The Importance of Damping in Accurately Predicting Ship Capsizing in Random Waves

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ABSTRACT

Although heave and pitch motion are well predicted using linear ideal flow theory ship motions computer programs, roll motion is not. This is because roll damping is generally small and the ideal flow part is a small part of that. (see Faltinsen, 1990). Accurately predicting the roll damping is the key to accurately predicting the roll motion. The most common methods are to use 1) empirical data mostly from Japan based upon exclusive series model test (Ikeda, 2002) or 2) Free decay model tests of vessel to determine damping. In addition recent progress has also occurred in predicting ship roll damping using computational fluid dynamics (CFD) (e.g., Korpus & Falzarano, 1997). The Reynolds Averaged Navier – Stokes CFD alternative has the potential to accurately predict full-scale roll damping, but is completely dependent upon the turbulence modeling. In this paper we suggest a simple method to at least approximately extrapolate roll damping to full scale. Our approximate method borrows Froude hypothesis from ship resistance testing to separately scale friction and residuary damping (wave & eddy) using Reynolds and Froude scaling respectively.

Keywords: Dynamical Systems Analysis, Nonlinear Ship Rolling, Random waves, Capsizing, Roll Damping.

INTRODUCTION

In order to investigate the importance of damping, the large amplitude nonlinear rolling motion and capsizing behavior of an offshore supply vessel hull-form is analyzed. A dynamical perturbation technique (Vishnubhotla, Falzarano and Vakakis, 1998) is applied to the hull using various approximations to the roll damping. The offshore supply vessel (OSV) is probably one of the most common seagoing hull forms. The specific offshore supply vessel hull-form we study herein is designed and built by Tacoma Boat and Halter Marine and considered to be one of the best hull-forms for motion. This small vessel was required to operate in the severe waters of the North Atlantic during all weather conditions. The severity of the North Atlantic environment has suggested that a new analysis methodology be considered in lieu of traditional ship static stability analysis. This is due to the occurrence of extreme wind and waves and the resulting large amplitude dynamics response. As a result of non-linearities inherent in extreme response, a dynamics based analysis procedure must be used

In this work, our previously developed dynamical perturbation technique (Vishnubhotla, Falzarano and Vakakis, 1998) is applied to study the large amplitude nonlinear rolling motion of an offshore supply vessel (OSV). This approach makes use of a closed form analytic solution which is exact up to the first order of randomness, and takes into account the unperturbed (no forcing or damping) global dynamics. The result of this is that very large amplitude nonlinear vessel motion in a random seaway can be analyzed with similar techniques used to analyze nonlinear vessel motions in a regular (periodic) seaway. The practical result is that dynamic capsizing studies can be undertaken considering the true randomness of the design seaway. The capsize risk associated with operation in a given sea spectra can be evaluated during the design stage or when an operating area change is being considered. Moreover, this technique can also be used to guide physical model tests or computer simulation studies to focus on critical vessel and environmental conditions which may result in dangerously large motion amplitudes.

BACKGROUND

In this paper, our dynamical perturbation technique is applied to an offshore supply vessel hull-form. This specific hull that is being studied has been previously studied extensively at the UNO Marine Dynamics Lab in the past with regards to several aspects. These include coupled six degree of freedom nonlinear ship rolling motion (Taz Ul Mulk, and Falzarano, 1994), combined steady-state and transient single degree of freedom analysis (Falzarano, et al, 1995) and most recently the effect of water-on-deck (Falzarano, et al, 2002). In Falzarano, et al (1995) we found the importance in damping in not only quantitatively predicting the roll amplitude but we also found that the amount of damping may also qualitatively alter character of the roll magnification curves and determine weather or not the magnification curve has multiple steady state solution.

Analysis of non-linear ship and platform rolling motion using dynamical systems approaches have become common (Falzarano, et al, 1992). However, most studies are limited to single degree of freedom and regular wave (periodic) excitation with few exceptions. It is well known that roll cannot always be decoupled from the other degrees of freedom but more importantly it is well known that sea waves are not regular but in fact are random. It is common in the design of ships and offshore platforms to make narrow banded assumptions and predict extremes using the Rayleigh Probability Density Function (PDF). However when critical capsizing motions are involved the response is not at all linear but highly non-linear. In this study, we model the highly non-linear near-capsizing behavior of an offshore supply vessel (OSV) in a random seaway by using an analytical solution to the differential equation. The availability of such a closed form solution allows us to generate the safe basin boundary curves. An alternative approach is to solely use model testing or time domain simulation.

However, such tank testing and computer simulation techniques are expensive and computer time intensive so an alternative is the application of analytic phase plane techniques. The dynamical systems techniques can be used in isolation or in conjunction with these other methodologies. The basis of the analytic phase plane technique is the identification of critical dynamics in terms of the important ship and environmental characteristics. Identification of such critical dynamics can provide a more focused model test or numerical simulation program. Moreover, recent advances in the dynamical systems techniques have allowed for the consideration of random excitation.

PHYSICAL SYSTEM MODELING

The focus of this study is on highly non-linear rolling motion possibly leading to capsizing and the effect of damping and sea state on the safe basin. Roll is in general coupled to the other degrees of freedom, however under certain circumstances it is possible to approximately de-couple roll from the other degrees of freedom and to consider it in isolation. This allows us to focus on the critical roll dynamics. The de-coupling is most valid for vessels which are approximately fore aft symmetric; this eliminates the yaw coupling. Moreover by choosing an appropriate roll-center coordinate system, the sway is approximately de-coupled from the roll. For ships, it has been shown in previous studies that even if the yaw and sway coupling are included the results differ only in a quantitative sense. The yaw and sway act as passive coordinates and do not qualitatively affect the roll (Zhang & Falzarano, 1993).

The other issue is the modeling of the fluid forces acting on the hull. Generally speaking the fluid forces are subdivided into excitations and reactions. The wave exciting force is composed of a part due to incident waves and another due to the diffracted waves. These are strongly a function of the wavelength / frequency (In the current paper the function is derived as a curve fit). The reactive forces are composed of hydrostatic (restoring) and hydrodynamic reactions. The hydrostatics are most strongly non-linear and are calculated using a hydrostatics computer program. In order that the zeroth order solutions are expressed in terms of known analytic functions, the restoring moment curve needs to be fit by a cubic polynomial. The hydrodynamic part of the reactive force is that due to the so called the radiated wave force. The radiated wave force is subdivided into added mass (inertia) and radiated wave damping. These two forces are also strongly a function of frequency. However since the radiated wave damping is light, and for simplicity, constant values at a fixed frequency are assumed. Generally, an empirically determined nonlinear viscous damping term is included. The focus of this paper is the effect of the additional non-ideal flow damping on the motion and this is discussed in the next section. The resulting equation of motion is:

The focus of this study is non-linear vessel rolling motion in a realistic wave excitation due to a random seaway. The sea spectral model used for the waves is the Pierson-Moskowitz (PM) Sea State descriptions, which was originally developed for a fully developed seaway in the North Atlantic. The PM model is used because it corresponds to a typical random seaway encountered in the North Atlantic winter. The effect of seaway intensity is considered. The sea spectra is multiplied by a roll moment excitation Response Amplitude Operator (RAO)(See Figure 1) squared in order to yield a roll moment excitation spectra (Equation 2a). The roll response RAO calculated was approximated by the smooth curve depicted in Figure 1a. The resulting sea spectra are decomposed into periodic components with random phase angles. The time dependent forcing function would then assume the form shown in Equation 2b.

$$S_R^+(\omega) = |RAO|^2 \; S^+(\omega) \qquad (2a)$$

$$F(t) = \sum_{i=1}^{N} F_M(\omega_i) \mathrm{Cos}(\omega_i t + \gamma_i) \qquad (2b)$$

where,

$$F_M(\omega_i) = \sqrt{2\, S_R^+(\omega_i)\, \Delta\omega} \qquad (2c)$$

Figures 1b and 1c show the excitation spectra and the corresponding time dependent force (in non-dimensional form) for a Pierson-Moskowitz sea spectra sea state 3. Figure 1d is the response spectra are for Pierson-Moskowitz Sea State 7.

Roll Damping Prediction from Free-Decay Tests

It is well known that linear ideal flow ship motions theory works well for heave and pitch motion but roll is not well predicted using a purely ideal flow hydrodynamic model. Ship roll motion is not as well predicted as heave and pitch because predicting roll damping is not as simple. This is because the ideal flow roll damping may be a small part of the total damping. The additional damping is dominated by rotational and viscous effects.

In order to estimate the model scale roll damping for this model, several roll free decay tests were performed in the UNO Tow Tank. The model testing was performed as part of a student laboratory during the Fall semester 2003. The primary purpose of that lab was to compare head seas heave and pitch linear ship motions theory computer results to model tests data. As expected the comparison between head seas computer prediction and model test results were quite good in both regular waves and in a random seaway. Although beam a sea testing in waves was not performed as part of this series the model was sallied in order to estimate the model's real fluid flow roll damping coefficients. The tests were performed using a six degree of freedom inertial package by Cross-Bow (three accelerometers and three rate gyros). This pack is extremely convenient for student laboratories since it has associated with it additional hardware and software which allows the motion to be directly displayed on a lap-top computer.

In order to estimate the linear and nonlinear damping we use two methods 1) from Faltinsen (1990) and 2) from Blagoveshchensky (1962). Each method allows both the linear and quadratic damping coefficient to be estimated. For a roll equation of motion in the following form,

$$\ddot{x} + p_1\dot{x} + p_2\dot{x}\,|\,\dot{x}\,| + p_3 x = 0$$

The linear p_1 and nonlinear p_2 roll damping coefficients can be estimated by comparing successive peaks X_n and knowing the damped natural period T_m.

Figure 2a shows a typical roll decay time history and Figure 2b and 2c show the results plotted in order to estimate the damping coefficients. The two methods give very similar estimates of damping coefficients. However, Blagoveshchensky's method shows better convergence during the curve fitting process. We use an empirical method based upon Himeno (1981) to predict the non ideal flow model roll damping and by adding a small bilge keel representing the chine we are able to exactly match the experimental results.

What was done to extrapolate the full-scale roll damping was to essentially use the Froude's hypothesis from ship resistance model testing. To start, we take away from the model roll damping the model damping due to skin friction (Chakrabarti, 2001). Next we scale the remainder (residuary) using Froude scaling and then add back to the scaled residuary damping the full scale skin friction. Using this method, we found the damping ratio for the full scale to be smaller than the model damping ratio. This method may or may not be accurate and we plan to validate it using a full scale sallying experiment and/or using Reynolds-Averaged

Navier-Stokes (RANS) computational fluid dynamics (CFD), see e.g. Korpus and Falzarano, (1997).

THE DYNAMICAL PERTURBATION METHOD

The focus of this investigation is to study the nonlinear dynamics of an offshore supply vessel (OSV) due to pseudo-random wave excitation. Considering that random excitation is a realistic model for ship and platform motions at sea, we developed our method to consider the case of random wave excitation as approximated by a finite summation of regular (periodic) wave components and determined the critical basin boundary curves.

The solution to equations such as Equation (1) with softening characteristics exhibit two greatly different types of motions depending upon the amplitude of the forcing. For small forcing amplitude, the first type of motion is an oscillatory motion which is bounded and well-behaved. For large amplitudes of forcing, the motion can be such that a uni-directional rotation occurs. The boundary between these two types of motions is called in the terminology of nonlinear vibrations, the separatrix. This curve literally separates the two qualitatively different motions. In the language of nonlinear dynamical systems, these curves are called the (upper and lower) saddle connections. The saddles are connected as long as no damping and forcing are considered in the system. Once damping is added to the system, the saddle connection breaks into stable and unstable manifolds. The stable manifolds are most important because they form the basin boundary between initial conditions which remain bounded and those that become unbounded. When periodic forcing is added to the system, these manifolds oscillate periodically with time and return to their initial configuration after one period of the forcing. This forcing period is chosen for the Poincaré sampling time of such a periodic system.

In this investigation, the random wave forcing is approximated by a summation of periodic components with random relative phase angles. Although this representation approximates the true random excitation as N64, and Δω60, for finite N this does not occur. Actually, the "random" signal repeats itself after $T_R = 2\pi/\Delta\omega$. Another relevant time period is the average or zero crossing period T_o. Assuming the wave excitation spectrum is narrow banded this might also be a good reference period for a Poincaré map. In lieu of Poincaré maps, we choose to trace out single solution paths which are contained in the stable manifolds. These are then projected onto the phase plane.

The critical solutions lying in the stable manifolds are calculated using our approach. This method is a perturbation method which begins with the undamped and unforced separatrix which for a softening spring is known in closed form, i.e.

$$\ddot{x} + x - k\,x^3 = 0 \tag{3}$$

$$x(\tau) = \frac{1}{\sqrt{k}}\mathrm{Tanh}\left(\frac{\tau - \tau_o}{\sqrt{2}}\right) \tag{4a}$$

$$\dot{x}(\tau) = \frac{1}{\sqrt{2k}}Sech^2\left(\frac{\tau - \tau_0}{\sqrt{2}}\right) \tag{4b}$$

The first order solution is determined by using the method of variation of parameters. The original Equation (1), is scaled into the following form,

$$\ddot{x} + x - kx^3 = \varepsilon(-\gamma\dot{x} - \gamma_q\dot{x}\,|\,\dot{x}\,| + F(t) \tag{5}$$

Having scaled the original equation, the solution method basically involves expanding the solution in a perturbation series as,

$$x(t) = x_0(t) + \varepsilon\,x_1(t)... \tag{6}$$

The second order equation to be solved is actually a linear equation with time varying coefficients. The coefficients are obtained from the zeroth order solution known from Equations (3) and (4) i.e.,

$$\ddot{x}_1 + x_1 - 3k\,x_1 x_0^2 = \hat{F}(x_0, t) \tag{7}$$

Solution to the zeroth and first order solution terms yields the perturbed manifolds which are the boundary between the bounded and unbounded motions. This method explicitly determines the critical solutions which separate the bounded steady state oscillatory motions from the unbounded motions. These solutions are determined by solving equations (3) and (5) and using them in (6).

The approach taken in this paper although different from our previous analysis is similar enough that all the details need not be completely repeated herein. The basin boundaries correspond to the stable manifolds associated with the positive and negative angles of vanishing stability and are just the damped and forced extensions to the upper and lower separatrices respectively which were previously discussed. These stable manifolds form the basin boundary between bounded (safe, non-capsize) and unbounded (capsizing) solutions.

Although this method was originally developed by Vakakis (1993) to study intersections of stable and unstable manifolds for equations for which the Melnikov method could not be used, this method is applied herein because it is general enough to yield exact solutions to general equations such as the multiple frequency forcing case being studied herein.

RESULTS

The results herein are for parameters representing this typical offshore supply vessel (OSV) rolling in beam seas in a mild and severe Pierson Moskowitz sea spectrum. As can be seen, when the seaway intensity increases, the vessel's dynamics may change qualitatively. The upper stable manifolds distance to the roll axis changes as the intensity of the seaway increases. As the wave amplitude increases the magnitude of the unstable periodic orbit in the neighborhood of the angle of vanishing stability increases.

The results are application of the dynamical perturbation technique to the hull form with the following hydrodynamic models as follows: 1) linear hydrodynamic coefficients based upon ideal flow theory alone, 2) roll damping coefficients modified using empirical prediction (Himeno, 1981), 3) roll damping coefficients obtained from free-decay model tests performed at the UNO Tow Tank 4) roll damping coefficients extrapolated to full scale.

The emphasis of this paper is on comparing the results of the offshore supply vessel (OSV)'s dynamics in various intensity sea states without and with viscous damping and with and without bilge keels.

Figures 3a and 3b are the safe basin projections for the vessel with linear potential flow damping exposed to a Pierson Moskowitz sea state 3 and 7, respectively. Figures 4a and 4b are the safe basin projections for the vessel with linear potential flow damping supplemented by empirical prediction of the nonlinear viscous and rotational damping exposed to a Pierson Moskowitz sea state 3 and 7, respectively. Figures 5a and 5b are the safe basin projections for the vessel with roll damping as predicted by the free-decay model test exposed to a Pierson Moskowitz sea state 3 and 7, respectively. Figures 6a and 6b are the safe basin projections for the vessel with roll damping as predicted by the free-decay tests and extrapolated to full scale as described previously exposed to a Pierson Moskowitz sea state 3 and 7, respectively. The final figures, Figures 7a and7b are the extended phase planes of the solution curves for case 2 for the vessel exposed to a Pierson Moskowitz sea state 3 and 7, respectively.

CONCLUSIONS

The method utilized herein is quite powerful and capable of handling rather general systems. Although the application herein required that the zeroth order solution to be known in closed form; this is not a requirement and actually it could be known numerically.

The results demonstrate the effect of random external excitation and the amount of nonlinear roll damping on the global nonlinear roll dynamics of this vessel. The sensitivity of the response to seaway intensity and the amount of damping is dramatic. The amount of damping and in our case the value of the predicted damping affects the size of the safe basin. Moreover, the intensity of the seaway also affects the size of the safe basin. It should be noted that the results given in the figures are non-dimensional, using the non-dimensionalization implied by equation (5).

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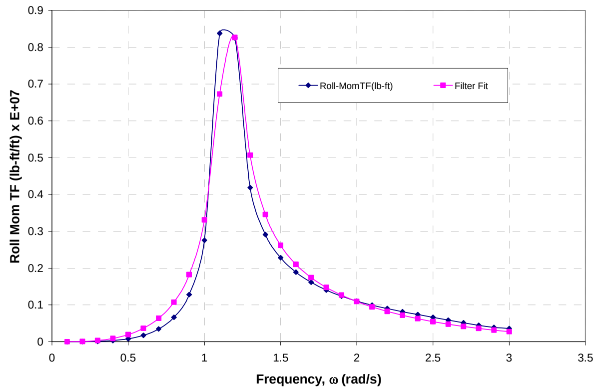

Fig 1a *OSV* Roll Moment Excitation Transfer Function

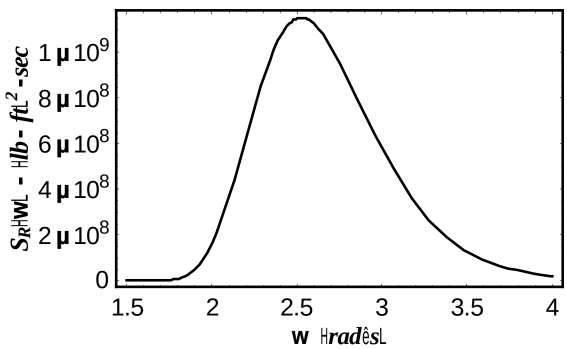

Fig 1b. *OSV* Roll Moment Excitation Spectra for Wind Speed, $U_W = 9\ fts^{-1}$ (Sea State 3)

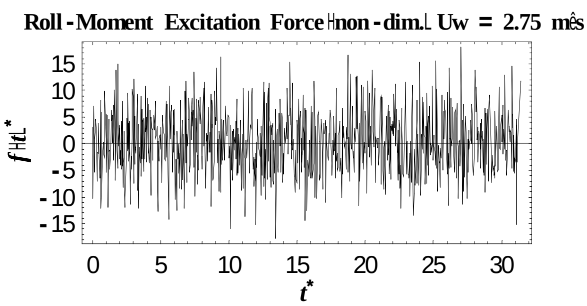

Fig 1c. *OSV* Corresponding Roll Moment Excitation Time History (non-dim), $U_W = 9\ fts^{-1}$

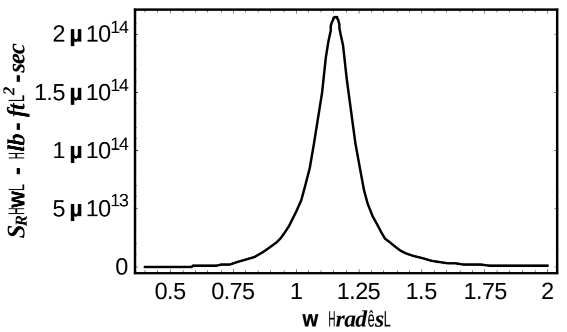

Fig 1d *OSV* Large Amplitude Roll Moment Excitation Spectra, U_W = 32.8 fts^{-1} (Sea State 7)

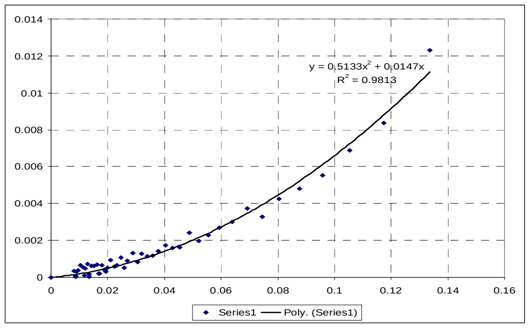

Fig 2c Analysis of Free Decay Test (Blagoveshchensky, 1962)

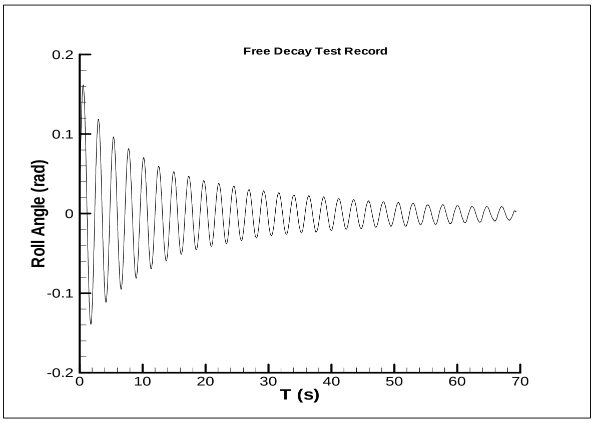

Figure 2a Typical Free-Decay Test

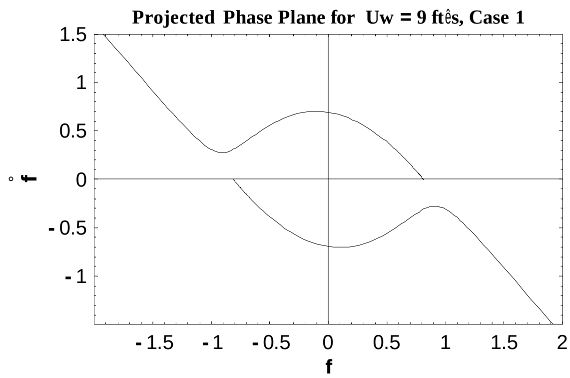

Fig 3a. *OSV* Projected Phase Plane for P-M Spectra, U_W = 9 fts^{-1} , Rolldamping – Linear (radiation, ideal flow)

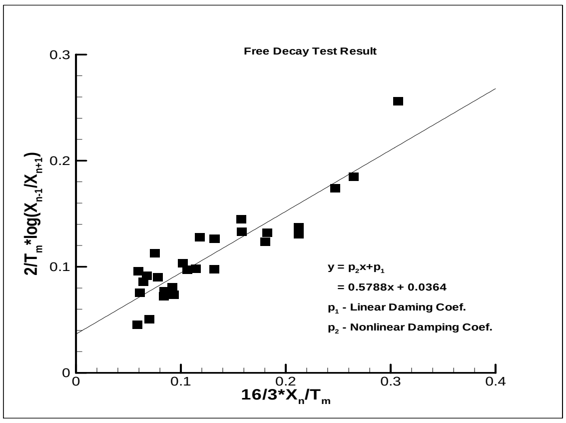

Figure 2b Analysis of Free Decay Test (Faltinsen, 1990)

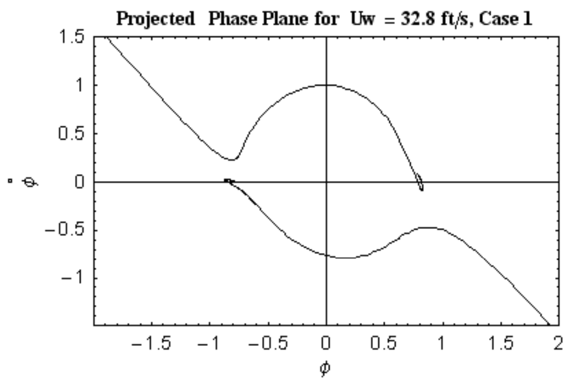

Fig 3b. *OSV* Projected Phase Plane for P-M Spectra, U_W = 32.8 fts^{-1} : Roll damping – Linear (radiation, ideal flow)

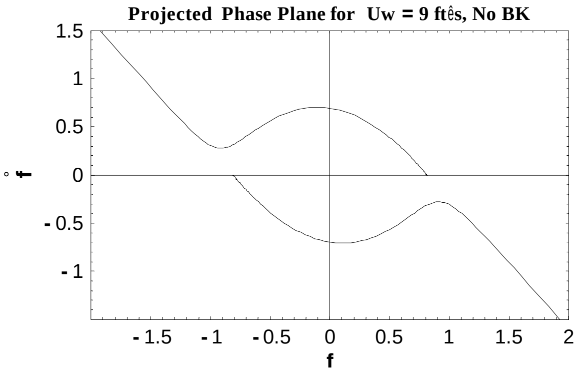

Fig 4a. *OSV* Projected Phase Plane for P-M Spectra, U_W = 9 fts^{-1} , No Bilge Keels, Prediction of total damping – Linear (radiation, friction) + Non-linear (viscous)

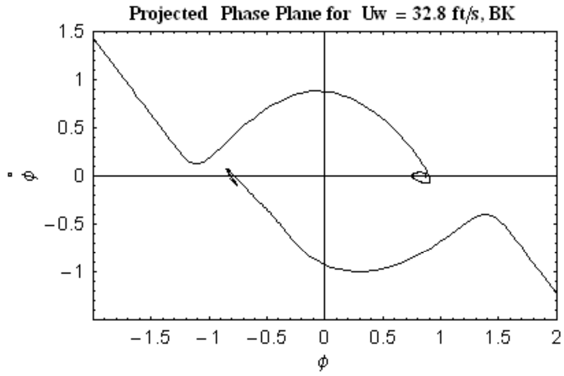

Fig 5b. *OSV* Projected Phase Plane for P-M Spectra, U_W = 32.8 fts^{-1} , With Bilge Keels, Prediction of total damping – Linear (radiation, friction) + Non-linear (viscous)

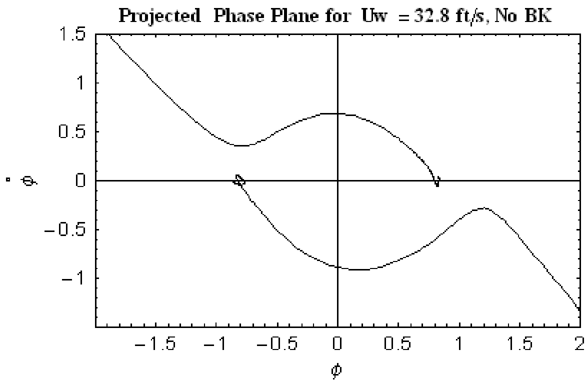

Fig 4b. *OSV* Projected Phase Plane for P-M Spectra, U_W = 32.8 fts^{-1} , No Bilge Keels, Prediction of total damping – Linear (radiation, friction) + Non-linear (viscous)

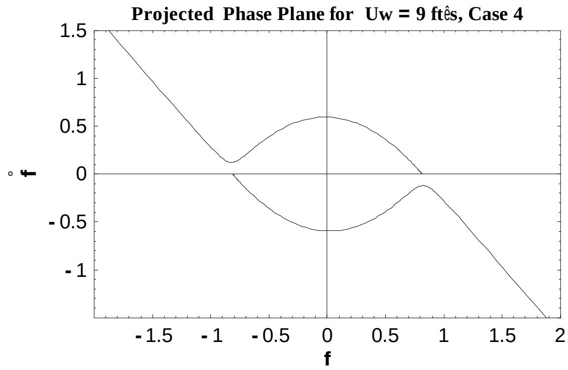

Fig 6a. *OSV* Projected Phase Plane for P-M Spectra, U_W = 9 fts^{-1} , With Bilge Keels: Roll damping – Linear (Full scale Extrapolation) + Non-Linear (viscous)

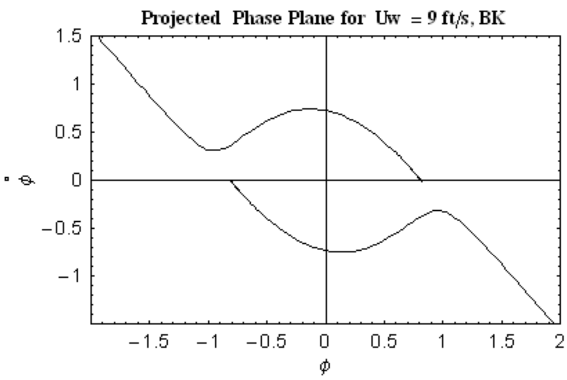

Fig 5a. *OSV* Projected Phase Plane for P-M Spectra, U_W = 9 fts^{-1} , With Bilge Keels, Prediction of total damping – Linear (radiation, friction) + Non-linear (viscous)

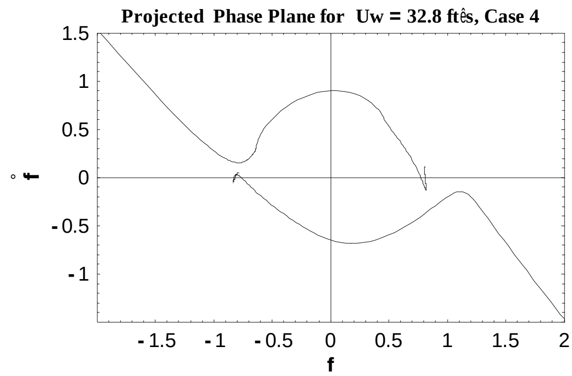

Fig 6b. *OSV* Projected Phase Plane for P-M Spectra, U_W = 32.8 ms^{-1} , With Bilge Keels: Roll damping – Linear (Full scale Extrapolation) + Non-Linear (viscous)

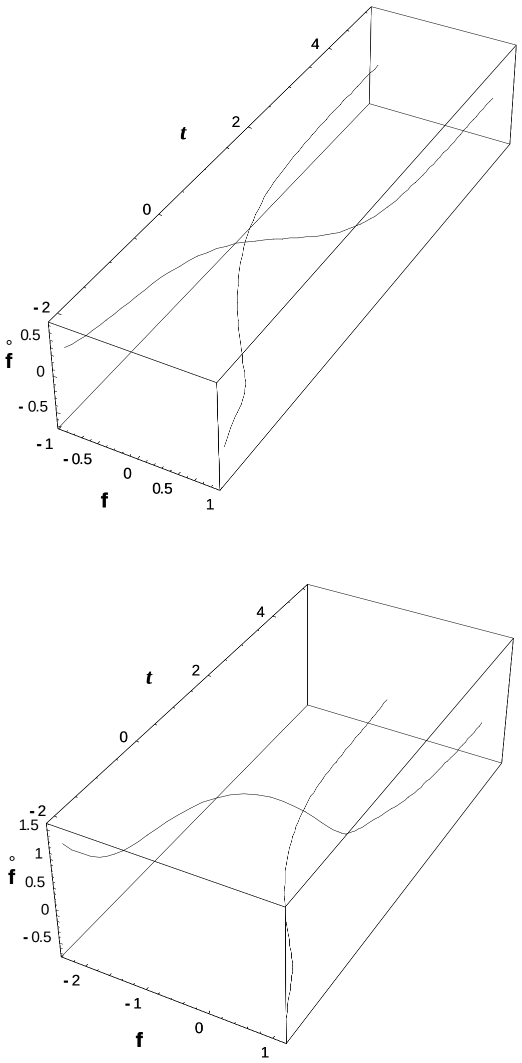

Fig 7a&b *OSV* Extended Phase Space showing solutions contained in upper stable, $W^{+s}(t)$ and lower stable manifold $W^{-s}(t)$, No Bilge Keels for U_W = 9 & 32.8 fts^{-1}