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Ship Motion Control Using Multi-Controller Structure

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ABSTRACT

The aim of this paper is to design a ship autopilot using multi-controller structure by introducing several working points. The control design of each local controller has been achieved using internal model control structure. An illustrative example shows the application of proposed control structure to container ship in real condition of navigation.

1. Introduction

The multi-controller approach has met with great interest in the automation field in the past few decades (Pagès et al, 2002). Any complex system will generally operate in multiple environments. The design of an efficient controller is often difficult because it must ensure satisfactory performance in all environments. Nevertheless, if the different environments are characterized separately, a local controller may be designed for each of them but of course, this requires knowledge about the behaviour of the system in each environment. At any given instant during the operation of the system, the main task will be to determine which model approximates the best plant in order to apply the corresponding controller. Three inherent parts are necessary in the multi-controller approach: the first one is a set of local controllers; the second one is a 'switching system' whose task is to design the control law from the set of local controllers and the last one is a supervisory system which controls the 'switching system' and more precisely, it indicates the most appropriate controller to the latter as shown in Figure 1.

The supervisory system can be omitted when the switching system, from measured information, can choose the best local controller.

Oceans and seas are particular environments, sailed by humankind for economic, social and strategic motives. In the economical point of view, ships have been operating for centuries and the ability to move goods from country to country and between continents by sea has always been the basis of international trade.

Every ship has a domain of operation, outside of which its use is tricky: this domain is defined according to the sea state, the wind, the currents, the speed and the course of the ship. According to its purposes, every ship is equipped by devices

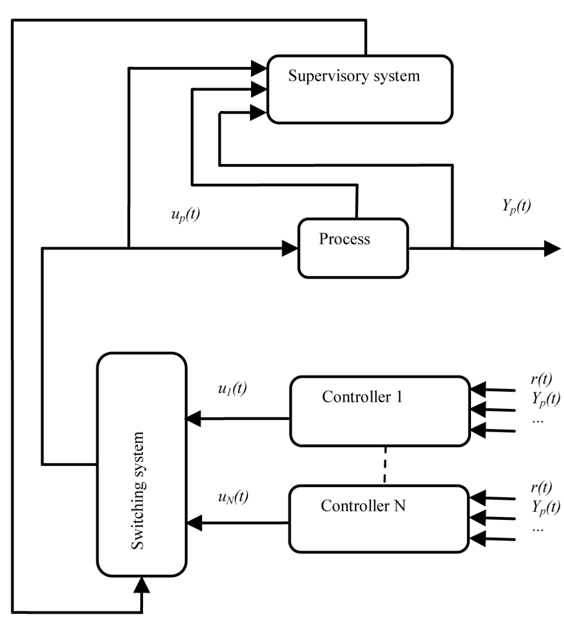

Figure 1. Multi-controller structure.

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permitting to automate maneuvers: automatic pilot, maneuvering device in low speed, dynamic positioning system, stabilization system… etc.

The increase of the maritime traffic and the development of technologies required to use bigger and bigger and more sophisticated ships. Therefore, due to the complexity of these ships, the classical control techniques became obsolete when it comes to ensure their efficient control, and therefore the security of passengers and goods. New control techniques were then necessary.

Our work consists on the use of a rudder control system in order to monitor the course of a container ship in real conditions of navigation. Since the ship model is nonlinear and depends especially from environment, we have opted to use a multi-controller structure to control its maneuvers. In fact, the ship model can be linearized around several working points that depend from the speed of the ship. For each submodel a linear controller is designed and based from the measurement of the speed of the ship, the switching system chooses the adequate controller.

This paper is divided into 3 sections. The first one concerns the modeling of the ship. The study of the movement of the vessel will be conducted first in order to deduce its nonlinear model. Since the latter is difficult to use for computing the control law, a linear model imposes itself. Afterwards, we shall approach the modeling of the rudder which is the main actuator. In section 3, we shall tackle the problem of the design of the ship autopilot using multi-controller structure and then end this paper by some simulation results.

2. Ship Modeling

The environment in which evolves a ship is very variable depending on weather conditions, the season and the geographical location. All these changes in the environment of the vessel have important influence on its movement and its behavior. Therefore, the study of the ship movement is very complex. Obtaining a reliable model requires the determination of the set of the parameters intervening in its dynamics.

Several researches have been made in the domain of the ship modeling and have been mentioned in (Blanke, 1981; Blanke, 1986; Blanke and Christensen, 1993; Blanke and Jensen, 1997; Källström, Wessel and Sjolander, 1988; Pérez and Blake, 2002; Pérez, 2005). Some of the main outcomes are: In 1975 the Abkowitz described the movement of the ship with four degrees of freedom. We have also taken note of the results published by Son and Nomoto in 1982 (Son and Nomoto, 1982) which presented a model obtained by combining planar motion mechanism test data for lateral motion, using different values of static heel for model under test, with independent roll motion tests. There is also the model presented in 1983 by Kälström and Ottersons, which combined a lateral planar motion mechanism model with theoretical estimates of roll coefficients, using free sailing model tests to calibrate the roll parameters.

In this section we present models based on experimental results in the unique 4-DOF and developed by the Danish Maritime institute (Pérez, 2005) that allow model testing with full dynamic interaction between motions in roll, sway, yaw and surge. The models have also been subject to validation via full-scale sea trials (Yang, 1998).

2.1. Non linear model of the ship

We adopt in this paper the SNAME conventions (Society of Naval Architects and Marine Engineers) (SNAME, 1950). The motion of a ship in six degrees of freedom is considered as a translation motion (position) in three directions: surge, sway and heave, and as a rotation motion (orientation) about three axes: roll, pitch and yaw. To determine the equations of motion, two reference frames are considered: inertial or fixed to earth frame *Oxyz* that may be taken to coincide with the ship-fixed coordinates in some initial condition and the body-fixed frame $O_0\, x_0\, y_0\, z_0$ (Figure 2). For surface ships, the most commonly adopted position for body-fixed frame that which gives hull symmetry about the $O_0\, x_0\, z_0$ plane and approximate symmetry about $O_0\, y_0\, z_0$ plane while the origin of the z_0 axis is defined by the calm water surface.

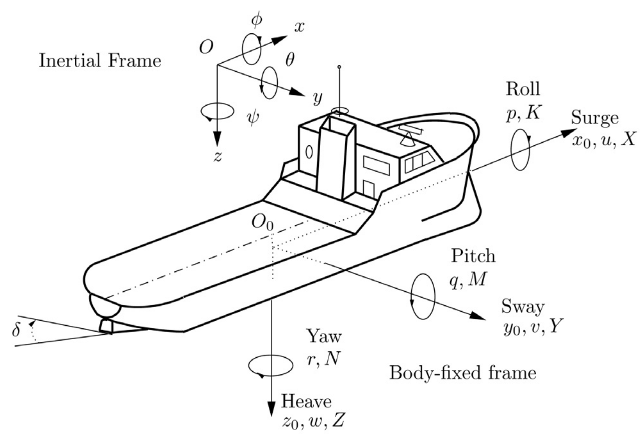

Figure 2. Ship motion description.

The Newtonian approach gives the equations of vehicle in the body fixed frame and considering that motions in pitch and heave can generally be neglected in comparison with the other motions for conventional surface ships and for ship maneuvering studies the oscillating wave force effect (roll) is often neglected.

Hence, ship maneuvering is treated as a horizontal plan motion, and only the surge, sway and yaw modes are considered. The following approximations are set up:

$$\begin{bmatrix} m & 0 & 0 \\ 0 & m & mx_G \\ 0 & mx_G & I_{zz} \end{bmatrix} \begin{bmatrix} \dot{u} \\ \dot{v} \\ \dot{r} \end{bmatrix} = \begin{bmatrix} X \\ Y \\ N \end{bmatrix} + \begin{bmatrix} m(vr + x_G r^2) \\ -mur \\ -mx_G ur \end{bmatrix} \quad (1)$$

$$\dot{\psi} = r \quad (2)$$

where m is mass of the ship, I_{zz} is inertia about z_0 axis, and x_G is the x co-ordinate of the center of gravity.

X, Y and *N* denote the hydrodynamic forces and moment. They result from the movement of the ship on the surface of the water. They depend on the speed, on the weight and on the profile of the hull but also on the effect of waves.

The structure of these forces and moment is shown below, by equations (5) to (7). The results are given as non dimensional quantities using *the prime system* (SNAME, 1950). In this model, the non-dimensional relative surge speed

$$u'_a = \frac{U - U_{nom}}{U} \tag{3}$$

is used in hydrodynamic terms, where U is the ship absolute speed

$$U = \sqrt{u^2 + v^2} \tag{4}$$

It should be noted that u'_a is different from non-dimensional surge velocity $u' = u / U$.

The non-dimensional surge equation is

$$X_{hyd} = X_{\dot{u}}\dot{u}'_a + X_u u'_a + X_{uu} {u'_a}^2 + X_{uuu} {u'_a}^3 + X_{vr} v'r' + X_{rr} r'^2 + X_v v' + + X_{vv} v'^2 \tag{5}$$

The non-dimensional sway equation is

$$\begin{aligned} Y_{hyd} = {} & Y_{\dot{v}}\dot{v}' + Y_{\dot{r}}\dot{r}' + Y_v v' + Y_{vv} v'^2 + Y_{v|v|} v'|v'| + Y_{v|r|} v'|r'| + Y_{vrr} v'r'^2 + Y_r r' + Y_{r|r|} r'|r'| \\ & + Y_{rrr} r'^3 + Y_{r|v|} r'|v'| + Y_{rvv} r'v'^2 + Y_0 + Y_{0u} u'_a \end{aligned} \tag{6}$$

The non-dimensional yaw equation is

$$\begin{aligned} N_{hyd} = {} & N_{\dot{v}}\dot{v}' + N_{\dot{r}}\dot{r}' + N_v v' + N_{vv} v'^2 + N_{v|v|} v'|v'| + N_{v|r|} v'|r'| + N_{vrr} v'r'^2 + N_r r' + N_{r|r|} r'|r'| \\ & + N_{rrr} r'^3 + N_{r|v|} r'|v'| + N_{rvv} r'v'^2 + N_0 + N_{0u} u'_a \end{aligned} \tag{7}$$

The forces and moments due to the rudder acting on the hull are given by:

$$X_{rudder} = X_\delta \delta' + X_{\delta\delta} \delta'^2 + X_{\delta u} \delta' u'_a + X_{\delta\delta u} \delta'^2 u'_a + X_{v\delta} v'\delta' + X_{v\delta\delta} v'\delta'^2 \tag{8}$$

$$Y_{rudder} = Y_\delta \delta' + Y_{\delta\delta} \delta'^2 + Y_{\delta\delta\delta} \delta'^3 + Y_{\delta v} \delta' v' + Y_{\delta vv} \delta' v'^2 + Y_{\delta u} \delta' u'_a + Y_{\delta\delta u} \delta'^2 u'_a + Y_{\delta\delta\delta u} \delta'^3 u'_a \tag{9}$$

$$N_{rudder} = N_\delta \delta' + N_{\delta\delta} \delta'^2 + N_{\delta\delta\delta} \delta'^3 + N_{\delta v} \delta' v' + N_{\delta vv} \delta' v'^2 + N_{\delta u} \delta' u'_a + N_{\delta\delta u} \delta'^2 u'_a + N_{\delta\delta\delta u} \delta'^3 u'_a \tag{10}$$

2.2. Linear ship model

The full nonlinear model is too complex to be used for designing the controller, so a linear model is chosen. It is easily obtained from equations (1) and (2) with hydrodynamic effects in non-dimensional form (5)-(10).

It is a common practice to decouple the surge equation from the others to analyze the linearized models. Thus, we consider a given service speed $\bar{u}$ and the reduced state vector $z = [v \;\; r \;\; \psi]^T$.

The linearized model is obtained at $\bar{z} = [0 \; 0 \; 0]^T$ and $\bar{\delta} = 0$ as

$$H\,\dot{z} = F\,z + G\,\delta \tag{11}$$

where

$$F = \begin{bmatrix} Y_v & (Y_r - m\bar{u}') & 0 \\ N_v & (N_r - m x_G \bar{u}') & 0 \\ 0 & 1 & 0 \end{bmatrix}$$

$$H = \begin{bmatrix} (m - Y_{\dot{v}}) & (m x_G - Y_{\dot{r}}) & 0 \\ (m x_G - N_{\dot{v}}) & (I_{zz} - N_{\dot{r}}) & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

$$G = [Y_\delta \quad N_\delta \quad 0]^T$$

Taking the Laplace transform of (11), one obtains:

$$z = (sH - F)^{-1} \; G \; \delta \tag{12}$$

It can be easily verified that from (12), one obtains the following reduced relation between r and δ:

$$\frac{r(s)}{\delta(s)} = \frac{K(1 + T_3 s)}{(1 + T_1 s)(1 + T_2 s)} \tag{13}$$

Eq. (13) is known as the second order Nomoto model, where K is the static yaw rate gain, and T_1, T_2 and T_3 are time constants.

Sea trial data-based identification results indicate that the values of parameters T_2 and T_3 in (13) are not very different. This suggests further simplification of (13) is possible and 1st order Nomoto model follows:

$$\frac{r(s)}{\delta(s)} = \frac{K}{1 + Ts} \tag{14}$$

where $T = T_1 + T_2 - T_3$ is called the effective yaw rate time constant.

The first order Nomoto model defined by (14) is widely employed in the ship steering autopilot design. The yaw dynamics is characterized by the parameters K and T, which can be easily identified from standard maneuvering tests. In practice, ship steering autopilots are designed for heading angle control. Hence, it is the transfer function relating the heading angle ψ to the rudder angle δ being needed in the autopilot design. Since the yaw rate is actually the time derivative of ψ, the required transfer function can be readily obtained by adding an integrator $1/s$ to the transfer function model defined by (14) and become:

$$\frac{\psi(s)}{\delta(s)} = \frac{K}{s(1 + Ts)} \tag{15}$$

2.3. Model of Steering Machine

The rudder is the main actuator in the control scheme. The mathematical model of the rudder mechanism most commonly used in computer simulations and autopilot design is the simplified model presented by (Amerongen, 1982). In Figure 3, we see a block diagram representation of the model. It contains two limiters, one describing the limitation of the rudder angle and the other describing the limitation of the rudder speed. The rudder limit is either determined by the rudder-angle constraints of the autopilot, or by the mechanical constraints. The maximum rudder speed is determined by the maximum valve opening and the pump capacity of the steering machine. The classification companies and the SOLAS convention (International Convention for the Safety of Life at Sea) require that the rudder should be able to move from 35° port to 35° starboard within 30 seconds. A maximum rudder speed of as low as 2.5 degrees per second is sufficient to meet this requirement.

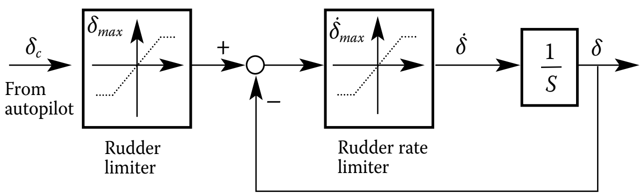

Figure 3. Simplified bock diagram of the steering machine model.

3. Multi-controller Design for ship motion control

Control of a ship system has been widely studied in the recent years (Engeln et al., 2004). The behaviour of these systems is nonlinear, however to simplify the problem, ship model can be linearized around several working points that depend from the speed of the ship. A linear approximation for each subsystem is then applied. Therefore, for each sub-system, a linear control system is designed. The design of linear local controllers is based on internal model control method. It is a model-based approach, which is characterized by explicit dependence between the plant model parameters and the controller parameters (Morari and Zafiriou, 1989). The number of each controller is based upon the number of sub-models of the system. Switching from one controller to another needs logical rules regarding the environment and different conditions of the ship. Selecting different controllers with respect to different sub-models of the system requires intelligent/logical rules.

3.1 Internal Model Control (IMC).

The IMC structure (Morari and Zafiriou, 1989) is shown in Figure 4, where G is the plant to be controlled, $\hat{G}$ is the plant model, Q is the design transfer function, r the reference input, u is the control input, y is the system output, d_i is the input disturbance and d_o is the output disturbance. The dash line box in Figure 4 actually represents the controller C.

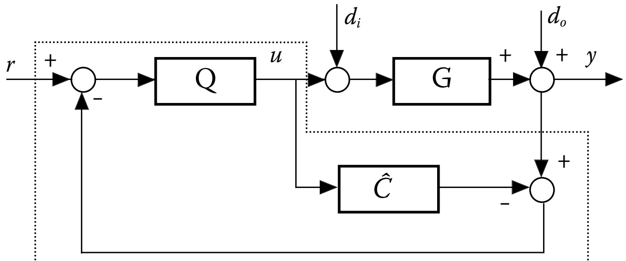

Figure 4. Internal Model Control System structure.

After some simplifications, the dash line box in Figure 4 reduces to a single transfer function C defined by $C = \frac{Q}{1 - Q\hat{G}}$ and the IMC structure of Figure 4 is easily transformed into the classical feedback control structure shown in Figure 5. Because the plant model $\hat{G}$ is imbedded in the controller C, this explains adoption of terminology internal model control.

The IMC structure shown in Figure 5 highlights the reasons for existence of feedback control. Specifically, if G and $\hat{G}$ are the same and if d_o and d_i are both zero, the signal in feedback path vanishes and the system is essentially in open loop state. Namely, if there is neither modeling errors nor disturbances, there is no need to feedback. The definition $C = \frac{Q}{1 - Q\hat{G}}$ appears in Figure 5 shows that there is an explicit relationship between the controller C and the model $\hat{G}$. Once the plant model $\hat{G}$ is fixed and selection of the design transfer function Q is made, the controller C follows immediately. Hence characterization of the controller parameters in terms of the plant model parameters is straightforward, and this is a useful feature for adaptive autopilot application.

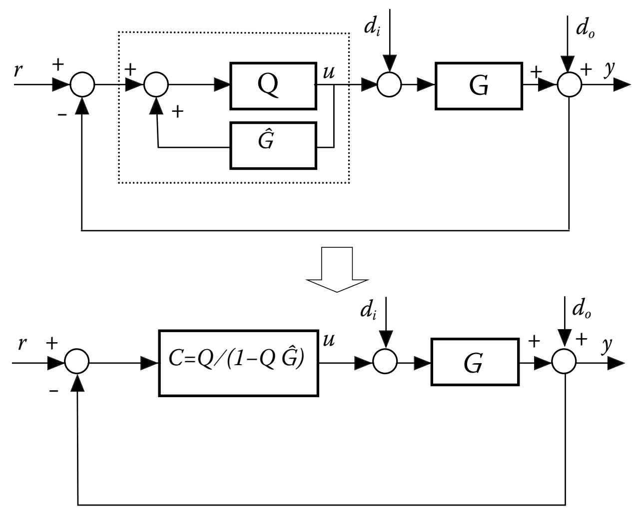

Figure 5. Classical Feedback Control Structure.

3.2. The IMC Design procedure

From Figure 4, the IMC-based controller C is given by:

$$C = \frac{Q}{1 - Q\hat{G}} \tag{17}$$

where Q is a design transfer function defined by:

$$Q = F\hat{G}_{inv} \tag{18}$$

where $\hat{G}_{inv}$ is the approximate inverse model of the plant and F is used in adjusting the closed-loop behavior, called the modulating filter defined as follows (Morari and Zafiriou, 1989):

$$F(s) = \frac{1}{(1 + \beta s)^n} \tag{19}$$

where n is a positive integer, chosen to make Q bi-proper (the order of the denominator equals the order of the numerator) and β is a design parameter that determines the speed of response of the close-loop system.

Direct inversion of the plant model may result in system instability. Hence, it is necessary to decompose the plant model into a stable part and an unstable part, and perform certain approximations in finding the inverse model. This is explained below. Given the plant model

$$\hat{G}(s)=\frac{\hat{B}_s(s)\hat{B}_u(s)}{\hat{A}_s(s)\hat{A}_u(s)} \tag{20}$$

then

$$\hat{G}_{inv}(s)=\frac{\hat{A}_s(s)\hat{A}_u(s)}{\hat{B}_s(s)\hat{B}_u(s)\Big|_{s=0}} \tag{21}$$

where the subscripts in $\hat{A}_s(s)$ and $\hat{A}_u(s)$ indicate the stable and the unstable terms. If the system has right-half-plane zero (i.e. non-minimum system), direct inversion leads to system instability. In (21), only the DC-gain of unstable term $B_u(s)$ is retained. This approximate approach captures part of exact inverse model properties and avoids the instability problem.

3.3. The unstable IMC design

To ensure internal stability of the closed loop system, the following four sensitivity functions are required to be stable [9].

$$\hat{T}=Q\hat{G} \tag{22}$$

(The nominal complementary sensitivity function)

$$\hat{S}_o=1-Q\hat{G} \tag{23}$$

(The nominal output sensitivity function)

$$\hat{S}_i=\left(1-Q\hat{G}\right)\hat{G} \tag{24}$$

(The nominal input sensitivity function)

$$\hat{S}_u=Q \tag{25}$$

(The nominal control sensitivity function)

From (22) to (25), it is easy to see that a stable system $\hat{G}$, selection of a stable design transfer function Q guarantees internal stability. However, for unstable systems, selection of a stable Q is not enough to ensure internal stability. In addition, it is required that the unstable poles of $\hat{G}$ appear as zeros of Q, and the unstable poles of $\hat{G}$ also appear as zeros of $\hat{S}_i=(1-Q\hat{G})\hat{G}$

3.4. Controller design

The followings show the controller design with respect to the marginally unstable ship model given by (15).

$$\hat{G}(s)=\frac{\psi(s)}{\delta(s)}=\frac{K}{s(1+Ts)} \tag{26}$$

According to the design procedures and for (Lee and Tzeng, 2004)

$$F(s)==\frac{3\beta s+1}{(1+\beta s)^3} \tag{27}$$

$$C(s)=\frac{Q(s)}{1-Q(s)\hat{G}(s)}=\frac{F(s)\hat{G}_{inv}(s)}{1-F(s)\hat{G}_{inv}(s)G(s)}=\frac{3\beta Ts^2+(3\beta+T)s+1}{\beta^3ks^2+3\beta^2ks} \tag{28}$$

which can be recast in PID controller format:

$$C_{PID}(s)=K_p+\frac{K_i}{s}+\frac{K_d s}{1+\gamma s}=\frac{(K_d+K_p\gamma)s^2+(K_p+K_i\gamma)s+K_i}{s(1+\gamma s)} \tag{29}$$

where the term $\left(\frac{1}{1+\gamma s}\right)$ is used in eliminating the undesirable differentiating effect at high frequency regions.

Equating (28) to (29) leads to the following PID controller gains:

$$K_i=\frac{1}{3\beta^2k} \tag{30}$$

$$K_p=\frac{\frac{8}{3}\beta+T}{3\beta^2k} \tag{31}$$

$$K_i=\frac{\frac{8}{3}\beta T-\frac{8}{9}\beta^2}{3\beta^2k} \tag{32}$$

$$\gamma=\frac{\beta}{3} \tag{33}$$

From the above derivations, it is clear that for a 2nd order plant (such as (26), the PID controller is a natural choice. Most industrial plants seem to be dominated a pair of conjugate poles; namely, their behaviors are dominated by a 2nd order process. This helps in explaining why the PID controller has been successful in these applications. It can be seen from (30) to (33) that under the IMC structure, the controller gains depend explicitly on the plant parameters (K,T) and the design parameter β. Once the plant model is chosen and the design parameter β is selected, the controller follows immediately. Moreover, the controller obtained via the IMC method is easily realized in the PID format.

4. Simulation results

The simulation concerns a container ship of L=230.66 m length, m = 46.710[6] kg mass and U_{nom} = 12,75 m/s nominal speed. The other parameters necessary for modeling the ship can be found in (Perez and Blake, 2002; Lee and Tzeng, 2004). The container ship will do a crossing from Oran port (Algeria) to Tunis port (Tunisia) via Barcelona (Spain) and La Spezia (Italy) as shown in Figure 6.

The distance from Oran to Barcelona is 648 Km with 70.4° of heading and the distance from Barcelona to La Spezia is 699 Km with 28.4° of heading. Finally, the distance from La Spezia to Tunis is 802 Km with -86.1° of heading.

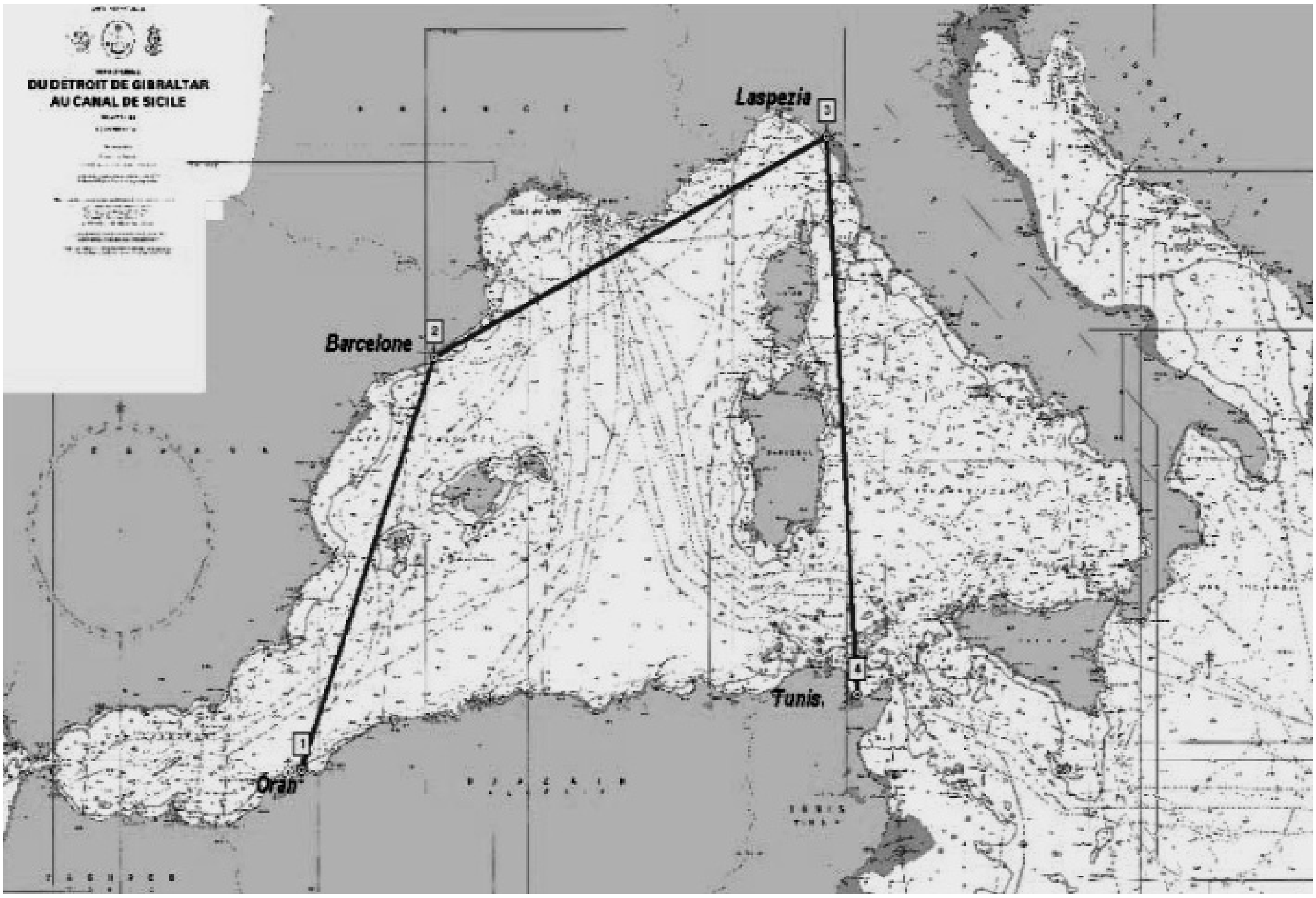

Figure 6. The crossing of the container ship.

During the crossing, the ship will be able to change its speed and its heading at any time.

The ship model is linearized around four set points; ($\bar{z}=[0\,0\,0]^T$, $\bar{\delta}=0$, and $\bar{u}=\{0.5\ U_{nom}, 0.8\ U_{nom}, 1.2\ U_{nom}, 1.5\ U_{nom}\}$) as shown in II.*B.* We obtain the following four simplified sub models (15) and theirs four local controllers (29):

	1st point	2nd point	3rd point	4th point
K	-0.0129	-0.0254	-0.553	-0.1046
T	6.6299	12.1056	20.7414	33.2830
β	40	40	40	40
K_p	-1.8328	-0.9737	-0.4797	-0.2787
K_i	-0.0162	-0.0082	-0.0038	-0.002
K_d	11.5670	1.0736	-2.9795	-4.238
γ	13.33	13.33	13.33	13.33

Table 1. Local model and controller parameter

The switching system will be able to select one of the controllers depending on the speed of the ship:

- If $\bar{u} \leq 0.5 \cdot U_{nom}$ then controller 1 is chosen
- If $0.5\ U_{nom} < \bar{u} \leq 0.8 \cdot U_{nom}$ then controller 2 is chosen
- If $0.8\ U_{nom} < \bar{u} \leq 1.2 \cdot U_{nom}$ then controller 3 is chosen
- If $\bar{u} > 1.2 \cdot U_{nom}$ then controller 4 is chosen.

Figure 7 shows the changes of yaw angle during the crossing. Figure 8 shows the rudder angle. One can see that the rudder angle is far from the limits of saturations except when changing the yaw reference. Figure 9 shows the switching between the different controllers during the crossing. The switching system selects, when the speed of the ship varies, the adequate controller in order to achieve the control goal. Finally, Figure 10 shows the simulation of the crossing of the container ship between the different ports cited above.

The time to go from Oran to Barcelona is 16 hours, from Barcelona to La Spezia is 20h 32m and from La Spezia to Tunis is 19h 28m.

5. Conclusion

In this paper, we have described the movements of the ship by the differential equations. These movements are generated by different phenomena: the propulsion of the ship, the effect of waves and the rudder motion. From the mechanical equations one gets the state representation of the ship that is given in nonlinear, and then we deduce the linear model that it used for synthesizing a control law. By introducing several working points, a multi-controller structure was introduced to improve the motion of the ship. The control design of each local controller has been achieved using IMC structure. The IMC design method is easy to follow and it is straightforward to recast the controller into PID format, which allows direct implementation with existing software. Thus, a family of PID controllers

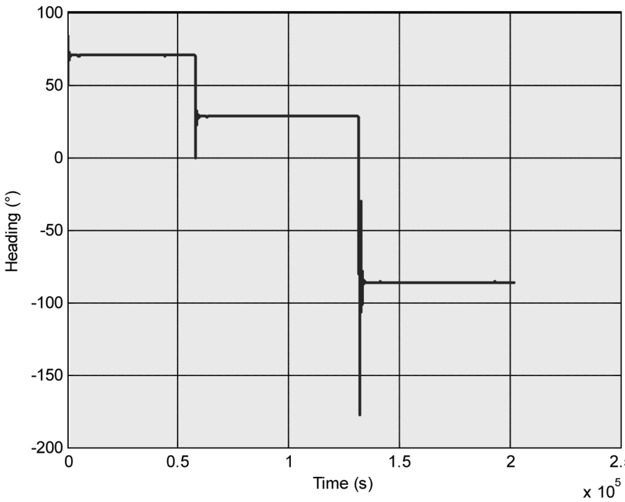

Figure 7. Yaw angle.

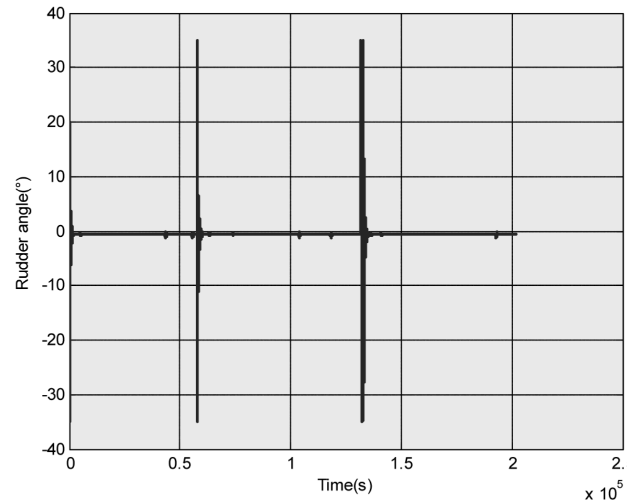

Figure 8. Rudder angle.

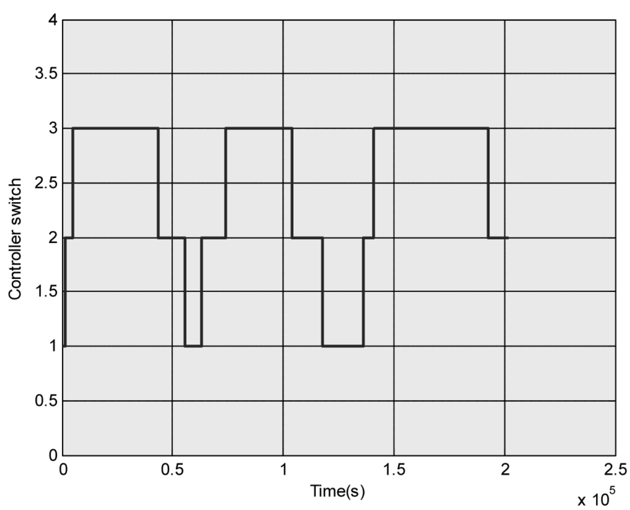

Figure 9. Controller switches during the crossing.

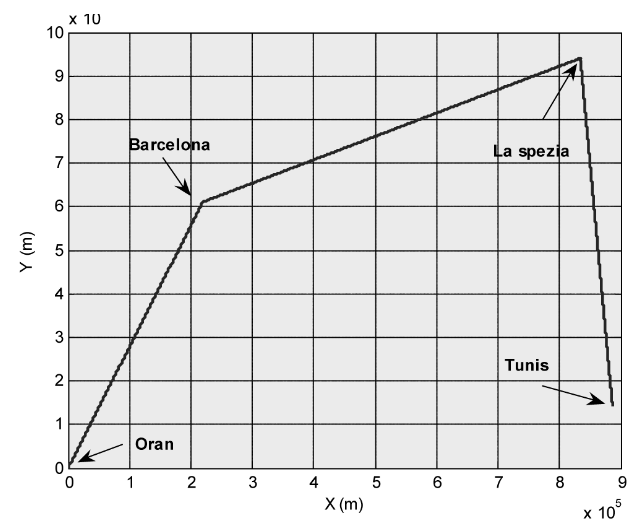

Figure 10. Simulation of the crossing of the container ship.

has been designed. Each controller can be switched to another depending to the environment, especially the speed of the ship. This has been achieved by considering a set of rules.

To test the efficient of the control law, one has simulated the crossing of the container ship around several Mediterranean ports.

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