Performance of a Navy Ship Roll Stabilisation System

Sérgio Ribeiro e Silva, *Technical University of Lisbon*

Nuno Fonseca, *Technical University of Lisbon*

C. Guedes Soares, *Technical University of Lisbon*

ABSTRACT

This paper presents a methodology of assessment of a roll stabilisation system for a naval patrol vessel on a certain ocean area where the ship is intended to operate. The 670 tonnes naval patrol vessels considered in this paper is intended to perform law enforcement, fisheries protection, defence operations, and search and rescue primary missions within the Portuguese Economic Exclusive Zone at a maximum speed of 25 knots. The initial roll stabilisation systems under consideration included bilge keels, passive "U" type tanks, activated fins, and combined systems. Ship operational effectiveness is statistically presented by means of an operability index. The calculation of the operability index, which represents the percentage of time during which the ship is able to accomplish a specific task, depends on the wave climate, the dynamic response of the ship's bare hull to the waves, the dynamic response of the roll stabilisation system, and the criteria adopted.

Keywords: *Ship Roll Stabilisation Design, Passive Tanks, Active Fins*

1. INTRODUCTION

The operability of a naval vessel is affected by the motions and accelerations due to rolling, which can reduce both crew and propulsion efficiencies. However, a variety of roll stabilisation systems are nowadays available to reduce this degradation to carry out the mission, in some cases with appreciable economic savings. Hence, a problem that was addressed during the preliminary ship design stage of a naval patrol vessels considered in this paper was to decide whether bilge keels alone were adequate to keep motions and accelerations within acceptable limits or whether these should be supplemented or replaced by another more cost effective stabilisation system. Soon it was assumed that despite the existing financial constraints, additional stabilisation was required to fulfil the vessel's operational requirements. Therefore, considering the transit & patrol main mission and other specific tasks conducted at lower ship speeds, it was necessary to decide whether a simple passive tank would suffice or whether it should be complemented with an active device. Moreover, if an active stabiliser was selected, it should be decided as well how much heeling moment it should be capable of applying to the ship, and the manner in which the stabiliser controller could be designed to the particular motions and speed profile of the vessel.

So it was necessary to represent the range of sea states which the naval patrol vessels might encounter in design conditions, and to develop a numerical model that could be utilised to specify the capacity required by the stabiliser to give acceptable motions and acceleration due to rolling for each proportion of the sea time of the vessel allocated to a given task.

Therefore, this paper presents a methodology to calculate the operability index for any particular naval vessel to the pre-selected seakeeping criteria, which then allows the designer to define the most suitable passive or active roll stabilisation system for a certain mode of operation.

2. THEORETICAL BACKGROUND

2.1 Evaluation Methodology of the Roll Stabilisation System

The method outlined here allows designers to assess whether a given hull with certain roll stabilisation system is suitable for its intended role. The first step is to obtain a top-level description of the ship's general missions and activities, with their associated frequency distributions and speed profiles so that designers can select or define criteria for assessing the effects of ship motions and related phenomena on the ship's operability The method basically follows four steps:

1) Calculate the transfer functions of the absolute ship motions and of some derived responses such as accelerations and relative motions;

2) Secondly it is necessary to calculate the ship responses to the complete range of short-term seastates;

3) Select seakeeping criteria and calculate curves of maximum significant wave height in which the ship can operate;

4) Calculate the percentage of time that the ship is operational (seakeeping index) in a given design condition.

The difference in the amount of time that the stabilised or unstabilised vessel can stay within specified motion limits or that the crew can perform efficiently can be easily estimated from results of step 4.

2.2 Seakeeping Performance

This section briefly presents the theory behind the method to calculate the seakeeping performance of ships. The method was first used by Fonseca and Guedes Soares (2002) to investigate the seakeeping performance of a fishing vessel and a container ship. More recently, after some full scale validation trials conducted by Ribeiro e Silva et al. (2005), enhanced capabilities have been added to the frequency domain computer simulation codes in order to model and assess stabilised ship responses fitted with bilge keels, passive "U" type tanks or active fins.

2.2.1 Ship Responses to Regular Waves

Newton law governs the vessel dynamics:

$$[M]\{\ddot{\xi}\} = [F] \tag{1}$$

The excitation forces $[F]$ and the motions $\{\xi\}$ can be conveniently represented on a right handed Cartesian coordinate system, $X = (x, y, z)$, fixed with respect to the mean position of the ship and origin in the plane of the undisturbed free surface. As shown in figure 1, the translatory displacements in the x, y, and z directions are respectively surge ξ_1, sway ξ_2, and heave ξ_3, while the rotational displacements about the same axis are respectively roll ξ_4, pitch ξ_5, and yaw ξ_6.

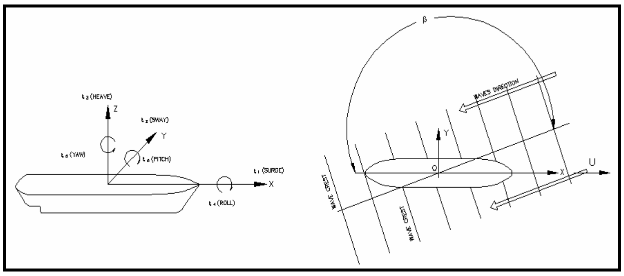


Figure 1 The right-hand coordinate system and six modes of ship motion, and definition of the ship heading angle.

In an approximate way, radiation and wave excitation forces are then calculated at the equilibrium waterline using a standard strip theory, where the two-dimensional frequency-dependent coefficients of added mass and damping, and the sectional diffraction forces are computed by the Frank close fit method.

Under the assumptions presented previously, all hydrodynamic forces are linear and when these are combined with the inertia forces, five linear coupled differential equations of motion are obtained, given by:

$$\sum_{j=2}^{6}\left\{\left(M_{kj}+A_{kj}\right)\ddot{\xi}_j+B_{kj}\dot{\xi}_j+C_{kj}\xi_j\right\}=F_k\,,\quad k,j=2,\ldots,6 \qquad (2)$$

with the subscripts, k, j indicating forces in the k-direction due to motions in the j-mode. M_{kj} are the components of the mass matrix for the ship, A_{kj} and B_{kj} are the added mass and damping coefficients, C_{kj} are the hydrostatic restoring coefficients and F_k are the complex amplitudes of the exciting forces.

The harmonic j-th response of the vessel, ξ_j will be proportional to the amplitude of the exciting force, at the same frequency but with phase shift, θ_j, and is then given by:

$$\xi_j(t)=\xi_j^a\cos\left(\omega_e t+\theta_j\right),\ j=1,\ldots,6 \qquad (3)$$

If the ship travels at a speed U making an angle β with the direction of incoming waves (see figure 1), she will encounter regular wave crests with a frequency of encounter, given by:

$$\omega_e=\omega-kU\cos\beta \qquad (4)$$

The encountered free surface is given by:

$$\zeta_w=\zeta_w^a\cos k\left[x\cos\beta+y\sin\beta-\left(c-U\cos\beta\right)t\right] \qquad (5)$$

This study uses a seakeeping code to calculate the potential flow hydrodynamic coefficients and harmonic wave exciting forces. However, it should be noted that it is essential to have an accurate prediction of the viscous roll damping in order to realistically calculate the roll responses amplitudes.

2.2.2 Roll Stabilisation Systems Performance

More detailed information about the three roll stabilisation systems described below, are provided by Ikeda et al (1975), Lloyd (1998) and Conolly (1969), respectively.

• Bilge Keels – these appendages work by generating drag forces and pressure fields onto the hull surface that, by means of an increase of hull damping forces, will oppose the rolling motion of the ship. Using a component based prediction method the effect of bilge keels and the total roll damping of a ship is estimated. The bilge keels component is divided into two sub-components, the sub-component due to normal force of the bilge keels and the sub-component due to pressure on the hull surface created by bilge keels. The normal sub-component is derived from experimental results of oscillating flat plates, whose drag coefficient can be expressed as follow:

$$C_D=22.5\frac{b_{BK}}{\pi r\theta_0 f}+2.4 \qquad (6)$$

where θ_0 is the roll amplitude, b_{BK} is the breadth of bilge keel, r is the distance between the roll axis and the bilge keel, and f is a correction factor to take into account the flow speed increase at the bilge;

• Passive "U" Type Tank – passive stabilisers are designed to provide good dynamic coupling between the stabiliser and ship, by proper selection of the natural frequency.

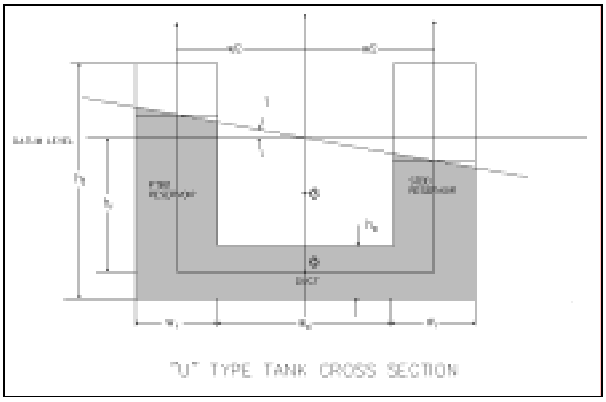


Figure 2 Definition of the passive "U" type

tank dimensions.

The tank natural frequency (ω_t) is determined by the values of tank dimensions (shown in figure 2), and the mass of working fluid (m_t), given by:

$$\omega_t = \sqrt{\frac{c_{\tau\tau}}{a_{\tau\tau}}} = \sqrt{\frac{2gh_d}{w_r w + 2h_r h_d}} \tag{7}$$

$$m_t = \rho_t x_t \left(w h_d + 2 h_r w_r \right) \tag{8}$$

Where $c_{\tau\tau}$ and $a_{\tau\tau}$ denote the displacement and the acceleration coefficients of the applied roll moment necessary to sustain a steady tank angle (τ) of the fluid.

In the numerical model adopted, the tank angle may be regarded as an additional degree-of-freedom (DOF) in the equations of motion for the ship. Its effects are taken into account by including additional terms into the lateral plane equations of motion.

Introducing the effect of a passive "U" type tank stabiliser into (1), noticing that only $k = 2$, 4, 6 directions are affected and that these horizontal plane motions may be decoupled from the vertical plane ones (Abkowitz, 1972), the following four DOF - sway, roll, yaw and motion of the fluid in the tank – equations are obtained:

$$\begin{bmatrix} M + A_{22} & A_{24} & A_{26} & a_{\tau 2} \\ A_{42} & M_{44} + A_{44} & A_{46} & -a_{\tau 4} \\ A_{62} & A_{64} & M_{66} + A_{66} & a_{\tau 6} \\ a_{\tau 2} & a_{\tau 4} & a_{\tau 6} & a_{\tau\tau} \end{bmatrix} \begin{bmatrix} \ddot{\xi}_2 \\ \ddot{\xi}_4 \\ \ddot{\xi}_6 \\ \ddot{\tau} \end{bmatrix} + \begin{bmatrix} B_{22} & B_{24} & B_{26} & 0 \\ B_{42} & B_{44} & B_{46} & 0 \\ B_{62} & B_{64} & B_{66} & 0 \\ 0 & 0 & 0 & b_{\tau\tau} \end{bmatrix} \begin{bmatrix} \dot{\xi}_2 \\ \dot{\xi}_4 \\ \dot{\xi}_6 \\ \dot{\tau} \end{bmatrix} + \begin{bmatrix} 0 & 0 & C_{26} & 0 \\ 0 & C_{44} & C_{46} & -c_{\tau 4} \\ 0 & 0 & C_{66} & 0 \\ 0 & 0 & 0 & c_{\tau\tau} \end{bmatrix} \begin{bmatrix} \xi_2 \\ \xi_4 \\ \xi_6 \\ \tau \end{bmatrix} = \begin{bmatrix} F^a_{W2} \sin(\omega_e t + \theta_2) \\ F^a_{W4} \sin(\omega_e t + \theta_4) \\ F^a_{W6} \sin(\omega_e t + \theta_6) \\ 0 \end{bmatrix} \tag{9}$$

Notice should be given to the fact that the tank stabilising moment ($a_{\tau 4}\ddot{\tau} + c_{\tau 4}\tau$) has to be always subtracted to left-hand-side of equation;

• Active Fins – the performance of fin stabilisers is determined by the fin static roll angle and hence primarily by the fin projected area. Anti-roll fins are used to reduce roll motion for medium to high speed and are usually ineffective below 10 knots. Consequently, these devices should be installed in conjuction either with bilge keels or the passive tanks referred before. Active fin stabilisers are fin-type control surfaces, which are installed onto the ship's hull in a position just above the turn of bilge, near amidships, port (PTBD) and starboard (STBD). Figure 3 shows the forces and moments applied to the ship, by a pair of fins, at an angle of incidence α, to the seawater flow. Each fin develops a lift force given by:

$$F_L = C_L \frac{1}{2} \rho \mathrm{U}^2 \mathrm{A_F} = \left(\frac{\mathrm{dC_L}}{\mathrm{d}\alpha} \right)^{3D} \alpha \frac{1}{2} \rho_{SW} \mathrm{U}^2 \mathrm{A_F} \tag{10}$$

Thus, the fins exert a roll moment $F_{F4} = 2F_L r_F$ about the centre of gravity, where the fin lever arm r_F is measured from the axis, through the centre of gravity, to the lift vector (assumed to be acting at the centre of pressure of each fin, taken to be placed at a distance of one third of the span of the fin measured from its root).

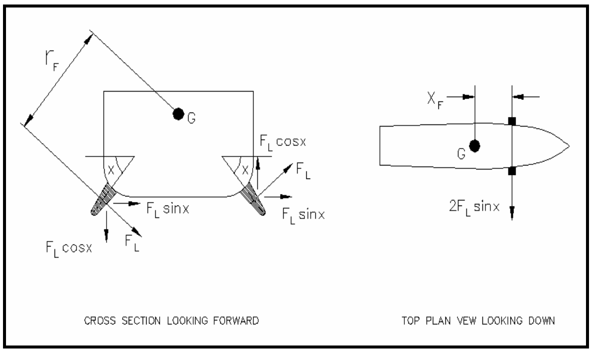


Figure 3 Definition of the fins position and orientation angles over the ship's hull.

If a simple 1 DOF uncoupled rolling model is adopted for controller design, then the stabilised roll equation of motion is modelled as a linear second-order differential equation, given by:

$$\left(M_{44} + A_{44}\right)\ddot{\xi}_4 + B_{44}\dot{\xi}_4 + C_{44}\xi_4 = F_{W4} + F_{F4} \qquad (11)$$

and with a Proportional, Integral and Derivative (PID) controller, the stabiliser displacement is proportional to the sum of the roll, displacement, velocity and acceleration, given by:

$$\alpha_D = K_1\xi_4 + K_2\xi_4 + K_3\xi_4 \qquad (12)$$

where: $K_1 =$ roll angle sensitivity, $K_2 =$ roll velocity sensitivity, and $K_3 =$ roll acceleration sensitivity.

2.2.3 Ship Responses to Irregular Waves

The method described in the previous section allows the assessment of either stabilised or unstabilised transfer functions, i.e. the response of the ship to harmonic exciting waves of unit amplitude. To determine the response of the ships to real seastates a spectral formulation is adopted. The Pierson-Moskowitz spectral form for fully developed seas describes the irregular seastates in terms of significant wave height H_S and peak period T_P.

In irregular seas the encountered wave profile is given by:

$$\zeta_w = \sum_{n=1}^{N} \zeta_{w_n}^a \cos\left[\begin{array}{l} \dfrac{\omega_n^{\ 2}}{g}\left(x\cos\beta + y\sin\beta\right) \\ -\left(\omega_n - \dfrac{\omega_n^{\ 2}}{g}U\cos\beta\right)t + \varepsilon_n \end{array}\right] \qquad (13)$$

where N is the number of component waves, ω_n the circular frequency, ε_n the random phase angle and $\zeta_{w_n}^a$ the amplitude of the n-th component waves, which are determined from the wave spectrum $S_W(\omega)$.

Because the system is linear, the relationship between the wave spectrum and that of the j-th response and is given by:

$$S_{\xi_j}(\omega) = \left|H_j(\omega)\right|^2 \ S_W(\omega) \qquad (14)$$

where $H_j(\omega)$ is the transfer function from wave elevation to the j-th mode.

The variance of a record is given by the zero order moment of each ordinate, as follows:

$$m_{0_j} = \sigma^2 = \int_0^{\omega} S_{\xi_j}(\omega)\ d\omega_e \qquad (15)$$

which is applicable to both the input and the response spectrum since the seasptate is modelled as a stationary, zero mean, Gaussian process and because the responses are linear, the same model describes the response process. This implies that a Rayleigh distribution describes the amplitudes or the peaks of the processes, according to which the probability of exceeding the level r is given by:

$$Q_S(r) = \exp\left(-\frac{r^2}{2\sigma^2}\right) \qquad (16)$$

Different statistics can be derived from the assumption of the Rayleigh distribution. For example the average of the one-third larger amplitudes, usually called the significant value r_S, is given by:

$$r_S = 2\sigma \qquad (17)$$

The most probable maximum value in N successive cycles is obtained from (16):

$$r_{\max} = \sqrt{2\sigma^2 \ln N} \qquad (18)$$

2.2.4 Seakeeping Assessment

In irregular seas, short term and long term distributions can be used to estimate the most probable maximum values of the responses. However these results alone are not a good measure of the seakeeping quality of a ship. The seakeeping quality will be quantified by an operability index that measures the degradation of the ship ability to carry out its mission comparatively to the calm water condition. This way the operability index represents the percentage of time during which the ship responses are bellow to those defined by the criteria.

If the criterion is defined as a probability of exceeding a critical value p_{CR}, then the corresponding root mean square of the response pis obtained from equation (18) and given by:

$$\sigma_{CR} = \sqrt{\frac{r_{\max}^2}{2\ln(1/p_{CR})}} \tag{19}$$

where $r_{\max}$ is the limiting magnitude of the response which has the probability p_{CR} of being exceeded. As an example, for green water on deck phenomenon $r_{\max}$ is usually the freeboard on the bow.

Using all the relevant seakeeping criteria it is possible to calculate the distribution of $H_{S\max}(T_z, \beta)$ for all mean wave periods of interest. Finally with the probability distribution of short-term seastates for a given ocean area, it is possible to select all seastates where the ship is operational. Summing up the probabilities of occurrence of these seastates, one obtains the expected probability that the ship operates satisfying the defined criteria. This is the operability index.

3. CASE STUDY

A 670 tonnes semi-displacement patrol vessel is considered as an example of the selection and evaluation procedures suggested. The vessels have been designed to perform law enforcement, fisheries protection, defence operations, and search and rescue primary missions within the Portuguese Economic Exclusive Zone at a maximum speed of 25 knots.

3.1 Operational Profile and Ship Main Characteristics

3.1.1 Operating Profile

The patrol vessel will be based either in Portuguese mainland coastal zones or in the Portuguese archipelagos of Madeira and Azores; her missions may last up to 10 days and the expected time at sea is 2,000 hours per year. It is estimated that 70% of the time at sea will be allocated for offshore and fisheries protection activities, which might include inspections onboard fishing vessels using a Rigid-Hull Inflatable Boat (RHIB) to transport the boarding team.

The wave climate statistics used are based on visual observations (Hogben, da Cunha and Olliver, 1986), and the vessel is expected to operate on area 16 from the referred Global Wave Statistics data base. From the scatter diagram, a mean significant wave height and zero crossing wave period of $H_S = 3.24$ (m) and $T_z = 8.42$ (sec) were obtained.

The frequency of ship speed and course might be represented as well by the scatter diagrams. For the naval patrol vessels the choice of speed and course will depend upon operational scenario under consideration. As shown in figure 4, in this study a simplified distribution of speeds and headings has been adopted where most courses are equally likely and a shorter range of speeds is demanded.

The regular crew size is only 20 persons because these vessels will have a high level of automation and control to run the propulsion system and the platform. The control station will be installed at the bridge, where three persons will be permanently carrying out essentially intellectual work.

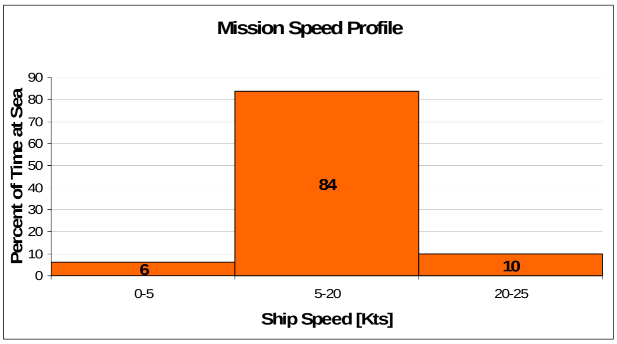


Figure 4 Simplified mission speed profile of the patrol vessel.

The vessel operates at low speed during offshore protection and fisheries inspection missions. The offshore protection might involve launch-and-recovery of a 10 metres RHIB in length and capable of attain a top speed 45 knots, designated as Rapid Assault Craft (RAC) from the stern dock. As illustrated in figure 5, at the stern there is a dock with two vertical hinged doors, and surrounded by several tanks, that can be ballasted in 15 minutes to change trim by the stern by 1 meter and therefore allow launch-and-recovery operations of the RAC by the stern. Realistic model testing in varied wave conditions are due in a near future to assess procedures for capturing the RAC.

Figure 5 Profile view of the patrol vessel.

During offshore and fisheries protection missions, the vessel might need to operate at the top speed of 25 knots for interception purposes and eventually the electronically stabilised gun located in the fore station might need to be reloaded by personnel. Most frequently, one of the two 6.5 and 8 metres RHIBs located on the starboard and portboard sides of the funnel will be launched and then utilised by the crew to conduct outboard inspections in the low speed regime (0-5 knots).

3.1.2 Main Characteristics

The patrol vessel is a slender twin-controllable-pitch propeller hull with deep-V forms in the fore body, and a deadrise at the aft body. The vessel has a main section with rising floor and tumbled sides. Figure 6 presents the bodylines and a 3D view of the ship's hull, while table 1 presents the ship's main particulars.

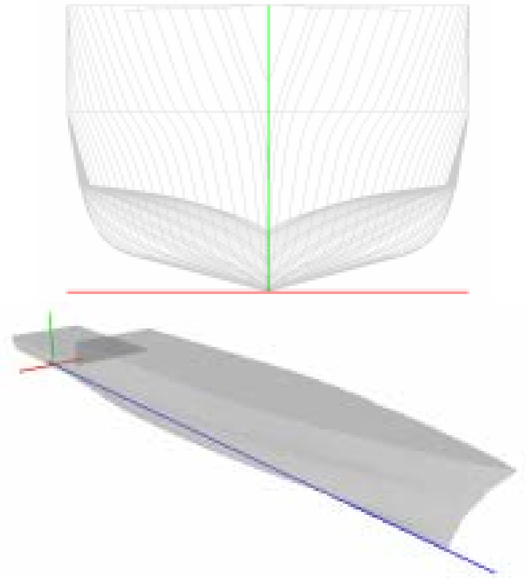


Figure 6 Body plan and 3D view of the hull of the patrol vessel.

Table 1 Main particulars of the patrol vessel.

Length betw. perp., L_{pp} (m)	57.5
Beam at waterline, B (m)	8.5
Draught, T (m)	2.70
Displacement, Δ (ton)	674
Long. pos. of CG (m)	-1.58
Vert. pos. of CG (m)	1.44
Block coefficient, C_b	0.50
Roll gyr. Radius (K_{xx}/B)	0.36
Metacentric height (m)	0.59
Natural roll period (sec)	9.6
Stabiliser design speed (kts)	15.4

3.2 Estimates of Roll Damping and Roll Stabilisation System Selection

3.2.1 Estimates of Roll Damping (B_{44}^*)

From wave making roll damping coefficient (B_W) obtained from strip theory, total roll damping coefficient (B_{44}^*) has been estimated with an iteration routine based on the Ikeda's components method. The components of viscous roll damping that are accounted for in this method include those due to shedding, skin friction and the appendage drag/lift forces. This method finds a linearised roll damping at a certain ship's speed and natural roll period for the maximum roll amplitude. Figure 7 displays the effect of ship's speed and bilge keels on roll damping, using the Ikeda's method. Roll damping due to these appendages includes a definition of their position and size, which has been calibrated by Ribeiro e Silva et al. (2005).

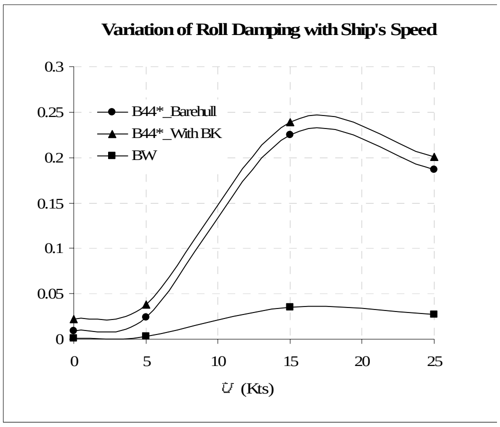


Figure 7 Effects of ship's speed and bilge keels over total roll damping (using Ikeda's components method).

As expected, unappended and appended hulls have larger roll damping at higher speeds. However, as can be seen in figure 7 this general trend has also variations imposed by the wave damping component at each speed.

3.2.2 Roll Stabilisation System Selection

The following solutions have been considered in the assessment:

- A pair of triangular shape bilge keels, 0.5 metres wide with its trailing edge located 25.11 metres in front of the aft perpendicular and 0.73 metres above the baseline. The length of the bilge keels is approximately 9 metres;
- A passive "U" type tank stabiliser has been also selected for these vessels due to low speed requirements for a large number of launch-and-recovery operations; and the chance of swing utilisation between the roll stabilisation system and the stern ballast system. After review of available space and deadweight, a "U" type tank with 3 metres in length has been selected to be installed around the stern dock, located aft of the aft machinery space. As shown in table 2, to achieve a natural frequency of the passive tank slightly higher than vessel's natural roll frequency of 0.65 (rad/sec), a connecting duct between the two reservoirs with 0.4 metres height has been selected.

Table 2 Characteristics of the passive "U" type tank.

Width of each reservoir (w_r)	*2.54 [m]*
Width of the connecting duct (w_d)	*3.5 [m]*
Height of reservoirs (h_r)	*1.5 [m]*
Heigth of the connecting duct (h_d)	*0.4 [m]*
Height of the tank (h_t)	*3.2 [m]*
Length of the tank (x_t)	*3.0 [m]*
Working fluid mass (m_t)	*28 [ton]*
Tank natural frequency (ω_t)	*0.68 [rad/sec]*

Notice should be given to the fact that to allow swing of utilisation and to prevent saturation of these passive "U" type tanks, at high frequencies of encounter with waves, the 28 tonnes of working fluid (saltwater) can easily be pumped outboard by two ballast pumps of 300 (m^3/hr), each;

- Anti-roll active fins have been also selected for these vessels due to medium to high speed requirements, which can play a significant role during offshore protection missions, and SAR operations.

Table 3 shows the main characteristics of the active anti-roll fins proposed to the Portuguese

Navy by the manufacturer Rolls-Royce of the Gemini type, size 10. Estimated lift force generated on each fin at 15 knots is $F_L = 3.9$ tonnes-force. Due to commercial in confidence nature of this type of technical information, general, speed-dependent, displacement, velocity and acceleration gains utilised in the design of the PID controller are not be presented in here.

Table 3 Characteristics of the active anti-roll fins, Rolls-Royce type Gemini, size 10.

Profile Area (A_F)	*1.4 [m2]*
Outreach (b_F)	*1.17[m]*
Mean Chord ($\bar{c}$)	*1.2 [m]*
Fin Span (b_F)	*1.17 [m]*
Fin Lever (r_F)	*4.1 [m]*
3D Lift Coefficient Slope	*0.043 [1/deg]*
Section Shape	*NACA 0015*

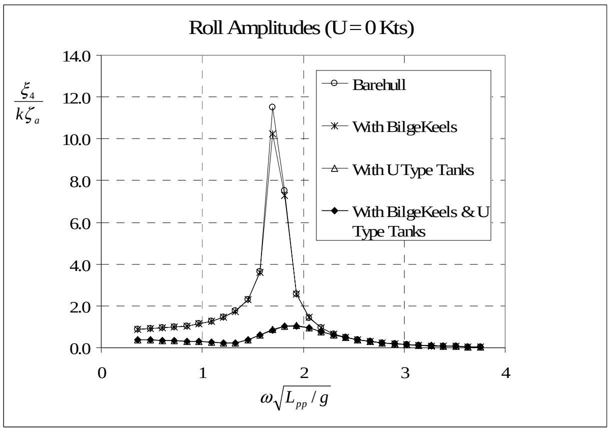
3.3 Seakeeping Performance

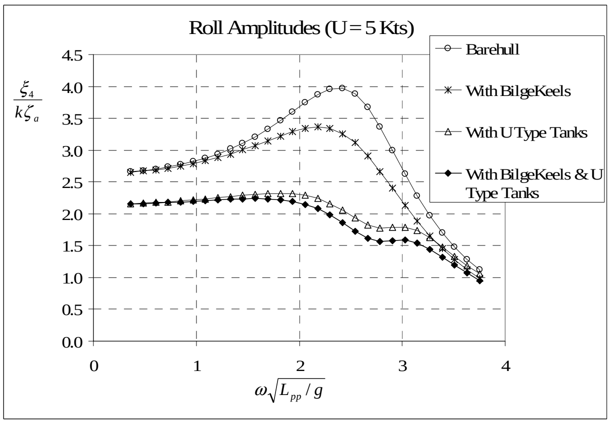
Calculations

The methodology presented in section 2.2 is applied to calculate the operability indices of the patrol vessel for the pre-defined mission profiles.

The first step is to calculate the relevant ship response transfer functions for all directions between following waves (0º) and head waves (180º). In practice a separation between headings of 45º is sought as adequate, resulting in five headings. The transfer functions include absolute ships motions, and derived responses at selected positions on the vessel, such as relative motions and vertical and lateral accelerations. This type of results is illustrated in figure 8 that presents the transfer functions of roll in beam seas for the naval patrol vessel with different types of roll stabilisation systems. The graphs include different speeds and the most adequate types of roll stabilisers for each regime.

Additionally, vertical and lateral accelerations at the working stations (bridge, gun reload station, RHIB stations,) are used to define additional criteria for the patrol vessel. The criteria are defined either in terms of the root mean square of the response (rms) or in terms of permitted probability of occurrence (prob).





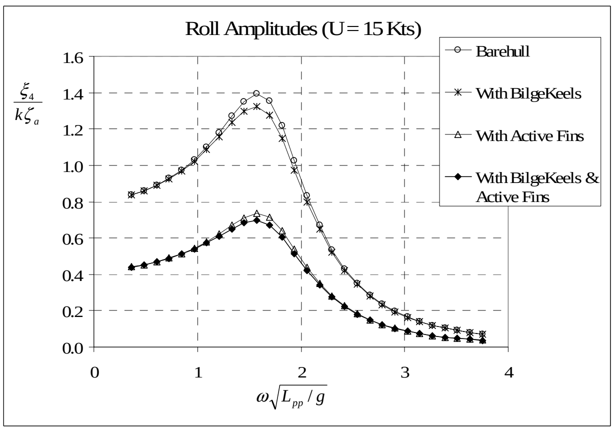


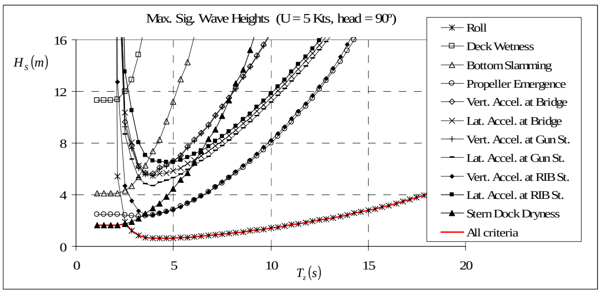
Figure 8 Transfer function amplitudes of unstabilised and stabilised roll responses in beam waves for the patrol vessel at speeds of 0,

5 and 15 knots.

It is assumed that green water on deck occurs when the relative motion is larger than the freeboard on the bow. A bottom slam occurs when the relative motion is larger than the draft at 10% of the L_{BP} from the bow. A propeller emergence occurs when 1/4 of the propeller diameter comes out of the water. Finally, stern dock becomes dry when the stern transom position corresponding to the new ballasted-waterline comes out of the water.

Table 4 Seakeeping criteria for the patrol vessel applied at different mission profiles.

Response	**Location** (m) ***x*,*y*,*z***	**Criterion**
Whole body dynamics		
Roll	-	3º (rms)
Green water on deck	24.58, 0, 6.9	5% (prob)
Bottom slamming	21.70, 0, -2.7	3% (prob)
Propeller emergence	-27.15, 2.15, -1.2	15%(prb)
Low to high speed mission		**0-25 Kts**
Vert. accel. at bridge	3.33, 0, 8.3	0.1g (rms)
Lat. accel. at bridge	3.33, 0, 8.3	0.05g (rms)
Vert. accel. gun st.	15.33, 0, 4.95	0.2g (rms)
Lat. accel. gun st.	15.33, 0, 4.95	0.1g (rms)
Low speed missions		**0-5 Kts**
Vert. accel. at RIB st.	-11.67, 4.0, 6.3	0.2g (rms)
Lat. accel. at RIB st.	7.6, 0, 4.5	0.1g (rms)
Stern dock dryness	-8.5, 0, 2.5	5% (prob)



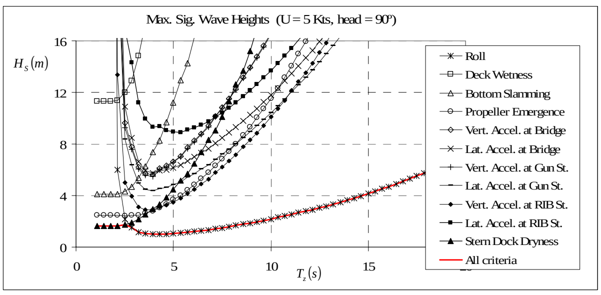
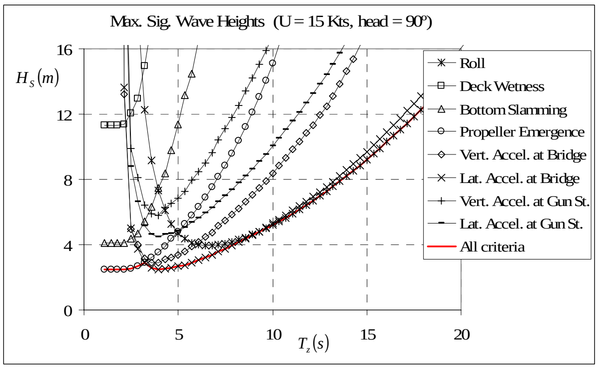


Figure 9 Maximum allowed significant wave heights for the patrol vessel without roll stabilisation system (top) and with passive “U” type tanks (bottom), heading beam waves at low speeds.

Next it is necessary to calculate the ship responses to stationary seastates of unit significant wave height for each of the responses. The calculations are done for the all range of mean wave periods. Comparing the resulting root mean square of the responses with the seakeeping criteria, it is possible to compute curves of maximum significant wave height as function of the mean wave period. Figure 9, presents these results for the patrol vessel without and with roll stabilisation at the advance speed of 5 knots. The vessel is operational for the seastates that are below all the curves.

The operational limit curves and the impact of the roll stabilisation systems on these operational limits at the medium speed regime are shown in figure 10.



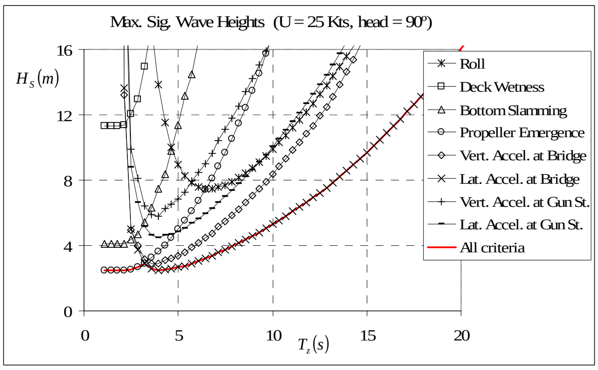


Figure 10 Maximum allowed significant wave heights for the patrol vessel without roll stabilisation system (top) and with active fins (bottom), heading beam waves at medium speeds.

The graphs show clearly which are the unstabilised and stabilised responses that limit most the ship operability in beam seas. In this case-study for low speeds of 5 knots, the most critical response is stern dock dryness for head and following waves, and the roll motion for beam, bow and quartering waves. In the medium and high speed regimes of 15 and 25 knots the operability is limited mostly by the vertical accelerations at the bridge for head waves and propeller emergence in following waves.

For beam waves and the headings around beam waves, roll motion becomes the most critical if low level of roll stabilisation is considered. Otherwise, lateral accelerations at the bridge start to play a significant role.

For more accurate and realistic calculation of the ship operability, it would be necessary to account for the distribution of wave directionality, and the expected ship heading when crossing the design-point for the associated percentage of time that the ship travels at each speed. Here the procedure was simplified by neglecting the wave directionality on the wave climate statistics. Additionally it is assumed that all ship headings relatively to waves are equally likely. Although the results may not be completely realistic, as it will be seen next it is possible to infer useful concepts that can provide a rational decision basis on selecting the most adequate roll stabilisation system for these naval patrol vessels.

Table 5 summarises the roll responses and the operability indexes for different levels of roll stabilisation installed onboard at low, medium and high speed regimes. These tables are very comprehensive since they show how much correlated are the roll responses with the overall seakeeping indexes for different levels of roll stabilisation. The last columns show the results when the ship satisfies all the seakeeping criteria. The tables present also the indexes for beam waves, and the indexes that result when an average is calculated for all the headings defined. With this set of results it is possible to detect a large influence of roll stabilisation levels over the overall seakeeping criteria at low and medium speeds, which then becomes smaller at high speeds, as shown in figure 12. Roll motion is a problem at low speeds because in this case roll damping is small and rms responses are large.

Table 5 Roll responses and operability indexes for the naval patrol vessel operating in West Coast of Portugal with four different levels of roll stabilisation at 5 knots (top), 15 kts (middle) and 25 kts (bottom).

Roll Stabilisation System		**Roll rms (deg) Sea Direction**			**Operability Indexes Criteria**		
Level	Type	90º	135º	45º	90º Roll	Aver. Roll	Aver. All Crit.
0	Barehull	**9.77**	**10.98**	**27.32**	**0.062**	**0.425**	**0.242**
1	Bilge Keels	**8.21**	**7.40**	**18.70**	**0.081**	**0.442**	**0.259**
2	U Tk	**2.19**	**2.59**	**3.22**	**0.948**	**0.940**	**0.686**
3	BK + U Tk	**2.17**	**2.57**	**3.17**	**0.948**	**0.941**	**0.686**

Roll Stabilisation System		**Roll rms (deg) Sea Direction**			**Operability Indexes Criteria**		
Level	Type	90º	135º	45º	90º Roll	Aver. Roll	Aver. All Crit.
0	Barehull	**2.44**	**1.66**	**4.83**	**0.809**	**0.857**	**0.687**
1	Bilge Keels	**2.33**	**1.62**	**4.78**	**0.846**	**0.865**	**0.687**
2	Active Fins	**1.24**	**0.87**	**2.54**	**0.988**	**0.985**	**0.985**
3	BK + Active Fins	**1.23**	**0.85**	**2.52**	**0.993**	**0.985**	**0.985**

Roll Stabilisation System		**Roll rms (deg) Sea Direction**			**Operability Indexes Criteria**		
Level	Type	90º	135º	45º	90º Roll	Aver. Roll	Aver. All Crit.
0	Barehull	**2.73**	**1.20**	**4.40**	**0.761**	**0.857**	**0.477**
1	Bilge Keels	**2.59**	**1.18**	**4.37**	**0.796**	**0.864**	**0.484**
2	Active Fins	**1.44**	**0.63**	**2.32**	**0.978**	**0.984**	**0.556**
3	BK + Active Fins	**1.37**	**0.62**	**2.30**	**0.986**	**0.987**	**0.556**

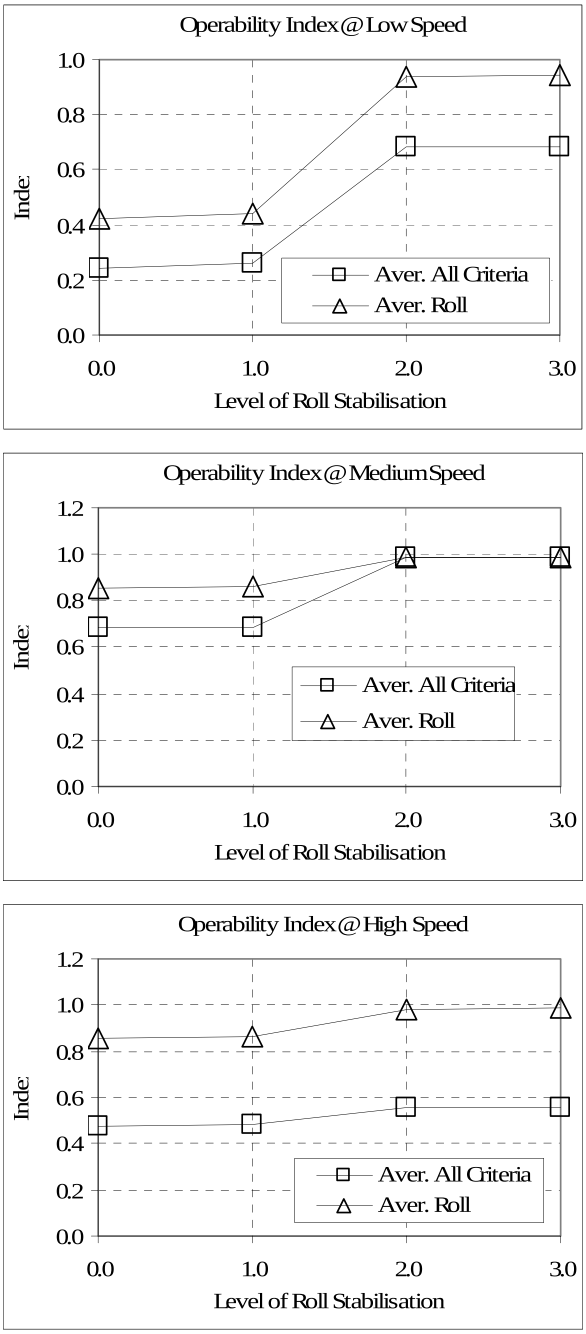


Figure 12 Variation of the operability index with the level of roll stabilisation of the patrol vessels at 5 (top), 15 (middle) and 25 knots (bottom).

The operability index for the naval patrol vessel designed to level 2 of roll stabilisation is of 0.92 when the operability indices at each speed are averaged according to the pre-defined mission speed profile. This means that the ship is fully operational (satisfies all seakeeping criteria) 92% of the time during the year. For levels 3, 1 and 0 the operability indexes are 0.92, 0.64 and 0.64, respectively.

Note that although the definition being used establishes that the ship is operational when all criteria are satisfied at a given ship's speed. In practice, these vessels might be operational for longer periods than the estimated values, because the Captain may change course to avoid stormy seas, and he may also reduce the speed or change the heading to alleviate the ship responses.

Table 5 also shows that although roll stabilisation level 2 is adequate, the operability of the patrol vessels at 25 knots is reduced at beam seas, mainly because of the roll damping reduction associated with a strict criteria applied to the lateral accelerations at the bridge (0.05g's). Moreover, level 3 of roll stabilisation is disregarded due to two factors: one is no effective (in statistical terms) gain in performance; and another is an additional fuel consumption associated with added resistance generated by bilge keels if these devices are appended to the ship's hull.

4. CONCLUSIONS

Good seakeeping performance can be more easily achieved by means of an adequate roll stabilisation system that allows ships to operate in adverse weather conditions with minimum degradation to their mission effectiveness.

The objective of this paper was to present an analysis that should be performed at the earliest stages of the design process when selecting the most appropriate roll stabilisation system.

The obtained results allow the patrol vessels operator to have confidence that this class of vessels will be capable of meeting their operational requirements.

5. ACKNOWLEDMENTS

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