuver must be executed just before the low-water. The vessel will then navigate along the north of the bridge with a weak N current, reducing the risk to be pushed towards the bridge.

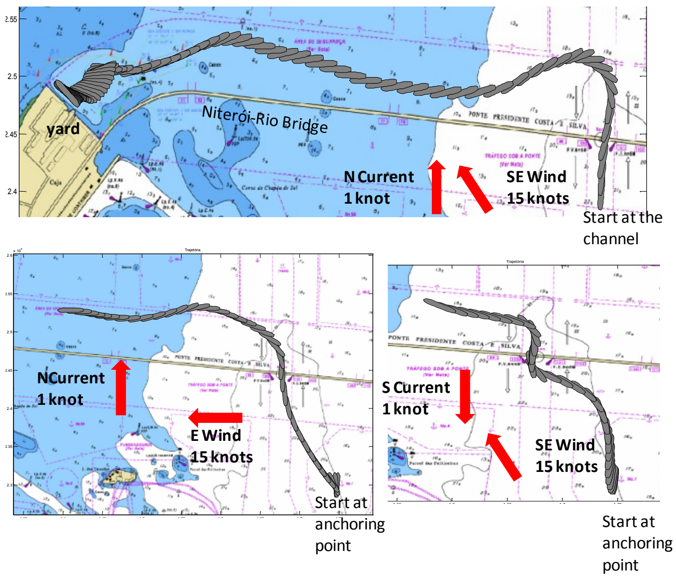

Figure 13 VLCC Simulations in Guanabara bay

CONCLUSIONS

This paper presented the mathematical model of the real-time simulator for low-speed maneuvering developed by the University of São Paulo Transpetro and Petrobras, and presented illustrative results of simulations executed with the cooperation of the pilots.

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REFERENCES

ABS (American Bureau of Shipping). Guide for Vessel Maneuverability, 2006 (Available online)

ANKUDINOV, V.K., MILLER, E.R., JAKOBSEN, B.K., DAGGETT, L.L. .Manoeuvring performance of tug/barge assemblies in restricted waterways, Proceedings MARSIM & ICMS 90, Japan, pp. 515-525, 1990.

ANKUDINOV, V. ; KAPLAN, P. ; JACOBSEN B. K., Assessment and Principal Structure of the Modular Mathematical Model for Ship Maneuverability Prediction and Real-time Maneuvering Simulations, MARSIM 93, Canada, 1993.

ARANHA, J.A.P. A formula for 'wave damping' in the drift of a floating body, Journal of Fluid Mechanics, Vol.275, pp.147-155, 1994.

ARANHA, J.A.P.; FERNANDES, A.C. On the second-order slow drift force spectrum, Applied Ocean Research, Vol.17, pp.311-313, 1995.

ASHRAUON, H. ; MUSKE, K. R. ; MCNINCH, L. C. . Review of nonlinear tracking and setpoint control approaches for autonomous underactuated marine vehicles. In American Control Conference, 2010.

FALTINSEN, O.M. Sea Loads on ships and offshore structures, Cambridge University Press, 1990.

FOSSEN, T.I. ; SMODELI, O. N. ; Nonlinear Time-Domain Strip Theory Formulation for Low-Speed Manoeuvering and Station-Keeping, Modeling Identification and Control, Vol. 25, No. 4, 201-221, 2004.

FUCATU, C. H. ; NISHIMOTO, K.; An Empirical Model of Current Shadow Effect on FPSO-Shuttle Tanker. In: The 23rd International Conference on Offshore Mechanics and Arctic Engineering, OMAE, Vancouver, Canada, 2004.

FUCATU, C. H.; Dynasim, Theoretical Manual", 2010 (In Portuguese).

HENSEM H. . Tug Use in Ports, A Practical Guide, 2nd Edition, The Nautical Institute, 2003.

HOGBEN, N.; COBB, F.C. Parametric modeling of directional wave spectra, In: Proceedings of 18th Offshore Technology Conference (OTC), 1986.

ICORELS (International Commission for the Reception of Large Ships). Report of Working Group IV, PIANC Bulletin, No. 35, Supplement, 1980.

KOH, K.K. ; YASUKAWA, H. ; HIRATA, N. ; KOSE, K. Maneuvering simulations of pusher–barge systems. J. Mar. Sci. Technol. 13, 117–126, 2008.

LEWIS, E.V. Principles of Naval Architecture, The Society of Naval, Jersey City, 1998.

MOLLAND, A. F. ; TURNOCK, S. R., Marine Rudders and Control Surfaces: Principles, Data, Design and Applications, Elsevier, 2007

NISHIMOTO, K. ; KASTER, F. ; MASETTI, I. Q. ; MATSUURA, J. ; ARANHA, J. A. P. . Full Scale Decay Test of a Moored Tanker: Field Data and Theoretical Analysis, Ocean Engineering, Vol.26, pp.125-145, 1999.

NISHIMOTO, K.; FUCATU, C.H.; MASETTI, I.Q.. Dynasim - a Time Domain Simulator of Anchored FPSO. Journal of Offshore Mechanics Arctic Engineering ASME, v. 124, n. 4, p. 203-211, 2002.

OORTMERSEN, G.V; The Motions of Moored Ship in Waves, PhD Thesis, Delft University of Technology, The Netherlands,1976.

OBOKATA, J.; On the basic design of single point mooring systems (1st report), Journal of the Society of Naval Architects of Japan, vol161, June, 1987.

OCIMF, "Predictions of wind and current loads on VLCCs", Oil Companies International Marine Forum, 1994.

PIANC . Approach Channels, A Guide for Design, Final report of the Joint PIANC-IAPH Working Group II-30, June, 1997.

PINKSTER, J.A., Low Frequency Second Order Wave Exciting Forces on Floating Structures, PhD Thesis, Delft University of Technology, The Netherlands, 1988.

QUEIROZ FILHO, A.N. ; TANNURI, E. A. . DP Offloading Operation: a Numerical Evaluation of Wave Shielding Effect. In: 8th Conference on Maneuvering and Control of Marine Craft (IFAC-MCMC'2009), Guarujá, Brazil, 2009.

QUEIROZ FILHO, A.N. ; ZIMBRES, M. ; TANNURI, E. A.. Development and Validation of a Customizable DP System for a Full Bridge Real Time Simulator, In: The 33rd International Conference on Offshore Mechanics and Arctic Engineering, OMAE, San Francisco, USA, 2014 (under revision).

SILVA, D. F. C. . Virtual Testing for Resistance, Wind and Current Loads on a Supply Boat, In: 31st International Conference on Ocean, Offshore and Arctic Engineering, OMAE,, Rio de Janeiro, Brazil, 2012.

SIMOS, A.N. ; TANNURI, E. A. ; PESCE, C.P. ; ARANHA, J. A. P. . A quasi-explicit hydrodynamic model for the dynamic analysis of a moored FPSO under current action. Journal of Ship Research, v. 45, n.4, p. 289-301, 2001.

SOUZA JR., J. A. R. ; TANNURI, E. A. ; OSHIRO, A.T. . Development and Application of a Ship Maneuvering Digital Simulator for Restricted Waters. In: 8th Conference on Maneuvering and Control of Marine Craft (IFAC-MCMC'2009), Guarujá, Brazil, 2009.

WATAI, R.A. ; RUGGERI, F. ; TANNURI, W. A. ; WEISS, J. . Evaluation of empirical and numerical methods on the prediction of hydrodynamic loads involved in the passing ship problem. In: 3rd International Conference on Ship Maneuvering in Shallow and Confined Water: with non-exclusive focus on Ship Behavior in Locks, p. 167-175, Ghent, Belgium, 2013.

YOSHIMATSU, E. S. ; MOREIRA, L. G. L.. Autopilot for surface vessels, Graduation Thesis, Escola Politécnica, University of São Paulo, 2013 (In Portuguese).

WEBSTER, W.C. . Shiphandling Simulation: Application to Waterway Design", National Academies Press, 1992.

