INFLUENCE OF DAMPING ON THE ROLL MOTION OF SHIPS

Emre PESMAN¹, Deniz BAYRAKTAR² and Metin TAYLAN³

ABSTRACT

This paper analyzes the effect of damping on nonlinear roll motion of ships advancing in beam seas. As it is known, roll damping is a very important parameter in estimating ship responses in calm water and waves. Therefore, it has been studied by many researchers in different ways. However, it seems that it has been far from being complete and much work needs to be done towards thorough understanding of the phenomenon. This work estimates damping coefficients of a ship by using available methods. Damping coefficients are broken down into various categories such as friction, wave making, eddy making etc. Influence of forward speed is also considered in the analysis. Frequency domain solution of roll motion equation is then found incorporating linear and nonlinear damping coefficients. Nonlinearities are introduced in damping and restoring terms in the equation. It has been concluded that damping plays a very important role on the roll motion of a ship and reduces peak amplitudes to a considerable level. Therefore, it must be treated with utmost care since it dictates motion amplitudes directly.

1. INTRODUCTION

Roll motion is the most important phenomenon for ships, coupled with a few others, which may lead to capsizing. Therefore, many researchers have been studying it in all aspects to find satisfactory answers to the physical phenomenon. Damping on the other hand is the most important parameter in roll motion equation among others since it controls magnitude of the amplitudes. However, it is the most difficult parameter to estimate because of its complex nature. There are several different components in roll damping such as wave damping, lift damping, friction damping, eddy making damping and bilge keel damping. The above-mentioned forms of damping were presented by Ikeda et al (1978) in a very comprehensive analysis of damping study. He introduced empirical methods in order to estimate various damping components. Himeno (1981) analyzed many aspects of roll damping by conducting a series of model experiments towards better understanding of roll damping. Schmitke (1978) also set out interesting ways to predict roll damping of naval ships. Haddara and Cumming (1990) emphasized that inviscid damping was a function of forward speed. Haddara and Zhang (1994) carried out extensive experimental work with fishing vessel models to predict damping characteristics of ships. They suggested a modification to Ikeda’s lift damping formula based on the results of the experiments. Experimental investigation has been carried out by other researchers as well.

This work makes use of the available roll damping theories to solve a nonlinear roll motion model. It also utilizes numerical simulation techniques to apply the chosen model to a sample ship. The details of the roll motion model, damping and other hydrodynamic characteristics and sample model are given in the following sections.

2. SAMPLE VESSEL

The sample vessel is a mid-size, twin screw fishing vessel. The characteristics and body plan of the ship are given below:

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>LBP</td>
<td>64.0 m.</td>
</tr>
<tr>
<td>B</td>
<td>11.6 m.</td>
</tr>
<tr>
<td>Depth</td>
<td>7.32 m.</td>
</tr>
<tr>
<td>Draft</td>
<td>4.48 m. (loaded)</td>
</tr>
<tr>
<td>Δ</td>
<td>1556 tons</td>
</tr>
<tr>
<td>GM</td>
<td>0.78 m.</td>
</tr>
<tr>
<td>Cb</td>
<td>0.449</td>
</tr>
<tr>
<td>Cm</td>
<td>0.852</td>
</tr>
<tr>
<td>LCF</td>
<td>2.02 m. (aft)</td>
</tr>
<tr>
<td>LCB</td>
<td>1.17 m. (aft)</td>
</tr>
</tbody>
</table>

The ship is fitted with a set of bilge keels, shaft brackets and fin stabilizers. Various loading conditions and corresponding hydrostatic and hydrodynamic characteristics of the ship have been used throughout the analysis.

The fin stabilizers and rudders were not considered in the analysis because of their functionalities. That means they are controlled actively to react to opposing forces and moment when needed. Other appendages on the other hand, work as passive anti rolling devices and cannot be controlled like others.

Various hydrostatic and stability characteristics of the ship were computed by using a commercial software package.

¹ Istanbul Technical University, Faculty of Naval Architecture and Ocean Engineering 34469 Maslak, Istanbul Turkey.
² Istanbul Technical University, Faculty of Naval Architecture and Ocean Engineering 34469 Maslak, Istanbul Turkey.
³ Istanbul Technical University, Faculty of Naval Architecture and Ocean Engineering 34469 Maslak, Istanbul Turkey.
3. ROLL MOTION MODEL

In general, equation of nonlinear roll motion can be written as follows;

\[ A \ddot{\phi} + B(\phi, \dot{\phi}) + C(\phi) = M(t) \]  

(1)

In the above equation, over dot denotes differentiation with respect to time. More specifically, Equation (1) may be expressed as follows under the influence of regular sinusoidal waves;

\[ (I_{xx} + \delta I_{xx}) \ddot{\phi} + B(\phi, \dot{\phi}) + \Delta GZ(\phi) = \omega_c^2 \alpha_m I_{xx} \cos \omega_c t \]  

(2)

In the literature, representation of damping and restoring terms has been handled differently. The restoring term which is chosen as an odd-order polynomial is also nonlinear in nature. Restoring term in the equation may appear as cubic or quintic polynomials depending on the character of the GZ curve in question. Sometimes, even higher order polynomials may best represent a particular GZ curve, e.g. a seventh degree polynomial. However, higher order polynomials cause somewhat bulky manipulations throughout the solution scheme.

If an equation of nonlinear roll motion is to be established under the above-mentioned assumptions, one may end up with the following form:

\[ (I_{xx} + \delta I_{xx}) \ddot{\phi} + B(\phi, \dot{\phi}) + B_N \phi \phi \dot{\phi} + \Delta \left( C_1 \phi + C_3 \phi^3 + C_5 \phi^5 \right) = \omega_c^2 \alpha_m I_{xx} \cos \omega_c t \]  

(3)

Dividing Equation (3) throughout by \( (I_{xx} + \delta I_{xx}) \), and substituting the values of coefficients \( C_1, C_3 \) and \( C_5 \) are substituted respectively, it takes the form;

\[ \ddot{\phi} + b_L(\phi, \dot{\phi}) + b_N \dot{\phi} \phi + \omega_c^2 \phi + m_3 \phi^3 + m_5 \phi^5 = \lambda \omega_c^2 \alpha_m \cos \omega_c t \]  

(4)

where;

\[ \omega_c^2 = \frac{\Delta G M}{I_{xx} + \delta I_{xx}} \]  

(5)

\[ m_3 = \frac{4 \omega_c^2}{\phi_v^2} \left[ \frac{3 A_{\phi_v}}{G M \phi_v^2} - 1 \right] \]  

(6)

\[ m_5 = -\frac{3 \omega_c^2}{\phi_v^2} \left[ \frac{4 A_{\phi_v}}{G M \phi_v^2} - 1 \right] \]  

(7)

\[ b_L = \frac{B_L}{I_{xx} + \delta I_{xx}} \]  

(8)

\[ b_N = \frac{B_N}{I_{xx} + \delta I_{xx}} \]  

(9)

Evaluation of linear and nonlinear damping terms \( b_L \) and \( b_N \) are assessed by Ikeda’s and Himeno’s approach. More elaborate evaluation of the damping terms and their effects on the motion characteristics are supplied in the following sections. Other related coefficients of the roll motion equation are given in Equations (5) through (12).

Equation (4) incorporates various effects of ship’s dynamic and environmental parameters including damping, restoring and wave excitation. The righting arms curve is approximated as a quintic \( GZ = C_1 \phi + C_3 \phi^3 + C_5 \phi^5 \) polynomial. Coefficients \( C_1, C_3 \) and \( C_5 \) of the polynomial are determined by a number of dynamic and static characteristics of the GZ curve namely, metacentric height \( G M \), angle of vanishing stability \( \phi_v \), and area under the curve \( A_{\phi_v} \) as follows:

\[ C_1 = \frac{d(GZ)}{d\phi} = GM \]  

(10)

\[ C_3 = \frac{4}{\phi_v^2} \left( 3 A_{\phi_v} - GM \phi_v^2 \right) \]  

(11)

\[ C_5 = -\frac{3}{\phi_v^2} \left( 4 A_{\phi_v} - GM \phi_v^2 \right) \]  

(12)

Solution of the equation can be made by a perturbation method, Taylan (1989, 1996). Numerical simulation may
also be carried out easily by 4th order Runge-Kutta solver.

4. LOADING AND ENVIRONMENTAL CONDITIONS

Basically, there are two different displacement values used in the analysis as 1556 tons and 1909 tons. However, three metacentric height values 0.43 m, 0.61 m and 0.788 m were taken into account. Therefore, four different stability conditions are considered in the systematic analysis, as follows;

1. \( \Delta = 1556 \) tons, \( GM = 0.43 \) m.
2. \( \Delta = 1556 \) tons, \( GM = 0.61 \) m.
3. \( \Delta = 1556 \) tons, \( GM = 0.788 \) m.
4. \( \Delta = 1909 \) tons, \( GM = 0.788 \) m.

As was mentioned earlier, the vessel is fitted with various appendages. Only bilge keels were taken into account for simplicity. Geometric details of the bilge keels are given below;

Length 32.00 m.
Depth 0.38 m.

In order to see the effect of bilge keels, damping values and corresponding roll amplitudes were calculated with and without the appendages. Thus, eight different test conditions in total have been analyzed in total.

As far as environmental conditions are concerned, linear sinusoidal wave characteristics were used disregarding any phase lag between the waves and the motion. It is obvious that wave slope plays an important role on the right hand side of the equation. Three distinct wave height/wave length ratios as 1/25, 1/50 and 1/60 are selected within the scope of study.

Finally, speed of the ship has varied between 0 and 10 knots in order observe the influence of speed on roll damping and eventually on roll amplitudes.

5. DAMPING EVALUATION

Estimation of roll damping moments or coefficients are extremely ambiguous owing to highly nonlinear nature of the motion. Although extensive theoretical research, experimental work and numerical simulation studies have been conducted on the matter, it is still far from being complete. Unfortunately, it is not feasible to estimate roll damping correctly from radiation theory only. Viscosity of the fluid plays an important role in predicting roll motion of ships. Haddara and Zhang (1994) have carried out series of model test to show the influence of forward speed in roll damping. Therefore, contributions from various sources and interaction between them make prediction of roll damping a difficult task.

Ikeda (1978) has broken down roll damping into five constituents:

a. Friction
b. Lift
c. Wave
d. Eddy making
e. Bilge keel

As a result of his work, empirical formulas were introduced to estimate different components listed above. Furthermore, it is more appropriate to divide above-mentioned components of roll damping as linear and nonlinear as stated in Equation (3). According to this classification, lift and wave damping will be regarded as linear whereas friction, eddy making and bilge keel damping will be regarded as nonlinear damping.

The effects of each damping component are investigated on the sample fishing vessel by using numerical simulation. Empirical formulations by Ikeda (1978) and by Himeno (1981) have been utilized in the simulation.

The procedure and relevant formulation to estimate damping components for this particular fishing vessel is supplied in the Appendix. Based on the numerical simulation, the damping characteristics of the sample ship are shown in Figures 2 and 3.

Figure 2. Linear and nonlinear damping coefficients with respect to velocity.

Figure 3. Non-dimensional damping coefficients components with respect to velocity.
It is interesting to note that Figures 2 and 3 depict similar trends as indicated by Ikeda and Himeno.

6. EFFECT OF DAMPING ON ROLL AMPLITUDES

The main purpose of determining damping values is to analyze their effects on roll amplitudes. Therefore, several parameters such as ship displacement and GM, wave characteristics and speed were altered systematically and equation of motion was solved both in time and frequency domains. Since combination of those variables yielded so many different alternatives and solutions in turn only a few are included in this paper due to space constraints. Even though, a sample of graph is given for each main parameter, similar trends have been obtained for other alternative simulations.

Figure 4. shows the effect of speed for a specific loading condition. 15% increase is observed between peak amplitudes for 5 knots with compared to 0 knots. It should be noted that damping values are also susceptible to speed variations. Figure 5 compares the roll amplitudes for a particular loading condition with and without bilge keels. The bilge keels damp the motion for about 35% at resonance frequency. The effect may be different in magnitude for different test conditions.

Figure 6. Effect of GM for Δ=1556 tons, V=0 knot and H/L=1/25.

Figure 7. Effect of H/L for V=10 knot, Δ=1556 tons and GM=0.778m with bilge keels.

Figure 6 depicts the effect of changing GM for a constant displacement. Increasing GM not only reduces peak amplitudes but also shifts resonant frequencies to higher values with no forward speed. Increased wave steepness has a noticeable impact roll amplitudes in the negative sense. About 40% reduction is expected going from 1/25 to 1/60, Figure 7. Effect of increased displacement on the peak roll amplitudes for the highest wave slope is shown in Figure 8.

Figure 8. Roll response amplitude when H/L=1/25, Δ=1909 tons and GM=0.778m without bilge keels.
As expected, bilge keels are conventional and simplest form of roll reduction devices. They may be expected to reduce roll amplitudes by 30%-40% for this particular sample ship.

Finally, the above assessment reveals that roll damping is a very critical parameter in motion characteristics of a ship. Therefore meaningful estimation of roll damping may lead to more accurate prediction of roll characteristics of a ship. In principle, it may be explained as the energy balance between damping and excitation forces. Since wave and wind excitation can not be controlled, one may play around the damping forces to enhance stability and motion qualities of ships to an extent. It may be incorporated in the present weather criterion towards better evaluation of dynamic stability of ships.

REFERENCES
APPENDIX

Following Ikeda (1978) and Himeno (1981), the following empirical expressions have been utilized in the calculation of damping coefficients;

**Skin Friction:**

\[
S = L(1.7D + C_B B) \\
S = (0.887 + 0.145C_B \left( \frac{S}{L} \right) + 2|KG - D|) \\
r_e = \frac{0.512[v_{r_a} \theta^2]}{v} \\
C_f = 1.328\left(\frac{r_{en}^{0.5} + 0.014r_{en}^{-0.114}}{n_{en}}\right) \\
B_{f_1} = 0.5\rho r_a^3 SC_f (1 + 4.1 \frac{V}{\omega L}) \\
B_F = \frac{8}{3\pi} \phi A w B_{f_1}
\]

**Lift Damping:**

\[
C_{SL} = LD \\
C_{LO} = 0.3D \\
C_{LR} = 0.5D \\
C_{OG} = |KG - D| \\
C_M = \left\{ \begin{array}{ll}
0 & \text{if } C_M \leq 0.92 \\
0.3 & \text{if } C_M > 0.97 \\
0.97 & \text{if } 0.97 > C_M > 0.92
\end{array} \right.
\]

\[
k_n = \frac{2\pi D}{L} + \kappa \left( \frac{4.1B}{L} - 0.045 \right) \\
B_i = \frac{1}{2} \rho C_{SL} V C_{KN} C_{LO} C_{LR} \left\{ 1 + 1.4 \frac{C_{OG}}{C_{LR}} + 0.7 \frac{C_{OG}^2}{C_{LO} C_{LR}} \right\}
\]

**Wave Damping:**

\[
\xi_d = \frac{\sigma^2 D}{g} \\
\tau = \frac{V \omega}{g}
\]

\[
A_1 = 1 + \xi_d^{-1.2} e^{-2\xi_d} \\
A_2 = 0.5 + \xi_d^{-1.0} e^{-2\xi_d} \\
B_w = \frac{1}{2} B_{w0} \left\{ \left( A_2 + 1 \right) + (A_2 - 1) \tanh(20(\tau - 0.3)) \right\} + (2A_1 - A_2 - 1)e^{-150(\tau - 0.25)^2}
\]

**Eddy Making Damping:**

\[
B_{CLR} \text{ is the bilge radius and is the function of } D_X, H_OX \text{ and } B_X \\
H_{OX} = \frac{B_X}{2D_X} \\
H_{OX}' = \frac{H_{OX} D}{D - C_{OG}} \\
\sigma = \frac{A_X}{B_X D_X} \\
\sigma' = \frac{\sigma D - C_{OG}}{D - C_{OG}} \\
A_1, A_2, M \text{ and } H_1 \text{ are functions of extinction coefficients from fitting the extinction curve in roll with a third degree polynomial with respect to roll angle;}
\]

\[
C_{f_3} = 1 + 4e^{-1.65 \times 10^6(1-\sigma)^2} \\
C_{f_1} = \frac{1}{2} (1 + \tanh(20(\sigma - 14))) \\
C_{f_2} = \frac{1}{2} \left( 1 - \frac{1}{2} \left( 1 \cos(\sigma \pi) - 1.5 \left( 1 - e^{5-5\sigma} \right) 0.5 - 0.5 \cos(2\pi \sigma) \right) \right) \\
\gamma = \frac{\sqrt{\pi C_{f_3}}}{2(D - C_{OG}) H_{OX}' \sigma} \left[ R_{MAX} + \frac{2M}{H_1} \sqrt{A_1^2 + A_2^2} \right] \\
C_p = \frac{1}{2} \left( 0.87 e^{-5} - 4e^{-0.187\gamma^2} + 3 \right) \\
C_R = \left( \frac{R_{MAX}}{D_X} \right)^2 \left( 1 - C_{f_1} B_{CLR} \right) \left( 1 - C_{f_1} B_{CLR} \right) \\
+ \left( C_{f_2} H_{OX} - C_{f_1} B_{CLR} \right)^2 \\
B_{ex} = \frac{1}{2} \rho D_X^4 C_{p} C_R \text{ this is the effect unit length}
\]

\[
B_e \text{ is the value of non-dimensional eddy making coefficient along the ship;}
\]

\[
C_K = \frac{V}{0.04\omega L} \\
B_K = \frac{8}{3\pi} \phi_a \omega B_e \frac{1}{1 + C_K^2}
\]

**Bilge-keel Damping:**

\[
B_{BK} = B_{BK} + B_{BK}
\]
The normal force component per unit length is written as;

$$B_{BKN} = \frac{8}{3\pi} \rho \omega^3 b_{BK} o B_{CRL} f^2 C_D$$

Equivalent drag force;

$$C_D = 22.5 \frac{b_{BK}}{\pi e B_{CRL} f} + 2.4$$

$$f = 1 + 0.3e^{-160(1-\sigma)}$$

The pressure component of damping per unit length due to hull surface was obtained from the pressure measurement on 2-dimensional hull surface, which was caused by the presence of the bilge keels.

$$B_{BKH} = \frac{4}{3\pi} \rho \omega^2 b_{BK} D^2 o B_{CRL} f^2$$

$$\left\{-22.5 \frac{b_{BK}}{\pi e B_{CRL} f} - 1.2 \right\} A_2 + 1.2 B_2$$

$A_2$ and $B_2$ are the functions which depend on bilge-circle radius, ship dimensions and the dimensions of the stations per unit length.

**NOMENCLATURE**

- $L$: Ship length
- $B$: Beam of ship
- $D$: Draft
- $C_B$: Block coefficient
- $C_M$: Mid-section area coefficient
- $GM$: Metacentric height
- $GZ$: Righting arm
- $LCB$: Longitudinal center of buoyancy
- $LCF$: Longitudinal center of floatation
- $KG$: Vertical position of center of gravity
- $I_{xx}$: Inertia of ship
- $\delta I_{xx}$: Added inertia of ship
- $\Delta$: Displacement
- $\rho$: Density of sea water
- $\alpha_m$: Maximum wave slope
- $\nu$: Kinematic viscosity
- $\omega$: Frequency
- $\omega_e$: Encouter wave frequency
- $\phi_A$: Roll amplitude
- $R_e$: Reynolds number
- $C_f$: Friction coefficient
- $B_{W0}$: Radiation damping
- $B_{CRL}$: Bilge radius
- $B_L$: Linear damping coefficient ($B_L = B_l + B_W$)
- $B_N$: Nonlinear damping coefficient ($B_N = B_e + B_f + B_{BK}$)
- $b_{BK}$: Breadth of the bilge keel
- $f$: Correction factor to take into account for the increase in the flow speed at the bilge keel
- $S$: Wetted Surface Area
- $B_v$: Beam of any station
- $D_v$: Draft of any station
- $A_v$: Area of any station
- $r_{cb}$: The mean distance from the roll axis