Measurement and Prediction of Induced Large-Amplitude Oscillations of a Tumblehome Hullform

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ABSTRACT

Parametric roll caused by the periodic change in transverse stability that occurs as a ship encounters waves, and the prediction of the associated large-amplitude motions have proven to be difficult and include large uncertainty. Without sufficient damping, the motion can grow to very large amplitudes. One of the reasons for the difficulty in predicting parametric roll is the limitation in roll damping models. Additionally, there is a general lack of knowledge of resultant forces and moments for a ship at large roll angles. Recently, a study of the forces and motions induced by large amplitude roll motions was undertaken. This effort included both experimental measurement and Computational Fluid Dynamics (CFD) simulation and prediction of the forces, moments, and motions of a tumblehome hullform undergoing large amplitude roll oscillation. This work was performed over a range of roll amplitudes, up to 50 degrees, roll frequencies, and Froude numbers (0 to 0.4).

1.0 INTRODUCTION

In October 1998, the post-Panamax C11-class containership APL China experienced either loss or damage to two thirds of its deckload of about 1300 containers [1]. The ship was en route from Kaohsiung to Seattle when it encountered a violent storm. The ship's captain reduced speed and maneuvered to take the incoming waves on the ship's starboard bow, as his experience and training instructed him to. However, even in head seas, the ship experienced 35-40 degree rolls, well beyond the design limit of the container lashings. Subsequent model testing of the APL China at MARIN showed the model pitching and rolling very little when running in head seas. But once the vessel took a small roll to one side, which could be induced by small rudder motions or a wave from another direction, the roll motion would increase to 30 degrees in five roll cycles and then maintain this large amplitude motion, known as parametric roll.

Parametric roll is caused by the periodic change in transverse stability that occurs as a ship encounters waves. For parametric roll resonance to occur the change in transverse stability must occur at twice the natural roll period of the ship, and the roll damping must not be sufficient to dampen out the motion induced by the wave encounter. Without sufficient damping, the motion can then grow to very large amplitudes. The prediction of the large-amplitude motions, associated with parametric roll, has proven to be difficult and include large uncertainty. One of the reasons for this difficulty is the limitation in roll damping models. Currently, roll damping models are empirical, based on experimental data performed within a small range of roll amplitudes. Additionally, the lack of knowledge of resultant forces and moments for a ship at large roll angles has limited the validation and development of higher order computational fluid dynamics (CFD) codes. Recent efforts by the Office of Naval Research have led to a substantial increase in the efficacy of these tools in the prediction of ship motions while maneuvering in waves.

Tumblehome hullforms present an additional complexity. As opposed to a typical flared ship hullform, whose righting moment continues to increase as roll angle increases until the deck is submerged, the righting moment of a tumblehome hull may actually decrease at large roll angles, as shown in Figure 1. The ship in waves problem is a fully-coupled, nonlinear problem. To better understand the kinematics of large amplitude motions and capsize events and to aid in the development of state-of-the-art CFD codes, a series of model tests were performed with the model undergoing forced large amplitude roll motion. Additionally, two different CFD codes were used to model the flow field and the induced forces and motions.

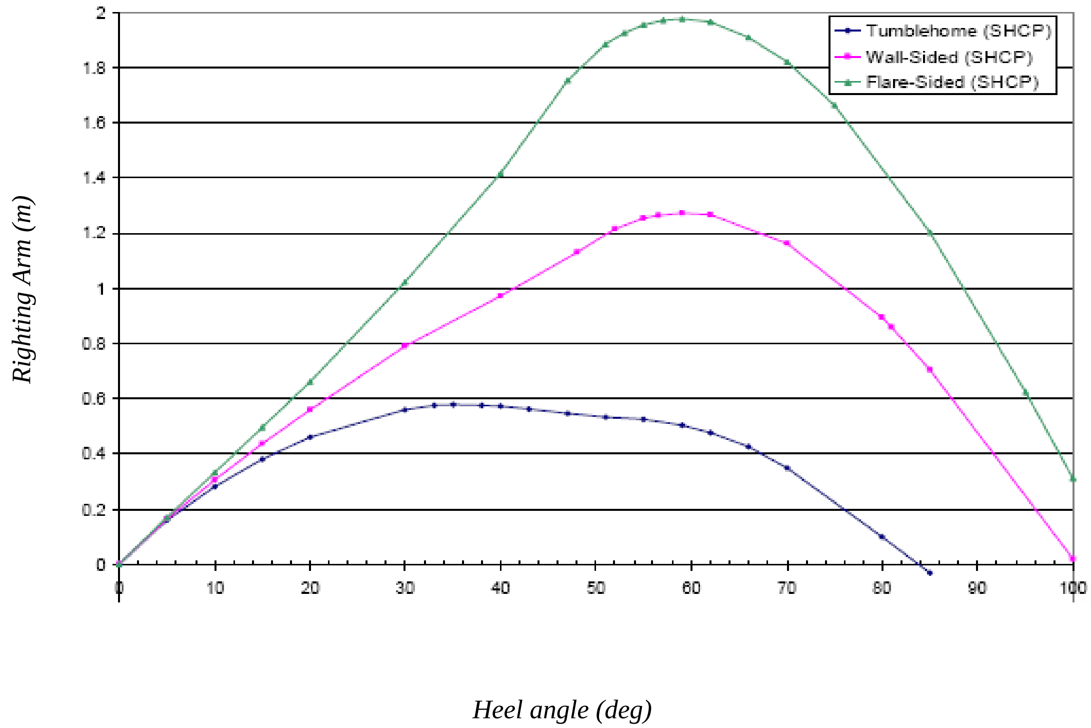


Figure 1: Righting arm curves for tumblehome, wall-sided and flared hull forms

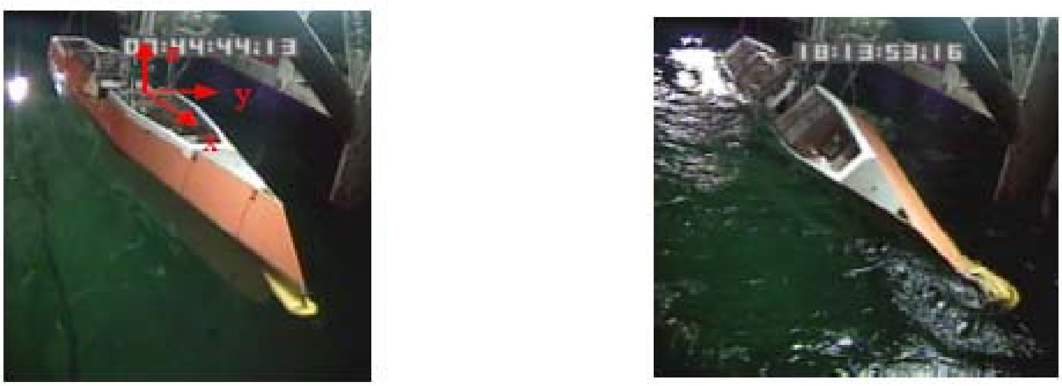
2.0 EXPERIMENT

2.1 Description

In 2005, the Naval Surface Warfare Center, Carderock Division (NSWCCD), tested NSWC Model 5613, a tumblehome hullform, with the primary objective of obtaining model-scale constrained seakeeping results to provide information necessary to perform verification of surge, sway, heave forces and motions, and roll, pitch, and yaw moments and motions acting on a surface combatant hull during large amplitude motions and capsize events [2]. A previously tested [3] modern surface combatant model (λ= 32, Model 5613) with 10-degree tumblehome sides was towed in the Deep Water Basin, Towing Carriage 2 at NSWCCD, and forced in roll using a motor-driven mechanism. The roll mechanism was used to oscillate the model through large roll amplitudes of up to 50 degrees to port and starboard while also controlling the roll frequency. Force, moment and motion data were recorded during the testing. The model was tested with and without bilge keels, at seven roll amplitudes ranging from 0 to 50 degrees, five roll periods centered about the natural roll period of 2 seconds, and four Froude numbers (based on length) ranging from 0 to 0.4. The specific values of the various test conditions are shown in Table 1. Additionally, data was acquired with the yaw motion both constrained and unconstrained. Figure 2 shows two views of the model during testing, where the figure on the left shows no roll angle and the figure on the right shows the model undergoing a 50-degree roll.

Table 1: Test Matrix

Roll Amp. (deg)	Roll Period (seconds)	Carriage Speed (Fn, (m/s))
0	1,1.5, 2, 2.5, 3	0 (0) ,0.15 (1.0), 0.25 (1.7), 0.4 (2.7)
5	1,1.5, 2, 2.5, 3	0 (0) ,0.15 (1.0), 0.25 (1.7), 0.4 (2.7)
10	1,1.5, 2, 2.5, 3	0 (0) ,0.15 (1.0), 0.25 (1.7), 0.4 (2.7)
20	1,1.5, 2, 2.5, 3	0 (0) ,0.15 (1.0), 0.25 (1.7), 0.4 (2.7)
30	1,1.5, 2, 2.5, 3	0 (0) ,0.15 (1.0), 0.25 (1.7), 0.4 (2.7)
45	1,1.5, 2, 2.5, 3	0 (0) ,0.15 (1.0), 0.25 (1.7), 0.4 (2.7)
50	1,1.5, 2, 2.5, 3	0 (0) ,0.15 (1.0), 0.25 (1.7), 0.4 (2.7)



(a) Model in zero roll position. **(b) Model in 50° roll position.**

Figure 2: Model during testing.

The model was also tested in a fixed condition, where the only allowed motion was the forced roll. The resultant forces were then measured. This testing was performed over a smaller range of roll amplitudes, as shown in Table 2.

Table 2: Fixed condition test matrix

Configuration	Froude Number (Fn)	ω (rads)	θ_a (deg)
BH+SK	0,0.1,0.2,0.3	0,1.18,2.17,3.81,4.83	5,15,30
BH+SK+BK	0,0.1,0.2,0.3	0,1.18,2.17,3.81,4.83	5,15,30

2.2 Model

The model used for this test was the 1/32 scale NSWC Model 5613. This model had 10 degree tumblehome sides. A rendered profile of Model 5613 is shown in Figure 3, and the waterline and body plan are shown in Figures 4 and 5, respectively. A summary of the hull design characteristics for the model are shown in Table 3. Due to the weight of the roll-forcing mechanism, the tested draft was 2.54 cm (1 in) greater than the design draft. The model was fitted with bilge keels of 1.25 m span (full scale), that were centered at midship; with a chord length equal to 1/3 the ship length. No other appendages were included in this test.

Table 3: Hull Characteristics

	Full-Scale (15C, SW)		1/32 Model-Scale (20C, FW)	
Lpp	154 m	505 ft	481 cm	15.8 ft (189.6 in)
Beam	18.8 m	61.7 ft	58.8 cm	1.93 ft (23.2 in)
L/B	8.2	8.2	8.2	8.2
Max. Depth	14.5 m	47.6 ft	45.3 cm	1.49 ft (17.8 in)
Max. Freeboard	9.00 m	29.5 ft	28.1 cm	0.92 ft (11.1 in)
Draft	5.50 m	18.0 ft	17.2 cm	0.56 ft (6.77 in)
Displacement	8790 tonnes	8650 LT	261 kg	575 Lbs
LCB (aft of FP)	79.6 m	261 ft	249 cm	8.16 ft
VCB (above BL)	3.26 m	10.7 ft	10.2 cm	0.33 ft (4.01 in)
KM_T	9.74 m	32.0 ft	30.4 cm	1.00 ft (12.0 in)

Figure 3: Profile of the Tumblehome Model 5613

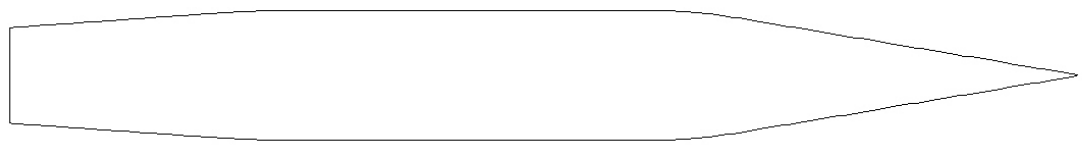


Figure 4: Waterline for Model 5613

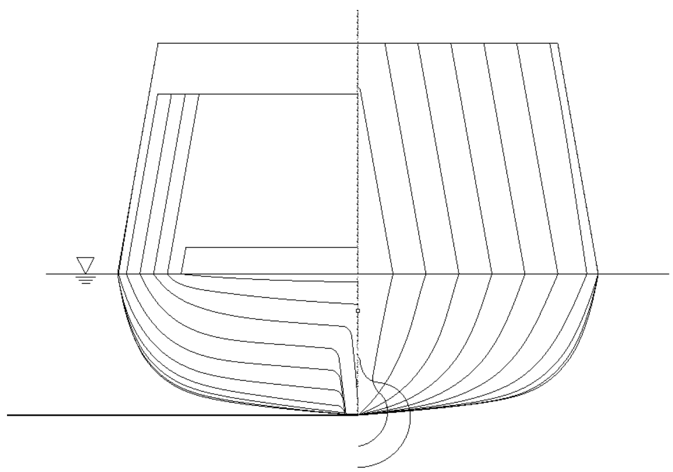


Figure 5: Body Plan of 10 Degrees Tumblehome Topside Design

2.2.1 Appendage Descriptions and Locations

The model was tested with bilge keels centered at midship that have a chord length equal to 1/3 the ship length and a span equivalent to 1.75 m full-scale. The bilge keel location can be seen in Figure 6.

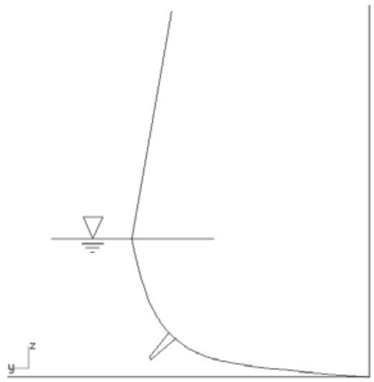


Figure 6: Station 10 Cut with Bilge Keel Cross-Section

2.3 Instrumentation

Model 5613 was fitted with a roll/pitch mechanism (Figure 7) that forced the model to a maximum roll angle of 50 degrees, while allowing the model to pitch freely to 25 degrees in both directions. The mechanism was located at the planned center of gravity of the model. A heave post allowed the model to move in heave, while a yaw mechanism (Figure 8) permitted the model to either be fixed in yaw or free to yaw up to 15 degrees. The section of the model containing the roll/pitch mechanism was separated from the rest of the model and filled with foam to prevent water from entering. The rest of the model sections were fitted with a Lexan cover to keep water out.

Three-component force and moment measurements were made using a Kistler force gage, which was mounted to the model interior underneath the roll/pitch mechanism, as shown in Figure 9. The Kistler gage was used to measure the forces and moments resulting from the constrained motions, including the sway force, the drag force, and the yaw moment for the fixed yaw configuration. The amplitudes and accelerations of the free motions (heave, pitch, and yaw for the free yaw configuration) were measured using a motion package assembled at NSWCCD. The motion package was mounted inside the hull forward and above the planned center of gravity; motions were corrected in post-processing. Standard frame rate (30 fps) video cameras were used to visually document ship motions from multiple views.

Figure 7: Roll/Pitch Mechanism

Figure 8: Yaw Mechanism

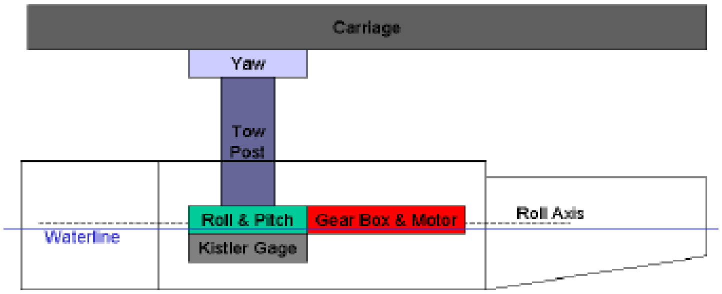


Figure 9: Schematic of Model Setup

2.4 Results

Figure 10 shows the roll motion (black dashed line) and the measured roll moment (red solid line) of the model with a Froude number of 0.25 (1.7 m/s, 3.3 kts) for the case of 5 degrees of roll (top panel), 30 degrees of roll (middle panel), and 50 degrees of roll (bottom panel), all with a 2 second period. The roll moment is closely in phase with the roll motion, and the magnitude of the moment increases with increased roll amplitude.

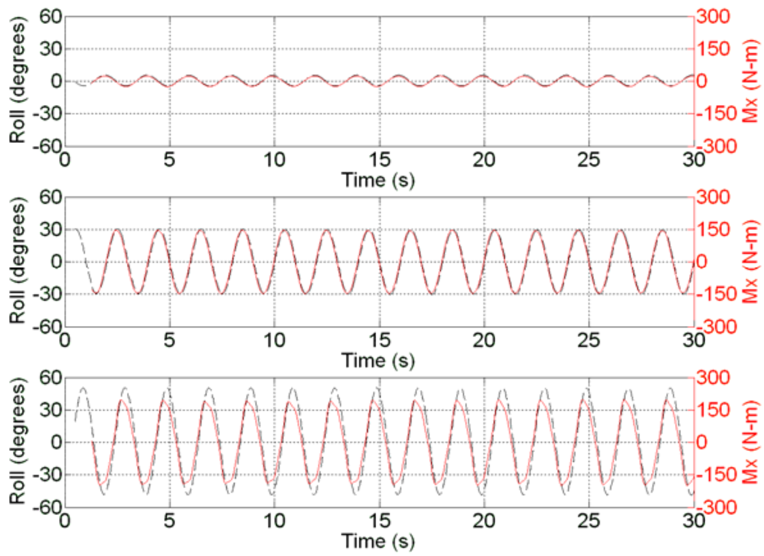


Figure 10: Roll Moment on Model for 5 degree roll (top panel), 30 degree roll (middle panel), and 50 degree roll (bottom panel) for a 2 second roll period at Fn of 0.25 (1.7 m/s, 3.3 kts).

Figure 11 shows the maximum roll moment for each roll amplitude, roll period and Froude number (obtained through harmonic analysis). The peak roll moment amplitude is lowest for the 1s roll period, and increases with increased roll period. The roll moment trend appears to be fairly linear below 30^{o} roll amplitude for all roll periods, except the one second period.

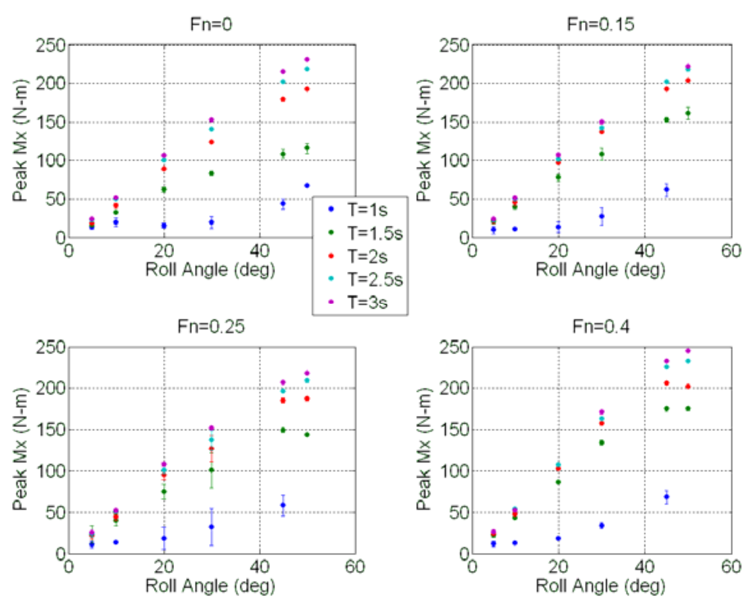


Figure 11: Maximum Roll Moment Amplitude

The equation for uncoupled roll motion is given in Equation 1 [4]:

$$(I + a_{44})\ddot{\phi} + b_{44}\dot{\phi} + c_{44}\phi = M(\phi) \qquad \text{(Eq. 1)}$$

where I is the mass moment of inertia of the model with respect to the x-axis, a_{44} is the added moment of inertia coefficient in roll, b_{44} is the roll damping coefficient, c_{44} is the roll damping coefficient, ϕ is the roll angle, and M is the amplitude of the forced moment. By performing a least squares fit of the measured data to this equation, the non-dimensional added inertia and roll damping coefficients can be computed.

3.0 PREDICTIONS

3.1 Computational Fluid Dynamics Codes

Two computational tools were used to simulate the large-amplitude force motions described above. These two CFD codes are Numerical Flow Analysis [5] and CFDShip-Iowa [6].

3.2 Numerical Flow Analysis

The Numerical Flow Analysis (NFA) code provides turnkey capabilities to model breaking waves around a ship, including both plunging and spilling breaking waves, the formation of spray, and the entrainment of air. NFA uses a Cartesian-grid formulation with immersed body and volume-of-fluid (VOF) methods. The governing equations are formulated on a Cartesian grid thereby eliminating complications associated with body-fitted grids. The sole geometric input into NFA is a surface panelization of the ship hull. Dommermuth, et al [5] describes the code and recent applications to naval problems. The forces, moments, and motions of Model 5613 for some of the conditions tested, were computed using NFA and more standard RANS codes. Figure 12 shows the visualization of a typical NFA result for the model undergoing large amplitude roll oscillations. Animations of this simulation and others are available at http://www.saic.com/nfa.

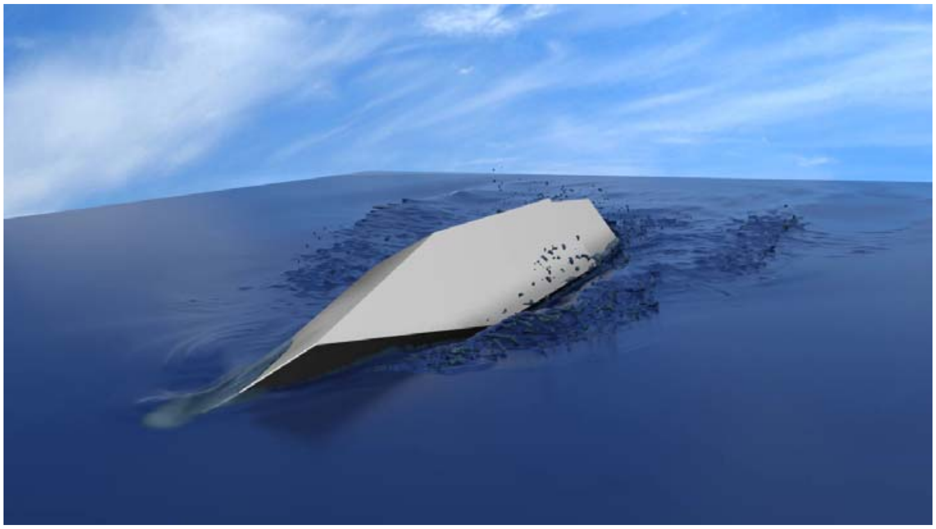


Figure 12: Visualization of the simulation of Model 5613 undergoing 65° forced roll.

Figure 13 shows forced roll results for the force in pitch. The Froude number is 0.3, and the roll amplitude and frequency are respectively 30° and 0.672 rad/sec. The skeg is included, but the bilge keels are not included. All motions except roll are constrained. Results shown are full scale. Two experimental results are compared to the numerical predictions. The phase difference in the experimental results is due to registration issues. The numerical predictions start the roll motions from rest and ramp up to the full amplitude of motion. This start up process is evident in the beginning for times less than 10 seconds. Even though the steady-state condition is not quite met in the numerical predictions, the predicted amplitude of the moment agrees with experiments.

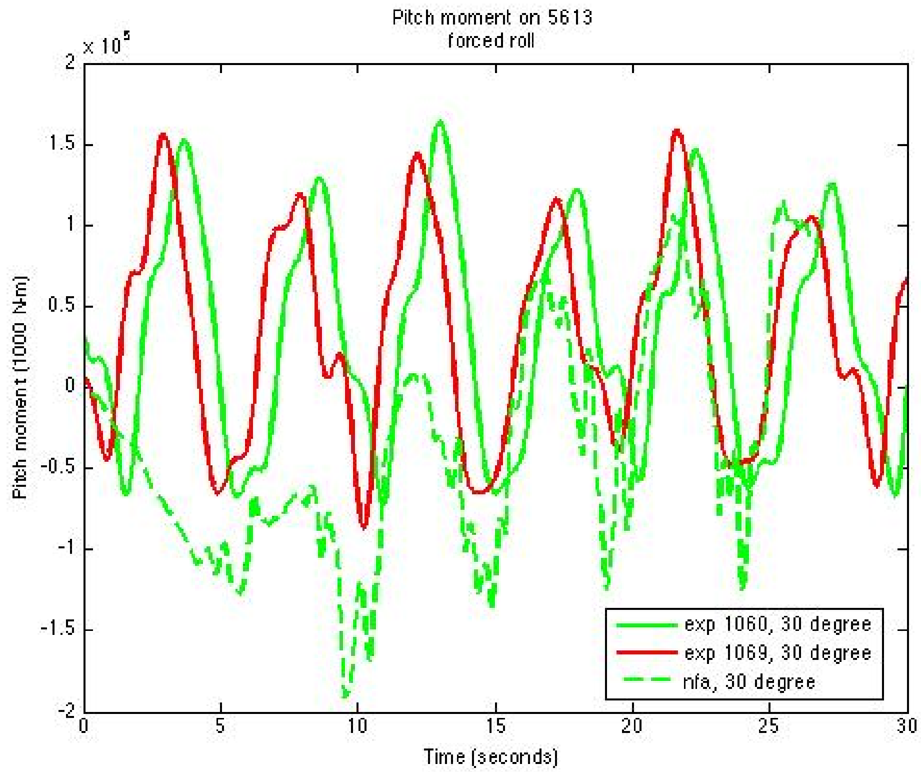


Figure 13: Force in pitch due to 30° forced roll.

3.4 Reynolds Averaged Navier-Stokes

NSWCCD and the University of Iowa have been working together for many years on the development of the CFD code CFDShip-Iowa [6]. While several RANS codes are currently in use at NSWCCD, CFDShip-Iowa has become the main code for surface ship calculations, most recently being used on the Office of Naval Research High-Speed Sea-Lift (HSSL) program [7].

The code is a Reynolds average Navier-Stokes code with overset gridding capability for realistic hull forms and motions and the ability to move the ship hull as part of the solution. This motion can be unsteady, prescribed, and/or predicted motion. The code utilizes free surface capturing methods and some attempt has been made to verify its capabilities in predicting roll motions. In 2002, Miller and Gorski [8] predicted the roll motion of a circular cylinder with bilge keels and showed good comparison to the forced motion cases.

Figures 14 and 15 show the CFDShip-Iowa forced-roll results. The Froude number is 0.3, and the roll amplitude ranges form 0 to 30 degrees. The roll frequency for the case shown is 0.0 rad/sec. The skeg is included, but the bilge keels are not included. All motions except roll are constrained. Results of the comparison of measured and calculated horizontal force on tumblehome hull NSWC Model 5613 are shown in Figure 14, and the vertical force in Figure 15. Figure 16 shows a visualization of the wave field and vorticity field at t=220, 236, and 244.

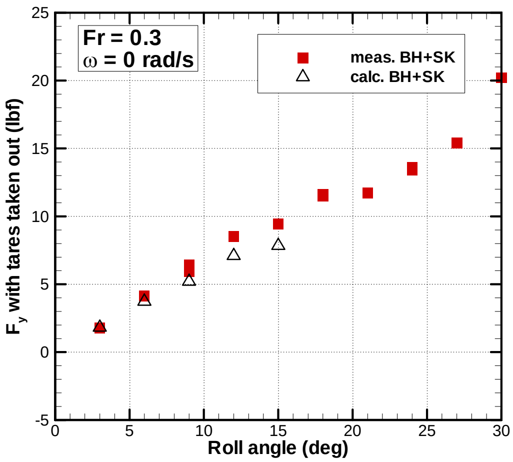


Figure 14: Comparison of the measured and predicted horizontal force on NSWC Model 5613 undergoing forced roll.

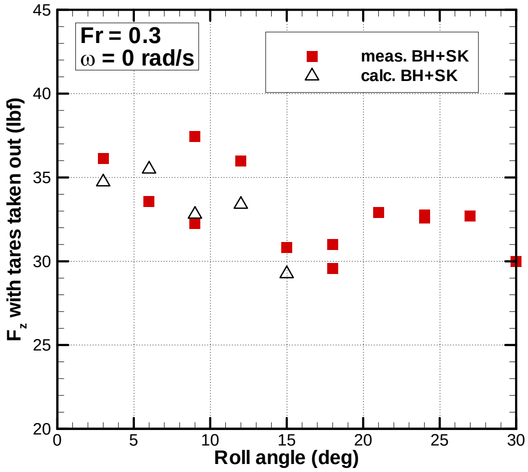
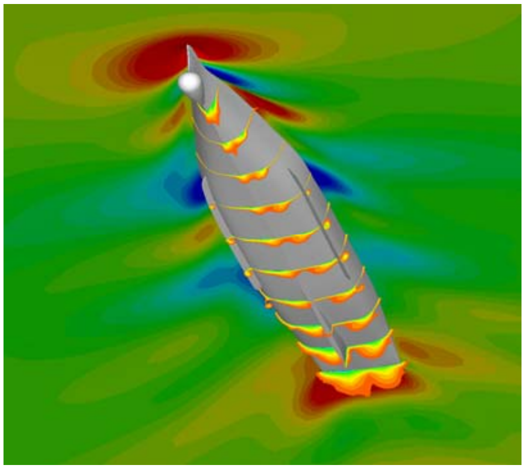
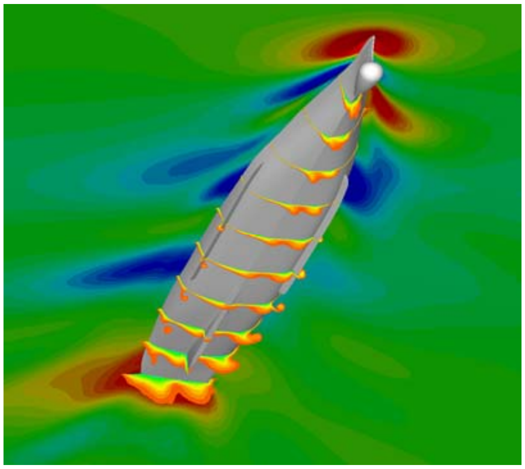


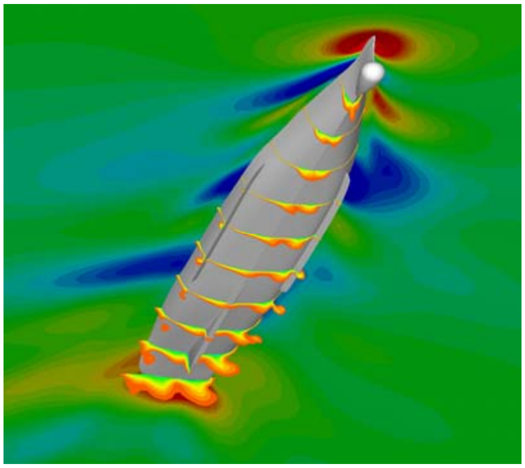
Figure 15: Comparison of the measured and predicted vertical force on NSWC Model 5613 undergoing forced roll.



a) Time t=220



b) Time =236



c) Time = 244

Figure 16: Visualization of the wave field and vorticity field at t = 220, 236, and 244.

4.0 SUMMARY

Though the motions induced from large amplitude roll oscillations are not as complex as full parametric roll, the ability to accurately predict the forces, moments and induced motions and to understand the physics is a necessary, simpler first step in modelling a ship undergoing parametric roll. A detailed set of measurements of the forces, moments and motions of a tumblehome hullform undergoing forced roll was performed at NSWCCD in 2005. This data was compared to CFD calculations made utilizing two higher order CFD codes, NFA and CFDShip-Iowa. Overall the comparison of the predicted and measured forces shows good agreement.

5.0 ACKNOWLEDGEMENTS

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