

Application of Advanced System Identification Technique to Extract Roll Damping from Model Tests in Order to Accurately Predict Roll Motions

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Abstract

In this paper our previously developed advanced system identification technique [1] has been applied to extract the frequency dependent roll damping from a series of model tests run in irregular (random) waves. It is shown that this methodology accurately models the roll damping which can then be used to produce accurate predictions of the ships roll motion. These roll motion predictions are not only more accurate than the potential flow predictions but more accurate than potential flow models corrected using either empirical prediction methods [2] and even those corrected using roll damping obtained from free decay sallying experiments. This methodology has the potential to significantly improve roll motion prediction during full scale at sea trials of vessels in order to dramatically improve safety of critical operations such as helicopter landing or ship to ship cargo transfer.

Keywords: Reverse-Multiple Input Single Output, Roll damping, System identification, Partial coherence, Conditional Spectra, R/V Melville, Environmental ship motion forecasting

1. Introduction

For most displacement vessels roll is the most critical of the six degrees of freedom. Unlike damping in the other five degrees of freedom, roll damping is not only frequency dependent, but also nonlinear and strongly dependent upon viscous effects.

Some of the complex dynamics exhibited by roll mode of motion where viscous roll damping plays a crucial role include dead ship capsizing in beam seas [3], second order roll motion [4, 5] and parametrically excited roll motion [6, 7, 8]. As the linear roll damping coefficient is reduced, Falzarano showed that the peak of the beam sea roll motion magnification curve comes close to the angle of vanishing stability and exhibits complex dynamics [9, 10]. An extension of this problem for random beam sea excitation was investigated by Su and Falzarano [11, 12] where further it was shown that reducing the damping increased the probability of capsize. In order to apply similar techniques to the problem of parametric roll Somayajula and Falzarano developed [13, 14] and compared [15, 16] simplified models to determine which model accurately captured the relevant dynamics of the problem. With the application of techniques from nonlinear dynamical systems and stochastic dynamics to the simplified model two independent analytical stability criteria were formulated. Both these approaches demonstrated that the dynamics of parametric roll in irregular seas are dependent on two factors - the strength of the tuned excitation and the nonlinear quadratic damping present in the system [17]. From all these examples it is clear that one of the critical factors in the accurate prediction of ship rolling motion is the accurate prediction of roll damping.

The standard approach to predict roll damping is a combination of potential flow wave interaction models and corrections for viscous effects. The correction for viscous effects can be done using standard empirical

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methods [2, 18, 19] or using free decay model tests. However, both methods have their limitations. The empirical method is based upon separating the damping into components. However, with any empirical method it is limited by the range of vessel characteristics considered. Moreover, the free decay test can only extract the damping at the natural frequency and depends upon building and testing a physical model [2]. To overcome these limitations some sort of forced rolling apparatus have been developed. However, such devices are extremely expensive to deploy. Moreover, forced rolling physical model tests techniques are generally limited to model scale.

An alternative would be to somehow extract the roll damping of a ship model or full scale vessels in waves. Such a capability can be obtained by utilizing an advanced system identification technique. The area of system identification is quite vast with many different approaches. Some of the more popular methods include Restoring Force Surface (RFS) [20, 21], NARMAX (Nonlinear Auto-Regressive Moving Average with eXogenous inputs) [22, 23], Hilbert transform [24, 25], Volterra system [26, 27, 28] and the Reverse-Multiple Input Single Output technique (R-MISO) [29, 30, 31, 32, 33]. A more detailed literature review of various system identification techniques applied to marine systems and a discussion of the advantages and limitations of each method can be found in Somayajula and Falzarano [1].

Somayajula and Falzarano [1, 34] previously developed a system identification tool based on the R-MISO technique. Using this technique it is possible to extract the actual nonlinear viscous frequency dependent roll damping from model tests or full scale measurements at sea. In this paper this capability will be demonstrated by extracting the actual roll damping from model tests of *R/V Melville* conducted at the seakeeping and maneuvering basin of Naval Surface Warfare Center Carderock Division (NSWCCD) [35]. The accuracy of this method will be demonstrated by calculating the roll motion and comparing it with experimentally measured time histories. It will be shown that the roll motion predictions from this approach are significantly better than those predicted using other methods. The larger objective is to investigate the effectiveness of this method for incorporation into the Environmental Ship Motion Forecasting (ESMF) project being pursued by the Office of Naval Research (ONR) [36, 37, 38].

2. Reverse Multiple Input Single Output (R-MISO) theory

This section documents the theory of the R-MISO technique which will be used in the next section to extract viscous roll damping from experimental data.

A MISO system is characterized by $q > 1$ inputs $x_i(t)$, $i = 1, 2, \dots, q$ and a single output $y(t)$ as shown in Figure 1. The inputs and outputs are assumed to be zero mean ergodic processes. The system is defined by a set of linear operators $H_{iy}(\omega)$, $i = 1, 2, \dots, q$ which relate contributions of each of the inputs $x_i(t)$ to the output $y(t)$. Although the input measurements are assumed to be devoid of noise, a system noise $n(t)$ is assumed at the output. In most physical scenarios it is also reasonable to assume that the noise $n(t)$ is uncorrelated with each of the inputs $x_i(t)$. Notice that the inputs can be correlated with each other which implies that the total contribution from any input $x_i(t)$ to the output $y(t)$ is the sum of the output through $H_{iy}(\omega)$ and other correlated outputs through other operators $H_{jy}(\omega)$ where $j \neq i$.

System identification is commonly associated with solving the inverse problem where the inputs $x_i(t)$, $i = 1, 2, \dots, q$ and output $y(t)$ are the known quantities, usually measured from experiments and the objective of the analysis is to determine the optimum filters $H_{iy}(\omega)$, $i = 1, 2, \dots, q$ such that the noise $n(t)$ is minimized. Bendat and Piersol [29] showed that determining optimum filters such that the noise is minimized mathematically leads to the inputs $x_i(t)$, $i = 1, 2, \dots, q$ being uncorrelated with the noise $n(t)$. When all the inputs are uncorrelated with each other, the system can be decomposed into q Reverse-Single Input Single Output (R-SISO) systems which can be solved easily. However, in general the inputs $x_i(t)$ are correlated with each other which makes solving for $H_{iy}(\omega)$ difficult.

2.1. Conditioning the input signals

It is much easier to solve for the system operators when the inputs, in addition to being uncorrelated with the noise are also uncorrelated with each other. Thus the first step of the approach is to convert the given system into a new system with uncorrelated inputs which can then be solved easily. For this, the inputs

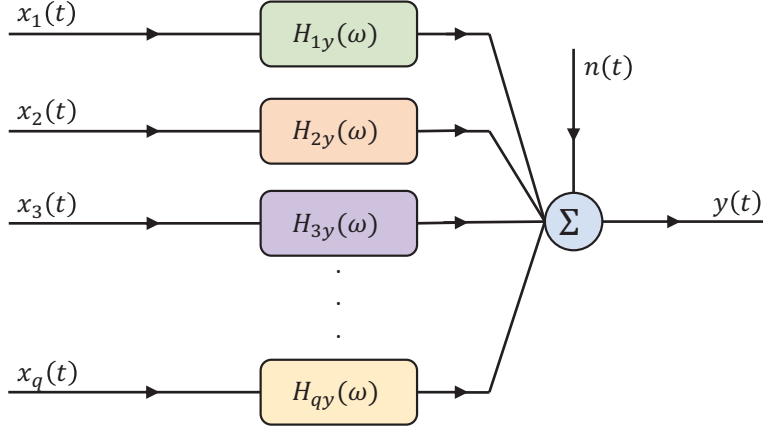


Figure 1: A general MISO system with q inputs $x_i(t)$, $i = 1, 2, \dots, q$ and a single output $y(t)$

need to be formulated into a set of q uncorrelated signals which is obtained by conditioning an input over all the previous inputs. For example, consider $x_2(t)$ which can be assumed to composed of two components - $x_{2:1}(t)$ and $x_{2,1}(t)$ as shown in (1) where $x_{2:1}(t)$ is the part of $x_2(t)$ correlated with $x_1(t)$ and $x_{2,1}(t)$ is the part uncorrelated with $x_1(t)$.

$$x_2(t) = x_{2:1}(t) + x_{2,1}(t) \quad (1)$$

By a similar process $x_3(t)$ can be expressed as a sum of $x_{3:2!}(t)$ (component correlated with $x_1(t)$ and $x_2(t)$) and $x_{3,2!}(t)$ (component uncorrelated with both $x_1(t)$ and $x_2(t)$) as shown in (2). Extending this process for other inputs, $x_i(t)$ can be expressed as a sum of $x_{i:(i-1)!}(t)$ (part of $x_i(t)$ correlated with $x_1(t), x_2(t), \dots, x_{(i-1)}(t)$) and $x_{i,(i-1)!}(t)$ (part of $x_i(t)$ uncorrelated with $x_1(t), x_2(t), \dots, x_{(i-1)}(t)$) as shown in (3). Thus a new set of uncorrelated inputs can be specified as $x_{i,(i-1)!}$, for $i = 1, 2, \dots, q$.

$$x_3(t) = x_{3:2!}(t) + x_{3,2!}(t) \quad (2)$$

$$x_i(t) = x_{i:(i-1)!}(t) + x_{i,(i-1)!}(t) \quad (3)$$

The original system in Figure 1 can now be expressed in terms of the uncorrelated inputs as shown in Figure 2. Note that the linear operators $L_{iy}(\omega)$ for this modified inputs are different from the original system operators $H_{iy}(\omega)$. Thus the estimation of $H_{iy}(\omega)$ is now separated into two steps. First one is the estimation of $L_{iy}(\omega)$ and the second is to get $H_{iy}(\omega)$ from $L_{iy}(\omega)$.

2.2. Solving the MISO system with uncorrelated inputs

Taking a Fourier transform of each of the inputs, output and noise, the MISO system can be represented in the frequency domain by Figure 3. The inputs $x_{i,(i-1)!}(t)$, output $y(t)$ and noise $n(t)$ are replaced by their corresponding Fourier transforms $X_{i,(i-1)!}(\omega)$, $Y(\omega)$ and $N(\omega)$ respectively. In the frequency domain the output $Y(\omega)$ can be expressed as a linear sum contributions from inputs and noise as shown in (4).

$$Y(\omega) = \sum_{i=1}^q L_{iy}(\omega) X_{i,(i-1)!}(\omega) + N(\omega) \quad (4)$$

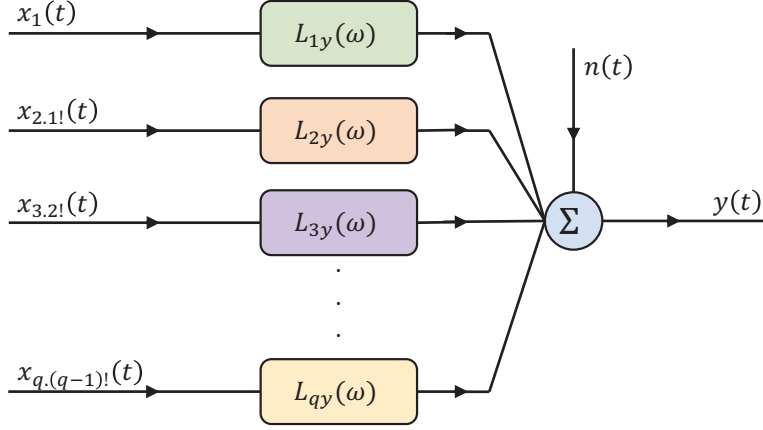


Figure 2: Alternate formulation of MISO system with q uncorrelated inputs $x_{i.(i-1)!}(t)$, $i = 1, 2, \dots, q$ and a single output $y(t)$

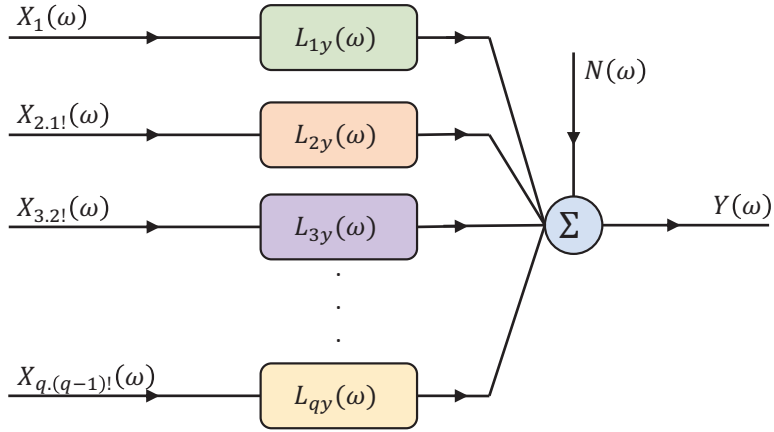


Figure 3: R-MISO system in frequency domain with inputs $X_{i.(i-1)!}(\omega)$, $i = 1, 2, \dots, q$, output $Y(\omega)$ and noise $N(\omega)$

Since each of the inputs $X_{i.(i-1)!}(\omega)$ and noise $N(\omega)$ are uncorrelated with each other, the system is equivalent to q R-SISO (Reverse-Single Input Single Output) systems. Since it is assumed that the inputs and outputs are ergodic processes, the cross spectra between any of the two inputs can be calculated using the Fourier transforms and is shown in (5) and (6). Since the inputs of conditioned system (Figure 3) are uncorrelated with each other, their cross spectra will be zero.

$$\lim_{T \rightarrow \infty} \frac{2}{T} E[X_{j.(j-1)!}^* X_{i.(i-1)!}] = \begin{cases} 0 & \text{if } i \neq j \\ S_{jj.(j-1)!} & \text{if } i = j \end{cases} \quad (5)$$

$$\lim_{T \rightarrow \infty} \frac{2}{T} E[X_{j.(j-1)!}^* N] = S_{jn.(j-1)!} = 0 \quad \text{for all } j = 1, 2, \dots, q \quad (6)$$

Note that X^* denotes the complex conjugate of X and $E[\]$ denotes the expected value operator. Multiplying (4) by $\frac{2}{T} X_{j.(j-1)!}^*$ and taking expected value under the limit $T \rightarrow \infty$ results in an expression for the linear operators L_{jy} as shown in (9).

$$\lim_{T \rightarrow \infty} \frac{2}{T} E[X_{j.(j-1)!}^* Y] = \lim_{T \rightarrow \infty} \frac{2}{T} \sum_{i=1}^q L_{iy}(f) \underbrace{E[X_{j.(j-1)!}^* X_{i.(i-1)!}]}_{=0 \text{ if } i \neq j} + \lim_{T \rightarrow \infty} \frac{2}{T} \underbrace{E[X_{j.(j-1)!}^* N]}_{=0} \quad (7)$$

$$S_{jy.(j-1)!} = L_{jy} S_{jj.(j-1)!} \quad (8)$$

$$L_{jy} = \frac{S_{jy.(j-1)!}}{S_{jj.(j-1)!}} \quad (9)$$

The conditional cross-spectral density function between $X_{i.r!}$ and $X_{j.r!}$ is defined by

$$S_{ij.r!} = \lim_{T \rightarrow \infty} \frac{2}{T} E[X_{i.r!}^* X_{j.r!}] \quad \text{for } i > r, j > r \quad (10)$$

However, additionally the following relations can be derived which are useful in simplifications.

$$S_{ij.r!} = \lim_{T \rightarrow \infty} \frac{2}{T} E[X_{i.r!}^* X_{j.r!}] + \lim_{T \rightarrow \infty} \frac{2}{T} \underbrace{E[X_{i.r!}^* X_{j.r!}]}_{=0} = \lim_{T \rightarrow \infty} \frac{2}{T} E[X_{i.r!}^* X_j] \quad (11)$$

$$S_{ij.r!} = \lim_{T \rightarrow \infty} \frac{2}{T} E[X_{i.r!}^* X_{j.r!}] + \lim_{T \rightarrow \infty} \frac{2}{T} \underbrace{E[X_{i.r!}^* X_{j.r!}]}_{=0} = \lim_{T \rightarrow \infty} \frac{2}{T} E[X_i^* X_{j.r!}] \quad (12)$$

This results in the identity shown in (13). Note that this identity has been used in the simplification of the left hand side of (7).

$$S_{ij.r!} = \lim_{T \rightarrow \infty} \frac{2}{T} E[X_{i.r!}^* X_{j.r!}] = \lim_{T \rightarrow \infty} \frac{2}{T} E[X_{i.r!}^* X_j] = \lim_{T \rightarrow \infty} \frac{2}{T} E[X_i^* X_{j.r!}] \quad (13)$$

Similar to (4) any general input $x_j(t)$ can also be conditioned over $r < j$ inputs as shown in (14). The R-MISO system conditioning $x_j(t)$ over $r < j$ inputs is shown in Figure 4. Note that $X_{j.r!}(\omega)$ represents the component of $X_j(\omega)$ uncorrelated with $X_i : i = 1, 2, \dots, r$.

$$X_j(\omega) = \sum_{i=1}^r L_{ij}(\omega) X_{i.(i-1)!}(\omega) + X_{j.r!}(\omega) \quad (14)$$

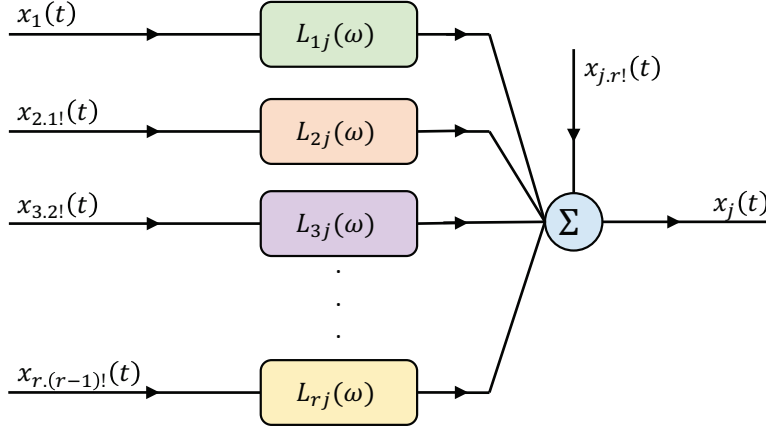


Figure 4: R-MISO system in frequency domain with inputs $X_{i,(i-1)!}(\omega)$, $i = 1, 2, \dots, r$, $X_{j,r!}(\omega)$ and output $X_j(\omega)$

It can be seen that (4) is a special case of (14) where $j = q+1$, $r = q$ and we additionally define $X_{q+1} = Y$ and $L_{i(q+1)} = L_{iy}$. If $X_j(\omega)$ was conditioned over $(r-1)$ inputs instead of r inputs, then the system would have been described by (15).

$$X_j(\omega) = \sum_{i=1}^{r-1} L_{ij}(\omega) X_{i,(i-1)!}(\omega) + X_{j,(r-1)!}(\omega) \quad (15)$$

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Comparing (14) and (15) results in

$$X_{j,r!}(\omega) = X_{j,(r-1)!}(\omega) - L_{rj}(\omega) X_{r,(r-1)!}(\omega) \quad (16)$$

Similar to the expressions for L_{jy} obtained in (9), the transfer functions $L_{rj}(\omega)$ are given by

$$L_{rj} = \frac{S_{rj,(r-1)!}}{S_{rr,(r-1)!}} \quad (17)$$

Substituting $L_{rj}(\omega)$ into (16) results in (18). Multiplying both sides by $\frac{2}{T} X_{i,r!}^*$ and taking expected value under the limit $T \rightarrow \infty$ results in (19) where $i > r$, $j > r$ and $r = 1, 2, \dots, q$.

$$X_{j,r!} = X_{j,(r-1)!} - \frac{S_{rj,(r-1)!}}{S_{rr,(r-1)!}} X_{r,(r-1)!} \quad (18)$$

$$S_{ij,r!} = S_{ij,(r-1)!} - \frac{S_{rj,(r-1)!}}{S_{rr,(r-1)!}} S_{ir,(r-1)!} \quad (19)$$

2.3. Solving for the operators of the correlated MISO system

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The governing equation for MISO system described by Figure 1 in the frequency domain is given by (20).

$$Y(\omega) = \sum_{i=1}^q H_{iy}(\omega) X_i(\omega) + N(\omega) \quad (20)$$

Multiplying (20) by $\frac{2}{T}X_{j.(j-1)}^*$ and taking expected value under the limit $T \rightarrow \infty$ results in a system of equations for H_{iy} as shown in (22).

$$\begin{aligned} \lim_{T \rightarrow \infty} \frac{2}{T} E[X_{j.(j-1)}^* Y] &= \lim_{T \rightarrow \infty} \frac{2}{T} \sum_{i=1}^q H_{iy} \underbrace{E[X_{j.(j-1)}^* X_i]}_{=0 \text{ if } i < j} \\ &+ \lim_{T \rightarrow \infty} \frac{2}{T} \underbrace{E[X_{j.(j-1)}^* N]}_{=0} \end{aligned} \quad (21)$$

$$S_{jy.(j-1)!} = \sum_{i=j}^q H_{iy} S_{ji.(j-1)!} \quad (22)$$

Dividing (22) throughout by $S_{jj.(j-1)!}$ and substituting from (9) and (17) results in (23).

$$L_{jy} = \sum_{i=j}^q H_{iy} L_{ji} \quad j = 1, 2, \dots, q, \quad i \geq j \quad (23)$$

(23) provides a scheme to evaluate H_{jy} system from L_{jy} system by working backwards as shown below.

$$H_{qy} = L_{qy} \quad (24)$$

$$H_{(q-1)y} = L_{(q-1)y} - L_{(q-1)q} H_{qy} \quad (25)$$

$$H_{(q-2)y} = L_{(q-2)y} - L_{(q-2)(q-1)} H_{(q-1)y} - L_{(q-2)q} H_{qy} \quad \text{and so on} \quad (26)$$

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Concisely, it can be expressed as shown in (28).

$$H_{qy} = L_{qy} \quad (27)$$

$$H_{jy} = L_{jy} - \sum_{i=j+1}^q L_{ji} H_{iy} \quad \text{where } j = (q-1), (q-2), \dots, 2, 1 \quad (28)$$

2.4. Partial coherence function

120 One of the main advantage of the conditioning approach is the calculation of partial coherence functions which provide an insight into the contribution of various inputs $x_i(t) : i = 1, 2, \dots, q$ towards the final output $y(t)$. The partial coherence function $\gamma_{iy.(i-1)!}^2(\omega)$ is defined as (29) and denotes the contribution of conditional input $x_{i.(i-1)!}(t)$ towards the final output $y(t)$.

$$\gamma_{iy.(i-1)!}^2(\omega) = \frac{|S_{iy.(i-1)!}(\omega)|^2}{S_{ii.(i-1)!}(\omega) S_{yy}(\omega)} \quad (29)$$

The sum of all the partial coherence functions of a system is always less than equal to unity. The equality is true when the system has no external noise.

$$\sum_{i=1}^q \gamma_{iy.(i-1)!}^2(\omega) \leq 1 \quad (30)$$

The importance of partial coherence functions will be further discussed in the next section in the context of an example.

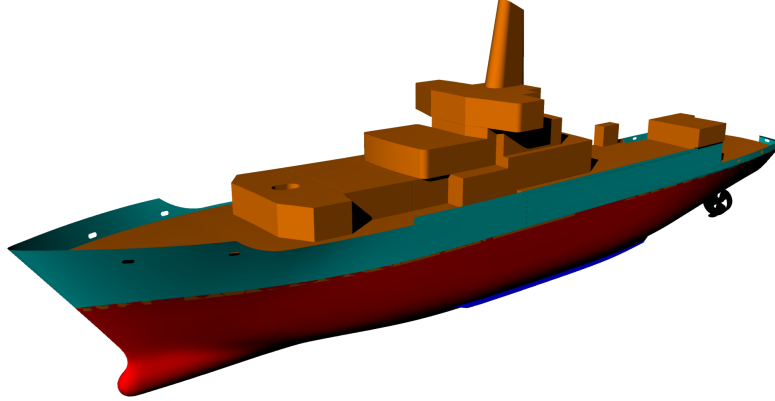


Figure 5: CAD model of *R/V Melville*

3. Application of R-MISO technique to Melville data

In this section the above described method is applied to the *R/V Melville* model test conducted at the Maneuvering and Seakeeping (MASK) basin of the Naval Surface Warfare Center Carderock Division (NSWCCD) to identify the roll damping in irregular waves. The CAD model of the vessel is shown in Figure 5 and the particulars are listed in Table 1. The details of the experimental setup and the description of the data collected can be found in the Naval Surface Warfare Center (NSWC) report [35].

The objective of this study is to demonstrate the applicability of the R-MISO technique to real physical model test data so that the developed tool can be later integrated into the Environmental Ship Motion Forecasting (ESMF) project being pursued by the Office of Naval Research (ONR) [37, 38, 36]. The ESMF project aims at predicting the motions of a ship in the near future in real time to assist maritime personnel in making decisions regarding certain operations like cargo transfer at sea or other sea-basing operations.

The first step involves the measurement of the wave environment around a vessel using a radar. Once the wave field data is collected, a forecasting model is used to evolve the wave field over a specified short period of time. This evolved wave field data is then provided as an input to a reduced order simulation model to predict the motions in the future. The magnitude of the predicted motions provide a better understanding towards whether certain operations can be undertaken or not.

It was found that the heave and pitch motion predictions from the reduced order models agreed well with the measured data. However, the roll motion predictions showed considerable differences from the measurements which suggested that some of the nonlinearities in the roll mode of motion were not being captured by the reduced order models. The current work on system identification reported in this article was undertaken in order to improve the reduced order models and bridge the gap between simulations and experiments.

Table 1: Particulars of *R/V Melville*

Particulars	Value
Length between perpendiculars L_{pp} (m)	77.40
Breadth B (m)	14.00
Mean Draft T (m)	4.95
Displacement $\Delta g = \rho \nabla g$ (kN)	28674
Vertical Center of Gravity $\overline{KG}$ (m)	5.89
Metacentric Height $\overline{GM}$ with Free Surface Correction (m)	1.09
Roll Natural Period T_n (sec)	11.25

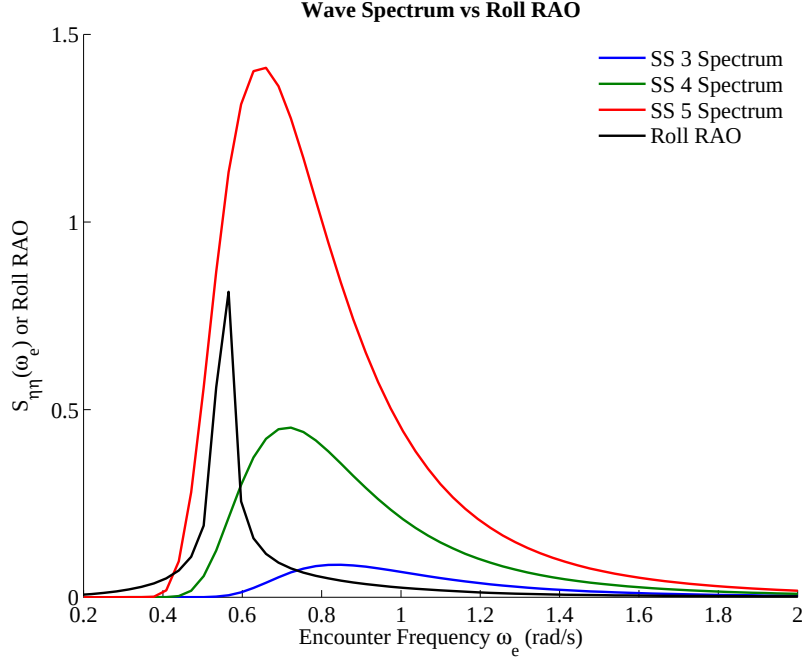


Figure 6: Comparison of wave spectra for different sea states

3.1. Beam sea test cases

Three different sea states were analyzed for every heading. All sea states were characterized by a Bretschneider spectrum. Sea state 3 (SS 3) is characterized by a significant wave height of 0.9 meters and a peak period of 7.5 seconds. Sea state 4 (SS 4) is characterized by a significant wave height of 1.9 meters and a peak period of 8.8 seconds. Sea state 5 (SS 5) is characterized by a significant wave height of 3.2 meters and a peak period of 9.7 seconds. Figure 6 shows the comparison of the wave spectra for SS 3, SS 4 and SS 5.

It can be seen that the roll RAO of the ship is well outside the frequency range where the wave energy is located for both SS 3 and SS 4. However, the peak of the spectrum for SS 5 coincides with the roll RAO peak. This indicates that for SS 5 significant roll motions must be observed as compared to the other sea states. It also means that for SS 3 and SS 4, the identification will be more challenging as the only a small amplitude of roll motion is expected.

3.2. Bridge versus carriage sensors

A number of wave probes were used to measure the wave elevation in the model basin. These included 6 stationary wave probes fixed on the bridge of the basin. Additionally 4 other wave probes were affixed to the carriage which moved along the bridge with a steady speed during the measurements. Originally these carriage fixed wave probes were assumed to be accurate. However a later comparison between the wave elevation from the bridge and the carriage wave probes indicated that the carriage wave probes had errors in measurements. A comparison of the wave elevation at the carriage javellin using the carriage and bridge wave probes is shown in Figure 7.

It can be seen that the wave elevation calculated from the carriage sensor data is much less than that from the bridge sensor data. Based on similar comparisons for other beam sea runs, it was concluded that the bridge sensor data will be used in the further study.

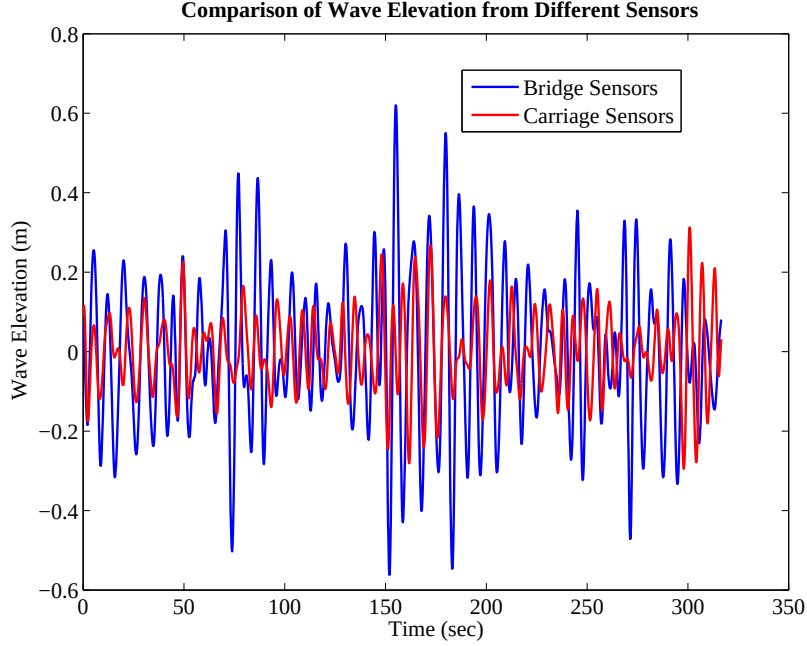


Figure 7: Comparison of the wave elevation at the carriage javellin computed using carriage and bridge wave probe measurements for Run 809 (sea state 3)

3.3. Drift of the model

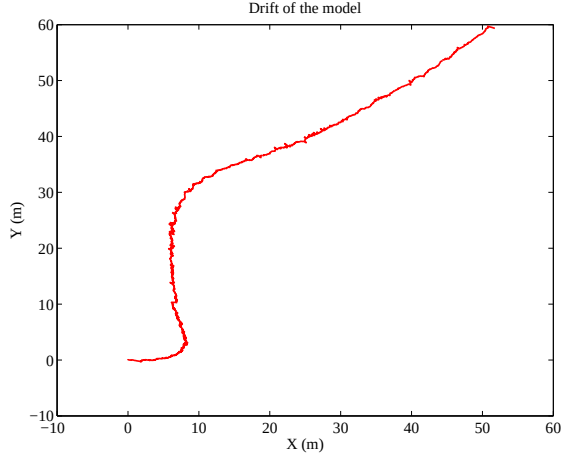
The model was a free running model controlled by a manual thrust throttle and a manual controller for adjusting the direction of the azimuthing propellers. The free running model test indicates that the measured motions of the model will include the drift of the model in addition to its wave induced motions. This drift arises due to the combination of wave drift forces and also the manual control of the thrust. This implies that the wave elevation measured from the probes must be transformed to provide the wave elevation at the origin of a body fixed coordinate system. The provided data includes the location of the model in both local and global coordinate systems. However, in course of analysis it was found that the location data in the local coordinate system did not match the corresponding location data in global coordinates. A plot of the location of the model based on the data measured in local coordinate system is shown in Figure 8a. The corresponding location of the model based on the measurements in global coordinate system is shown in Figure 8b. A similar trend has been observed in all the test runs, suggesting that the two measurement channels are not in agreement with each other and only one of them is correct.

In order to assess which of the two drifts are correct, a comparison of the partial coherence functions between the roll motion time series and the drift corrected wave elevation was performed. The comparison between the two cases is shown in Figure 9.

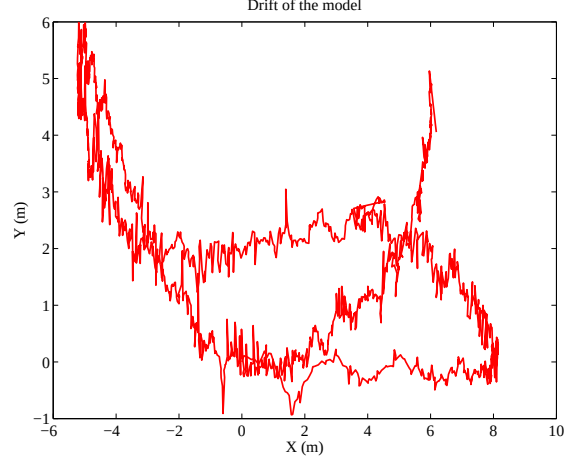
It can be seen that the coherence between roll motion and drift corrected wave elevation based on local location data is very low. However, a beam sea response should indicate a higher level of coherence as seen by the data based on global location. A similar trend was also found in all of the other beam sea cases implying that the global location data is correct.

3.4. Discussion of results from R-MISO analysis

For each sea state 5 different runs were performed for all three sea states. Since the analysis of each of the five runs are found to be similar, only the results of analysis of one run from each sea state are shown. The three runs chosen for which the results are shown in the paper are Run 809 (SS 3), Run 824 (SS 4) and Run 838 (SS 5). The wave elevation time series obtained using the bridge sensor data for the three different

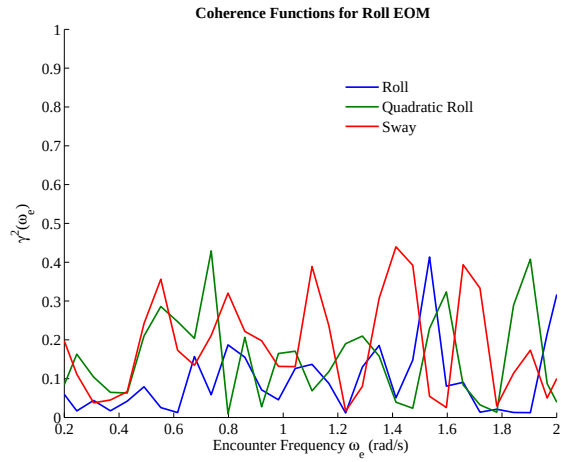


(a) Calculated using the local location data

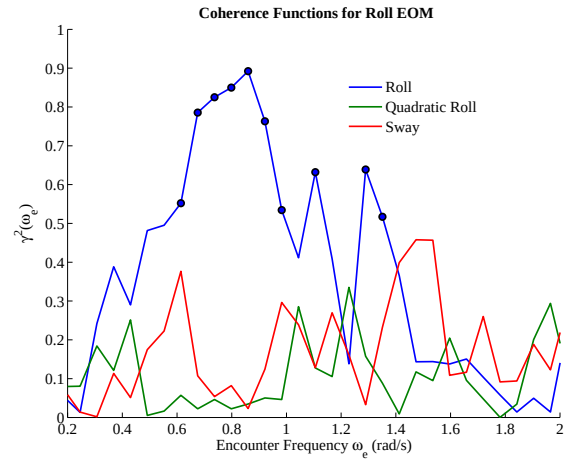


(b) Calculated using the global location data

Figure 8: Location of the model in a coordinate system moving steadily with the carriage and fixed at the location of the model at time $t = 0$ calculated using the local location data for Run 809



(a) Based on local location data



(b) Based on global location data

Figure 9: Partial Coherence between roll motion and drift corrected wave elevation for Run 809

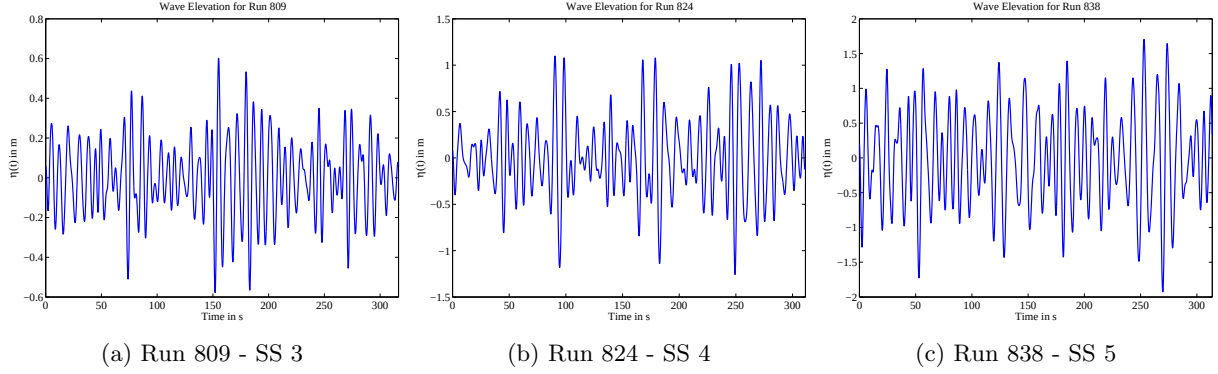


Figure 10: Beam sea wave elevation time series for different sea states

sea states is shown in Figure 10. The corresponding motion time histories of the ship for sea states 3, 4 and 5 are shown in Figure 11, Figure 12 and Figure 13 respectively.

In most of the beam sea runs it was found that the roll motion did not exceed 15 degrees which suggested, that the effect of nonlinearities should not be significant. In order to assess the importance of nonlinearities, both R-MISO and R-SISO analyses were performed for each case. For the R-MISO analysis, the roll displacement $\xi_4(t)$, quadratic roll velocity $\dot{\xi}_4|\dot{\xi}_4|$ and sway displacement $\xi_2(t)$ are used as the inputs and the roll diffraction (incident and scattering) moment is specified as the output. For the R-SISO system, only a single input of roll displacement $\xi_4(t)$ is specified while the output is the roll diffraction moment similar to the R-MISO analysis. The comparison of roll RAOs from R-SISO and R-MISO analyses for three sea states against the roll RAO obtained from potential theory are shown in Figure 14. The corresponding comparison of partial coherence functions for the same cases is shown in Figure 15.

A standard 3-D Green function based in-house panel method code - MDLHydroD - is used to generate the potential theory results [39]. This work was originally developed for a single body with zero speed and later extended to include forward speed effects [40], calculation of second order mean drift forces and moments [41] and finite water depth effects [42]. Recently this program has further been extended to allow the hydrodynamics analysis of multiple bodies [43, 44].

The partial coherence functions for the R-MISO cases show that the partial coherence of quadratic roll and sway motions is in general less than 50 % and is much smaller than the partial coherence due to the roll motion. The partial coherence due to sway motion for R-MISO analysis of Run 824 does show the sway coherence exceeding the roll partial coherence at high frequencies. However, there is hardly any roll motion at these high frequencies indicating that this is not a significant phenomenon.

Since the quadratic roll and sway contributions to the roll equation of motion are not significant, excluding them and performing the R-SISO analysis is expected to provide better capture of the roll RAO. This can be observed from Figure 14 where the R-SISO analyses yield better estimation of roll RAO as compared to the R-MISO analyses. The better capture of RAO by R-SISO analysis also confirms the fairly linear nature of the system.

The roll RAO obtained by R-SISO analyses in Figure 14 shows that the predicted roll RAO agrees fairly well in the tail region of the RAO. However, the peak from experiments in all three cases is less than the peak predicted from potential theory. This suggests that although the system does not exhibit the presence of quadratic roll damping, it does have a significant linear viscous damping. From the roll RAO obtained from potential theory in Figure 14 it can be observed that the roll natural frequency $\omega_n \approx 0.5 \text{ rad/s}$. From Figure 6 it can be observed that SS 4 and SS 5 wave frequency ranges overlap with the roll RAO peak. This means that roll motion coherence at the roll resonant frequency will be significant only for these sea states. This is also observed from Figure 15 that the partial coherence of roll motion around the roll natural frequency is very low for SS 3 and progressively increases for higher sea states.

The roll RAO in case of a R-SISO system is given by

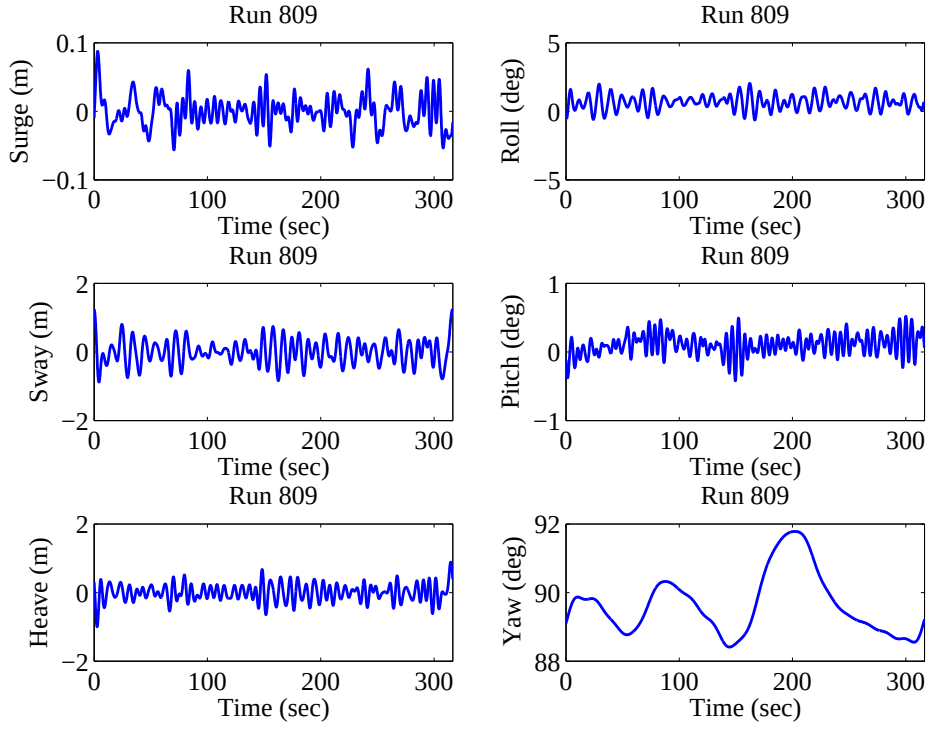


Figure 11: Run 809 - Motion time series for SS 3

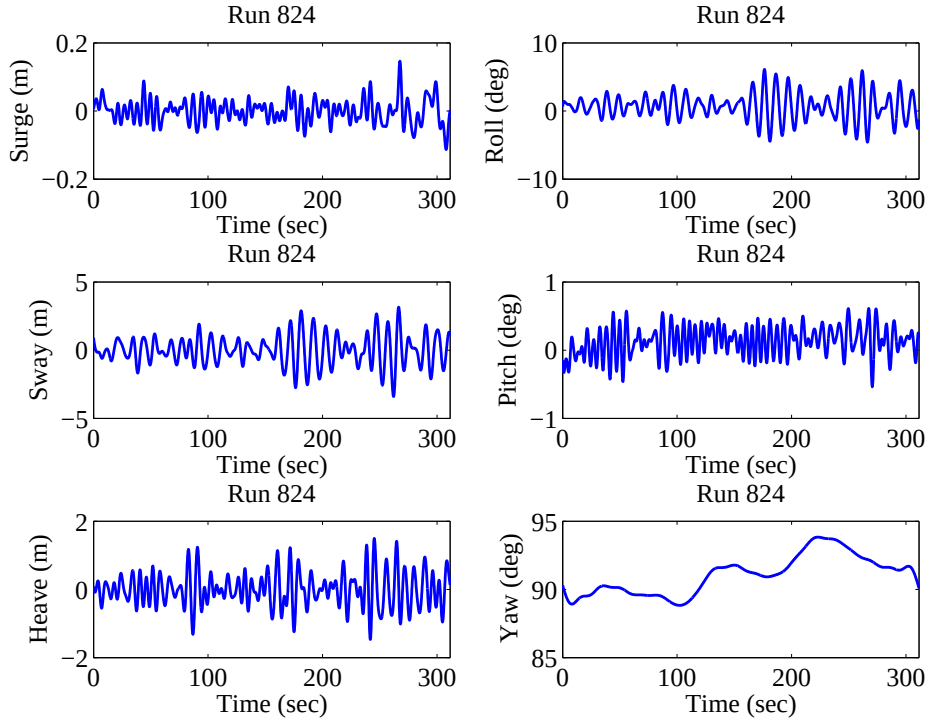


Figure 12: Run 824 - Motion time series for SS 4

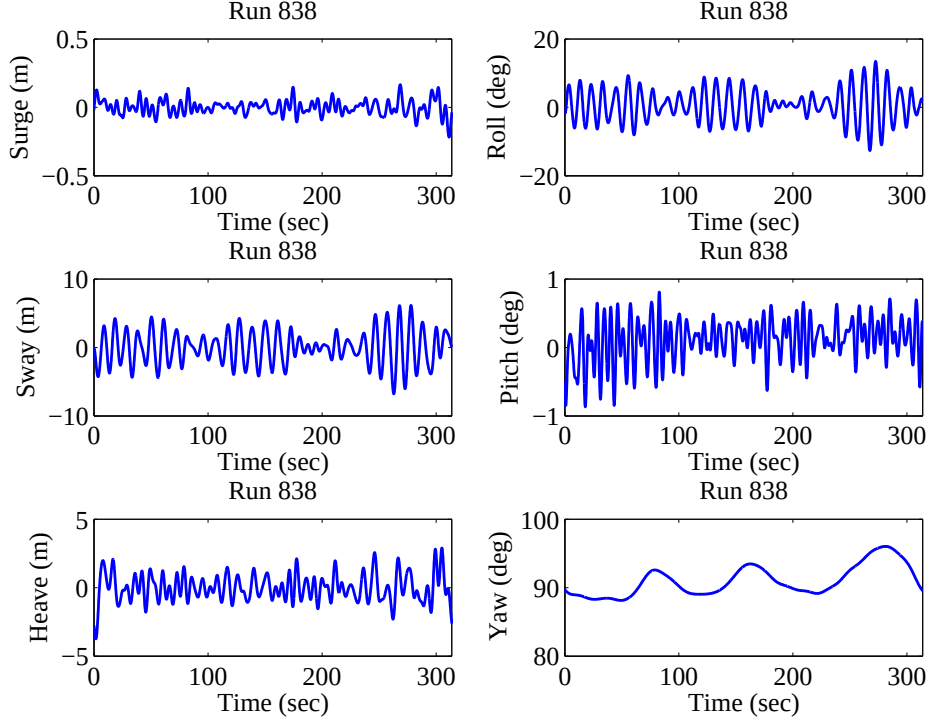


Figure 13: Run 838 - Motion time series for SS 5

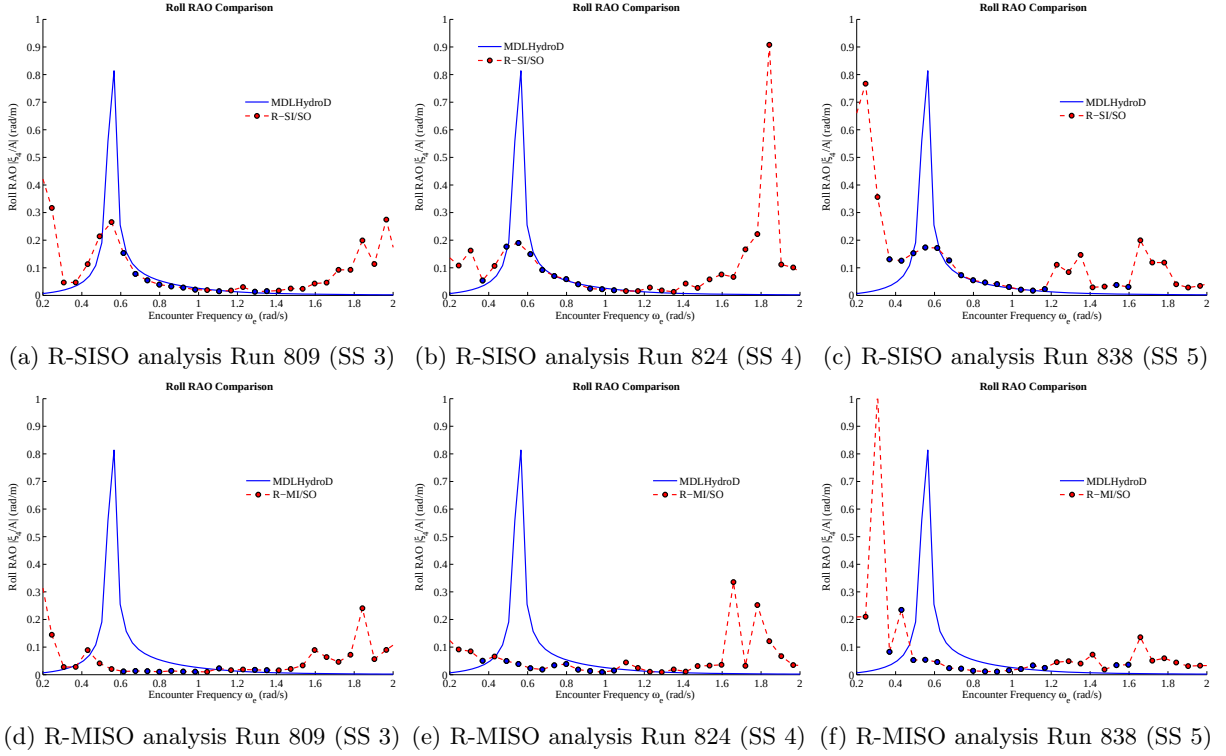


Figure 14: Roll RAOs for different sea states using R-MISO and R-SISO techniques

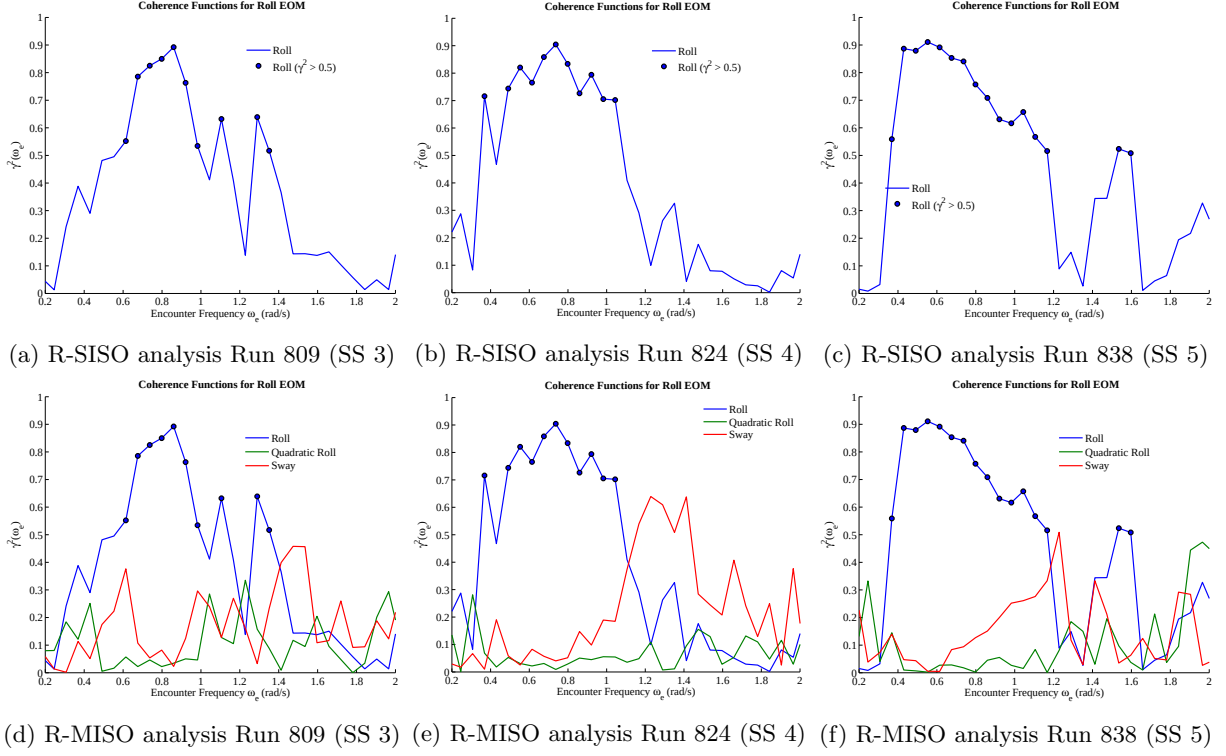


Figure 15: Partial coherence functions for different sea states using R-MISO and R-SISO techniques

$$RAO_4(\omega_e) = \frac{1}{H_{44}(\omega_e)} = \frac{1}{K_{44} - \omega_e^2(I_{44} + A_{44}(\omega_e)) + i\omega_e(B_{44}(\omega_e) + B_1(\omega_e))} \quad (31)$$

where K_{44} is the roll hydrostatic stiffness, I_{44} is the mass moment of inertia, $A_{44}(\omega_e)$ and $B_{44}(\omega_e)$ are the frequency dependent added mass and radiation damping and $B_1(\omega_e)$ is the frequency dependent linear viscous damping. In this scenario, $H_{44}(\omega_e)$ is obtained as the output of the R-SISO analysis whose inverse is plotted in Figure 14. Since the geometry of the vessel is known, it means that K_{44} , I_{44} , $A_{44}(\omega_e)$ and $B_{44}(\omega_e)$ are known quantities. Thus, (31) can be used to calculate the frequency dependent linear viscous damping.

A comparison of the mean total roll damping (viscous and radiation) obtained from 5 different experimental runs for each of the three sea state are shown in Figure 16a. The mean estimate of total damping from all data is shown in Figure 16b. Since the viscous damping only plays a significant effect on the roll dynamics around the resonant frequency, its estimate at the roll natural frequency will be the most reliable value.

In order to effectively compare the estimated damping against reality, a set of time domain simulations are performed and compared against the experimental time histories. The in-house developed time domain simulation program - SIMDYN [45, 42] is used to perform the simulations. The following 4 different damping models are simulated and compared against experimental time histories for Run 838 (SS 5):

1. **Free Decay Damping Model:** The viscous damping coefficient is obtained from the free decay analysis performed by Minnick et al [35] and is provided as an input to the simulation.
2. **Himeno Damping Model:** The viscous damping coefficient is obtained from empirical method prescribed by Himeno. The details of the estimation and implementation can be found in the works of Falzarano et al [2].

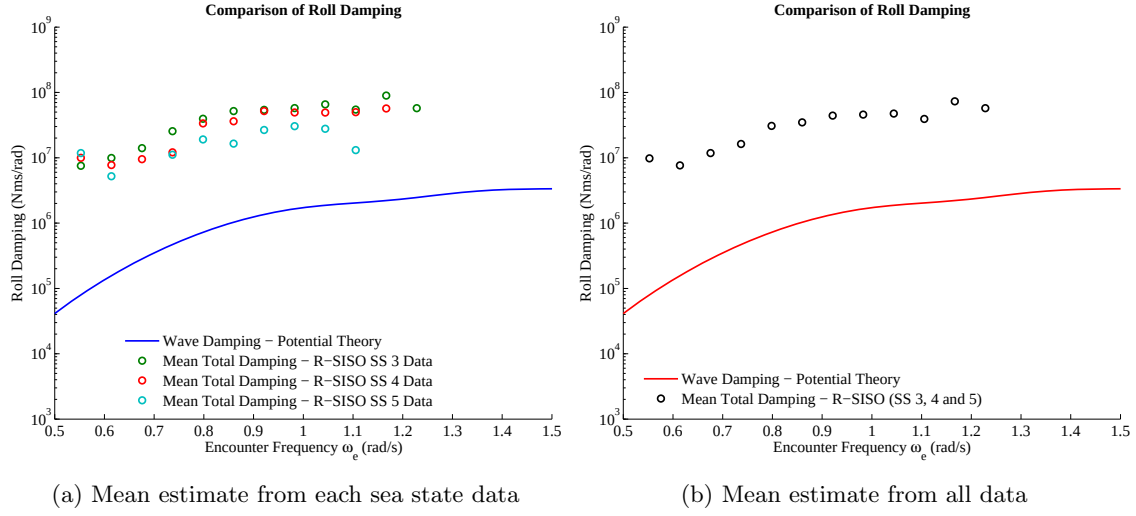


Figure 16: Estimated total roll damping coefficient from experimental data

Table 2: Comparison of simulated and experimental roll time histories of Run 838 (SS 5)

Damping Model	Normalized Cross Correlation Coefficient (%)	RMS Error (degrees)
No Viscous Damping Model	70.51	13.32
Himeno Damping Model	77.69	10.07
Free Decay Damping Model	86.70	7.60
R-SISO Improved Damping Model	93.07	5.84

3. **R-SISO Improved Damping Model:** The viscous damping coefficient is obtained from the R-SISO analysis as described above and is provided as an input to the simulation.
4. **No Viscous Damping Model:** No viscous damping model is included. The simulation only includes the radiation damping.

The comparison of the time histories from different models are shown in Figure 17. It can be seen that the free decay damping and R-SISO improved models are closer to the experimental time history as compared to the Himeno and no viscous damping models. In order to quantitatively compare the closeness of the simulated time history to the experimental time history, the normalized correlation coefficient is computed. The comparison of normalized cross-correlation coefficients and root mean square (RMS) error of all 4 damping models is shown in Table 2. It can be seen that the R-SISO improved model has a higher cross correlation with the experimental data as compared to the other models indicating its accuracy in capturing the viscous damping in irregular seas.

4. Conclusion

A detailed description of the theory of an advanced system identification tool based on Reverse-Multiple Input Single Output (R-MISO) technique has been provided. The paper also demonstrates the application of this tool to estimate the viscous roll damping from the irregular wave model tests of the vessel *R/V Melville*. An initial attempt to identify the quadratic roll damping shows that the effect of nonlinear viscous damping is insignificant in the model tests and the motions are fairly linear. In light of this information, further investigation is performed to extract the linear viscous roll damping in the system. The identified

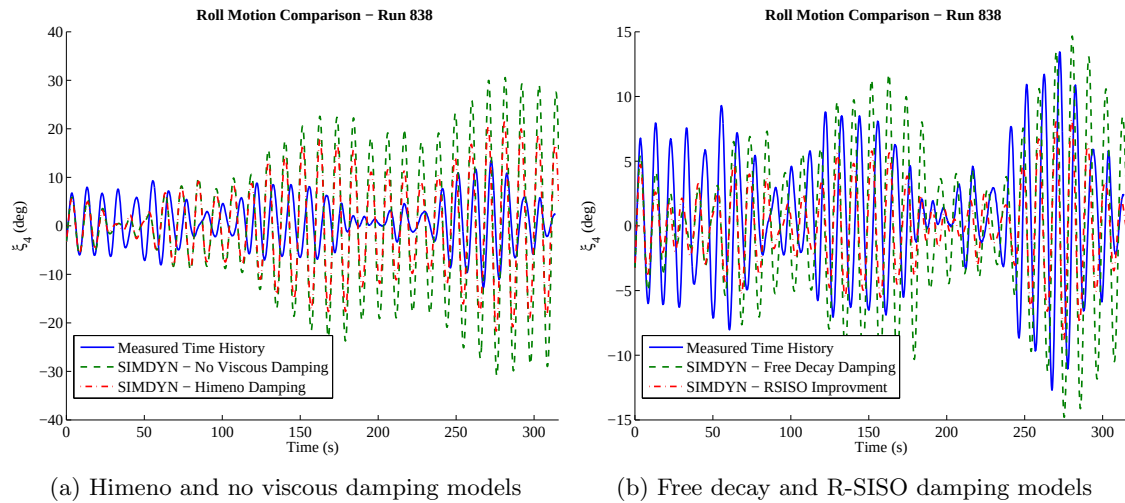


Figure 17: Comparison of different damping models with experiment

damping is used to produce accurate predictions of the ship's roll motion. It is also demonstrated that these roll motion predictions are not only more accurate than the potential flow predictions but more accurate than potential flow models corrected using either empirical prediction methods and even those corrected using roll damping obtained from free decay sailing experiments. This methodology has the potential to be used in significantly improving roll motion prediction during full scale sea trials of vessels in order to dramatically improve safety of critical operations such as helicopter landing or ship to ship cargo transfer.

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