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An overview of the prediction methods for roll damping of ships

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Abstract. Of all the six degrees of freedom, the roll motion of a ship is the most poorly understood and displays complicated phenomena. Due to the low potential wave damping at the natural frequency, the effective analysis of ship roll dynamics comes down to the accurate estimation of the viscous roll damping. This paper provides overview of the importance of roll damping and an extensive literature review of the various viscous roll damping prediction methods applied by researchers over the years. The paper also discusses in detail the current state of the art estimation of viscous roll damping for ship shaped structures. A computer code is developed based on this method and its results are compared with experimental data to demonstrate the accuracy of the method. While some of the key references describing this method are not available in English, some others have been found to contain typographic errors. The objective of this paper is to provide a comprehensive summary of the state of the art method in one place for future reference.

Keywords: roll damping; viscous damping; quadratic damping; equivalent linearized damping

1. Introduction

Ship rolling motion is the most critical vessel motion yet the most poorly understood of the six degrees of freedom. It is the most critical because roll amplitudes can become large and subsequently result in cargo shifting or even capsizing. Moreover, the roll restoring moment and the damping are both highly nonlinear. According to Vugts (1968) and repeated in the textbook by Faltinsen (1993), the radiated wave damping in roll for a typical 2-D rectangular vessel mid-ship sections is small and may even go to zero at certain frequencies. This is due to the cancelation of the waves produced by the sides and bottom. As a result of the potential radiated wave damping being small it becomes important to predict the viscous roll damping. Due to small overall damping roll motion exhibits significant dynamic magnification. Prediction of non-potential roll damping is a challenging effort. Work on roll damping goes back at least as far as William Froude who was employed by Brunel to design the bilge keels for the Great Eastern. He later continued his investigations into roll motions while working for the British Admiralty.

In addition to the effect on roll dynamic magnification (Response Amplitude Operator – RAO), the damping is also important in a variety of phenomena associated with roll motions such as broaching, surf-riding, parametric rolling (Moideen *et al.* 2014, 2013, 2012) etc. In the analysis of

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many unstable phenomena such as parametric rolling (Somayajula and Falzarano 2015b, 2014, Somayajula *et al.* 2014) or capsizing in beam seas (Su and Falzarano 2013, 2011), the damping determines the stability criteria to avoid catastrophic failure. Thus an accurate roll damping prediction model is a necessity to effectively analyze other complicated phenomena associated with ship rolling.

The rest of the paper is structured as follows. Section 2 discusses the historical developments of different roll damping prediction methods. The various methods of damping prediction currently used by the various commercially available simulation tools are described in Section 3. Section 4 gives the details of the various linearization techniques used in the prediction methods. Section 5 provides a comprehensive details on the current state of the art roll damping prediction methods which is also the current industry standard. The model testing and validation techniques are given in Section 6. The various devices and techniques used to reduce the roll motion are described in Section 7. The significant problems faced by the offshore industry in the prediction of damping for transport vessels are discussed in Section 8 and Section 9 summarizes the paper with conclusions.

2. Development of roll damping prediction methods

Although many researchers have investigated the topic of ship roll damping since Froude's investigations, it was the Japanese as far back as the 1950's and even before that investigated the various aspects of ship roll damping in a systematic and detailed manner. In the late 1970's Ikeda, Himeno and Tanaka all of Osaka Prefecture University in Japan published several papers which both summarized the work of others and presented a fundamentally new practical estimation technique for roll damping. This technique separated the roll damping into components and ignored their interactions (Ikeda *et al.* 1978). The components were wave, hull friction, eddy, lift and bilge keel. The bilge keel component can be further sub-divided into bilge keel normal and bilge keel hull components. The various components of damping are shown in Eq. (1).

$$\begin{aligned} B_{44} &= B_W + B_F + B_E + B_L + B_{BK} \\ B_{BK} &= B_{BKN} + B_{BKH} \end{aligned} \tag{1}$$

The wave damping is typically calculated using potential flow radiation-diffraction computer programs. The other components are calculated using the empirical methodology. In order for the method to be practical Ikeda *et al.* (1978) ignored the interactions between the various components and the coupling of roll to the other degrees of freedom. Moreover, by selecting an appropriate roll center, he was able to decouple the roll from the other degrees of freedom. In that early paper originally published in Japanese a computer program is provided (Ikeda *et al*. 1978). Unfortunately, most of the Japanese references were written in Japanese. However, a few of the more significant papers have eventually been translated in to English by the Ikeda group.

At about the same time Schmitke (1978) working at the Canadian Defense Research Establishment Atlantic (DREA) was also investigating new methods to predict the roll, sway and yaw motion of warships. His method used some of the earlier Japanese references such as Tanaka and Hishida (1960, 1959, 1957a, 1957b) and Kato (1965, 1958) combined with his own methodology. Interestingly, Prof. Himeno provided a comment to Schmitke SNAME Transactions paper (Schmitke, 1978) informing him of the similar work that had been occurring at Osaka Prefecture University on ship roll damping.

A few years later, in 1981, Prof. Himeno was visiting professor at the University of Michigan and during that visit produced his comprehensive report on ship roll damping (Himeno 1981). This report included a very complete literature survey of both Japanese and non-Japanese literature on this topic and two computer programs to predict ship roll damping. The first computer program is a simple method based only upon ship and bilge keel characteristics. However, the second computer program is more complete and involves the component-wise approach. As a result of the systematic and comprehensive approach of the Japanese this methodology is by far the most popular. Ikeda has continued work on roll damping prediction until this day (Kawahara *et al.* 2012). Unfortunately the Himeno report and associated computer programs are well-known to have numerous typographical errors and it is suggested to verify all equations with the original references. Another shortcoming of the method noted is that the methods are for general cargo ship hulls and may not be applicable to shallow draft high beam to draft ratio transport barges and other similar vessels.

In response to the wide use of transport barges in the offshore industry in the mid-1980's several investigations centered on a Noble Denton Joint Industry Project were undertaken. The focus of this effort was both model testing and predicting eddy damping of barges with sharp corner bilges using a vortex method. The development of this vortex method was done jointly by Standing, (1991) at British Marine Technology (BMT) and several faculty and graduate students at Imperial College London (Downie *et al.* 2006). Several papers described the method in various stages of development and also include comparisons to the Japanese empirical prediction methods of Ikeda *et al.* (1978) and Tanaka and Hishida (1960, 1959, 1957a, 1957b).

The British papers suggest that for vessels with square bilge corners the vortex method compares more favorably to experiments than the Japanese methods. Unfortunately, not all barges or shallow draft vessels have square bilge corners so an empirical correction is required for round bilges. In response to the criticism Ikeda (1984) has suggested that a simple equation Eq. (2) based upon the experimental and theoretical work of Yamashita and Katagiri (1980) can accurately predict the eddy damping of shallow draft sharp bilge hulls. Later, Ikeda et al. (1993) performed experiments to validate that for shallow draft sharp bilge hulls the newer formulation worked better than the older formulation.

$$B_e = \left(\frac{2}{\pi}\right)\rho L d^4 \left(H_0{}^2 + 1 - \frac{OG}{d}\right)\left(H_0{}^2 + \left(1 - \frac{OG}{d}\right)^2\right) R_0 \omega \tag{2}$$

The so-called Noble-Denton method of predicting roll damping is one of many vortex shedding based methods (Standing 1991). These methods are only strictly applied to sharp bilge corner transport barges. However, in an effort to make these methods more broadly applicable an empirical correction has been developed to account for a finite bilge radius (Robinson and Stoddart 1987). Unfortunately, the relatively good results achievable with these methods for square bilges are not generally achievable for barges of finite bilge radii. With the square bilge corner the separation point is the sharp corner. However, with the rounded bilge the separation point is not so well defined. Moreover, for a given barge the separation point may change depending upon a number of barge and motion parameters. There have been many other investigators that have applied vortex methods to ship roll damping including Braathen and Faltinsen (1988), Yeung *et al.* (2001) and Patel and Brown (1986).

It should be noted that various ship motions textbooks include discussion of ship roll damping to varying degrees of completeness. The 1989 Principles of Naval Architecture sea keeping chapter in Volume III has a section (Webster 1989) about the transverse motions (sway, roll and yaw) and briefly discusses roll damping prediction. The Lloyd (1989) sea keeping textbook has a relatively complete section on roll damping but not enough to undertake calculations. The textbook by Chakrabarti (2002) has a more complete section which generally follows the Chakrabarti (2001) journal paper on empirical prediction of roll damping. Although, both the paper and the textbook are up-to-date, they have certain typographical errors in describing the calculation of the eddy damping and should therefore be used with caution. The Himeno (1981) is the most complete treatment but it too suffers from various typographical errors.

3. Roll damping prediction in ship motion computer programs

The current situation for owners/operators requiring prediction of extreme roll motion and loads on transportation barges and vessels is the application of various standard commercial software. These software use a wide range of methods to estimate roll damping and may not even be well documented as to which exact method is being used in some cases. Most of the frequency domain strip theory computer programs seem to be using the Japanese empirical methods to some degree with the exception of the computer program AQWA which provides a reference to a RINA transactions paper which describes the UK vortex methods (Robinson and Stoddart, 1987). The MOSES program gives reference to the Schmitke-Tanaka method on its web-site and even provides a copy of the paper (Schmitke 1978). The DREA SHIPMO7 program had also used the Schmitke method but has since changed to the Himeno method (McTaggart 1997). The Journee Seaway program is using the Japanese methods as described in a conference paper (Journée 1992) and the program's user's manual (Journee and Adegeest 2003). As described in its user's manual the University of Michigan's SHIPMO program uses the Himeno (1981) method with corrections from the original papers. An additional Japanese reference (Yamashita and Katagiri 1980) for sharp corner bilges is also given in the manual. The program also uses further corrections derived from additional proprietary barge motion tests performed at Michigan. The US Navy's strip theory computer program SMP also uses a combination of the Japanese methods and Schmitke's method to predict roll damping at forward speed (Baitis *et al*. 1981). Other computer programs such as WAMIT allow input of a linear damping matrix but do not include a prediction method. Overall the documentation as to what method is used in each program is limited or non-existent. Moreover, the combination of inexperienced program users and poor documentation may further compromise the accuracy of the roll damping and motion results.

In excerpts of a MARIN report by Quemere (2012) the various roll damping calculation approaches used in the MARIN Shipmo program are described. There are three approaches that are used; the first is a somewhat standard Ikeda *et al*. (1978)- Schmitke (1978) approach, the second is the so-called FDS approach is for high speed vessel while finally the third is the Noble-Denton approach used for barges at zero speed.

4. Application of linearization techniques

Linearization for nonlinear and stochastic problems is well established in the field of nonlinear and random engineering vibrations (Socha 2008). The issue for ship roll damping is the form of the roll damping which involves both linear and non-linear quadratic and possibly cubic terms

(Dalzell 1976). Linearization of the nonlinear term involves either regular wave harmonic equivalent linearization or irregular wave stochastic linearization method. The harmonic linearization equates the energy loss per cycle and results in the following relationship.

$$B_e = B_1 + \frac{8}{3\pi}\omega R_0 B_2 + \frac{3}{4}\omega^2 R_0{}^2 B_3 \tag{3}$$

For prediction in irregular seas, the stochastic linearization originally described by (Kaplan 1966) is applied. The result assumes both the input and output to be Gaussian and minimizes the error between the linearized and actual system.

$$B_e = B_1 + \frac{8}{\pi}\sigma_{\dot{\phi}} B_2 \tag{4}$$

Where $\sigma_{\dot{\phi}}$ is the standard deviation of the response. This stochastic linearization method is used in the University of Michigan SHIPMO program. Recent studies by Su and Falzarano (2013, 2011) have suggested that cumulatant neglect and stochastic averaging may be more accurate.

5. The current state of the art ship roll damping prediction method

This section provides a comprehensive description of the empirical damping model suggested by Japanese researchers (Himeno 1981, Ikeda *et al*. 1978) which has become the de facto industry standard for estimating the roll damping of a ship shaped structure. Care has been taken to correct all the typographical errors in the existing literature.

The equivalent linear damping B_{eq} is assumed to be divided into 5 components – wave damping B_W, skin friction damping B_F, eddy damping B_E, lift damping B_L and bilge keel damping B_{BK}. It is assumed that the interaction between each of the components is negligible.

$$B_{eq} = B_W + B_F + B_L + B_E + B_{BK} \tag{5}$$

Figs. 1 and 2 show the comparison of the total roll damping as evaluated by the sum of the above components between the in-house roll damping prediction program and Ikeda's results.

5.1 Wave damping

Ikeda *et al*. (1978) specify a formulation for predicting the forward speed wave damping from the zero speed radiation wave damping. However, the use of panel methods to predict the forward speed added mass and radiation damping based on the theory developed by Salvesen *et al.* (1971) is theoretically accurate and should be used instead of the empirical formulae specified by Ikeda *et al.* (1978). The radiation wave damping at Marine Dynamics Laboratory at Texas A&M University is calculated by an in-house panel method code – MDLHydroD (Guha and Falzarano 2015).

Figs. 3 and 4 show the comparison of the wave damping obtained by potential flow method to that obtained by Ikeda's empirical formulation for a series 60 hull form. As can be seen from the results, there is a considerable error in the wave damping as predicted by Ikeda and as obtained from the potential theory.

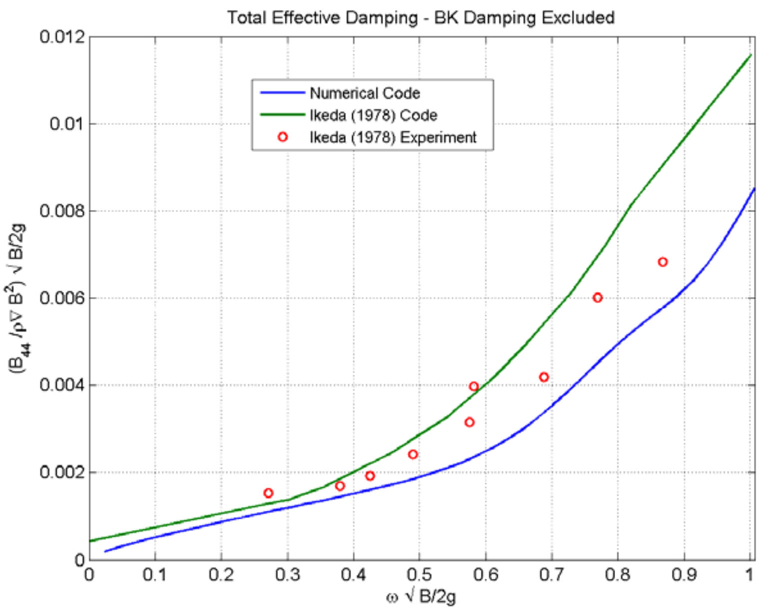

Fig. 1 Zero Speed Total Roll Damping for Series 60 Hull

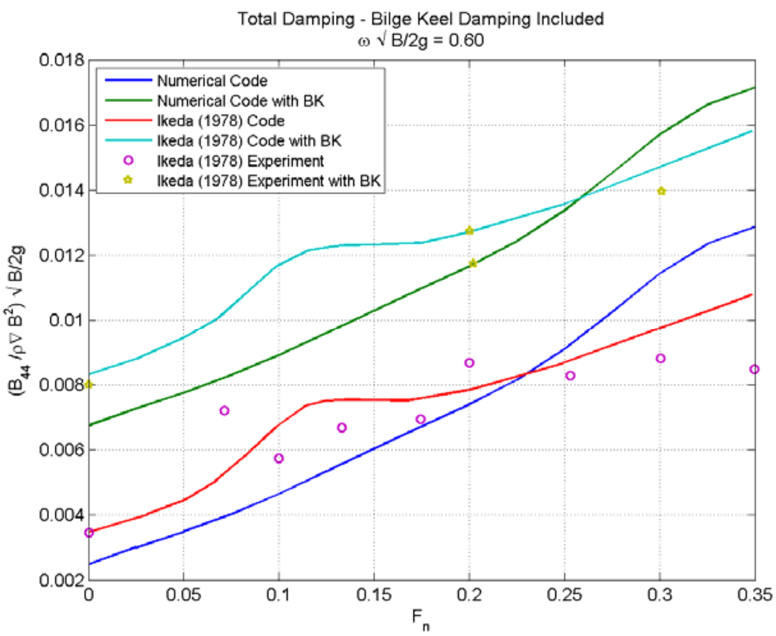

Fig. 2 Forward Speed Total Roll Damping for Series 60 Hull

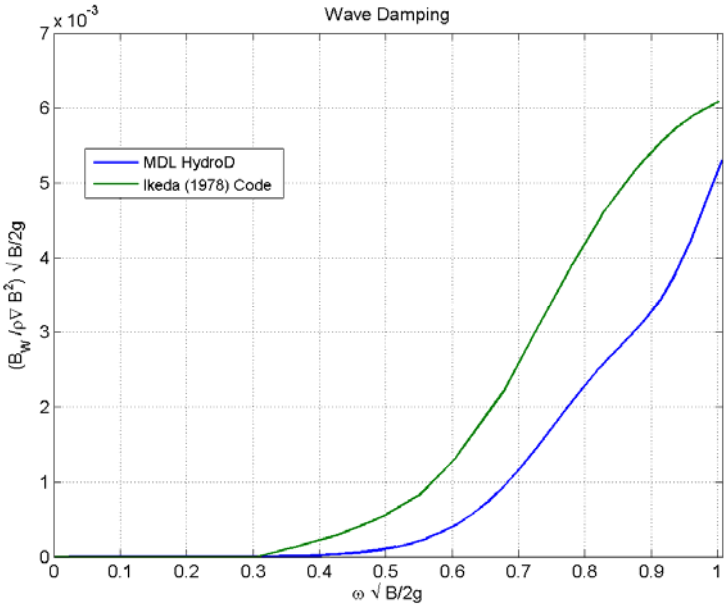

Fig. 3 Zero Speed Wave Damping for Series 60 Hull

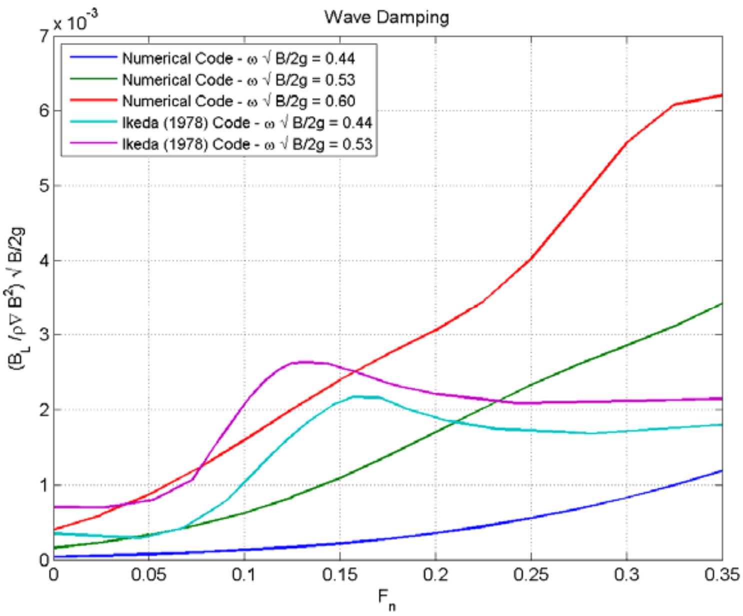

Fig. 4 Forward Speed Wave Damping

5.2 Skin friction damping

The skin friction drag is caused by the viscous skin friction stress acting on the hull surface. The empirical expression for skin friction damping coefficient for laminar flow was provided by Kato (1958) and is shown in Eq. (6).

$$B_{f0} = \frac{4}{3\pi}\rho S r_e^3 R_0 \omega C_f \tag{6}$$

$$C_f = 1.328\sqrt{\frac{2\pi\nu}{3.22 r_e^2 R_0^2 \omega}} \tag{7}$$

$$r_e = \frac{1}{\pi}\left[\left(0.887 + 0.145 C_B\right)\frac{S}{L} - 2OG\right] \tag{8}$$

$$S = L\left(1.7D + C_B B\right) \tag{9}$$

where

ρ is the density of the fluid (sea water for full scale ships and fresh water for models)
S is the wetted surface area which is empirically calculated as given by Eq. (9)
r_e is the effective bilge radius as given by Eq. (8)
R_0 is the roll amplitude
ω is the frequency of excitation
C_f is the friction coefficient given by Eq. (7)
ν is the kinematic viscosity of fluid
C_B is the block coefficient of the ship
L is the length of the ship
B is the breadth of the ship
D is the draft of the ship
OG is the distance between the roll center of the ship and the center of gravity

Although the model is subjected to laminar flow owing to its scale, the full scale ship experiences a turbulent flow and hence Eq. (6) requires a correction for turbulent flow which is given in Eq. (10). The second term is the correction factor to account for the turbulent flow.

$$B_{f0} = 0.787\rho S r_e^2\sqrt{\omega\nu}\left\{1 + 0.00814\left(\frac{r_e^2 R_0^2 \omega}{\nu}\right)^{0.386}\right\} \tag{10}$$

Schmitke (1978) provides modification in Eq. (11) for a ship moving with forward speed U.

$$B_F = B_{f0}\left(1 + 4.1\frac{U}{\omega L}\right) \tag{11}$$

Figs. 5 and 6 show the comparison of the in-house roll damping prediction program against the results obtained by Ikeda. Note that the plots in Ikeda's paper were not of high quality and did not always have perpendicular axes. Hence, the digitized results from his paper contain errors which cannot be avoided.

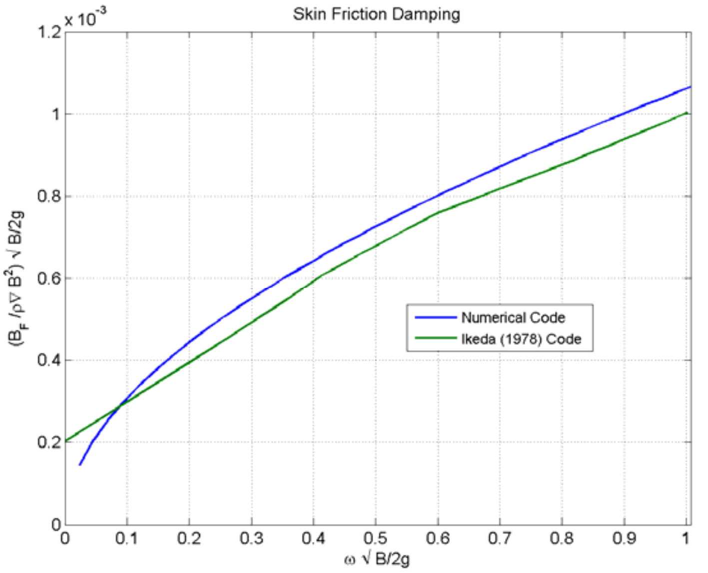

Fig. 5 Zero Speed Skin Friction Damping for a series 60 Hull

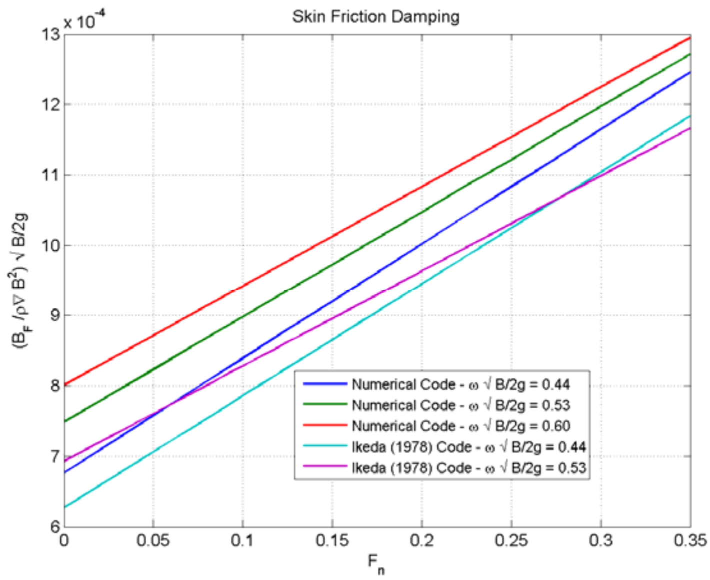

Fig. 6 Forward Speed Skin Friction Damping for Series 60 Hull

5.3 Lift damping

Ikeda *et al.* (1978) provide a simple empirical formulation for calculating the lift component of the roll damping as shown in Eq. (12). C_M represents the mid-ship cross section coefficient.

$$B_L = 0.075\rho ULD^3 k_N \left[1 - 2.8\frac{OG}{D} + 4.667\left(\frac{OG}{D}\right)^2\right] \tag{12}$$

$$k_N = 2\pi\frac{D}{L} + \kappa\left(4.1\frac{B}{L} - 0.045\right) \tag{13}$$

$$\begin{aligned} &\kappa = 0.0 \text{ for } C_M \leq 0.92 \\ &\kappa = 0.1 \text{ for } 0.92 \leq C_M \leq 0.97 \\ &\kappa = 0.3 \text{ for } 0.97 \leq C_M \leq 0.99 \end{aligned} \tag{14}$$

It may be noted from the expression in Eq. (12) that the lift damping coefficient varies linearly with the speed U and is independent of the frequency of roll motion. Fig. 7 shows the comparison of the lift damping evaluated by the in-house roll damping prediction program against the results from Ikeda. Note that all the “Numerical Code” lie on top of each other and are hence not visible in the plot. The slight digression in the results is due to the poor quality of the pictures from Ikeda’s paper which could not be digitized without slight errors.

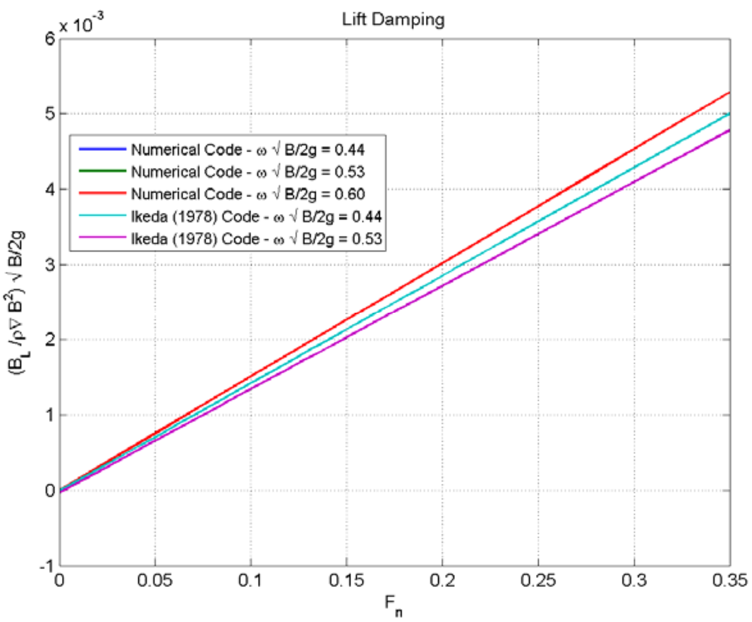

Fig. 7 Lift Damping for Series 60 Hull

5.4 Eddy damping

The eddy damping is caused by the separation of flow and the shedding of vortices around the bottom of the ship. For slender ships, the vortices are shed from the forward and the aft regions while for a vessel with fuller shape the mid ship region contributes significantly to the phenomenon. The empirical formula for estimating the eddy damping is similar to the estimation of drag force on a cylinder using a drag coefficient. The eddy damping per unit length for a cross-section is given by Eq. (15).

$$\frac{B_{E0}}{L} = \frac{4}{3\pi} D^4 \omega R_0 C_R \tag{15}$$

C_R is defined as shown in Eq. (16) where M_{RE} represents the eddy damping moment.

$$C_R = \frac{M_{RE}}{\frac{1}{2}\rho D^4 L \dot{\theta} \,|\, \dot{\theta} \,|} \tag{16}$$

Let H_0 and σ represent the half the beam-draft ratio and area coefficient at the underwater cross-section under consideration.

$$\begin{aligned} H_0 &= \frac{B}{2D} \\ \sigma &= \frac{A_{\sec}}{BD} \end{aligned} \tag{17}$$

The eddy damping moment M_{RE} is empirically estimated by

$$M_{RE} = \frac{1}{2}\rho L r_{\max}{}^2 D^2 \dot{\theta} \,|\, \dot{\theta} \,|\, \mathrm{C}_P \left\{ \left(1 - f_1 \frac{R_b}{D}\right)\left(1 - \frac{OG}{D} - f_1 \frac{R_b}{D}\right) + f_2 \left(H_0 - f_1 \frac{R_b}{D}\right)^2 \right\} \tag{18}$$

Where R_b is the bilge radius given by Eq. (19).

$$R_b = \left\{ \begin{array}{lll} 2D\sqrt{\dfrac{H_0(\sigma-1)}{\pi-4}} & \text{for} & R_b < D, R < \dfrac{B}{2} \\ D & \text{for} & H_0 \geq 1, \dfrac{R_b}{D} > 1 \\ \dfrac{B}{2} & \text{for} & H_0 \leq 1, \dfrac{R_b}{D} > H_0 \end{array} \right\} \tag{19}$$

$$\begin{aligned} f_1 &= \frac{1}{2}\left[1 + \tanh\{20(\sigma - 0.7)\}\right] \\ f_2 &= \frac{1}{2}\left(1 - \cos(\pi\sigma)\right) - 1.5\left(1 - e^{-5(1-\sigma)}\right)\sin^2(\pi\sigma) \end{aligned} \tag{20}$$

The coefficient C_P is further given by

$$C_P = \frac{1}{2}\left(0.87e^{-\gamma} - 4e^{-0.187\gamma} + 3\right) \tag{21}$$

$$\gamma = \frac{\sqrt{\pi} f_3}{2D\left(1-\frac{OG}{D}\right)\sqrt{H_0{}'\sigma'}}\left(r_{\max} + \frac{2M_1}{H_1}\sqrt{A_1^2 + B_1^2}\right) \tag{22}$$

where $H_0{}' = \frac{H_0 D}{D-OG}$ and $\sigma' = \frac{\sigma D - OG}{D - OG}$.

$$f_3 = 1 + 4e^{-1.65\times10^5(1-\sigma)^2} \tag{23}$$

$$\begin{aligned}
A_1 &= -2a_3\cos(5\psi) + a_1(1-a_3)\cos(3\psi) + \left\{(6-3a_1)a_3^2 + \left(a_1^2 - 3a_1\right)a_3 + a_1^2\right\}\cos(\psi) \\
B_1 &= -2a_3\sin(5\psi) + a_1(1-a_3)\sin(3\psi) + \left\{(6+3a_1)a_3^2 + \left(a_1^2 + 3a_1\right)a_3 + a_1^2\right\}\sin(\psi) \\
H_1 &= 1 + a_1^2 + 9a_3^2 + 2a_1(1-3a_3)\cos(2\psi) - 6a_3\cos(4\psi) \\
M_1 &= \frac{B}{2(1+a_1+a_3)} \\
r_{\max} &= M\sqrt{\left\{(1+a_1)\sin(\psi) - a_3\sin(3\psi)\right\}^2 + \left\{(1-a_1)\cos(\psi) + a_3\cos(3\psi)\right\}^2}
\end{aligned} \tag{24}$$

where

$$\psi = \begin{cases} \psi_1 = 0 & \text{for } r_{\max}(\psi_1) \ge r_{\max}(\psi_2) \\ \psi_2 = \frac{1}{2}\cos^{-1}\left(\frac{a_1(1+a_3)}{4a_3}\right) & \text{for } r_{\max}(\psi_1) < r_{\max}(\psi_2) \end{cases} \tag{25}$$

The coefficients a_1 and a_3 are the Lewis form parameters corresponding to the shape of the modified cylinder below the roll axis. The 3 dimensional eddy damping coefficient is obtained by integrating $\frac{B_{E0}}{L}$ over cross-sections along the length of the ship. In the presence of forward speed, the eddy damping rapidly decreases according to the empirical formula

$$B_E = B_{E0}\left[\frac{(0.04\omega L/U)^2}{1+(0.04\omega L/U)^2}\right] \tag{26}$$

Figs. 8 and 9 show the comparison of the zero speed and forward speed eddy damping of series 60 hull with the results presented by Ikeda. Although the same formulation as mentioned in Ikeda's paper is used to calculate the eddy damping, there is considerable difference in the results. We again attribute this difference to the non-perpendicular axes of the plots in Ikeda's paper.

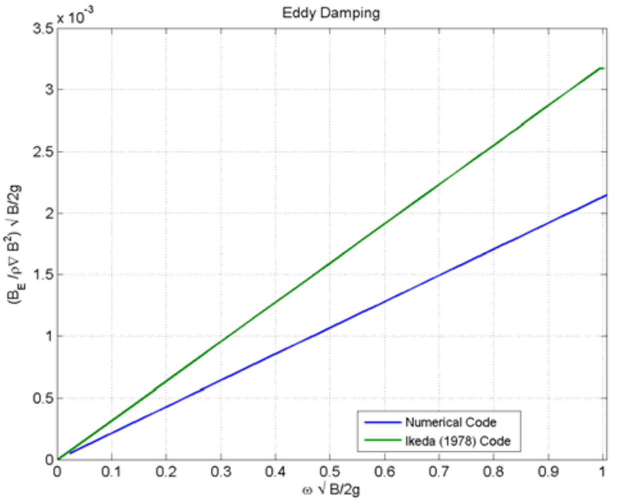

Fig. 8 Zero Speed Eddy Damping for Series 60 Hull

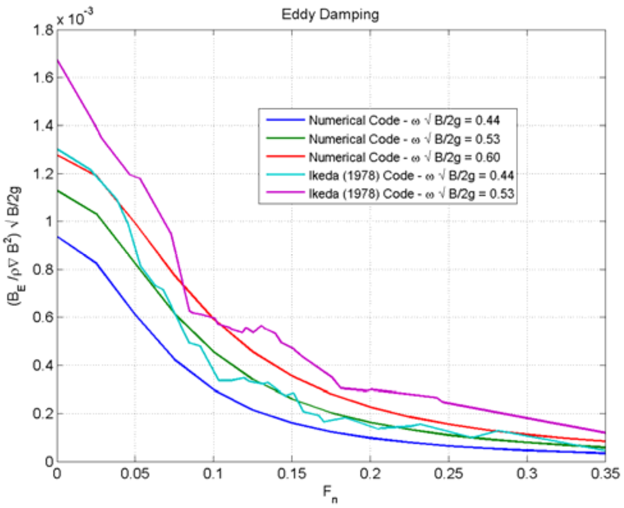

Fig. 9 Forward Speed Eddy Damping for Series 60 Hull

5.5 Bilge Keel Damping

The total bilge keel damping may be separated into two components – normal pressure damping and hull damping.

$$B_{BK} = B_{BKN} + B_{BKH} \tag{27}$$

The normal component of the damping per unit length is given by

$$\frac{B_{BKN}}{L} = \frac{8}{3\pi}\rho r_{cb}{}^{3} b_{BK}\omega R_0 f^2 C_D \tag{28}$$

where b_{BK} is the breadth of the bilge keel and r_{cb} is the mean distance from the roll axis to the bilge keel and is given by

$$r_{cb} = D\sqrt{\left\{H_0 - 0.293\frac{R_b}{D}\right\}^2 + \left\{1 - \frac{OG}{D} - 0.293\frac{R_b}{D}\right\}^2} \tag{29}$$

$$f = 1 + 0.3e^{-160(1-\sigma)} \tag{30}$$

$$C_D = 22.5\frac{b_{BK}}{\pi r_{cb} R_0 f} + 2.4 \tag{31}$$

The pressure component of damping per unit length is given by

$$\frac{B_{BKH}}{L} = \frac{4}{3\pi}\rho r_{cb}{}^{2} D^2 \omega R_0 f^2 \left\{-\left(-22.5\frac{b_{BK}}{\pi r_{cb} f R_0} - 1.2\right)A_2 + 1.2B_2\right\} \tag{32}$$

where

$$\begin{aligned} A_2 &= \left(m_3 + m_4\right)m_8 - m_7{}^2 \\ B_2 &= \frac{m_3{}^2}{3\left(H_0 - 0.215m_1\right)} + \frac{\left(1-m_1\right)^2\left(2m_3 - m_2\right)}{6\left(1-0.215m_1\right)} + \left(m_3 m_5 + m_4 m_6\right)m_1 \end{aligned} \tag{33}$$

$$\begin{aligned} m_1 &= \frac{R_b}{D} \\ m_2 &= \frac{OG}{D} \\ m_3 &= 1 - m_1 - m_2 \\ m_4 &= H_0 - m_1 \\ m_5 &= \frac{0.414H_0 + 0.0651m_1^2 - \left(0.382H_0 + 0.0106\right)m_1}{\left(H_0 - 0.215m_1\right)\left(1 - 0.215m_1\right)} \\ m_6 &= \frac{0.414H_0 + 0.0651m_1^2 - \left(0.382 + 0.0106H_0\right)m_1}{\left(H_0 - 0.215m_1\right)\left(1 - 0.215m_1\right)} \\ m_7 &= \begin{cases} \dfrac{S_0}{D} - 0.25\pi m_1 & \text{for } S_0 > 0.25\pi R_b \\ 0 & \text{for } S_0 \le 0.25\pi R_b \end{cases} \end{aligned} \tag{34}$$

$$m_8 = \begin{cases} m_7 + 0.414 m_1 & \text{for } S_0 > 0.25\pi R_b \\ m_7 + m_1\sqrt{2}\left(1-\cos\left(\dfrac{S_0}{R_b}\right)\right) & \text{for } S_0 \le 0.25\pi R_b \end{cases} \tag{35}$$

where S_0 is the constant pressure distribution length given by

$$S_0 = 0.3\pi\, fr_{cb} R_0 + 1.95 b_{BK} \tag{36}$$

The three dimensional bilge keel damping is obtained by integrating the normal and hull components of damping over the length of the bilge keel. Figs. 10 and 11 show the zero and forward speed bilge keel damping for a series 60 hull. Note that the in-house roll damping prediction program compares quite well Ikeda's results.

5.6 Comparison of roll damping prediction against experiments

The viscous roll damping formulation described above has been implemented in FORTRAN and incorporated into the in-house nonlinear time domain simulation tool – SIMDYN – developed at Marine Dynamics Laboratory, Texas A&M University (Somayajula and Falzarano 2015a). A numerical simulation of the free decay test of APL China at 5 knots forward speed is compared with the experimental data for the same ship obtained from MARIN. The results in Figs. 12 and 13 show a good agreement which also provides credibility to the roll damping prediction method described above.

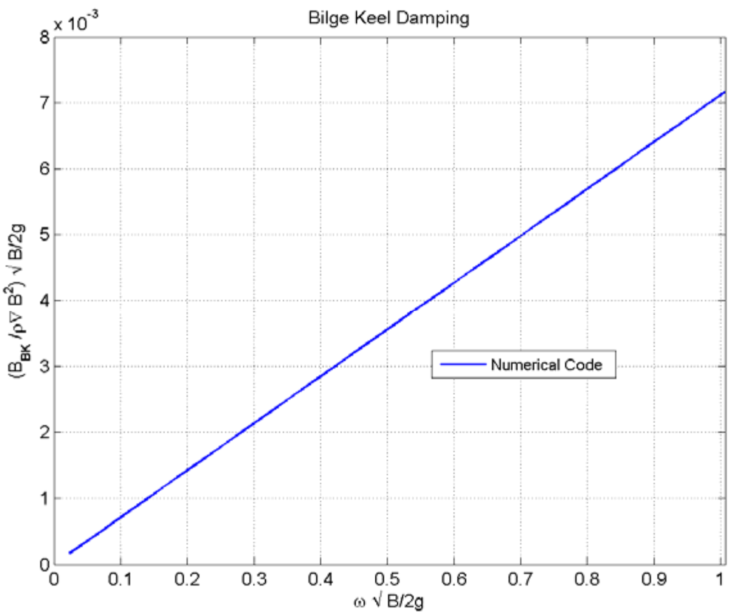

Fig. 10 Zero Speed Bilge Keel Damping for Series 60 Hull

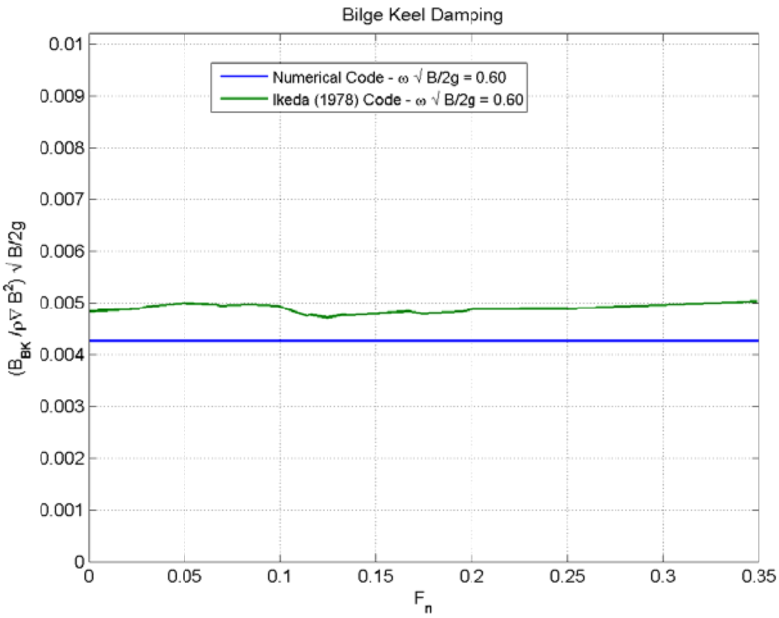

Fig. 11 Bilge Keel Damping for Series 60 Hull in Forward Speed

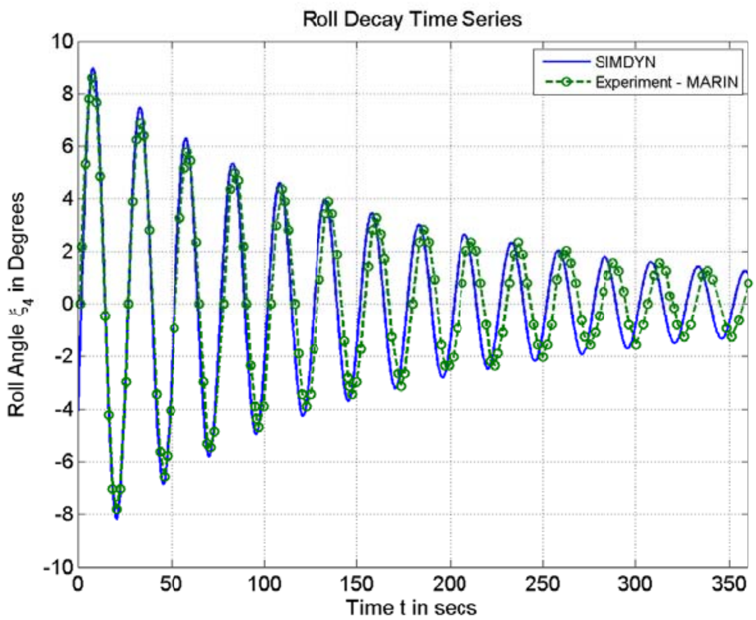

Fig. 12 Roll Decay Time Series for APL China

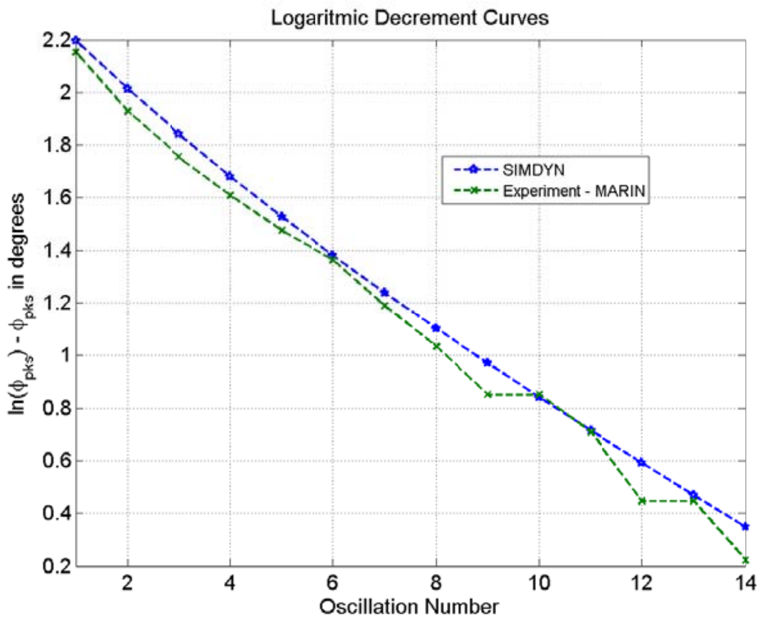

Fig. 13 Logarithmic Decrement Curve for APL China

6. Model testing to predict and validate roll damping predictions

A standard part of ship motion model testing preparation is to perform free-decay model tests. Using the simple log-decrement formula the linear roll damping can be obtained (Thomson and Dahleh 1988). Moreover, by using an extension of this technique (Faltinsen 1993), the nonlinear quadratic term can be obtained. Unfortunately free decay tests only give the roll damping at the damped natural frequency and we know the damping is frequency dependent. Another method is to use forced rolling to determine damping over a range of frequencies. Most model test facilities regularly undertake the free-decay test, far fewer have done forced rolling experiments (Kegakshi *et al.* 1957, Sugai and Yarnanouchi 1963, Takezawa *et al.* 1978). Model scale tests provide the opportunity to validate the component-wise damping prediction methodology for a specific hull form. Unfortunately extrapolating the model scale predictions to full scale may require additional assumptions. It should be noted that the Japanese and others developing empirical formulas have done extensive model scale validation (Himeno, 1981).

In a 2012 ASNE day paper, Judge and Beaver (2012), describe the development and use of a forced roll mechanism for planning hull models. The unit was developed at the US Naval Academy's hydromechanics lab. This mechanism can be used to statically hold a model at a particular heel angle or it can be used in a forced oscillatory rolling mode. In addition to recording the roll motion the instrumentation is able to measure the heave force, sway force and roll moment so that the added mass and damping and cross-coupling terms can be derived. In addition to the US Naval Academy design there are several Japanese language papers describing other forced rolling mechanisms.

7. Devices to reduce roll motion

The most common method to reduce roll motion by increasing roll damping is the installation of bilge keels. The bilge keel is effective at both zero speed and forward speed. It is also effective at small and large motion amplitudes. Other common methods include anti-roll tanks and active fins. The anti-roll tanks typically come in two types, i.e., u-tube and free surface tanks. Although U-tube tanks are simpler they are hard to adjust to varying loading conditions. The free surface tank is more versatile but more challenging to design and operate. Active fins are used for high speed and high value ships such as passenger ships and warships (Cox and Lloyd 1977, Miller *et al.* 1974).

7.1 Effect of hull form

The beam to draft ratio and the bilge radius are the most important hull form characteristics effecting roll damping. The smaller the bilge radius the higher is the damping. The limit is a sharp bilge corner with the highest damping. The beam to draft ratio is important mainly due limitations of the empirical methods which are generally based upon typical cargo ship B/T ratios (Himeno, 1981).

7.2 Effect of forward speed

Forward speed affects the individual roll damping components in various ways. The eddy damping is significantly reduced with forward speed while the lift component becomes dominant with forward speed. The bilge keel component is assumed to be un-effected by forward speed. The skin friction is also modified due to forward speed (Himeno 1981).

7.3 Prediction of full scale versus model scale

Most of the prediction work has involved empirical or physical model scale predictions. The general understanding is that scale weakly effects the eddy damping but strongly effects the friction. However, the friction is only a small part of the roll damping so scale effects can generally be ignored (Himeno, 1981). However, several studies have attempted to clarify and further quantify full scale roll motion with full scale observation and measurement, see e.g., Grant (2008), Cabezas and Rojas (1997), Stewart and Ewers (1979), Debord *et al.* (1987), Szajnberg *et al.* (1980).

8. Other issues associated with roll damping of transport vessels

Most prediction methods involve either ship shapes or shallow draft barges with sharp bilge corners. Transport vessels seem to have shallow draft as compared to typical merchant ships but non-sharp bilge corners as do the transport barges and therefore may require further analysis to verify the applicability and accuracy of the various empirical roll damping prediction techniques.

Application of Computational Fluid Dynamics (CFD) to ship roll damping has been done by many for many years. One of the earlier applications of Reynolds Averaged Navier Stokes (RANS) computer code was done by Korpus and Falzarano (1997). The conference and journal paper refer to some 2-D results without a free surface. Since that time many others have applied CFD to predict ship roll damping.

One of the important issues associated with transport barges or vessels is over-hanging cargo and the possibility of wave impacts. This requires a time domain simulation of the motion taking into account the water impacts of the cargo (Huang and Paulling 1993).

Although the emphasis of this study has been on the linear and nonlinear roll damping, it has been suggested that the non-linear restoring moment may be as important if not more important to accurately predict the roll motion (Denise 1983). It is therefore suggested that the effect of exact hydrostatics/Froude-Krylov, and 2nd and higher order hydrodynamics on the roll motion be investigated (Molin 2004). Moreover, for some vessels it may be important to investigate the difference between the 3-D radiation-diffraction computer program results and versus those of strip theory programs.

It is well known that the empirical prediction techniques are restricted at least to the amplitude of the roll and do not consider deck edge immersion and bilge emergence. Recently several investigators have focused on these effects. Bassler has described the very large amplitude effects as experience by advanced hull form US Navy warships (Bassler *et al.* 2010).

The group at the Federal University of Rio de Janerio and Lab Ocean (Fernandes and Oliveira, 2009) has recently made some unique observations on roll damping stating that it actually decreases for large angles.

Additional specific issues associated with Transportation vessels are discussed in the proceedings of three conferences sponsored by the Royal Institution of Naval Architects (RINA 2012, 2008, 2005). Unfortunately, very little discussion of the details of motion prediction methods are included in these three conference proceedings. Although some of the papers do describe the operability analysis of some specific vessels.

9. Conclusions

A comprehensive literature survey of various methods of roll damping prediction has been provided which would help the reader understand the nuances of each method and their inherent assumptions. In addition the complete description of the state of the art roll damping prediction based on the methods of Himeno, (1981) has been given. This method has been coded to generate a computer program to predict the roll damping of ship shaped structures. The results of this code have been validated against existing literature and available experimental data.

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