OFT TOC

Proceedings of OMAE'03:
22nd International Conference on Offshore Mechanics and Arctic Engineering
08-13 June 2003, Cancun, Mexico

OMAE2003-37066

Assisted Dynamic Positioning of A Moored FPSO: Robustness Aspects Regarding Current Forces Modeling

Eduardo A. Tannuri(eduat@usp.br) [1]   **Alexandre N. Simos (alesimos@usp.br)** [1]

Celso P. Pesce (ceppesce@usp.br) [2]   **Décio C. Donha (decdonha@usp.br)** [2]

[1] Department of Naval Architecture and Ocean Engineering
[2] Department of Mechanical Engineering

University of São Paulo
São Paulo, SP, Brazil

ABSTRACT

This work treats the problem of assisted dynamic positioning (DP) of moored FPSOs, focusing on robustness issues regarding the modeling of hydrodynamic current forces and moment. Modern DP systems usually employ Extended Kalman Filters associated to LQG or Adaptive controllers to estimate current forces and velocity, performing a direct compensation of these environmental loads. Such a procedure guarantees better performance of the controller under a wide range of environmental conditions, which is known as a "large environmental window". However, it demands a large number of control parameters, usually requiring a time consuming tuning process. More recently, Sliding Mode Controllers (SMC) with an alternative compensation technique [1] have also been investigated and proved to be an appropriate choice. Robustness properties of SMC allow the use of a simple hydrodynamic current force modeling and rough measurements of current velocity and direction to perform a real time estimation of current loads. Robust performance and stability are guaranteed even in the presence of current measuring and modeling errors. This procedure preserves the advantages of current compensation and avoids the parameter tuning process. In the present paper the so-called 'Short Wing-Cross Flow Current Model' [2] is used. This current force model has the advantage of being semi-explicit, in the sense that it depends on only three experimental coefficients, obtained from simple rotating-arm experiments in a towing tank.

Nevertheless, it is a common practice in Ocean Engineering to use the well-known 'Hydrodynamic Derivative Models', for low maneuvering speeds as, e.g., the 'Takashina model'. Those models require a large number of towing tank tests, and are usually believed to represent current loads with very small errors. The present work shows simulations of the SMC technique (with the simpler Short Wing Current Model), controlling a tanker that is otherwise modeled with a 'Hydrodynamic Derivative Model'. The results confirm that, for control purposes, the 'Short Wing-Cross Flow Current Model' current model can be used, since performance and stability requirements are preserved.

Keywords: DP systems, FPSO, Mooring, Nonlinear Control, Robustness, Hydrodynamic Current Force Models.

INTRODUCTION

Control algorithms define the overall performance of a Dynamic Positioning System. However, several characteristics of offshore systems turn the controller design a hard task.

A first problem is related to non-linear terms in mathematical models that describe vessel dynamics and environmental forces. Early DP systems used linear control techniques, such as PID or LQG controllers, and performance degradation were verified, unless the states remain sufficiently close to that about which the dynamics were linearized [3]. One possible approach would be the use of several linear controllers, each one designed with the model linearized about the actual state of the ship. However, such controller would be extremely complex and the stability of such "gain scheduled" system would be questionable [4]. Another approach is the application of non-linear controllers, that use full non-linear model of the system during control design ([5] e [6]).

Another serious problem is related to the way controllers consider hydrodynamic disturbances. This topic distinguishes, basically, two classes of controllers. A first approach considers these actions as disturbances in the control loop, and designs the controller to compensate it. Control parameters are tuned by means of sea-trials or numerical simulations, and the performance can only be assured for environmental conditions similar to those used in tuning process. The narrow “environmental window” presented by such controllers motivated the development of a second class of controllers, which have, internally, models for environmental agents.

This approach benefits from all information contained in the available models of environmental forces. Such “model based” controllers present simplified models of hydrodynamic forces and algorithms for estimating or adapting its coefficients and the environmental conditions ([7], [8]). The estimated forces are then used in a *feedforward* control path, and directly compensate the environment action. This approach guarantees the performance under a wider variation in environmental condition, because the controller “knows” the actual situation.

As described in [9], model based controllers may present some operational problems. Since “the mathematical model for a vessel is a description of the vessel’s hydrodynamic characteristics…it may be necessary for the manufacturers to attend the vessel for fine-tuning of the system.” In other words, the large number of control and model parameters to be adjusted or estimated requires an extra effort during DP commissioning.

An alternative model based controller was developed in [1], where a robust non-linear control methodology was applied to the problem. The controller does not present estimation or adaptation algorithms for model parameters, but its robustness properties assures performance and stability even with limited errors in such parameters. The controller is based on Sliding Mode Control (SMC) theory, originally proposed in [10], and adapted in [11].

This paper focuses on SMC robustness issues regarding the modeling of hydrodynamic current forces and moment. Robustness properties of SMC allow the use of a simple hydrodynamic current force modeling and rough measurements of current velocity and direction to perform a real time estimation of current loads. Robust performance and stability are guaranteed even in the presence of current measuring and modeling errors. This procedure preserves the advantages of current compensation and avoids the parameter tuning process.

The controller uses the so-called 'Short Wing-Cross Flow Current Model' [2]. This current force model has the advantage of being semi-explicit, in the sense that it depends on only three experimental coefficients, obtained from simple rotating-arm experiments in a towing tank. Experimental validations of this model is presented in [12].

Nevertheless, it is a common practice in Ocean Engineering to use the well-known 'Hydrodynamic Derivative Models', for low maneuvering speeds as, e.g., the 'Takashina model'. Those models require a large number of towing tank tests, and are usually believed to represent current loads with very small errors. However, the utilization of such models in the controller must be avoided, due to the large number of parameters, what would lead to the commissioning problems early described.

The present work shows simulations of the SMC technique (with the simpler Short Wing Current Model), controlling a tanker that is otherwise modeled with a 'Hydrodynamic Derivative Model’. The results confirm that, for control purposes, the 'Short Wing-Cross Flow Current Model' current model can be used, since performance and stability requirements are preserved.

CURRENT FORCES AND MOMENT MODELING

Short Wing-Cross Flow Current Model

Consider the floating body moving with velocity $u\mathbf{i} + v\mathbf{j}$ related to the fluid, without yaw rotation r (r=0), and let

$$\begin{aligned} &U = \sqrt{u^2 + v^2}; \\ &a = p + \arctan(v/u). \end{aligned} \qquad (3)$$

In the reference system moving with the body one sees a current with intensity U incident in a direction that makes an angle α with the longitudinal axis. Using the expressions derived in [13] for the current forces in a FPSO one has that

$$\begin{aligned} &X_{1C}(u,v,0) = \frac{1}{2} rU^2 LT \cdot C_{1C}(a); \\ &X_{2C}(u,v,0) = \frac{1}{2} rU^2 LT \cdot C_{2C}(a); \\ &X_{6C}(u,v,0) = \frac{1}{2} rU^2 L^2 T \cdot C_{6C}(a), \end{aligned} \qquad (4)$$

where X_{1C} is surge current force, X_{2C} is sway current force, X_{6C} is yaw current moment and r is water density. The coefficients $\{C_{1C}(\alpha);C_{2C}(\alpha);C_{6C}(\alpha)\}$ are given in terms of the ship main dimensions by the expressions (B: beam; T: draft; L: length; C_B: block coefficient):

$$\begin{aligned} &C_{1C}(a) = C_f(R_e)\cos a + \frac{pT}{8L}(\cos 3a - \cos a); \\ &C_{2C}(a) = \left(C_Y - \frac{pT}{2L}\right) sin a\,|\,sin a\,| + \frac{pT}{2L} sin^3 a + \\ &\qquad + \frac{pT}{L}\left(1 + 0.4\frac{C_B B}{T}\right) sin a\,|\cos a\,|; \\ &C_{6C}(a) = -lC_Y sin a\,|\,sin a\,| - \frac{pT}{L} sin a \cos a - \\ &\qquad - \frac{pT}{2L}\left(\frac{1 + |\cos a\,|}{2}\right)^2 \left(1 - 4.8\frac{T}{L}\right) sin a\,|\cos a\,|. \end{aligned} \qquad (5)$$

As anticipated, the parcel that depends on (u,v) depends only on three hydrodynamic coefficients: the friction coefficient C_f(Re), the cross-flow drag coefficient C_Y and the cross-flow moment coefficient[1] lC_Y. These coefficients can be easily obtained by towing tank captive tests, or can be estimated by approximations described in [2].

The forces and moments that depends on hull rotation ($r\neq 0$) are also evaluated using heuristic developments, and also depends on ship main dimension and the hydrodynamic coefficients previously exposed. A full description of the Short Wing Cross-Flow Model is presented in [2].

Hydrodynamic Derivative Current Model

Following the procedure described in [14], current forces and moment can be written using the following Taylor series:

$$\begin{aligned} X_{1C}(u,v,r) &= \left(X_u\bar{u} + \frac{1}{2}X_{uu}\bar{u}^2 + \frac{1}{6}X_{uuu}\bar{u}^3 \right) 1/2\, rL^2U^2 \\ X_{2C}(u,v,r) &= \left(Y_v\bar{v} + \frac{1}{6}Y_{vvv}\bar{v}^3 + Y_r\bar{r} + \frac{1}{6}Y_{rrr}\bar{r}^3 \right) 1/2\, rL^2U^2 \\ X_{6C}(u,v,r) &= \left(N_v\bar{v} + \frac{1}{6}N_{vvv}\bar{v}^3 + N_r\bar{r} + \frac{1}{6}N_{rrr}\bar{r}^3 \right) 1/2\, rL^3U^2 \end{aligned} \quad (6)$$

The coefficients of expression (6) are called “hydrodynamics derivatives”, and are obtained by exhaustive towing-tank tests. Several authors propose variations in the terms of (6) and in the experimental procedures to obtain the hydrodynamic derivatives, such presented in [15], [16], [17] and [18]. However, the main features of these model philosophy are preserved, namely, the large number of experimental parameters and the high sensitivity to variations in some of them [19].

ASSISTED DYNAMIC POSITIONING DESIGN USING SMC TECNIQUE

In an assisted dynamic positioning of moored FPSOs, the controller must attenuate slow-drift oscillations in all horizontal motions of the ship. However, surge and sway static forces due to environmental action must be counteracted by the mooring lines, and not by the controller, in order to minimize fuel consumption. The control system must only counteract environmental yaw static moment.

So, the controller presents different functions for the translational and rotational channels. In the first case, the controller must only increase the damping of the system to decrease slow-drift oscillations; the final equilibrium position will not be affected by control action and will be given by the balance between static environmental forces and mooring action. In the yaw channel the controller must keep the ship heading near the set point, obtained in the first layer. The heading control is crucial because, as already explained, determines the first order vertical oscillation in the mooring lines and risers. This control scheme was proposed in [20] and is appropriate for assisted dynamic positioning systems.

[1] Here l represents the longitudinal distance between the origin of the local coordinate system (mid-ship section) and the center of pressure for beam incidence.

The following dynamic model gives horizontal motions of a moored FPSO:

$$\begin{aligned} &(M+M_{11})\ddot{x}_1 - (M+M_{22})\dot{x}_2\dot{x}_6 - M_{26}\dot{x}_6^{\,2} = X_{1E} + X_{1M} + X_{1T}; \\ &(M+M_{22})\ddot{x}_2 + M_{26}\ddot{x}_6 + (M+M_{11})\dot{x}_1\dot{x}_6 = X_{2E} + X_{2M} + X_{2T}; \\ &(I_Z+M_{66})\ddot{x}_6 + M_{26}\ddot{x}_2 + M_{26}\dot{x}_1\dot{x}_6 = X_{6E} + X_{6M} + X_{6T}. \end{aligned} \quad (7)$$

where M is the mass of the FPSO; I_z is the moment of inertia about the vertical axis; M_{ij} is the added mass tensor; X_{1E}, X_{2E}, X_{6E} are surge, sway and yaw environmental loads (current, wind and waves), X_{1M}, X_{2M}, X_{6M} are mooring forces and moment and X_{1T}, X_{2T}, X_{6T} are propulsion system forces and moment. The variables $\dot{x}_1$, $\dot{x}_2$ and $\dot{x}_6$ are the midship surge, sway and yaw absolute velocities (Fig 1).

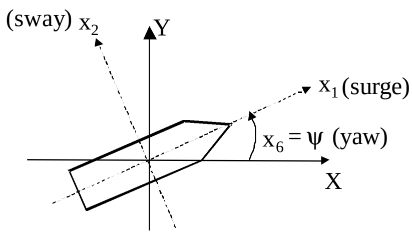

Fig 1 Coordinate systems

Solving (7) for the accelerations it is obtained:

$$\ddot{x}_1 = f_1^{dyn}(\dot{\mathbf{x}}) + \frac{1}{M+M_{11}}X_{1Total}, \quad (8a)$$

$$\ddot{x}_2 = f_2^{dyn}(\dot{\mathbf{x}}) + \frac{I_Z+M_{66}}{D}X_{2Total} - \frac{M_{26}}{D}X_{6Total}, \quad (8b)$$

$$\ddot{x}_6 = f_6^{dyn}(\dot{\mathbf{x}}) - \frac{M_{26}}{D}X_{2Total} + \frac{M+M_{22}}{D}X_{6Total}, \quad (8c)$$

where : $\dot{\mathbf{x}} = (\dot{x}_1, \dot{x}_2, \dot{x}_6)$, $X_{iTotal} = X_{iE} + X_{iM} + X_{iT}$,

$$f_1^{dyn}(\dot{\mathbf{x}}) = \frac{M+M_{22}}{M+M_{11}}\dot{x}_2\dot{x}_6 + \frac{M_{26}}{M+M_{11}}\dot{x}_6^2,$$

$$f_2^{dyn}(\dot{\mathbf{x}}) = \frac{(M+M_{11})(I_Z+M_{66}) - M_{26}^2}{D}\dot{x}_1\dot{x}_6,$$

$$f_6^{dyn}(\dot{\mathbf{x}}) = \frac{(M_{11}-M_{22})M_{26}}{D}\dot{x}_1\dot{x}_6,$$

$$D = (M+M_{22})(I_Z+M_{66}) - M_{26}^2.$$

The control of translation motions must only increase the total damping, since the mooring system is responsible for the counteraction of the mean environmental forces. The functions $f_i^{dyn}(\dot{\mathbf{x}})$ are known with high accuracy, being related to the dynamics of a rigid body immersed in an ideal fluid and the added masses are calculated using only the potential flow theory. So, for such motions a feedback linearization control is used, with an extra term proportional to the velocity, according to:

$$X_{1T} = (M + M_{11})\left[- f_1^{dyn}(\dot{\mathbf{x}}) - c_1\dot{x}_1\right],$$
$$X_{2T} = \frac{D}{I_Z + M_{66}}\left[- f_2^{dyn}(\dot{\mathbf{x}}) - c_2\dot{x}_2\right], \qquad (9)$$

where the coefficients c_1 and c_2 are calibrated to guarantee adequate damping to the motions. The surge and sway dynamics in closed loop are then given by:

$$\ddot{x}_1 + c_1\dot{x}_1 - \frac{1}{M + M_{11}} X_{1M} = \frac{1}{M + M_{11}} X_{1E},$$
$$\ddot{x}_2 + c_2\dot{x}_2 + \left(- \frac{I_Z + M_{66}}{D} X_{2M} + \frac{M_{26}}{D} X_{6M} \right) =$$
$$= \frac{I_Z + M_{66}}{D} X_{2E} - \frac{M_{26}}{D}(X_{6E} + X_{6T}), \qquad (10)$$

which is equivalent to oscillators with non-linear restoration due to mooring lines. Furthermore, the damping and drag effects due to mooring lines and risers will conservatively increase the total damping of the systems. The environmental forces are the external excitation, and the balance between such forces and the mooring system restoration will determine the equilibrium position of the oscillators.

Due to the coupling between sway and yaw, the control moment X_{6T} also acts in sway motion as an external excitation. Nevertheless, even with a stable yaw dynamics (what is assured by the yaw sliding mode controller exposed latter), the coupling between sway and yaw could turn sway dynamics unstable. A formal proof that this coupling do not degenerate sway dynamics is not given at this moment, but simulations under several environmental conditions enables us to infer such an effect as not significant. Possible justifications for these evidences are related to the high damping in sway motion and weak coupling between sway and yaw motions, due to the small bow/stern asymmetry.

The yaw control must counteract the environmental forces, in order to keep the actual heading of the ship as close as possible the desired value, even with uncertainty and errors in the models of such forces.

A sliding mode controller is then applied to the yaw dynamics, given in (8c). A theoretical review of this method is given in [21]. Isolating thrusters' forces and moment in (8c) one obtains:

$$\ddot{x}_6 = f_6(\mathbf{x},\dot{\mathbf{x}}) - \frac{M_{26}}{D} X_{2T} + \frac{M + M_{22}}{D} X_{6T}, \qquad (11a)$$

being

$$f_6(\mathbf{x},\dot{\mathbf{x}}) = f_6^{dyn}(\dot{\mathbf{x}}) - \frac{M_{26}}{D}(X_{2E} + X_{2M}) + \frac{M + M_{22}}{D}(X_{6E} + X_{6M}). \qquad (11b)$$

The sliding mode control with an extra term to eliminate the influence of the control sway force X_{2T} is then given by:

$$X_{6T} = \frac{M_{26}}{M + M_{22}} X_{2T} + \left[\frac{D}{M + M_{22}} (-\hat{f}_6(x,\dot{x}) + (\ddot{x}_{6d} - 2l\dot{\tilde{x}}_6 - l\tilde{x}_6) - k.signal(s) \right], \qquad (12)$$

where

$$\hat{f}_6(\mathbf{x},\dot{\mathbf{x}}) = f_6^{dyn}(\dot{\mathbf{x}}) - \frac{M_{26}}{D}(\hat{X}_{2E} + \hat{X}_{2M}) + \frac{M + M_{22}}{D}(\hat{X}_{6E} + \hat{X}_{6M}),$$

x_{6d} represents the reference heading angle trajectory, $\tilde{x}_6 = x_6 - x_{6d}$ represents the error and s is the scalar defined by $s = \dot{\tilde{x}}_6 + 2l\tilde{x}_6 + l^2\int_0^t \tilde{x}_6 dt$ representing a true measure of tracking performance added to an integral action. The terms with (^) represent the best estimate to the corresponding variable and are referred as nominal values. The controller parameter l is related to the closed-loop bandwidth of the system [21].

It can be verified that if the gain k is chosen such that $k \geq h + \max\left| f_6 - \hat{f}_6 \right|$, the time required to the variable s reaches 0 is given by $t_{reach} < |s(0)|/h$. After this time, the tracking error tends exponentially to zero [21].

So, the control will be totally defined if the maximum uncertainty about the function f_6 is known. It must be stressed that this function is related to the environmental and mooring sway forces and yaw moments. In the next section, the uncertainty about f_6 will be estimated for current and waves action.

The control law (12) has an on-off term that depends on the signal of the variable s, which is as more important as higher is the uncertainty. This term can cause undesired high frequency oscillation in the control action due to implementation delays, noise or some other practical issue. Such oscillation, known as *chatter*, may excite high frequency dynamics neglected in the model and might damage the actuators.

To avoid chattering, the *signal* function in (12) is replaced by the saturation function defined by:

$$sat(s/\Phi) = \begin{cases} s & |s| \leq \Phi \\ signal(s) & |s| > \Phi \end{cases}.$$

This modification, proposed by [11], defines a boundary for the variable s with width Φ. This boundary layer smoothens the signal transition.

Design of the controller, robust to current variations

When the ship is under current action only, the controller must guarantee the stability and performance requirements due to uncertainty about the direction and velocity of the current and about its model. Being y_C the angle between current direction and the X-axis (Fig 2) and a its maximum variation[2], it can be written:

$$U_{\min} < U < U_{\max}\ , \qquad p - a < y_C < p + a\ .$$

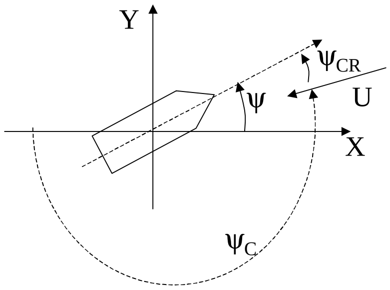

Fig 2 Current: definition of incidence angles

In the present formulation, it is considered that the whole state vector $(x, y, y\ , \dot{x}, \dot{y}, \dot{y})$ is measured or estimated. The best estimate of the function f_6 (referred as $\hat{f}_6$) is obtained using nominal environmental conditions ($\hat{U} = (U_{\min} + U_{\max})/2$ and $\hat{y}_C = p$) in (5b). The application of (6) requires the maximum error on f_6. In the present case, this error is caused by uncertainties on current direction and intensity and on the current and mooring line force models. It is possible to show that:

$$\max\left|f_6(\mathbf{x},\dot{\mathbf{x}}) - \hat{f}_6(\mathbf{x},\dot{\mathbf{x}})\right| \le \left|\frac{M_{26}}{D}\right|\left(\max\left|X_{2E} - \hat{X}_{2E}\right| + \max\left|X_{2M} - \hat{X}_{2M}\right|\right) + \\ + \left|\frac{M + M_{22}}{D}\right|\left(\max\left|X_{6E} - \hat{X}_{6E}\right| + \max\left|X_{6M} - \hat{X}_{6M}\right|\right) . \qquad (13)$$

Therefore, it is only required to evaluate $\max\left|X_{iE} - \hat{X}_{iE}\right|$ and $\max\left|X_{iM} - \hat{X}_{iM}\right|$, where i=2 and 3. Error calculation is performed using Short Wing-Cross Flow Current Model and mooring line restoring and damping models (see [1]).

The controller can be easily extended when the ship is also subjected to wave action , but it will not be exposed in the present paper.

Tuning of Controller Parameters

Surge and sway controller requires determination of the parameters c_1 and c_2, related to the damping of the closed loop system. When damping is increased, the amplitude of second order oscillations due to waves decreases, but the settle time of the system also increases. The trade off between these properties determines the best values for the constants c_1 and c_2. From simulations, the parameters were chosen as $c_1 = 0.03$ e $c_2 = 0.04$.

The yaw controller has three tuning parameters: h, related to the time required to variable *s* reaches zero; l, related to the bandwidth of the system and Φ, that represents the boundary layer *thickness*.

The control system must counteract static and slowly varying moments, which contains energy up to 0.02rad/s. So, the bandwidth of the closed-loop system must be higher than 0.02rad/s, what is reached with $l = 0.03$.

The parameter h can be obtained by requiring that the variable *s* reaches zero in 200s or less:

$$t_{reach} \le \frac{|s(0)|}{h} \Rightarrow h = \frac{l\,(x_{6d}(0) - x_6(0))}{t_{reach}} .$$

Using simulation results, the boundary layer thickness was chosen as $\Phi = 5\times10^{-4}$, leading to good tracking precision and avoiding chatter.

CASE STUDY

The characteristics and main dimensions of the VLCC used in this work are presented in Table 1.

It is supposed that the FPSO is equipped with 4 tunnel thrusters with thruster capacity of 400kN and one main propeller with 1000kN maximum trust capacity, as shown in Fig.3.

Table 1 VLCC dimensions (100% loaded)

Properties	Values
Mass (M)	321900 ton
Moment of inertia (I_Z)	2.0×10^9 ton.m^2
Length (L)	320 m
Draft (T)	21.5 m
Breadth (B)	54.5 m
Wetted Surface(S)	27340 m^2
Position of CG (x_G)	9.81 m
M_{11}	18480 ton
M_{22}	275000 ton
M_{66}	1.58×10^9 ton.m^2
M_{26}	2.21×10^6 ton.m
Block Coefficient(C_B)	0.83
Transversal force coef. (C_y)	0.78
Cross-flow moment coefficient (lC_Y)	0.045

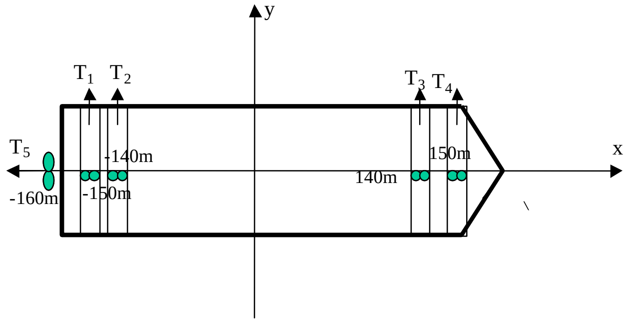

Fig.3 Thrusters distribution

---

[2] The estimated current direction was adopted as $\hat{y}_C = p$. One can always rotate the coordinate system to align the Ox axis with current direction.

The environmental condition considered in the simulations is represented in next figure.

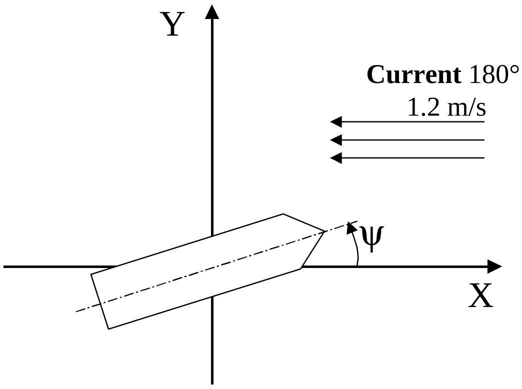

Fig. 4 Environmental condition

Current forces are evaluated by means of Hydrodynamic Derivative Current Model, with the following coefficients, extracted from [14].

Table 2. Hydrodynamic derivatives

Coefficient	**Value**	**Coefficient**	**Value**
Xu	-5.10^{-4}	Yrrr	0
Xuu	0	Nv	-0.0093
Xuuu	0	Nvvv	0.00611
Yv	-0.0261	Nr	-0.0035
Yvvv	-0.045	Nrrr	0.00611
Yr	0.0047		

Since the turret is installed near midship section, the vessel does not present weathervane capacity, and the stable position (without control) would be $\psi_{eq}>0$. In this case, a DP system is required, in order to recover the weathervane capacity of the vessel.

The first simulation is performed supposing a turret mooring system, composed by 9 symmetrically distributed lines, with the turret installed next to the midship section (the distance between the turret and midsection, x_p, is 64m).

Supposing initial ship heading $\psi_{initial}= 10^o$, the controlled system was simulated. The controller supposes a nominal current velocity of $\hat{U}$=1.0m/s and can handle current velocities between 0.8m/s and 1.2m/s and $\pm 5^\circ$ variations in its direction ($a = 5^o$). The corresponding variation in current force, as well as the best estimate of the function f_6 ($\hat{f}_6$) is evaluated using the Short Wing-Cross Flow Current Model.

Fig.5 shows yaw heading angle, Fig.6 shows X and Y positions, and thrust delivered by propellers 1, 3 and 5 are exposed in Fig.7. The overall performance of the systems is acceptable, with a small overshoot of approximately 2^o in yaw angle and a settling time of 750s. After stabilization, control forces remain small, since the position $\psi=0^o$ is an unstable equilibrium point. This example showed that, by means of a DP system, the turret moored FPSO recover its weathervane ability.

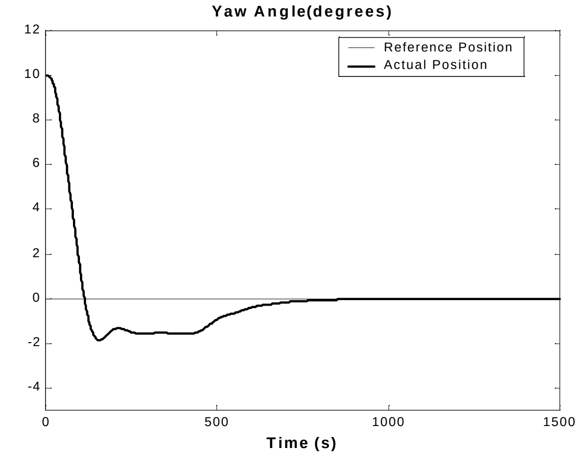

Fig.5 Yaw Angle of system modeled by Hydrodynamic Derivative Current Model

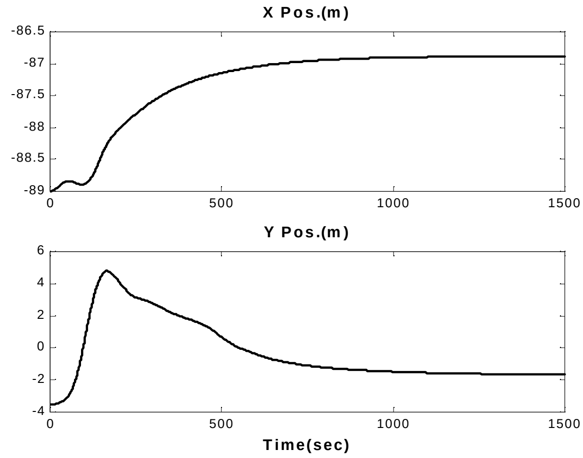

Fig.6 X and Y position of system modeled by Hydrodynamic Derivative Current Model

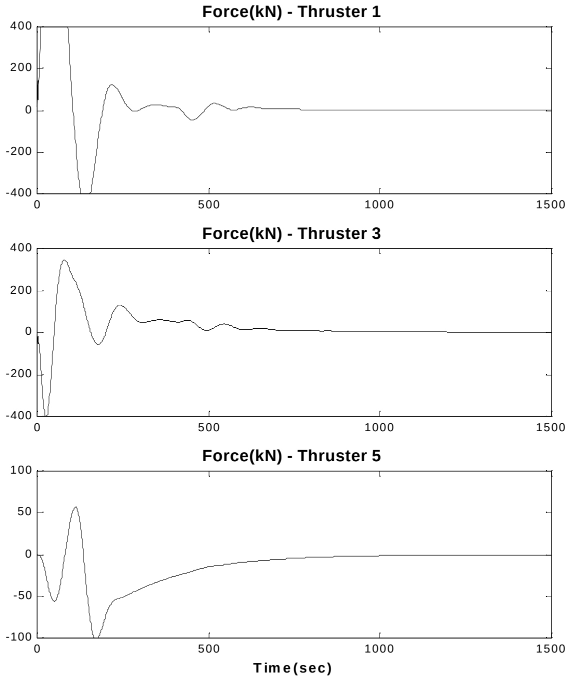

Fig.7 Forces in thrusters 1, 3 and 5 - system modeled by Hydrodynamic Derivative Current Model

In order to illustrate the fact that the controller is robust to current modeling, the same case was simulated using the Short Wing-Cross Flow Model in the system model. The results, showed in Figs.8, 9 and 10, are similar to those obtained when the system was modeled by Hydrodynamic Derivative, confirming control robustness.

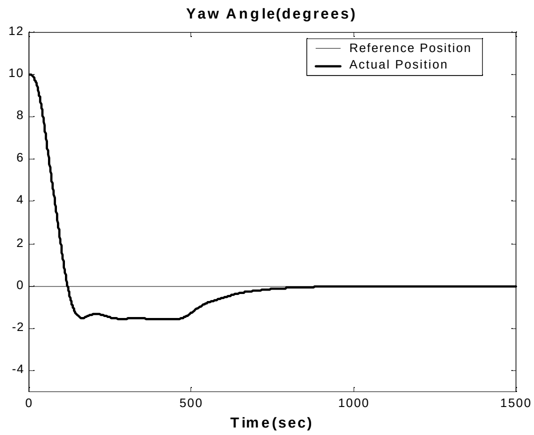

Fig.8 Yaw Angle of system modeled by Short Wing-Cross Flow Current Model

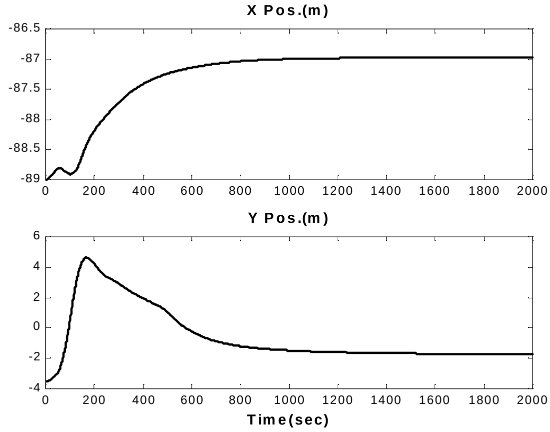

Fig.9 X and Y position of system modeled by Short Wing-Cross Flow Current Model

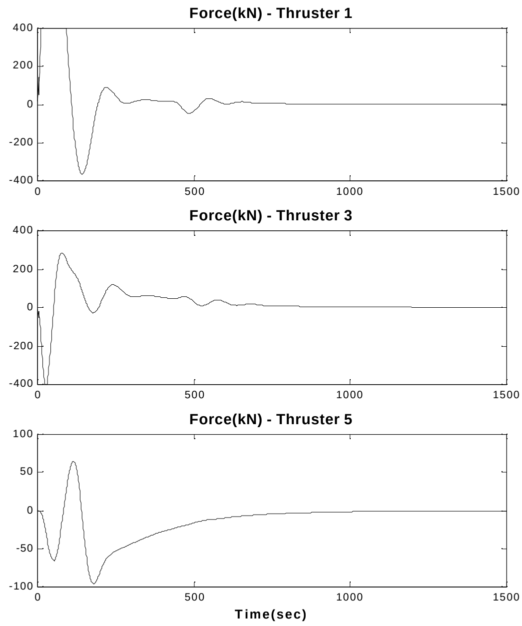

Fig.10 Forces in thrusters 1, 3 and 5 - system modeled by Short Wing-Cross Flow Current Model

It is considered, now, the case of a SPM moored tanker, with the same properties shown in Table 2. The hawser length is 224m and it is connected near vessel's bow. Under the environmental condition of Fig.4, the uncontrolled system would present a fishtailing oscillation. In Fig. 11, it is shown Y position and yaw heading of the ship, simulated using both Hydrodynamic Derivative and Short Wing Current Models.

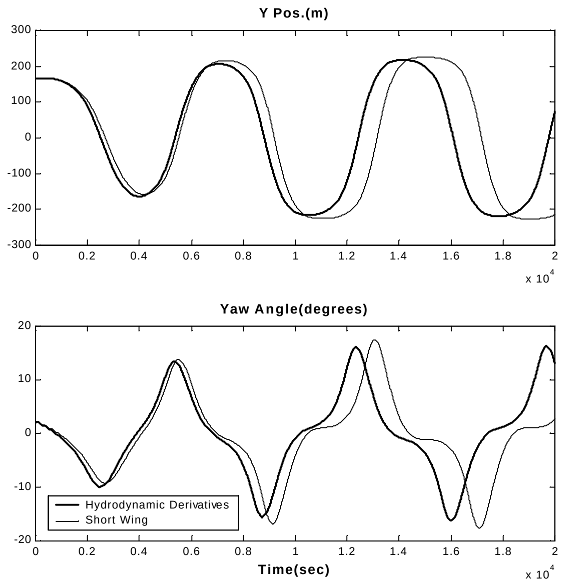

Fig.11 Y Position and Yaw Angle SPM moored tanker – fishtailing oscillation

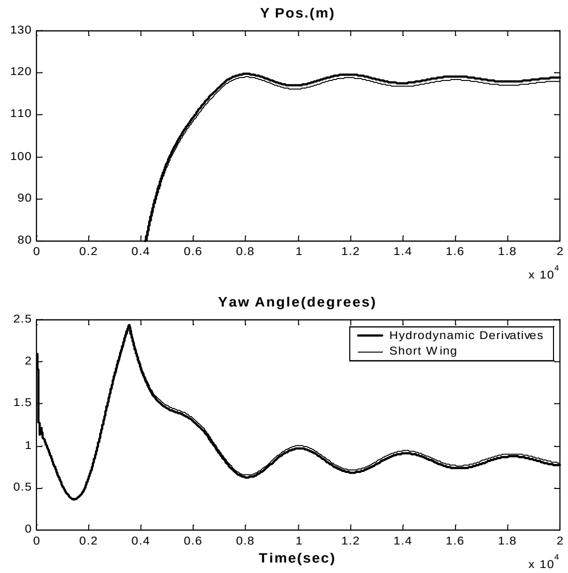

Fig.12 Y Position and Yaw Angle SPM controller moored tanker

Dynamical behavior predicted by both models are similar, but quantitative discrepancies may be observed due to high sensibility of this motion to current forces. An interesting analysis of such sensibility is done in [22].

Fig. 12 shows controlled ship response for both current models. The fishtailing oscillation is eliminated, and performance is quite similar in both cases. It is again confirmed controller robustness properties to current modeling variation.

CONCLUSIONS

The present work focused on robustness issues regarding the modeling of hydrodynamic current forces and moment in a DP controller. It was used a robust control methodology, which includes those models in order to calculate environmental forces and directly compensate them by means of propellers. This control strategy, generically known as “model based” controllers, presents better performance than the traditional ones, which consider environmental forces as unknown disturbances.

A “simple” model of hydrodynamic current forces and moment was used in the controller. This current force model, known as Short Wing-Cross Flow Model, has the advantage of being semi-explicit, depending on only three experimental coefficients, obtained from simple rotating-arm experiments in a towing tank. Although it is a “simple” model, it presents experimental validation and can recover hydrodynamic forces with high accuracy.

Nevertheless, it is a common practice in Ocean Engineering to use the well-known 'Hydrodynamic Derivative Models'. The present paper aimed to verify if the controller, containing the Short Wing-Cross Flow Model, can guarantee performance properties if the vessel being controlled is modeled by 'Hydrodynamic Derivative Model’.

Simulations of a moored tanker (in either SPM or turret configuration) indicated that robustness properties of the controller can “deal” with slightly modeling differences, and performance and stability requirements are preserved. Due to simplicity and small number of parameters of Short Wing-Cross Flow Model, it is a very suitable option for model based control design.

ACKNOWLEDGMENTS

This work has been supported by the State of São Paulo Research Foundation (FAPESP – Process no. 98/13298-6). A CNPq research grant, process no. 304062/85, is also acknowledged.

REFERENCES

[1] Tannuri, E.A., Donha, D.C., Pesce, C.P., 2001, “Dynamic Positioning of a Turret Moored FPSO Using Sliding Mode Control”, *Int J of Robust and Nonlinear Control*, Vol.11, pp.1239-1256, May.

[2] Simos, A.N. et al., 2001, “A quasi-explicit hydrodynamic model for the dynamic analysis of a moored FPSO under current action”, *Journal of Ship Research*, Vol.45, No.4, pp289-301.

[3] Tannuri, E.A., Pesce, C.P., 2002, “Comparing two different contrrol algorithms applied to dynamic positioning of a pipeline launching barge”, *Proceedings of 10th Mediterranean Conference on Control and Automation, MED2002*, Lisbon, Portugal.

[4] Yoerger, D.R., Newman, J.B., Slotine, J.J.E., 1986, “Supervisory Control System for the JASON ROV”, *IEEE Journal on Oceanic Engineering,* Vol. OE-11, No. 3, pp392-400.

[5] Aarset, M.F., Strand, J.P., Fossen, T.I., 1998, “Nonlinear Vectorial Observer Backstepping With Integral Action and Wave Filtering for Ships”, *Proceedings of the IFAC Conference on Control Applications in Marine Systems (CAMS'98)*, Fukuoka, Japão, pp.83-89.

[6] Fossen, T.I., Strand, J.P., 2001, “Nonlinear Passive Weather Optimal Positioning Control (WOPC) System for Ships and Rigs: Experimental Results”, *Automatica*, Vol.37, No. 5, pp701-715.

[7] Saelid, S., Jenssen, N.A., Balchen, J.G., 1983, “Design and Analysis of a Dynamic Positioning System Based on Kalman Filtering and Optimal Control”, *IEEE Transactions on Automatic Control*, Vol.AC-28, No.3, pp331-339.

[8] Donha, D.C., 1989, *Estudo, Implementação, Teste e Avaliação de um Sistema de Posicionamento Dinâmico*, São Paulo, 1989. 1v. PhD Thesis – Escola Politécnica, University of São Paulo (In Portuguese).

[9] Bray, D., 1998, *Dynamic Positioning*, The Oilfield Seamanship Series, Volume 9, Oilfield Publications Ltd. (OPL).

[10] Utkin, V.I., 1978, *Sliding Modes and their application to variable structure systems*, MIR Publishers, Moscow.

[11] Slotine, J.J.E., 1984, "Sliding controller design for non-linear systems", *Int. J. Control*, Vol.40, No.2, pp421-434.

[12] Tannuri, E.A. et al., 2001, "Experimental Validation of a Quasi-Explicit Hydrodynamic Model: Fishtailing Instability of a Single Point Moored Tanker in Rigid Hawser Configuration", *Journal of Ship Research*, Vol.45, No.4, Dezembro, pp302-314.

[13] Leite, A.J.P. et al., 1998, "Current forces in tankers and bifurcation of equilibrium of turret systems: hydrodynamic model and experiments", *Applied Ocean Research*, No.20, pp.145-56.

[14] Bernitsas, M.M., Kekridis, N.S., 1985, "Simulation and Stability of Ship Towing", *International Shipbuilding Progress*, vol 32 (369), May, 1985, pp.112-123.

[15] Abkowitz, M., 1964, *Lectures on ship hydrodynamics, steering and manoeuvrability*, Hydro-Og Aerodynamisk Laboratorium, Report Hy-5, Lyngby, Denmark, pp.113.

[16] Takashina, J., 1986, "Ship manouvering motion due to tugboats and its mathematical model", *Journal of the Society of Naval Architects of Japan*, v.160, p. 93-104.

[17] Wichers, J.E.W., 1987, "The prediction of the behaviour of single point moored tankers", *Developments in Marine Technology*, v. 4, Floating Structures and Offshore Operations, november, p.124-42.

[18] Sphaier, S.H., Fernandes, A.C., Correa, S.H.S., 2000, "Maneuvering coefficients from model testing for FPSO's station keeping behavior", *Proceedings of the 19th International Conference on Offshore Mechanics and Arctic Engineering (OMAE'00)*, 8p.

[19] Pesce, C.P., Tannuri, E.A., 1997, "Stability and Dynamics of Offshore Single Point Mooring Systems", *Revista Brasileira de Ciências Mecânicas*, vol. XIX (4), pp.531-552.

[20] Strand, J.P., Sørensen, A.J., Fossen, T.I., 1998, "Design of automatic assisted position mooring systems for ships", *Modeling, Identification and Control*, Vol. 19, No. 2, pp. 61-75.

[21] Slotine, J.J.E., Li, W., 1991, *Applied Nonlinear Control*, Prentice Hall, New Jersey.

[22] Simos, A.N., 2001, *Modelo hidrodinâmico heurístico para análise de navios petroleiros amarrados sujeitos à ação de correnteza*, São Paulo, 2001. PhD Thesis – Escola Politécnica, University of São Paulo.

