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THE EFFECTS OF SHIPBOARD STEERING MACHINERY DYNAMICS ON RUDDER ROLL STABILIZATION SYSTEMS

by

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The surface ship rolling motion equation is modeled as a second order system, with a natural frequency of $\omega_n = 0.4/\sec$ and a dimensionless damping ratio of $\zeta = 0.08$. The model is subjected to a random forcing function, which has a Gaussian probability distribution and can be considered as "white noise", and placed into State-Space form. State variable feedback of roll rate is applied and the system discretized to match digital control. Roll angle time histories are developed for a range of feedback gains and compared. Additionally, steering machinery dynamics are modeled by a first order system and time constants varied to determine the effects of rudder dynamics on the feedback system.

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The Effects of Shipboard Steering Machinery Dynamics on Rudder Roll Stabilization Systems

by

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ABSTRACT

The surface ship rolling motion equation is modeled as a second order system, with a natural frequency of $\omega_n = 0.4/.\text{sec}$ and a dimensionless damping ratio of $\zeta = 0.08$. The model is subjected to a random forcing function, which has a Gaussian probability distribution and can be considered as "white noise", and placed into State-Space form. State variable feedback of roll rate is applied and the system discretized to match digital control. Roll angle time histories are developed for a range of feedback gains and compared. Additionally, steering machinery dynamics are modeled by a first order system and time constants varied to determine the effects of rudder dynamics on the feedback system.

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NOMENCLATURE

A Plant Matrix

$b_{\delta r}$ Element B(2) in control matrix, sec^{-1}

B Control matrix, $[0 \; b_{\delta r}]^T$

k Feedback gain constant, seconds

RMS Root Mean Squared, $E^{.5}(\phi^2)$

RRS Rudder Roll Stabilization

t Time, seconds

t_n Time at "n-th" time sample, (n) Ts

Ts Sampling time, seconds

u(t) External input, continuous form

u(n) External input, discrete form

X State variables, $[\phi \; \dot{\phi}]^T$

δ_r Rudder control angle

ζ Dimensionless damping ratio

ϕ Roll angle, radians

τ Steering machinery time constant, seconds

ω_n Undamped natural frequency, radians/second

I. INTRODUCTION

Digitally controlled Rudder Roll Stabilization(RRS) systems are currently installed on the Hamilton class Coast Guard cutters and a prototype version has also been tested aboard a U.S. Navy Spruance class destroyer. Observations of these systems have shown that up to a 40 percent reduction in the vessel's root mean squared(RMS) roll amplitude value is obtainable. Unfortunately, the maximum benefits achievable by RRS are not known. This deficiency stems from the performance of the ship's rudder control system, which has a large degree of non-linearity and also operates with a dead band at small input command magnitudes [Ref 1: p. 46]. It is also known that the Spruance and Ticonderoga class of ships have a steering system which behaves with an effective time constant that varies almost inversely with amplitude demand. At low amplitudes, 2-3 degrees, the system can be modeled as having a 3 second time constant while at larger amplitudes, 7-9 degrees, the time constant reflecting the average behavior is closer to 1 second. In addition, the four separate steering systems on the ship, when performing at manufacturer's specifications, will still have distinctly different dynamics that will vary with time as machinery wear and crew servicing occurs.

Additional research in this area has been performed by Van Der Klugt [Ref. 2] and Powell [Ref. 3], to name a few. Van Der Klugt focused his attention on the incorporation of an RRS system into an all weather autopilot for a surface vessel. He revealed that the RRS system needed to be one that could be easily modified, to meet the changing weather conditions, for optimal performance. He also developed guidelines for the proper rudder slew rate that should be used in an RRS system to provide maximum performance.

Powell's work was an appraisal of RRS and compared this system to the fin stabilizer system. What this report exposed was the need for a larger rudder slew rate and increased rudder span. It also discredited proposed ship design changes, to improve RRS effectiveness, such as; altering the rudder Aspect Ratio, increasing the rudder foil area, angling the rudders outboard and increasing the rudder outreach. He also discussed the importance of developing an improved steering machinery system to handle the RRS demands and agreed, in principle, with Van Der Klugt on the rudder slew rate limit.

The purpose of this research was to estimate the influence of steering system dynamics, particularly system time constants, upon the RMS roll amplitude of a ship when the rudder is used in a stability augmentation system; i.e., RRS using state variable feedback of the roll rate. While most research in this field concentrates on ship design changes and rudder slew rate, this thesis examines machinery response to the control input and it's effect on the system performance. A secondary function was to also expose the importance of an adaptive filter which, when used in conjunction with the digital RRS, would bring uniformity to a wide variation of steering system dynamics thus providing a full realization of the stability augmentation capability of an RRS system.

The intended procedure to accomplish these tasks is:

1. Develop a system math model.
2. Discretize the model to match digital control.
3. Subject the model to a generated random forcing function and develop a roll angle time history.
4. Use state variable feedback of roll rate, RRS, and compare roll angle time histories.
5. Develop a steering system model and incorporate it into the system model.
6. Use the random function on the new system and determine, by roll angle history comparison, the effects of the various steering system dynamics on the RRS system.

II. VESSEL DYNAMIC MODELING

As a surface ship travels through the sea it is subjected to three primary disturbances; the wind, waves and its own rudder movement. These inputs create hydrodynamic forces and moments that cause the vessel to move about its horizontal axis, or roll. The equation that best describes this motion is a linearized fourth order state equation involving; sway velocity, yaw rate, roll rate and roll angle, subjected to the inputs of rudder deflection and sea state. Each of the aforementioned components in this equation has with it a hydrodynamic coefficient that is variable. The determination of these coefficients can be quite difficult and in this instance the sea state influence coefficients could never be found.

Even though the appropriate coefficients for the fourth order equation could not be found, it would still be possible to model the roll dynamics as a second order system. Granted, this model would be less stringent then the fourth order equation but it could still provide satisfactory and realistic results.

A. MODEL OF VESSEL DYNAMICS

A surface ship's roll dynamics due to rudder deflection and disturbances can be accurately modeled as a second order equation of motion under the conditions that the model provide the same or very similar results as the actual system under equal conditions. Also, the model results cannot be considered valid when used during instances where the actual system is known to provide conflicting results. As an example consider that a second order model may be stable under all feedback circumstances while the system it is modeling may be unstable under certain conditions of state variable feedback. Therefore, the model results cannot be used under the conditions of feedback where the actual system is unstable.

The second order model for the equation of motion describing a surface ship's behavior in roll is:

$$\ddot{\phi} + 2\zeta\omega_n\dot{\phi} + \omega_n^2\phi = b_{\delta r}\delta_r + u(t) \tag{1}$$

where

$$\ddot{\phi} = roll\ acceleration,\ d^2\phi/dt^2$$

$$\dot{\phi} = roll\ rate,\ d\phi/dt$$

$$\phi = roll\ angle$$

$$\zeta = dimensionless\ damping\ ratio$$

$$\omega_n = natural\ frequency$$

$$b_{\delta r} = rudder\ scaling\ factor$$

$$\delta_r = rudder\ deflection$$

$$u(t) = input\ disturbance$$

The variables ζ and ω_n are ship's speed dependent and were selected for correspondence to the Spruance class destroyer sea trial data at 15 knots. The estimated magnitudes were $\zeta = 0.08$ and $\omega_n = 0.4\ /sec$. To determine the magnitude of $b_{\delta r}$ one must first find the correlation between the ship roll angle and rudder deflection, or ϕ/δ_r. The sea trial data was again used and a value of 0.15 roll angle (radians) per rudder angle (radians) was found from steady turn observations.[Ref. 1: pp. 45-46]

Knowing the static sensitivity value, ϕ/δ_r , it is then possible to return to the second order model and determine $b_{\delta r}$ as follows:

$$\ddot{\phi} + 2\zeta\omega_n\dot{\phi} + \omega_n^2\phi = b_{\delta r}\delta_r$$

at steady state all transients decay to zero and

$$\omega_n^2\phi = b_{\delta r}\delta_r$$

with

$$\frac{\phi\omega_n^2}{\delta_r} = b_{\delta r}$$

therefore $b_{\delta r} = 0.024/\ \mathrm{sec}^2$.

With all variable coefficients determined and the disturbance $u(t)$ acting as wind and wave action the second order model becomes:

$$\ddot{\phi} + 0.064\dot{\phi} + 0.16\phi = 0.024\delta_r + u(t)$$

B. STATE-SPACE EQUATION

The second order model can be converted to the standard State-Space format:

$$\dot{x} = Ax + Bu$$

by recognizing that:

$$x = state\ variable, [\phi \quad \dot{\phi}]^T$$

$$A = plant\ matrix$$

$$b = control\ matrix$$

$$u = noise\ input$$

The State-Space Equation then becomes:

$$\begin{bmatrix} \dot{\phi} \\ \ddot{\phi} \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega_n^2 & -2\zeta\omega_n \end{bmatrix} \begin{bmatrix} \phi \\ \dot{\phi} \end{bmatrix} + \begin{bmatrix} 0 \\ b_{\delta r} \end{bmatrix} \delta_r + \begin{bmatrix} 0 \\ 1 \end{bmatrix} u(t) \tag{2}$$

The State-Space form provides a convenient means to determine and analyze the system's response to various manipulations. The addition of state-variable feedback or any other change can be accomplished with relative ease. The State-Space representation can be quickly converted to a discretized form to accommodate digital control.[Ref. 4: p. 10]

C. DISCRETIZATION

The RRS system used aboard these vessels is digitally controlled and thus our model should be represented and used in this manner. The governing criteria when using digital control and discrete time systems is the sampling frequency. The sampling frequency is how often the control system senses data input, or the amount of time that elapses between successive data samples. The choice of sampling frequency is determined by the period of the system to be measured. The Nyquist criteria says that the sampling frequency must be no less than two times the maximum frequency of the system being measured [Ref. 5]. It has been determined that the period of oscillation for the Spruance destroyer at 15 knots is 15.3 seconds which gives it a frequency of 0.0653/ sec [Ref. 1: p. 44]. A sampling time step of 0.333 seconds was chosen, which is greater than 45 times the frequency of the system and allows for a complete reconstruction of the continuous input disturbance function.

When dealing with discretization, state variables are predicted for each successive time step by the previous step's state variables and the previous input. A numerical representation of this is:

$$x(n+1) = \Phi(Ts)x(n) + \Gamma(Ts)u(n)$$

where

$$\Phi(Ts) = e^{ATs}$$

$$\Gamma(Ts) = \left[e^{ATs} - I\right]A^{-1}B$$

$$I = identity\ matrix$$

$$A = plant\ matrix$$

$$Ts = time\ step$$

$$x = state\ variables$$

$$B = control\ matrix$$

$$u = input\ disturbance$$

$$n = integer\ index,\ n = 0,1,2,3,....$$

By using this type of approximation a recursive relationship is developed and the state variable's time history can be easily calculated by a computer program.[Ref. 6]

D. FEEDBACK

State variable feedback can be used to alter the dynamic response of a system. In this case the desired effect is to decrease the roll angle magnitude of the ship by feeding back the roll rate increased by a multiplicative gain.[Ref. 4: p. 222]

To maintain the accuracy of the second order model it is necessary to look more closely at the actual fourth order Equation of Motion for this system to determine if and where that system may achieve maximum stability or become unstable. An estimated root locus plot of the fourth order system has been developed and the optimum value or position of greatest stability occurs when the dimensionless damping ratio, ζ, is equal to 0.4 for a ship speed of 15 knots [Ref. 1: p. 47]. This implies that the maximum damping ratio for the feedback compensated system model can not be expected to exceed $\zeta = 0.4$. Figure 1 displays the estimated root locus plot for the ship.

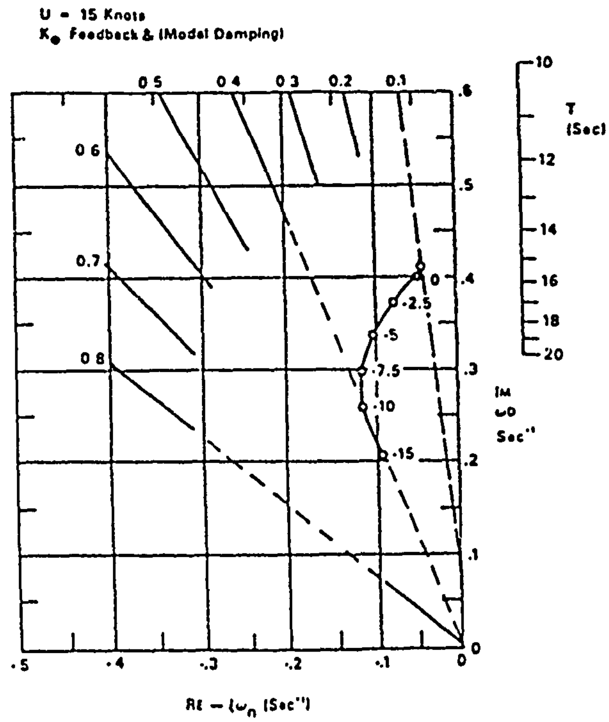

Figure 1. Root Locus plot

The State-Space Equation then becomes:

$$\begin{Bmatrix}\dot{\phi}\\ \ddot{\phi}\end{Bmatrix} = \begin{bmatrix}0 & 1\\ -\omega_n^2 & -2\zeta\omega_n - kb_{\delta r}\end{bmatrix}\begin{Bmatrix}\phi\\ \dot{\phi}\end{Bmatrix} + \begin{bmatrix}0\\ 1\end{bmatrix}u(t) \qquad (3)$$

where

$$-2\zeta_c\omega_n = -2\zeta\omega_n - kb_{\delta r}$$

and the maximum value of $\zeta_c = 0.4$. Therefore the maximum value of the gain k is 10.66. This system was discretized as discussed previously.

E. RUDDER CONTROL MODEL

The ship's electro-hydraulic rudder control system behaves non-linearly and is also accompanied by a dead band at small rudder input commands. Since the dead band is very small relative to the full scale it will be ignored in the model and the non-linear response will be characterized by a first order system with the various time constants representing the average first order behavior for different amplitudes of the machinery. The first order equation for the steering system is:

$$\dot{\delta}_r = \frac{1}{\tau}(\delta_{rc} - \delta_r) \tag{4}$$

$\dot{\delta}_r$ = *rudder deflection rate, sec*$^{-1}$

τ = *time constant, seconds*

δ_{rc} = *helm command*

δ_r = *rudder deflection*

From shipboard machinery performance tests it was determined that the time constants associated with the steering system ranged from one to three seconds and these values will be used in this discussion [Ref. 1: p. 46]. Considering that a ship usually travels in a straight path, it was decided to use a zero degree rudder command from the ship's helm. This is not a poor assumption, most vessels remain on the same course for long lengths of time. This reduces the rudder model to the time constant and the rudder deflection and deflection rate. The actual ship steering machinery has an approximate rudder demand rate limit of 6 degrees per second [Ref. 1: p. 47]. In the studies described herein, it was assumed that required rudder rates did not involve the influence of control-rate limiting. Figure 2 displays the rudder deflection with various time constants to a unit step input.

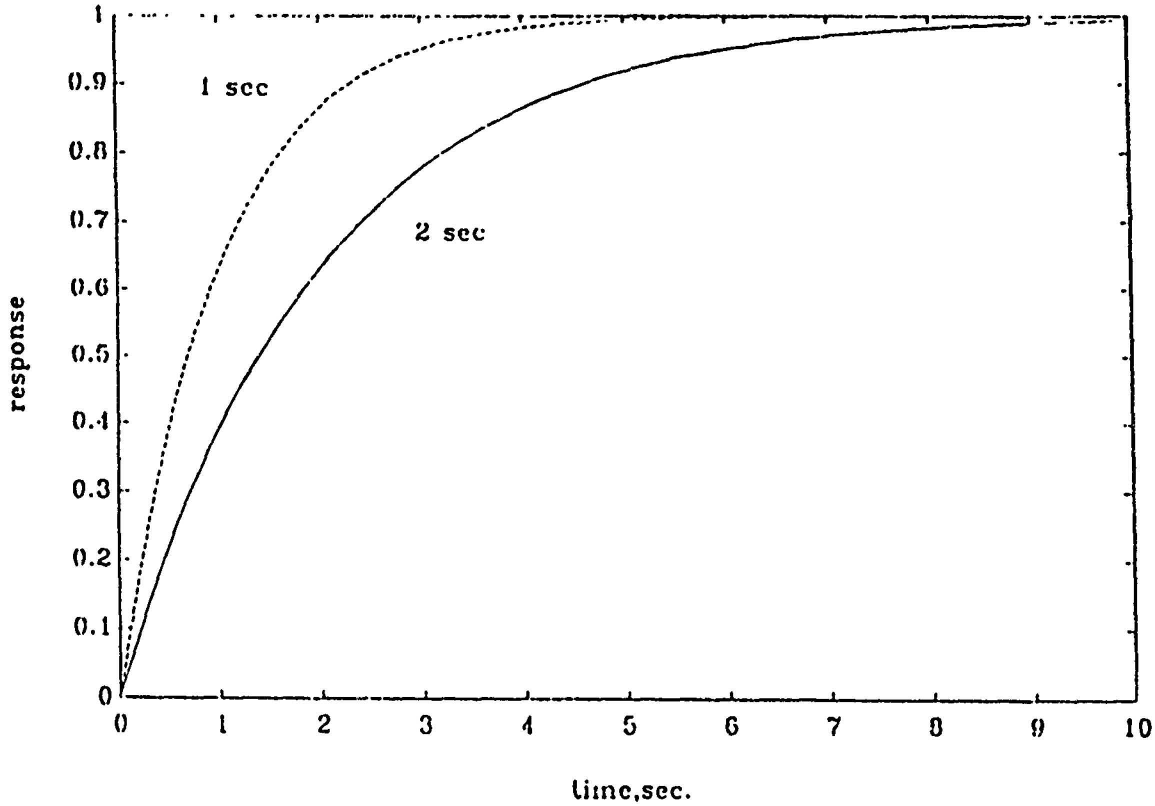

Figure 2. Rudder Response

F. THE COMPLETE SYSTEM

When the rudder deflection is included in this system, to account for the rudder dynamics, it assumes the role of a third state variable and the $\Phi(Ts)$ and $\Gamma(Ts)$ discussed previously in the second order system are now represented in a third order equation. The complete system including feedback and the steering system characteristics when placed in State-Space form becomes:

$$\begin{Bmatrix} \dot{\phi} \\ \ddot{\phi} \\ \dot{\delta}_r \end{Bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ -\omega_n^2 & -2\zeta\omega_n & b_{\delta r} \\ 0 & -k/\tau & -1/\tau \end{bmatrix} \begin{Bmatrix} \phi \\ \dot{\phi} \\ \delta_r \end{Bmatrix} + \begin{bmatrix} 0 \\ 1 \\ 0 \end{bmatrix} u(t) \tag{5}$$

III. RESULTS

The random function used as the disturbance from wind and wave action was developed by the Monte Carlo method [Ref. 7]. The characteristics of this function are that it has a Gaussian probability distribution and, because it represents a wide-band random input function, can be considered as white noise. In essence, the wide-band input function passes through a band-pass filter (the ship math model) and produces a narrow-band output similar to that of the actual ship's behavior. One must recognize, however, that the sea is not considered "white" and that this white noise function was used to simplify the problem. The same disturbance was used on all the systems thereby validifying the comparisons. A graphical representation of the disturbance function is shown in Figure 3.

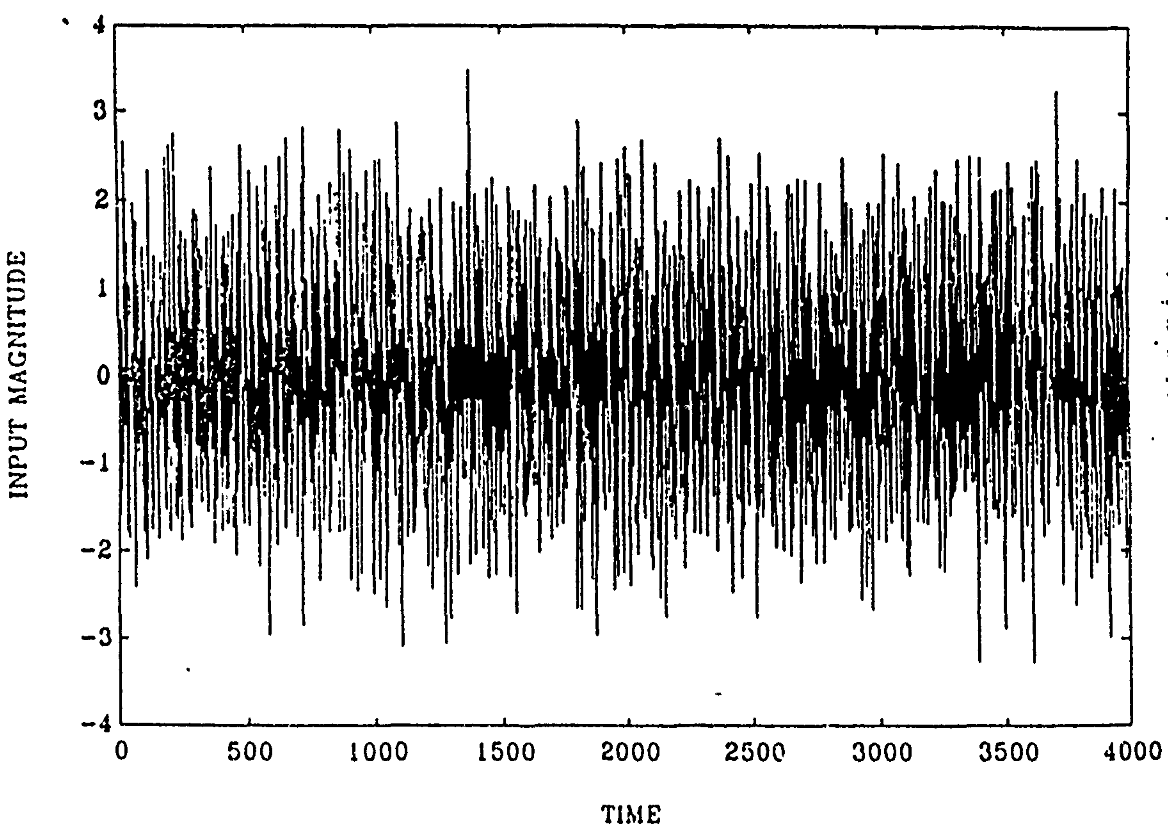

Figure 3. Disturbance function

A. FEEDBACK EFFECTS

To determine the effects of roll rate feedback, upon the system with ideal steering characteristics, a series of output responses were developed for a gain magnitude range of zero to 10.66. This corresponds to a damping ratio spectrum of 0.08(no gain) to 0.4(k = 10.66). Figure 4 displays the roll angle RMS for this spectrum and as expected the RMS magnitude significantly decreases from 3.641 to 1.692, a 53 percent reduction. Also plotted on this graph is the expected RMS behavior of a second order system subjected to white noise, RMS $\propto 1/\sqrt{\zeta}$. A comparison between the two curves shows that the model results are slightly higher. This can be accounted for by round off error in both the discretization and recursion relationship calculations. Since the output response of the ship model compares very closely to the theoretical response of the second order system to white noise, it can be assumed that the ship model is valid.[Ref. 8: pp. 503-505]

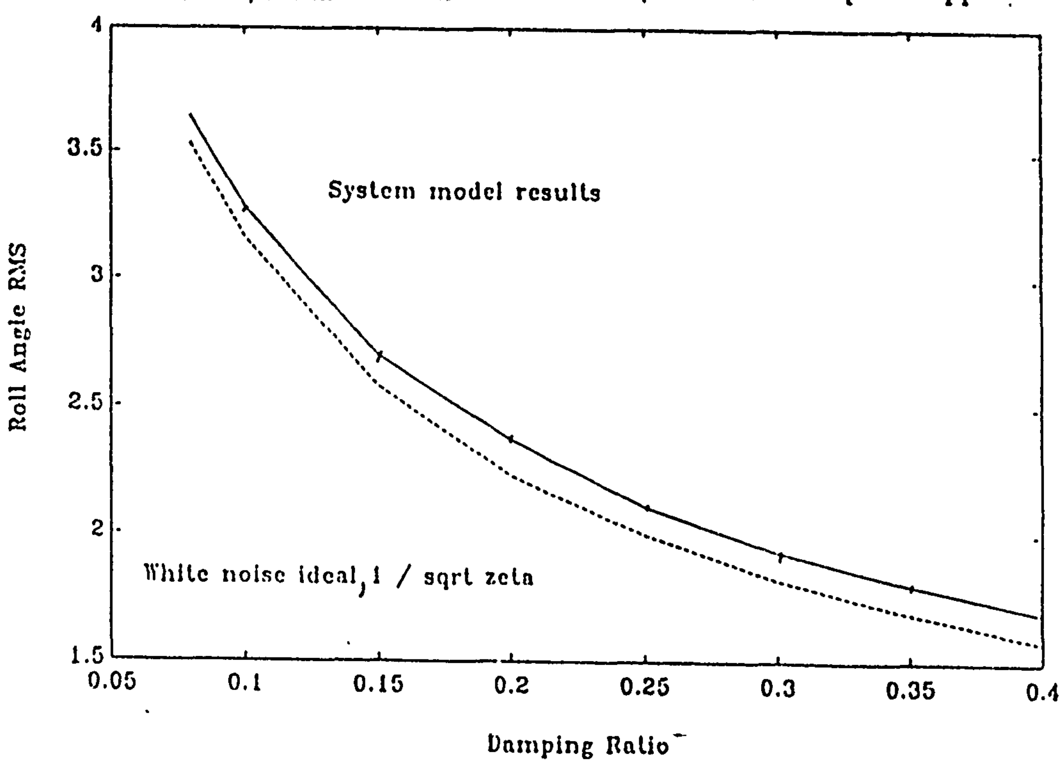

Figure 4. RMS vs. Damping ratio

Figures 5 and 6 are the actual roll angle time histories for the uncompensated and fully compensated systems. From these figures it is obvious that not only has the RMS value decreased but that the peak magnitudes have been drastically reduced. It is the reduction in peak magnitude that is really the most beneficial aspect of RRS, there is always going to be some rolling motion and if the size of the roll angle can be minimized then the ship and her crew can carry out it's mission more effectively.

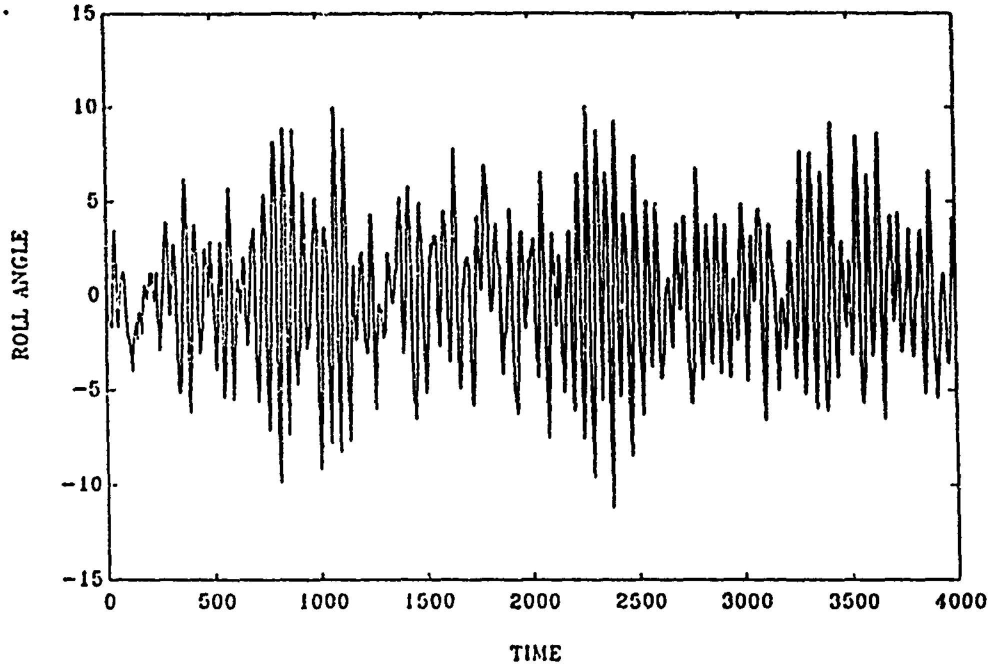

Figure 5. Uncompensated Roll Angle History, Spruance class Destroyer

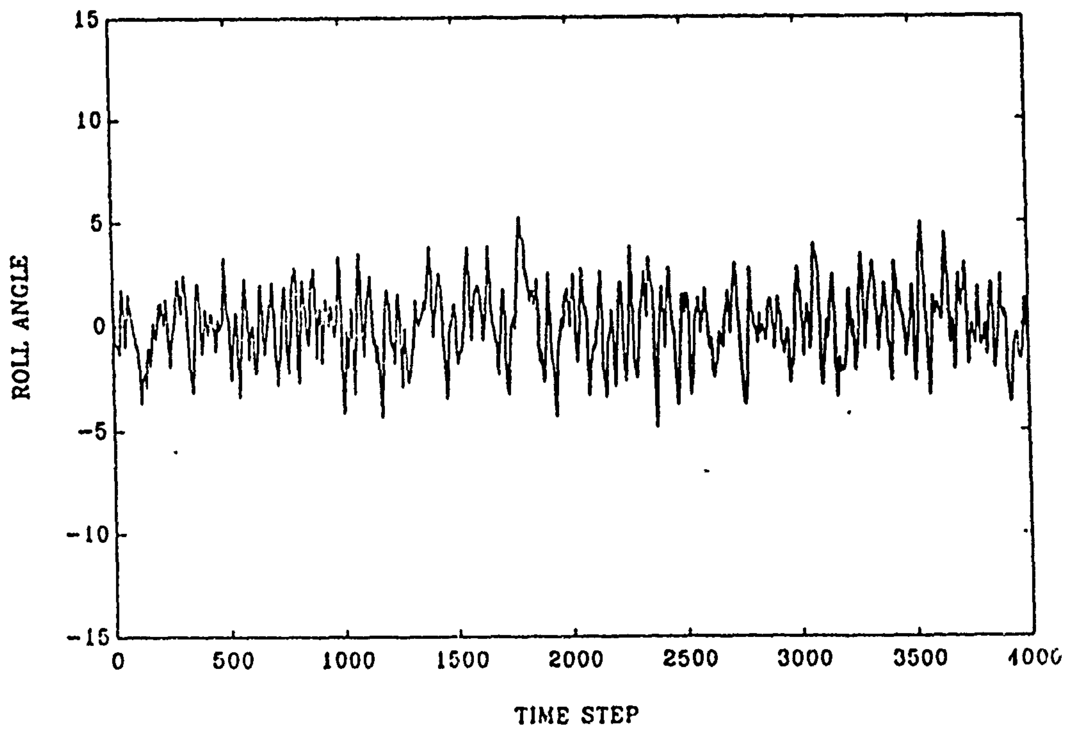

Figure 6. Fully Compensated Roll Angle History, Spruance class Destroyer

B. STEERING SYSTEM EFFECTS

Time constants of zero to three seconds were chosen for use with the system involving the maximum gain magnitude of 10.66 (damping ratio of 0.4). Figure 7 displays the relationship between the output RMS of the assumed ship model to the individual time constants. The RMS magnitude increases with the increase in the time constant thereby substantiating an intuitive feeling that steering machinery dynamics adversely effect the RRS system and more importantly, that the influence of the steering machinery can be quantified.

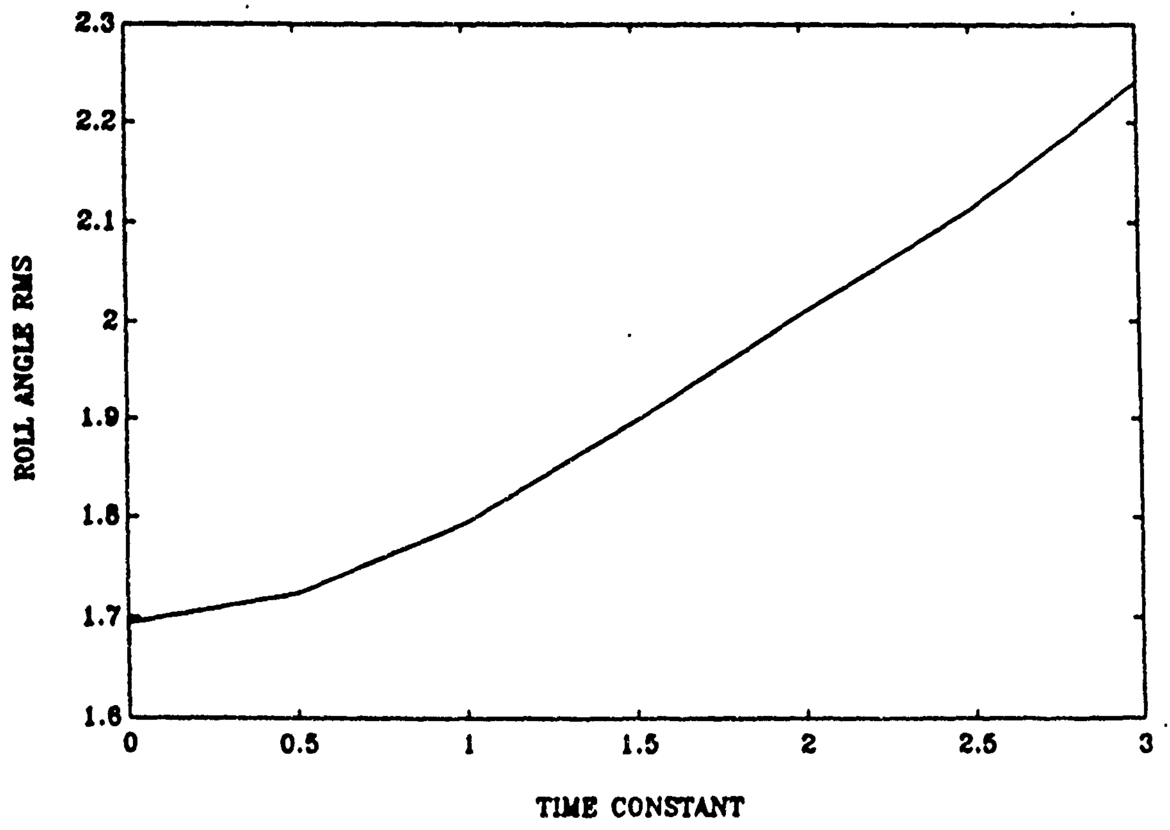

Figure 7. Roll Angle RMS vs. Machinery Time Constant

By taking a closer look at equation 5 one can fully realize the effects of the machinery time constant on the system behavior. Figure 8 is a plot of the characteristic roots of that equation based on various time constants and from this plot it is clear that as the time constant increases the effective system damping ratio is decreasing. This is more clearly displayed in Figure 9 where the damping ratio is plotted versus the time constants. It is because of this decrease in system damping, brought on by the addition of the time constants, that increases the RMS value of the roll angle.

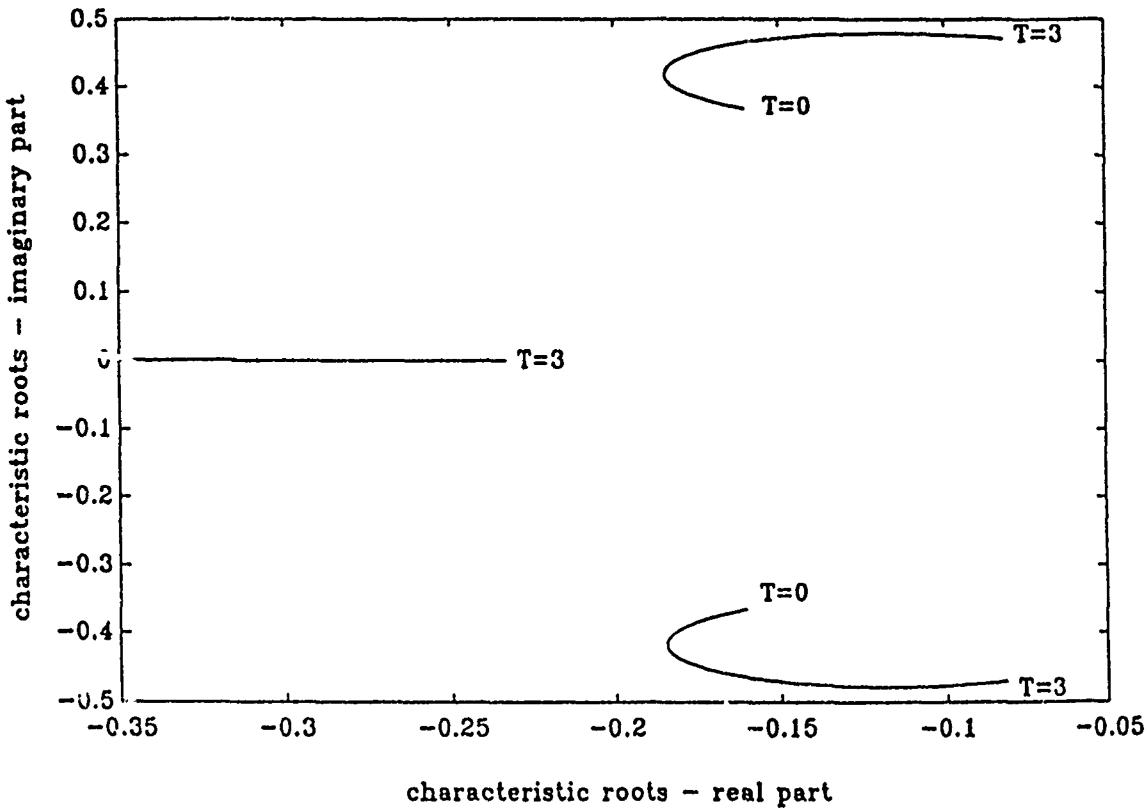

Figure 8. Characteristic Roots of Equation 5

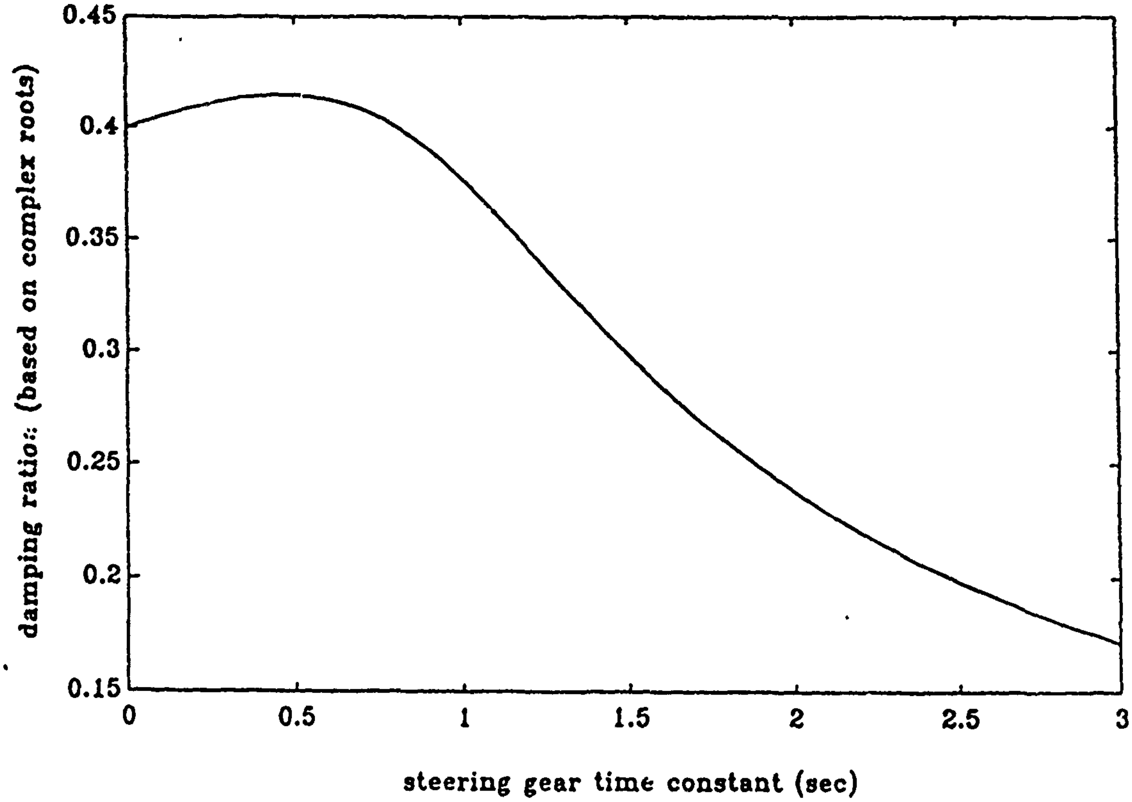

Figure 9. Damping Ratio VS. Time Constants

Figures 10 and 11 display the output response of the zero and 1.5 second time constant systems using the maximum gain of k = 10.66. From these figures we see an increase in peak magnitude and roll angle RMS as the time constant increases with no other significant differences. Table 1 shows the percentage of RMS reduction for time constants ranging from 1 to 3 seconds. Without steering machinery dynamics, the potential RMS roll reduction due to the assumed state- variable feedback was 53 percent. The effect of a one second time constant reduced the potential roll reduction by approximately 6 percent while a three second time constant reduced the potential roll reduction by 24 percent. The conclusion is that "sluggish" steering machinery dynamics has a perceptible effect upon ship roll stabilization by the use of the rudder.

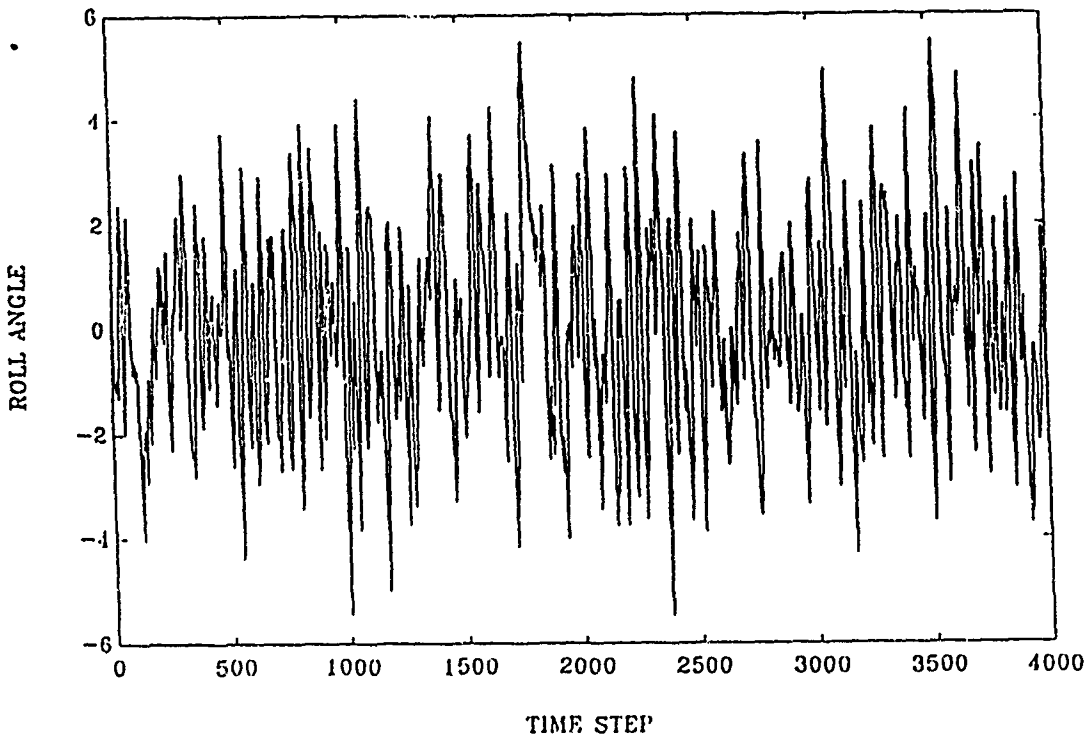

Figure 10. Roll Angle History, Ideal Steering Machinery

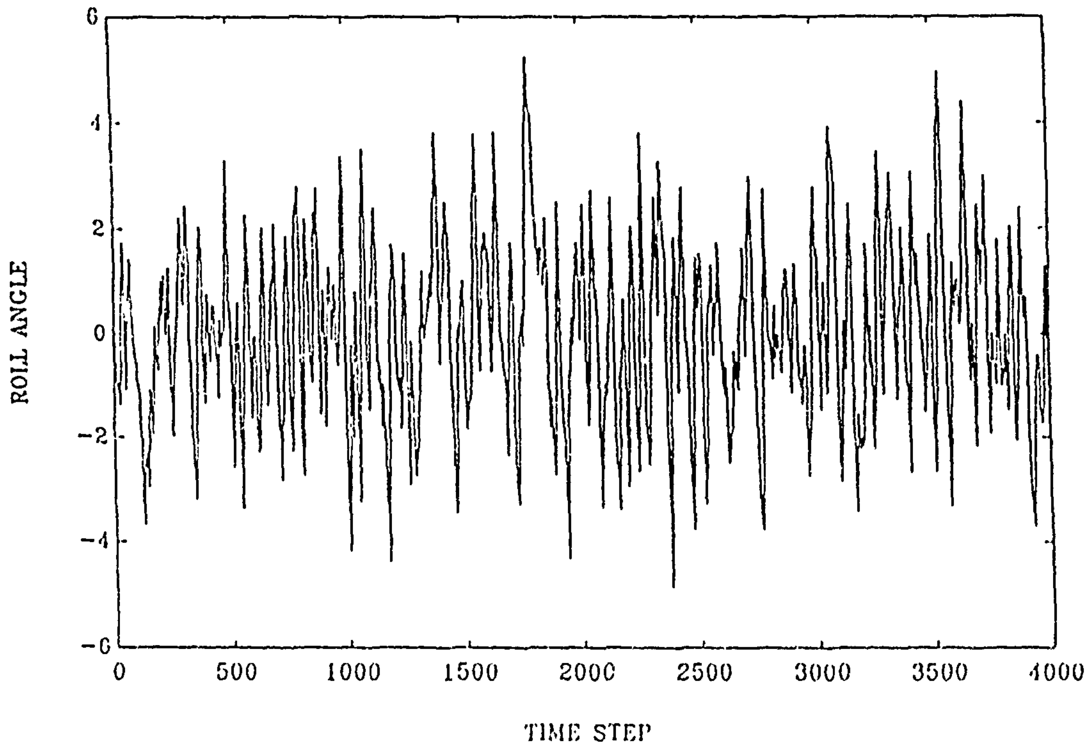

Figure 11. 1.5 Second Time Constant Roll Angle History

Table 1. RMS AND PERCENT REDUCTION VALUES FOR TIME CONSTANTS

Time constant in seconds	RMS	Percent reduction
0.0	1.692	53.5
0.5	1.719	52.3
1.0	1.794	50.7
1.5	1.897	47.8
2.0	2.008	44.8
2.5	2.113	41.9
3.0	2.240	38.4

Figures 12 and 13 display rudder deflection time histories for the fully compensated system with two different time constants. The point to be made is that as the time constant increases the magnitude of the rudder deflection decreases because the system becomes less responsive. Thus the rudder becomes less effective and it then cannot fully support the RRS system.

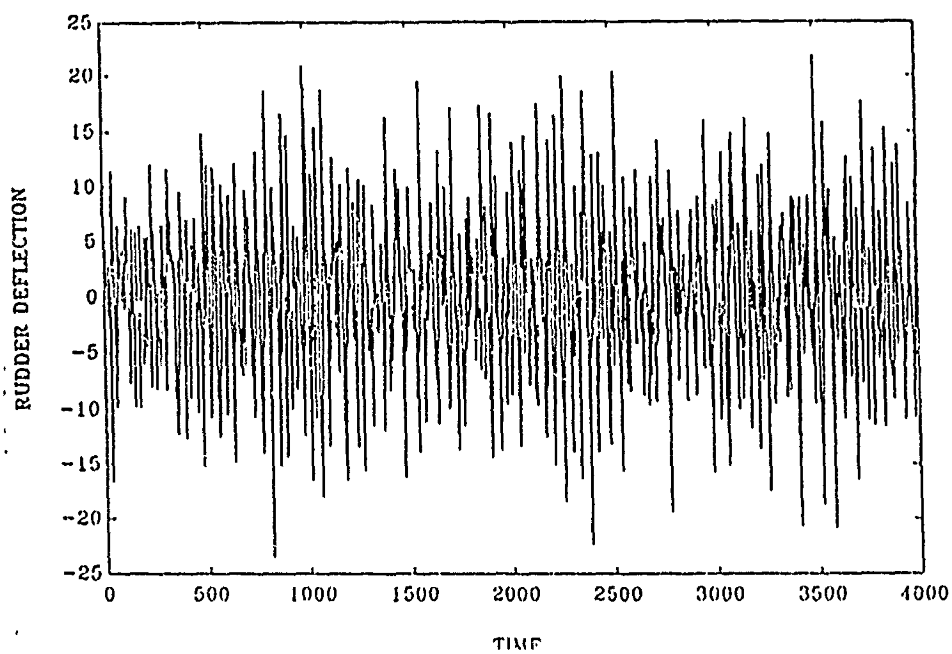

Figure 12. Rudder Activity, Machinery Time Constant of 0.5 Seconds

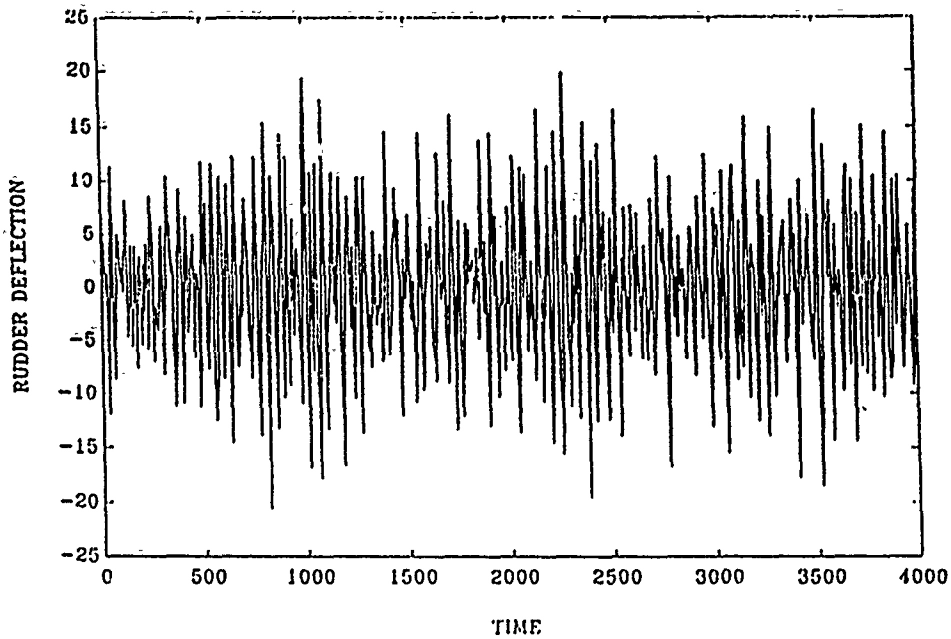

Figure 13. Rudder Activity, Machinery Time Constant of 1.5 Seconds

IV. CONCLUSIONS AND COMMENTS

It is obvious from the results that steering machinery dynamic characteristics adversely effect the Rudder Roll Stabilization system. Even with this hindrance the system can still reduce the roll angle RMS and peak magnitude values significantly.

Up to this point no mention of the effects of the RRS system upon the rudder itself has been made. Figures 12 and 13 show that the rudder is moving often and very quickly. Concerns directed towards machinery wear and reliability are justified and focus should be placed on designing these systems to handle the increased activity. If no design solution can be found then guidelines should be developed suggesting that the RRS system only be operated in harsh sea states thus reducing the overall operating time and possibility of material failure.

The choice of the maximum system gain was made purely on the basis of stability. Van Der Klugt in his paper, "Rudder Roll Stabilization: The Dutch Solution", states that there is a point, irrespective of the rudder speed, where the RRS system can longer reduce the roll activity. This point is a rudder speed of approximately 20 deg/sec which corresponds to a compensated damping ratio of 0.52 in this system.[Ref. 2 : p. 85] In this research that point could not be reached because the gain needed to get there would have driven the fourth order equation of motion unstable, as seen by Figure 1.

On the other hand, Powell's paper , "Rudder Roll Stabilization: A Critical Review," has determined that a rudder speed of approximately 10 deg/sec, which is an average value taken from the results of various sea states and wave encounter angles, is where the point of no further roll reduction takes place [Ref. 3: pp. 2.250]. Using that criterion, one sees that the second order model used in this research is reasonable and produces qualitativly correct results.

Another conclusion that can be drawn is that each piece of machinery is going to behave differently. This fact promotes the development of an adaptive filter that can be used to bring uniformity to the wide range of machinery dynamics thus enabling the RRS to reach its full potential as a stability augmentation device.

APPENDIX THE INVERSE PROBLEM

The original idea behind this research was to use the roll angle time histories developed by the sea trial team of Baitis and Schmidt et. al. and, working in an inverse manner, develop the sea state forcing function that caused this motion. This idea had good merits. The roll time history data was taken when the ship was on a steady course with minimal rudder movement, so the roll data could reasonably be considered from a rudder fixed condition which would mean that all motion was induced from the sea. The coefficients for the fourth order state equation were known with good confidence, except those for the sea state, so it seemed that the inverse problem could be solved by working backwards through the state equation. What stood in the way was the fact that three out of the four state variables were either measured inaccurately or not measured (observed). The yaw rate, sway velocity and roll rate were missing from the available data base and without them it seemed that any inversing process would be doubtful.

These concerns were substantiated by Gao and Hess who describe an inverse technique which is an iterative process requiring that all output states were measurable and that the number of inputs must equal or exceed the number of outputs [Ref. 9]. Since neither of these conditions were met, the research steered away from this path.

In retrospect, a solution to this problem may have been to use the actual sea trial roll data in conjunction with a second-order linear ship model to develop an approximation to a forcing function. Then the input function could be re-applied to the second-order model system with state variable feedback and steering system time constants, so that roll time histories could be developed. These time histories could then be compared and the impact of the steering system dynamics brought out. This approach is left for future consideration in follow on studies.

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