INFLUENCE OF DAMPING ON THE ROLL MOTION OF SHIPS

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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:

L_{BP} = 64.0 m.
B = 11.6 m.
Depth = 7.32 m.
Draft = 4.48 m. (loaded)
Δ = 1556 tons
GM = 0.78 m.
C_B = 0.449
C_M = 0.852
LCF = 2.02 m. (aft)
LCB = 1.17 m. (aft)

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.

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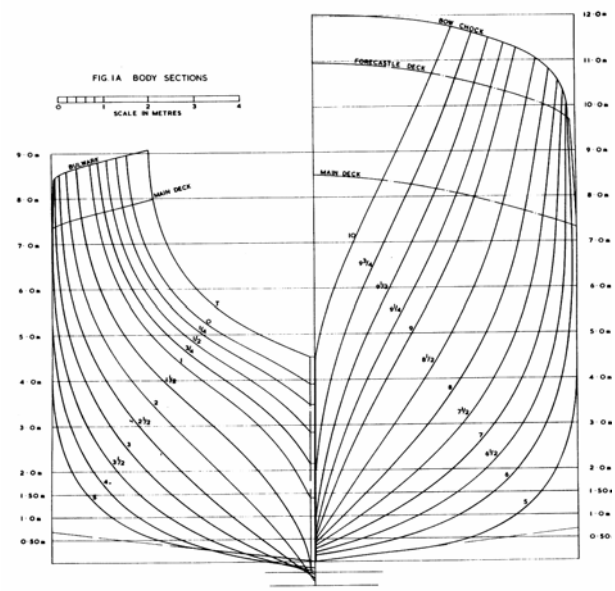

Figure 1. Body plan of the sample vessel.

3. ROLL MOTION MODEL

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

$$A\ddot{\phi} + B(\dot{\phi},\phi) + C(\phi) = M(t) \tag{1}$$

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;

$$\begin{aligned}&(I_{xx} + \delta I_{\mathbf{xx}})\ddot{\phi} + B(\dot{\phi},\phi) + \Delta GZ(\phi)\\ &= \omega_e{}^2 \alpha_m I_{xx} \cos\omega_e t\end{aligned} \tag{2}$$

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:

$$\begin{aligned}&(I_{xx} + \delta I_{\mathbf{xx}})\ddot{\phi} + B_L\dot{\phi} + B_N\dot{\phi}\left|\dot{\phi}\right|\\ &+ \Delta\left(C_1\phi + C_3\phi^3 + C_5\phi^5\right) = \omega_e{}^2 \alpha_m I_{xx} \cos\omega_e t\end{aligned} \tag{3}$$

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;

$$\begin{aligned}&\ddot{\phi} + b_L\dot{\phi} + b_N\dot{\phi}\left|\dot{\phi}\right| + \omega_\phi{}^2\phi + m_3\phi^3 + m_5\phi^5\\ &= \lambda\omega_e{}^2\alpha_m \cos\omega_e t\end{aligned} \tag{4}$$

where;

$$\omega_\phi{}^2 = \frac{\Delta GM}{I_{xx} + \delta I_{xx}} \tag{5}$$

$$m_3 = \frac{4\omega_\phi{}^2}{\phi_v{}^2}\left[\frac{3A_{\phi v}}{GM\phi_v{}^2} - 1\right] \tag{6}$$

$$m_5 = -\frac{3\omega_\phi{}^2}{\phi_v{}^4}\left[\frac{4A_{\phi v}}{GM\phi_v{}^2} - 1\right] \tag{7}$$

$$b_L = \frac{B_L}{I_{xx} + \delta I_{xx}} \tag{8}$$

$$b_N = \frac{B_N}{I_{xx} + \delta I_{xx}} \tag{9}$$

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 GM, angle of vanishing stability ϕ_v, and area under the curve $A_{\phi v}$ as follows:

$$C_1 = \frac{d(GZ)}{d\phi} = GM \tag{10}$$

$$C_3 = \frac{4}{\phi_v{}^4}\left(3A_{\phi v} - GM\phi_v{}^2\right) \tag{11}$$

$$C_5 = -\frac{3}{\phi_v{}^6}\left(4A_{\phi v} - GM\phi_v{}^2\right) \tag{12}$$

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. Δ = 1556 tons, GM = 0.43 m.
2. Δ = 1556 tons, GM = 0.61 m.
3. Δ = 1556 tons, GM = 0.788 m.
4. Δ = 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.

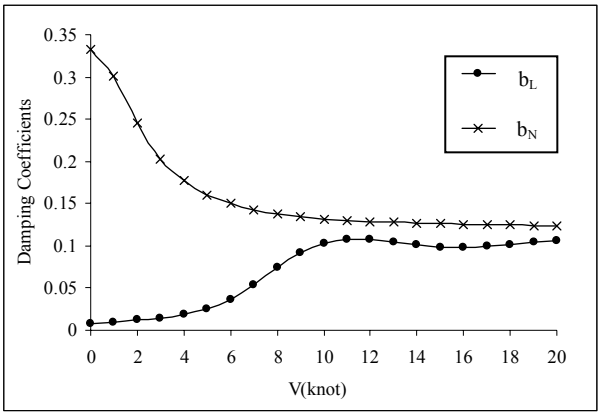

Figure 2. Linear and nonlinear damping coefficients with respect to velocity.

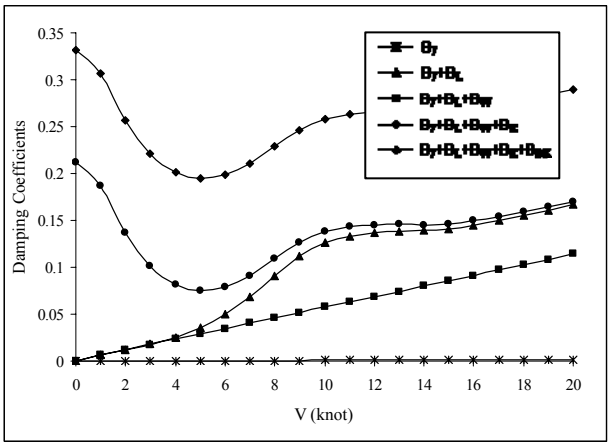

Figure 3. Non-dimensional damping coefficients components with respect to velocity.

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.

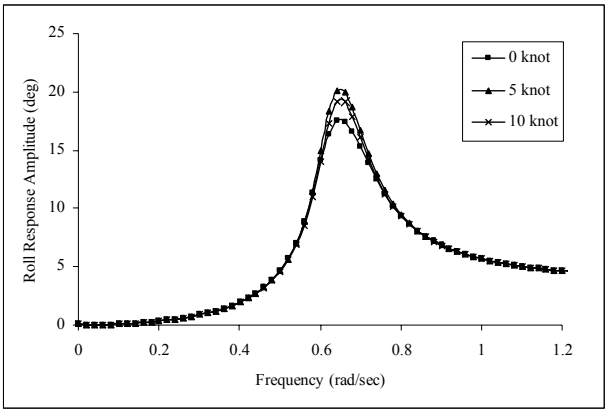

Figure 4. Effect of speed for H/L=1/50, Δ=1556 tons and GM=0.778m with bilge keels.

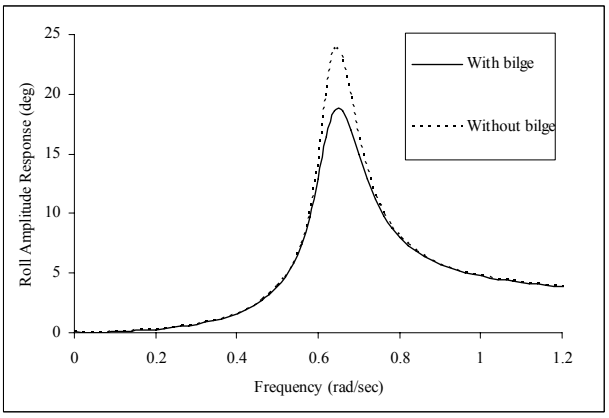

Figure 5. Effect of bilge keels for H/L=1/60, Δ =1556 tons and GM=0.778m.

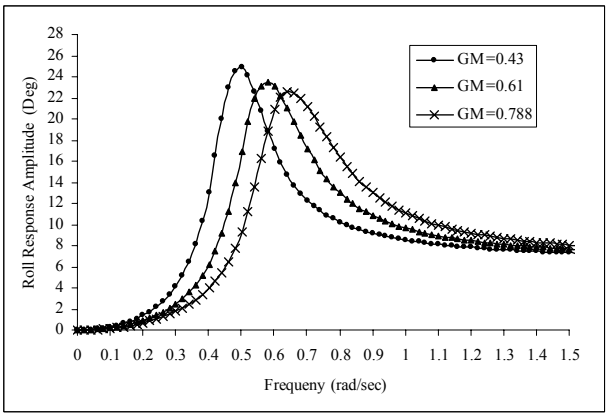

Figure 6. Effect of GM for Δ=1556 tons, V=0 knot and H/L=1/25.

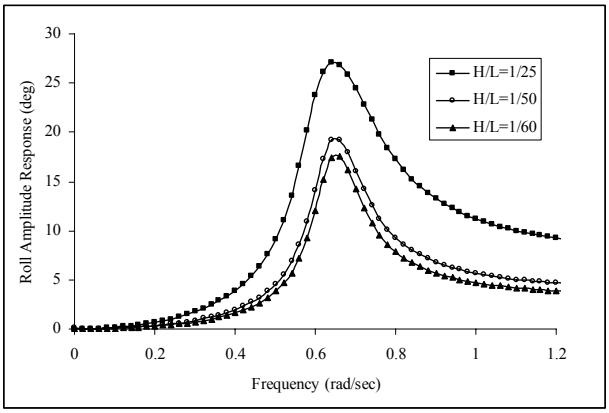

Figure 7. Effect of H/L for V=10 knot, Δ=1556 tons and GM=0.778m with bilge keels.

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.

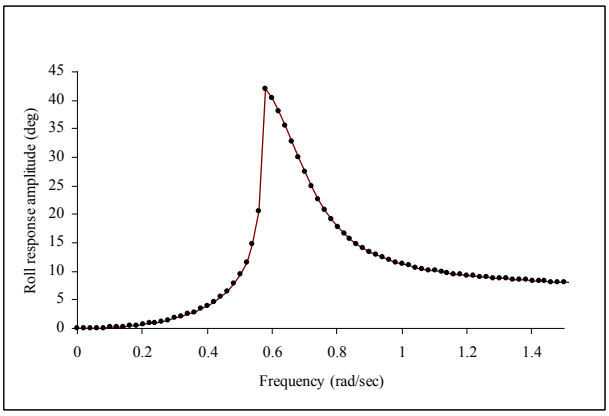

Figure 8. Roll response amplitude when H/L=1/25, Δ=1909 tons and GM=0.778m without bilge keels.

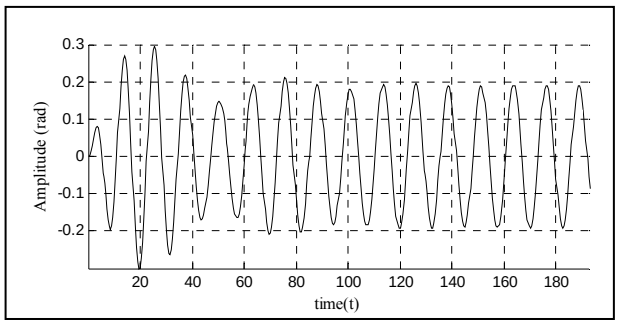

Figure 9. Time history of roll amplitudes at ω=0.5 rad/sec for H/L=1/25, Δ =1909 tons, GM=0.778m and without bilge keels.

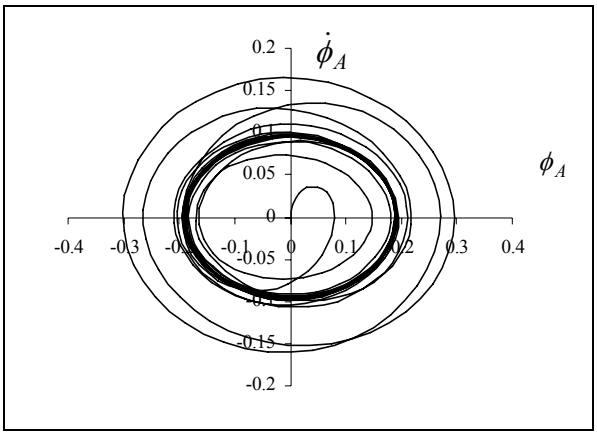

Figure 10. Phase diagram at ω=0.5 rad/sec for H/L=1/25, Δ =1909 tons, GM=0.778 m and without bilge keels.

Furthermore, time domain solutions and phase diagrams of the roll motion equation may also be obtained for each case analyzed. Figure 9 and Figure 10 are examples of such specific solutions.

7. CONCLUSIONS

In this paper the effect of damping, wave and stability characteristics and wave parameters on the roll motion of ships have been investigated by numerical simulation techniques for a sample vessel. It has been shown that, wave steepness plays an important role on especially resonant peak amplitudes regardless of variation of other parameters. The difference in the peak roll amplitudes may be in the range of %25 to 40% between the lowest and highest wave steepness values.

Increasing the speed from 0 to 10 knots also reduces peak roll amplitudes by about %50. It should be noted that damping values are also affected by increased speed in this manner. It is not possible to get a linear trend in this manner. Stability characteristics which may be translated into different weight and centers that a ship undergoes during her voyage also influence roll motion of a ship. Shift in resonant frequency can be seen along with the change in peak roll amplitudes.

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.

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APPENDIX

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

Skin Friction:

$$S = L(1.7D + C_B B)$$

$$r_e = \frac{(0.887 + 0.145C_B)\left(\frac{S}{L}\right) + 2*|KG - D|}{\pi}$$

$$\nu = \left(1.063 + 1.039(\rho - 1.025) + 0.02602(\rho - 1.025)^2 \cdot \left(10^{-6}\right)\right)$$

$$R_e = \frac{0.512\left(r_e \phi_A{}^2\right)\omega}{\nu}$$

$$C_f = 1.328\left(ren^{-0.5} + 0.014ren^{-0.114}\right)$$

$$B_{f_1} = 0.5\rho r_e{}^3 S C_f (1 + 4.1\frac{V}{\omega L})$$

$$B_F = \frac{8}{3\pi}\phi_A w B_{f_1}$$

Lift Damping:

$$C_{SL} = LD$$

$$C_{LO} = 0.3D$$

$$C_{LR} = 0.5D$$

$$C_{OG} = |KG - D|$$

If $C_M <= 0.92$ $\quad \kappa = 0$

If $C_M > 0.97$ $\quad \kappa = 0.3$

If $0.97 >= C_M > 0.92$

$$\kappa = 106(C_M - 0.91)^2 - 700(C_M - 0.91)^3$$

$$k_N = \frac{2\pi D}{L} + \kappa\left(4.1\frac{B}{L} - 0.045\right)$$

$$B_l = \frac{1}{2}\rho C_{SL} V C_{KN} C_{LO} C_{LR}\left(1 + 1.4\frac{C_{OG}}{C_{LR}} + 0.7\frac{C_{OG}{}^2}{C_{LO}C_{LR}}\right)$$

Wave Damping:

$$\xi_d = \frac{\omega^2 D}{g}$$

$$\tau = \frac{V\omega}{g}$$

$$A_1 = 1 + \xi_d{}^{-1.2} e^{-2\xi_d}$$

$$A_2 = 0.5 + \xi_d{}^{-1.0} e^{-2\xi_d}$$

$$B_W = \frac{1}{2}B_{W0}\left\{\begin{matrix}\left[(A_2 + 1) + (A_2 - 1)\tanh(20(\tau - 0.3))\right] \\ + (2A_1 - A_2 - 1)e^{-150(\tau - 0.25)^2}\end{matrix}\right\}$$

Eddy Making Damping:

B_{CRL} is the bilge radius and is the function of D_X, H_{OX} and B_X

$$H_{OX} = \frac{B_X}{2D_X} \qquad H_{OX}' = \frac{H_{OX}D}{D - C_{OG}}$$

$$\sigma = \frac{A_X}{B_X D_X} \qquad \sigma' = \frac{\sigma D - C_{OG}}{D - C_{OG}}$$

A_1, A_2, M and H_1 are functions of extinction coefficients from fitting the extinction curve in roll with a third degree polynomial with respect to roll angle;

$$C_{F3} = 1 + 4e^{-1.65x10^5(1-\sigma)^2}$$

$$C_{F1} = \frac{1}{2}\left(1 + \tanh(20\sigma - 14)\right)$$

$$C_{F2} = \frac{1}{2}\left(-1\cos(\sigma\pi)\right) - 1.5\left(1 - e^{5-5\sigma}\right)\left(0.5 - 0.5\cos(2\pi\sigma)\right)$$

$$\gamma = \frac{\sqrt{\pi}C_{F3}}{2(D - C_{OG})\sqrt{H_{OX}'\sigma'}}\left[R_{MAX} + \frac{2M}{H_1}\sqrt{A_1{}^2 + A_2{}^2}\right]$$

$$C_P = \frac{1}{2}\left(0.87e^{-\gamma} - 4e^{-0.187\gamma} + 3\right)$$

$$C_R = \left(\frac{R_{MAX}}{D_X}\right)^2\left(1 - \frac{C_{F1}B_{CRL}}{D_X}\right)\left(1 - \frac{C_{OG}}{D_X} - \frac{C_{F1}B_{CRL}}{D_X}\right)$$

$$+ C_{F2}\left(H_{OX} - \frac{C_{F1}B_{CRL}}{D_X}\right)^2$$

$$B_{ex} = \frac{1}{2}\rho D_X{}^4 C_P C_R$$ this is the effect unit length

B_e is the value of non-dimensional eddy making coefficient along the ship;

$$C_K = \frac{V}{0.04\omega L}$$

$$B_E = \frac{8}{3\pi}\phi_A \omega B_e \frac{1}{1 + C_K{}^2}$$

Bilge-keel Damping:

$$B_{BK} = B_{BKN} + B_{BKH}$$

The normal force component per unit length is written as;

$$B_{BKN} = \frac{8}{3\pi} \rho r_{cb}{}^{3} b_{BK} \omega B_{CRL} f^{2} C_{D}$$

Equivalent drag force;

$$C_{D} = 22.5 \frac{b_{BK}}{\pi r_{cb} B_{CRL} f} + 2.4$$

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

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.

$$B_{BKH} = \frac{4}{3\pi} \rho r_{cb}{}^{2} D^{2} \omega B_{CRL} f^{2} \cdot \left\{ -\left(-22.5 \frac{b_{BK}}{\pi r_{cb} f B_{CRL}} - 1.2 \right) A_{2} + 1.2 B_{2} \right\}$$

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
δI_{xx}	: Added inertia of ship
Δ	: Displacement
ρ	: Density of sea water
α_m	: Maximum wave slope
υ	: Kinematic viscosity
ω	: Frequency
ω_e	: Encounter wave frequency
ϕ_A	: Roll amplitude
R_e	: Reynolds number
C_f	: Friction coefficient
B_{W0}	: Radiation damping
B_{CRL}	: Bilge radius
B_l	: Lift damping
B_W	: Wave damping
B_E	: Eddy making damping
B_F	: Skin friction damping
B_{BK}	: Bilge keel damping
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_X	: Beam of any station
D_X	: Draft of any station
A_X	: Area of any station
r_{cb}	: The mean distance from the roll axis