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AN EFFICIENT NONLINEAR NUMERICAL SCHEME TO PREDICT SHIP ROLL RESPONSES IN HEAVY SEAS

Rahul Subramanian1
Aishwarya Jayaraman2
Jyothish P V3

Abstract

Estimation of the roll motion of a vessel plays a vital role in the safe and efficient operation of a ship. It gives the designer prior knowledge of the limiting characteristics for transportation of cargo or passengers in a given sea-way. It is also crucial for the design of efficient, onboard roll control devices such as active fin stabilizer systems for both passenger and defense vessel applications. Traditionally, classification society based rules and empirical methods have been used to estimate the motion characteristics at the design stage. Although useful for the early phases of design, they tend to be inaccurate and conservative for more advanced design applications. Over the past couple of decades, computer based simulations have proven to be a more useful, efficient and robust tool for the designer. Typically based on linear theory, although they tend to be fairly accurate for moderate waves, the accuracy can quickly diminish for more realistic or severe sea states. This paper demonstrates the application of an efficient nonlinear computer program to predict the rolling of a ship. The computational model has been developed based on the direct time-domain body-nonlinear strip theory. The nonlinear theory is important to estimate ship motions in a severe sea-state. To model the viscous damping, the fully nonlinear form of the semi-empirical Ikeda-Kawahara method is employed.

In order to validate the computational predictions, a 1:12.5 scale model of a 200t fuel transport vessel has been fabricated and experiments carried out to evaluate decay characteristics and response in regular beam seas at zero forward speed. The comparisons of the numerical predictions with the experiments show generally good agreement. Detailed analysis of the hydro-mechanical force components helps in bringing out the importance of nonlinearities. The methodology thus promises to be a useful tool to designers to accurately predict the nonlinear roll behavior of ships at the early phase.

Key words - Seakeeping, Nonlinear Strip-Theory, Large Amplitude Roll Response, Roll Decay Tests, Body-Exact Time-Domain.

1. INTRODUCTION:

Accurate estimation of the dynamical responses and characteristics of a floating body in the presence of waves is crucial for the design of floating structures such as ships and offshore platforms. In particular, proper prediction of the roll motion is of significant practical importance. It is crucial in determining not only the stability and safety aspects of the floating system, but also in determining the comfort levels and its operational limits. There is great commercial interest in the theoretical understanding of the roll
behavior from operators of containerships, heavy transport vessels and FPSOs. There exist several approaches to predict the roll behavior of vessels. They can broadly be classified into experimental and computational methods. Historically, scaled model tests carried out in the model basin have been used to predict the roll responses. A rigorously conducted experiment provides invaluable information and still remains as the most reliable tool to the designer. However, fabrication and labour costs can quickly rise if a large number of experiments are to be conducted to explore a variety of alternate designs. In addition, scale effects may affect the fidelity in extreme cases.

Over the last couple of decades, with the dramatic increase in computational power, computational techniques have evolved as a promising tool to the designer. They can be classified into ideal fluid and viscous methods. Ideal fluid potential flow theory has been the most popular approach to seakeeping. The method dates back to the 1950s starting with the work of (Korvin-Kroukovsky and Jacobs, 1957). The advantages of the method lie in the fact that it has proven to be fairly accurate, robust and computationally fast. Traditionally, the problems have been formulated in the frequency domain to solve the linearized Boundary value problem (BVP). Although, it can give fairly satisfactory results for small amplitude waves, the accuracy drops substantially for physical scenarios dominated by nonlinear behavior.

Since the early 2000s, viscous flow based CFD codes are being actively pursued to aid in the design of ships. These codes have shown to be capable of simulating highly complex fluid flows occurring in cases such as the roll motion. They can thus give very valuable insights about the local flow field to the designer. However, the enormous computational cost and time limit their application as a practical design tool. In addition, they also require accurate turbulence models and are known to be sensitive to the quality of geometry and grids. These factors affect their robustness and practicality. The direct time-domain approach has been receiving a lot of attention, following the pioneering work of (Longuet-Higgins and Cokelet, 1976), where the fully nonlinear water wave problem was solved using the Mixed Euler Lagrange (MEL) approach. Time-domain based methods have a major advantage in being able to model different degrees of nonlinearity, as reviewed in (Beck and Reed, 2001). The sophistication level of these so called “blended method” codes can be chosen based on the particular problem being solved, the required accuracy level and available computational resources. Validation
studies have shown increased levels of accuracy and capability in handling large motions when compared with linear frequency domain methods; with the downside being increased computational effort.

This paper presents the application of an efficient “blended method” scheme to predict the roll behavior of a ship in a large sea-state. A computer program based on the body-nonlinear time-domain strip theory (cf. (Bandyk, 2009), (Subramanian, 2012) and (Subramanian and Beck, 2015)) has been developed to perform the numerical computations. The strip theory approach allows for faster computational times and simplified body geometry definition, when compared to fully three-dimensional methods. The body-nonlinear direct time-domain approach accounts for nonlinear dynamics attributed to the changing wetted surface of the body, thus accounting for higher order radiation and diffraction wave loads. This ensures the validity of the method for large roll amplitudes. In order to validate the computational predictions, a 1:12.5 scale model of a 200 tonne oil transport vessel has been fabricated and experiments carried out at the model basin in the Department of Ocean Engineering at IIT Madras. To this end, the free decay and seakeeping experiment in regular beam seas at zero forward speed have been carried out.

2. MATHEMATICAL FORMULATION:
The objective is to predict the motions and forces on a ship in a seaway. Three different coordinate systems are used for solving the fluid flow problem as in shown in Figure 1; an earth fixed inertial axis \((x_e, y_e, z_e)\) is used to keep track of the position of the center of gravity of the ship and the Euler

Figure 1.: Schematic showing the different coordinate frames used
angles. A hydrodynamic frame \((x_h, y_h, z_h)\) translates in the horizontal calm water plane with translational velocities \(U, V\) and rotational yaw rate \(\psi\). It thus follows the ship such that its origin \(O_h\) is always in vertical line with the origin of the body frame, \(O_b\). This is the frame in which the boundary value problem is formulated. A body fixed frame \((x_b, y_b, z_b)\) rotates and translates in all 6-DOF with the body. The frame is used to compute the forces acting on the ship and to solve for the equations of motions.

The velocities \(U, V\) and \(\psi\) are the instantaneous body velocities resolved in the hydrodynamic frame.

The fluid flow is considered inviscid, irrotational, incompressible and unsteady. For such a flow, a velocity potential \(\Phi\), representing the perturbation potential for the absolute fluid velocity in the earth fixed frame is defined. The fluid particle velocity \(v\) can be written as:

\[
v = \nabla \Phi\\
\]

From Equation (1), the continuity equation for the conservation of mass of the fluid reduces to the Laplace’s equation

\[
\nabla^2 \Phi = 0\\
\]

The linearized kinematic and dynamic free surface boundary conditions written in the moving hydrodynamic frame are,

\[
\frac{\partial \eta}{\partial t} = \frac{\partial \Phi}{\partial z} + (U + \dot{\Psi} \times \mathbf{r}_h) \cdot \nabla \eta \quad \text{on } z = 0 \quad (3)
\]

And

\[
\frac{\partial \Phi}{\partial t} = -g \eta + (U + \dot{\Psi} \times \mathbf{r}_h) \cdot \nabla \Phi \quad \text{on } z = 0 \quad (4)
\]

Here, \(\eta\) and \(\mathbf{r}_h\) represent the free surface wave elevation measured from the calm water surface, and the position vector in the hydrodynamic frame respectively.

The body boundary condition ensures no normal flow through the body surface:

\[
\nabla \Phi \cdot \mathbf{n} = v \cdot \mathbf{n} \quad \text{on } S_B(t) \quad (5)
\]

where \(\Phi\) is the total three-dimensional perturbation potential, \(v\) is the absolute velocity of a node on the body surface with respect to the earth fixed frame including velocities.
due to rotational effects; \( \mathbf{n} \) is the unit normal vector positive out of the fluid (or into the body), and \( S_B(t) \) is the exact wetted body surface.

The above formulations are defined for a fully three-dimensional flow field. If it is assumed that the ship is slender such that the slope of the body surface in the longitudinal direction is smaller than the slopes in the transverse direction, the gradient \( \frac{\partial}{\partial x} \ll \frac{\partial}{\partial y}, \frac{\partial}{\partial z} \). This forms the basis for the strip theory approximation, where the three-dimensional problem is solved as a series of individual two-dimensional problems.

The strip wise two-dimensional potential satisfies the Laplace equation at each frame:

\[
\nabla^2 \phi(y,z,t;x) = 0
\]

(6)

Here \( \phi = \phi(y,z,t;x) \) shall henceforth refer to the two-dimensional potential for notational convenience.

The free surface boundary conditions take the following form from Equations (3) and (4):

\[
\frac{\partial \eta}{\partial t} = \frac{\partial \phi}{\partial z} + V \frac{\partial \eta}{\partial y} + x_h \frac{\partial \phi}{\partial y} \quad \text{on } z = 0
\]

(7)

\[
\frac{\partial \phi}{\partial t} = -g \eta + V \frac{\partial \phi}{\partial y} + x_h \frac{\partial \phi}{\partial y} \quad \text{on } z = 0
\]

(8)

Consistent with strip theory, the encounter frequency is assumed high such that, \( \frac{\partial}{\partial t} \gg U \frac{\partial}{\partial x} \) and the downstream free surface effects are ignored.

The far field radiation boundary condition is satisfied by incorporating an outer damping beach and modifying the free surface boundary conditions. The details are given in (Subramanian and Beck, 2015). The free surface equations (7) and (8) are integrated in time using a 4\textsuperscript{th} order Adams Bashforth-Moulton predictor-corrector scheme. All simulations are carried out in deep water and so the gradient of the perturbation potential vanishes as \( z \rightarrow -\infty \). This is automatically taken care of by the Rankine source function.
The perturbation potential can be broken down into different components for proper bookkeeping:

$$\phi(y,z,t;x) = \phi_I + \phi_D + \phi_R$$

where, $\phi_I$, $\phi_D$, $\phi_R$ refer to the incident, diffracted and radiated potential, respectively.

The incident wave potential and elevation are known at all time and the linear Airy wave theory is used. The analytical expressions for the potential and wave elevation with respect to the earth fixed inertial frame are:

$$\phi_I = \frac{i a \omega_0}{\omega_0} e^{-i k (x_c \cos \beta + y_c \sin \beta)} e^{i \omega_0 t} e^{k(z-\eta)}$$

$$\eta_I = a e^{-i k (x_c \cos \beta + y_c \sin \beta)} e^{i \omega_0 t}$$

(9)

(10)

Here, $a$, $\omega_0$, $k$, $\lambda$, and $\beta$ refer to the wave amplitude, frequency, wave number, wavelength and the heading angle relative to earth fixed $x$-axis, respectively. The incident wave potential formulation also includes the Wheeler stretching term.

The body boundary condition (5) is re-written in terms of its individual components in the two-dimensional frame

$$\nabla \phi_D \cdot N = -\nabla \phi_I \cdot N \quad \text{on } S_B(t)$$

$$\nabla \phi_R \cdot N = v \cdot N \quad \text{on } S_B(t)$$

(11)

(12)

Here the two-dimensional strip theory unit normal $N$ is used. $S_B(t)$ denotes the surface formed by the intersection of the instantaneous wetted body surface with the mean water level. Algorithms have been developed to find the intersection of the body surface and cut panels at each time step. Computations are only performed for sections that are wetted. The free surface evolution can either continue or be reset to zero. Thus, the present method allows for changing wetted geometry and emergence of sections.

The velocity $v$ used in Equation (12) is the velocity of a node on the body surface with respect to the earth fixed frame and includes all the three translational and rotational components, namely $(u,v,w,p,q,r)$. 

IIRE Publications: IJMRD

6
The mixed boundary value problem (Equations (6) - (12)) is solved for the perturbation potentials $\phi_R$ and $\phi_D$ and their derivatives. In the present work, a source distribution technique is used. Desingularised sources are placed above the free surface nodes and constant strength panels are used on the body. The desingularised method avoids complicated panel quadrature and can handle higher order derivatives in a straightforward manner (Cao, Schultz, and Beck, 1991). Details of the method are given in (Subramanian, 2012).

Once the potentials, pressure and forces acting on the body are determined, the equations of motion (EOM) are set up using the position and velocity of the body. The fully nonlinear 6-DOF EOM is used here. Details are given in (Subramanian, 2012). The velocities and displacements are time-stepped using a 4th order Adams-Bashforth time-stepping scheme. The new values of the variables are used to continue the evolution of the flow variables.

3. IKEDA ROLL DAMPING MODEL:
The Ikeda method estimates the roll damping coefficient by breaking it down into various components. These components are then individually modeled taking into account the physics governing each of them. The total roll damping moment $F^d_4$ is given by:

$$F^d_4 = -B^{(1)}_{44} \phi - B^{(2)}_{44} \dot{\phi} \ddot{\phi}$$

Here, the terms $B^{(1)}_{44}$, $B^{(2)}_{44}$ and $\dot{\phi}$ denote the linear damping, quadratic damping, and roll velocity, respectively. To use with linear frequency domain codes, the quadratic term is typically linearized by assuming the dissipated energy over a cycle to be equal.

This yields the following expression for the equivalent linear damping $B^{(1)}_{44eq}$,

$$B^{(1)}_{44eq} = \frac{8}{3\pi} \phi_a \omega B^{(2)}_{44}$$

Here, $\phi_a$ and $\omega$ denote the roll amplitude and encounter circular frequency respectively. Therefore, their product represents the roll velocity amplitude.
In the modified Ikeda method ((Kawahara, Maekawa, and Ikeda, 2009)), the ship hull geometry is represented by the Taylor Standard Series. Once the hull geometry is analytically represented, the integrations along the length can be carried out to obtain explicit expressions for the damping coefficients in terms of the hull main particulars. This approach is very useful to quickly estimate the roll damping coefficients at the preliminary phase of design when often the only details the designer has at his or her disposal, are the main particulars of the vessel. The damping coefficients consist of five components. These are the wave radiation, frictional, eddy, lift and bilge keel. A brief description and semi empirical formulas according to the modified Ikeda method are as follows.

3.1 Wave Radiation Damping:
The wave radiation component is the damping due to the radiated waves created due to roll. This is directly computed by the potential theory formulation. Therefore, the present method does not use this term. The detailed formulations are given in (Kawahara et al., 2009).

3.2 Frictional damping:
The frictional damping is attributed to the hull skin friction. This is estimated based on Kato’s formula from experimental results for cylinders.

3.3 Eddy damping:
The eddy component of the roll damping is created by small separation bubbles or shed vortices generated at the bilge of the midship section and large vortices generated at the relatively sharp bottom of bow and stern sections. Although vortex shedding from oscillating bluff bodies is usually governed by the Keulegan-Carpenter number ($K_C$ number), it was found by (Ikeda, Himeno, and Tanaka, 1978) that the viscous forces created by such small separation bubbles or small shedding vortices do not significantly depend on the $K_C$ number. In Ikeda’s prediction method, the distribution of the pressure created on a hull surface by such separation bubbles is assumed as a simple shape for each shape of cross sections on the basis of experimental results of pressure distribution on hull surfaces.
3.4 Lift damping:
The lift damping occurs due to lift forces generated by the hull surface when the ship has a non-zero forward speed. The present study is restricted to zero forward speed; hence this component is not considered at present.

3.5 Bilge keel damping:
It is quite typical to fit a bilge keel to most ocean going vessels. A well designed bilge keel can be an effective solution to reduce the roll response, where it can contribute more than 50% of the total roll damping. Vortices shed from the sharp edges of the bilge keels are responsible for the large viscous forces. Two primary mechanisms have been identified for the generation of quadratic damping moments. The first part is due to the normal force acting on the bilge keel, and the second sub-component, is due to the pressure on the hull surface created by the bilge keel.

4. EXPERIMENTAL SET-UP:
In order to validate the computational predictions, experiments have been carried out in the model basin in the Department of Ocean Engineering, Indian Institute of Technology Madras, India. A model of a 200 t oil vessel has been fabricated to a scale of 1: 12.5 using fiberglass. In addition to the geometric similarity, the dynamic similarity is ensured by properly distributing weights to obtain the correct inertia properties of the prototype. Superstructure and deck-house have been added to simulate correct mass distribution and account for dynamic effects due to deck-wetting and spray.

An inclining test was initially conducted to ascertain the GM of the loaded ship. All the model tests have been conducted for the fully loaded condition. After correctly loading the model, the free roll decay tests were conducted to access the roll natural period and damping characteristics. The ship model was given an initial roll displacement and released and allowed to settle down while undergoing damped oscillations. Care was taken to ensure that the initial velocity was zero and displacement was only in roll. The measurement system uses conductivity-type wave probes for wave measurements and ORE Motion Reference Unit (MRU) for measuring the roll. The MRU is placed on the deck level on the centerline of the vessel at the LCG position. The tests were performed in the wave flume measuring 90m in length, 4m in width, and 2.8m in depth; with
facilities for computer controlled wave generation. To evaluate the roll response of the vessel, regular waves of constant wave slope were generated for a range of frequencies. The model was oriented perpendicular to the wave fronts and placed beyond the region of transient waves and many wavelengths away from the parabolic wave absorbers at the far end to minimize effects of wave reflection. After each run, the model was reoriented to the beam sea configuration using two guide lines.

5. **NUMERICAL SET-UP:**

The geometry of the vessel is imported using the offset data of the ship. The strip theory formulation allows for effortless input and panel modeling. A cubic spline interpolation is used to obtain smooth contours on each of the sections. On the hull surface, an average of 60 flat panels are used per station. To accurately model the free-surface, 30 nodes are used per wavelength. To ensure mesh-independent solutions, the free-surface extends two wavelengths away from the body on each station. To enforce the far field boundary condition, a numerical beach is implemented extending two wavelengths. A time-step size of \( T/100 \) is used for the numerical integrations; \( T \) referring to the wave period. These values are chosen to minimize computation time and maximize accuracy, ensuring optimum numerical efficiency. These parameters have been set based on the extensive convergence studies performed in (Subramanian, 2012). To avoid large transients in the beginning of the simulations, a smooth ramp is used to ramp-up the incident wave amplitude. The ramp period duration is set to 2.5 wave periods.

6. **RESULTS AND DISCUSSION:**

The main particulars of the 200t oil vessel are given in Table 1. The body plan is shown in Figure 2. A photo of the model being prepared is shown in Figure 3. The outcome of the experimental roll decay test is shown in Figure 4. Multiple lines on the graph indicate multiple test runs, with different initial displacements. The roll decay has also been simulated numerically by giving the vessel a specified initial roll displacement and allowing the equations of motion solver to predict the behavior. The comparison of the two approaches is shown in Figure 5. The results show good agreement of
Figure.2.: Body plan, 200 t oil vessel

Figure.3.: Model of the oil vessel (Scale 1:12.5)

Table.1.: Main particulars of 200 t oil vessel

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<th>Model Scale</th>
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the numerical prediction with the experimental results. The natural period predicted by
the numerical method is 1.65 s, compared to the experimental value of 1.64 s. This also
indicates that the added mass term, $A_{44}$ is correctly predicted. Assuming linear behavior,
the non-dimensional damping coefficient, $k$, given by $\frac{B_{44}^{(1)}}{2\sqrt{(I_{44}+A_{44})C_{44}}}$, can be estimated
assuming an exponential decay of the roll oscillations. Here $I_{44}$, $C_{44}$, and $B_{44}^{(1)}$ refer to the
roll moment of inertia, hydrostatic stiffness and total linear roll damping term including
viscous effects, respectively.

The plot of successive peaks is shown in Figure 6. The numerical results show good
agreement with the experimental data. The smaller roll angles show small deviations,
with excellent match at higher amplitudes. In reality, because of the nonlinear nature of
roll, the damping coefficient varies with the amplitude. This is illustrated in Figures 7
and 8, which show the various roll damping coefficients of the 200 t oil vessel at the
roll natural frequency using the modified Ikeda method. The linearized damping
coefficient, $B_{44}$ is non-dimensionalised by

$$B_{44} ND^{(1)} = \frac{B_{44}^{(1)}}{\rho N B^2} \sqrt{\frac{B}{2g}}$$

Figure 7 shows the linearized damping coefficient is not constant but increases rapidly
with increasing roll amplitude, indicating the nonlinear characteristic. The nonlinear
(quadratic) coefficients are shown in Figure 8. It is seen that the coefficients are nearly
constant for $\theta > 5$ degs. As amplitudes get smaller, the damping coefficients increase
rapidly. This is attributed to the dependence of these viscous components to the
Reynold’s number and $K_C$ number. The plots also give a picture on the relative
contribution of these components in damping the roll motions. It is evident that the bilge
keel plays a major role in controlling the roll motion.

It is important to note that the only inputs to the computational program are the main
particulars of the vessel including the bilge keel geometry. Typically, the amount of
roll damping is “tuned” using the experimental results. In the present method, the roll
damping is completely modeled using the nonlinear form of the semi-empirical Ikeda
method described in Section 3.
The roll response in regular incident waves at 90 degs is shown in Figure 9. The incident wave slope is set to a value of $H/\lambda = 1/60$. $H$ and $\lambda$ denote the wave height and wavelength, respectively. The comparisons are favorable, with excellent agreement at and near the roll natural period. Good agreement is seen for larger wave periods. There is some deviation at smaller periods. The reason for this is unknown at present, and will be further investigated. Figure 10 presents the non-dimensionalised roll response computed for different incident wave slopes. The dramatic changes near the resonance

Figure 5: Comparison of numerical and experimental roll decay test.
**Figure 6.** Comparison of exponential decay characteristics

![Comparison of exponential decay characteristics](image)

**Figure 7.** Linear and linearized roll damping components using Ikeda method

![Linear and linearized roll damping components using Ikeda method](image)

**Figure 8.** Nonlinear roll damping components using Ikeda method

![Nonlinear roll damping components using Ikeda method](image)
Figure 9.: Roll response in regular waves, $H/\lambda = 1/60$

![Roll RAO - Model Scale, $H/\lambda$: 1/60](image)

- Amplitude (°/kN)
- Phase (°)
- Period (s)

Figure 10.: Variation of roll response with incident wave slope

![Roll RAO - Model Scale](image)

- Amplitude (°/kN)
- Phase (°)
- Period (s)

Figure 11.: Roll motion time history

![Roll motions](image)

- Motion (°)
- Time ($t/T$)

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**Figure 12.** Roll moment (Incident and Diffraction) time history

**Figure 13.** Roll moment (Radiated and Hydrostatic) time history

**Figure 14.** Roll moment (Viscous and $|\nabla \phi|^2$) time history
period highlights the strong nonlinear behavior. The nonlinear approach is capable of capturing this phenomenon. It is also interesting to note that the peak response shifts slightly, shifting towards lower period as the wave slope increases. The response also gets flatter as the wave slope increases. The time-history of the roll response for incident wave period, \( T = 1.66 \) s is shown in Figure 11. It is observed that the mean roll is non-zero due to the effect of higher order loads. As expected, this is more pronounced for the higher wave slope of \( H/\lambda = 1/40 \). Examining the roll moments acting on the vessel gives a better picture of the underlying dynamics.

The breakdown of the roll moment components is shown in Figures 12 through 14. It is clear that all components exhibit nonlinear characteristics, which can be attributed to the changing wetted surface of the body. The radiated component includes the damping due to potential wave radiation. The incident wave and hydrostatic forces are computed up to the exact incident wave surface. The viscous moments represent the contribution from the roll damping model. Although the damping coefficient is quadratic in nature, roll velocity varies with wave slope. This implies that the viscous moments can exhibit non-quadratic variation with wave amplitude (Figure 14). The term \( |\nabla \phi|^2 \) refers to the \( \nu^2 \) term contribution in the Bernoulli (pressure) equation.

7. SUMMARY AND CONCLUSIONS:
A computer program based on the time-domain body-nonlinear approach has been developed. Implementation of the strip-theory formulation allows for computational efficiency and simplified body geometry definition. It also allows for vastly efficient algorithms for dealing with dynamic wave-body intersection and panel cutting. The fully nonlinear form of the modified Ikeda-Kawahara roll damping method has been implemented to accurately predict the roll behavior. The combination of a higher-order time-domain approach and fully nonlinear damping technique allows for accurately predicting large amplitude roll response without the need for linearizing the coefficients.

In order to validate the numerical predictions, experiments have been conducted on a 200t fuel vessel at the model basin in the Department of Ocean Engineering at IIT Madras. The free decay simulations show excellent agreement with the experimental results. The roll natural frequency matches closely with the measured value. The
predicted roll responses for incidence angle of 90 degs also show good agreement with the measured values. The simulations are able to accurately predict the peak responses. There exists some deviation at higher wave frequencies. The exact reason for this is unknown at present and further studies will have to be done. The responses have been computed for various values of the wave slopes. The roll responses show the expected nonlinear behavior, including a mean component due to higher-order mean loads. Closer examination of the wave loads shows nonlinear contribution from all the components. This is attributed to the changing wetted body geometry, inclusion of exact Froude-Krylov and hydrostatics, and the nonlinear viscous damping.

The methodology promises to be beneficial to not only ship designers, but also offshore platforms such as FPSOs and FLNGs. The modified Ikeda method is particularly useful to estimate the roll response at the early stages of design, since the only inputs required are the main particulars of the vessel.

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About the Authors

Dr. Rahul Subramanian
Has received his bachelor’s degree in Ocean Engineering from the IIT, Madras in 2007 and earned his PhD degree in Naval Architecture and Marine Engineering from the University of Michigan in 2012. He is currently a faculty with the Ocean Engineering Department at Texas A and M University at Galveston, USA.

Ms. Aishwarya Jayaraman
Has received her B. Tech degree in Aerospace Engineering from Amrita School of Engineering, Coimbatore, India in 2012. She is currently an M.S Research Scholar. Her main areas of research interest are ship hydrodynamics and motion studies, and stabilization systems.

Mr. Jyothish Puthukkattil Vijayan
Has received his B.E. from the Aeronautical society of India, New Delhi, India, in 2012. Since 2014, he has been an MS research scholar with the Department of Ocean Engineering at the Indian Institute of Technology, Madras.
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