A Control System for a Hybrid Linear Actuator for a Flush Deck Hatch

Musoke H. Sendaula   Saroj K. Biswas   Joseph P. Teter   David J. Brady

Abstract—**This paper presents the control system design in the development of a hybrid linear actuator for opening the flush deck hatch CV/CVN ship class. We present a simple feedback control system for the hatch opening, which is described by a nonlinear model. The hybrid linear actuator consists of magnetostrictive thrusters and high force electric linear motors segments ganged together to drive a load shaft through of sets of self-locking, magnetically activated clutches. The control system is based on trajectory following method with a simple feedback-linearizing loop. Simulation results show excellent performance of the closed loop system.**
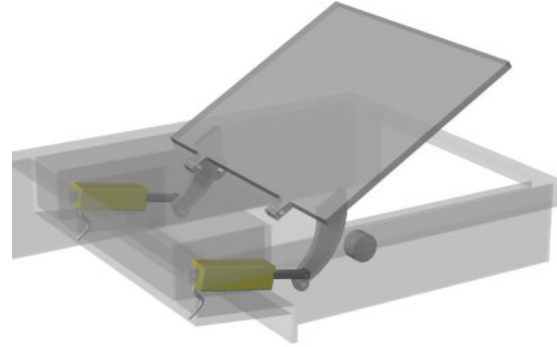
I. INTRODUCTION

Hydraulic drive systems commonly used in ships are expensive, both because of their long piping system as well as maintenance of their fluid system. As such, all-electric drives are preferred for modern electric ships. Transition from hydraulic drive to electric drive is a technological challenge since commercially available electric thrusters are not capable of generating forces in the range of 50,000 lbs and above that is required for replacement of hydraulic thrusters. A novel hybrid linear actuator (HLA) consisting of a magneto-strictive actuator and a linear motor has been developed [1, 2] that meets the above power density requirement. The HLA prototype develops an average of 49,000 lbs of thrust force, and a peak force of 98,000 lbs. This research presents the control system design for the actuator, and its application in opening of a flush-deck.

Musoke H. Sendaula is a Professor on Electrical and Computer Engineering, Temple University, Philadelphia, PA19122 (Ph: 215-204-7819, Fax: 215-204-5960, email: musoke@temple.edu)
Saroj K. Biswas is an Associate Professor of Electrical and Computer Engineering, Temple University, Philadelphia, PA19122 (email: sbiswas@temple.edu)
Joseph P. Teter is with the Naval Surface Warfare Center, Carderrock Division, Material Science Branch, West Bathesda, MD20817 (email: TeterJP@nswccd.navy.mil)
David J. Brady is with the Naval Surface Warfare center, Carderrock Division, Cargo handling Branch, Philadelphia, PA 19112 (email: BradyJD@nswccd.navy.mil)

Figure 1 Hybrid Linear Actuators for CV/CVN Flush Deck

Figure 1 shows a proposed installation of the hybrid linear actuators as replacements of the current elector-hydraulic actuators for the CV/CVN flush deck.

The hybrid linear actuator consists of a magneto-strictive thruster and multiple linear motor segments that are divided into groups and ganged together to drive the load through different sets of clutches. The different groups of thrusters and linear motor segments are energized in sequence to form a multi-phase drive system. The details of the design and its magnetically actuated clutches can be found in [1-2].

The challenge in designing the hybrid solid-state electric linear actuator is to be able to control the thrusters, linear motors and clutches while driving a time-varying load subject to random disturbances. Unlike elector-hydraulically-powered actuators that have inherent damping, the HLA has minimal damping. A precise control system will be needed in the presence of wind loads that may exceed the drive force when the hatch is fully open.

The outline of the paper is as follows: Section 2 develops a mathematical model for the hatch opening in the presence of random wind load and ship motion. A trajectory following control system is presented in Section 3, which is followed by simulation results in Section 4. Section 5 has some concluding remarks.

II. SYSTEM MODEL

The flush deck hatch represents a time-varying load whose magnitude varies as it opens due to changes in orientation, wind loading, and sea states. A dynamic model of the CV/CVN flush deck shown in Figure 1 was developed using the dimensions and variables given in

Figure 2 and Figure 3. Figure 2 shows the position of the hatch and the HLA, when the hatch is closed. As the HLA shaft extends, the hatch rotates though an angle θ(t). Figure 3 shows the HLA and the hatch as it opens.
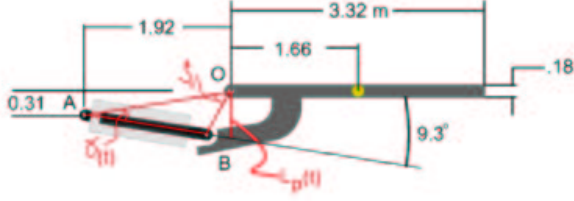

Figure 2: HLA with Hatch Closed

It follows from the CV/CVN flush deck hatch and actuator geometry in Figures 2 and Figure 3 that the actuator length $L_a(t)$ is given by:

$$L_a(t) = AB, \text{ and}$$

$$L^2_a(t) = AO^2 + BO^2 - 2AOBO\cos\left(\zeta_0 + \theta(t)\right).$$

So the moment arm of the actuator force $L_p(t)$ is given by:

$$L_p(t) = \frac{AOBO}{L_a(t)}\sin(\xi_0 + \theta(t)).$$

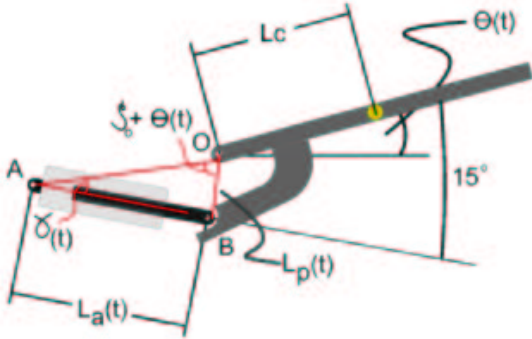Figure 3: HLA with Hatch Opening

The motion during the opening of the hatch can be considered as a rotating plate described by:

$$I_z\frac{d^2\theta}{dt^2} = L_pF_d - M_hgL_c\cos\theta + L_cW_L \quad (1)$$

where

θ = Angular position of the hatch door

$$\omega = \frac{d\theta}{dt} = \text{Angular velocity of the hatch}$$

M_h = Mass of the Hatch

I_z = Moment of inertia of the hatch for angular swing

W_L = Effective load due to deck wind and sea state

F_d = Drive force applied to the hatch
g = Acceleration due to gravity
L_p = Perpendicular distance of the drive arm from the hinge
L_c = Distance to hatch center of gravity

A. EFFECTS OF WINDLOAD

Strong winds on the ship deck may affect the hatch opening. Force applied to the hatch due to deck wind may be described by

$$W_{wind} = \rho k v^2 \sin^2\alpha \quad (2)$$

where v is the wind speed, and α is the angle of attack, i.e., angle at which wind hits the hatch, and k is a constant, and ρ is a wind direction factor. For the wind direction shown in the figure, deck wind actually aids the hatch opening so that we take $\rho = 1$. Deck winds in the opposite direction, $\rho = -1$, will obstruct the hatch opening.

The wind speed is a random process, which can be assumed to be a Gaussian process with an appropriate mean and a variance. The angle of attack for the wind direction varies as the hatch opens and as the ship oscillates due to random sea states so that we can express $\alpha = \theta + \xi$, where ξ is a random angle representing the effects sea states.
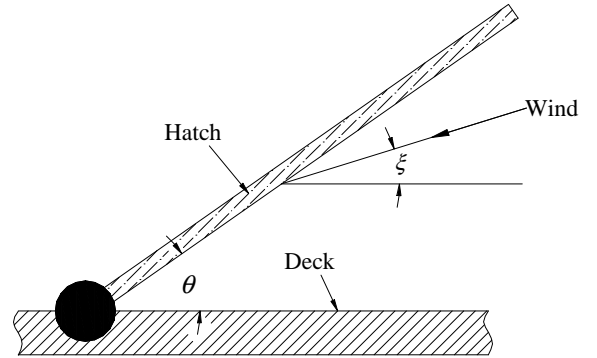

Figure 4: Angle of Attack of Deck Wind

Considering random variations discussed above, windload applied to the hatch can be represented as

$$W_{wind} = \rho k(v_m + \eta)^2 \sin^2(\theta + \xi) \quad (3)$$
$$= \rho k v_m^2(1 + \tfrac{\eta}{v_m})^2 \sin^2(\theta + \xi)$$

where v_m is the mean wind speed and θ is the actual hatch angle, and η and ξ are Gaussian random processes representing variations in wind speed and angle of attack, respectively. In the sequel, for simplicity, η is normalized as a fraction of the wind speed.

B. EFFECTS OF SEA STATE

Ship oscillations due to sea states affect the hatch opening in two different ways. First of all, it changes the angle of attack of the wind onto the hatch, which has been discussed earlier. Secondly, ship oscillations in the θ direction impart an inertial moment to the hatch that may affect the

hatch opening; this is essentially due to the fact that hatch reference frame is not stationary in the inertial space. Denote the orientation angle of the ship coordinate system in the θ direction by ϕ. Then the effect of ship motion is essentially equivalent to an additional moment in the right hand side of equation (1), which is given by

$$W_{seastate} = \text{Moment due to sea state} = -I_z \frac{d^2\phi}{dt^2} = \zeta \quad (4)$$

Note that ship oscillations are random so that the angle ϕ is also a random process. Thus the effects of sea states can be conveniently simplified to a single random process, ζ.

C. TOTAL RANDOM FORCE ON THE HATCH

Combining the wind load and the effects of sea state, the total force on the hatch is obtained as

$$W_L = \rho k v_m^2 (1+\eta)^2 \sin^2(\theta + \xi) + \zeta \quad (5)$$

For the wind speed term in equation (5), we observe that

$$(1+\eta)^2 = (1+\sigma^2) + (2\eta + \eta^2 - \sigma^2) \quad (6)$$

$$= (1+\sigma^2) + \psi_1 \quad (7)$$

with ψ_1 being a stochastic process with mean zero and the variance $4\sigma^2 + 2\sigma^4$, where σ is the covariance of the process η. Similarly, for small variations of the angle of attack, we have

$$\sin^2(\theta + \xi) \cong \sin^2\theta + \xi \sin 2\theta = \sin^2\theta + \psi_2 \quad (8)$$

where ψ_2 is a random process. Combining the above equations, the total effect of wind load and the sea state is summarized as

$$W_L = \rho k v_m^2 (1+\sigma^2) \sin^2\theta + \psi \quad (9)$$

D. MODEL NORMALIZATION

Assuming that the mass center of the hatch is located at the center of the hatch, we have

$$I_z \cong \frac{4}{3} M_h L_c^2 \quad (10)$$

We also normalize the various terms in system equation with respect to the weight of the hatch, $M_h g$. Define

$$F_d = M_h g F_a$$
$$W_L = w_L M_h g \quad (11)$$

Then equation (1) simplifies to

$$\frac{d^2\theta}{dt^2} = \frac{3g}{4L_c} \left[\frac{L_p}{L_c} F_a - \cos\theta + w_L \right] \quad (12)$$

The wind load (with hatch completely open) is normalized with respect to the weight of the hatch

$$\lambda = k v_m^2 / M_h g$$

and, for simplicity, denote

$$\beta = \frac{3g}{4L_c}$$

Then we have

$$\frac{d^2\theta}{dt^2} = \beta \left[\frac{L_p}{L_c} F_a - \cos\theta + \rho\lambda(1+\sigma^2)\sin^2\theta + \psi \right] (13)$$

where ψ is a random noise process. The mean dynamics of the system simplifies to

$$\frac{d^2\theta}{dt^2} = \beta \left[\frac{L_p}{L_c} F_a - \cos\theta + \rho\lambda(1+\sigma^2)\sin^2\theta \right] \quad (14)$$

This completes the development of the mathematical model of the hatch motion.

III. TRAJECTORY FOLLOWING CONTROL

Next we consider the design of a trajectory following controller that drives the hatch according to a desired profile. First consider the unperturbed system, equation (14). Since the system is inherently nonlinear, the control system consists of two components, a linearizing control and a trajectory generating control as shown in Figure 5. Assume that the drive force is given by

$$F_a = \frac{L_c}{L_p} \left[u^* + \cos\theta - \rho\lambda(1+\sigma^2)\sin^2\theta \right] \quad (15)$$

where u^* is a control signal for the linearized system. Substituting the above control law into equation (14), we obtain

$$\frac{d^2\theta}{dt^2} = \beta u^* \quad (16)$$

Assume that the desired trajectory for the hatch opening operation is given by

$$\theta = \frac{\omega}{4}\left(t - \frac{1}{\omega}\sin\omega t\right), \qquad \omega = \frac{2\pi}{T}, \quad (17)$$

where T is the total time to open the hatch. Then the required control u^* is obtained as

$$u^* = \frac{\omega^2}{4\beta}\sin^2\omega t \quad (18)$$

This gives us the total drive force as

$$F_a = \frac{L_c}{L_p}\left[\frac{\omega^2}{4\beta}\sin^2\omega t + \cos\theta - \rho\lambda(1+\sigma^2)\sin^2\theta\right] \quad (19)$$
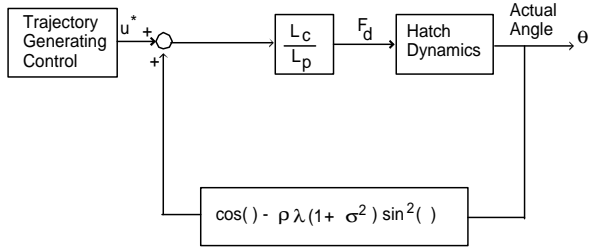
Figure 5   Trajectory Following Control System

Figure 6   Trajectory Following Control System with Error Correction for Noise

The configuration of the controller for the unperturbed system is shown in Figure 5. Note however that the system is perturbed by random wind and sea state so that additional compensation will be necessary.   As shown in Figure 6, the control signal for the drive is synthesized in three parts: a) a feedback component that compensates for load variations due to changes in hatch angular position, b) an error correction component that minimizes the randomness of wind loading and sea states, and c) a command generator for the desired trajectory of hatch motion. Recall that the effective load on the hatch varies with angular position of the hatch and the wind speed and direction.   Mean variations of wind loading is accommodated by the feedback linearzing block, and the error correction block provides additional compensation. A simple PID control could be used for the error compensation.

To design the PID controller, let

$$e_1 = \int (\theta - \theta^*) dt$$

$$e_2 = \theta - \theta^*$$

$$e_3 = \frac{d}{dt}(\theta - \theta^*)$$

where θ is the response of the actual system, equation (13), and θ^* is the response of the mean dynamics, equation (14). Also denote u as the control signal from the error correction block. Then the error dynamic is given by

$$\frac{d}{dt}\begin{bmatrix} e_1 \\ e_2 \\ e_3 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix}\begin{bmatrix} e_1 \\ e_2 \\ e_3 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \beta \end{bmatrix}u + \begin{bmatrix} 0 \\ 0 \\ \beta \end{bmatrix}\psi \qquad (20)$$

Denote

$$u = -\begin{bmatrix} k_i & k_p & k_d \end{bmatrix}\begin{bmatrix} e_1 \\ e_2 \\ e_3 \end{bmatrix} \qquad (21)$$

Various gains of the PID controller could be found using standard methods of design.
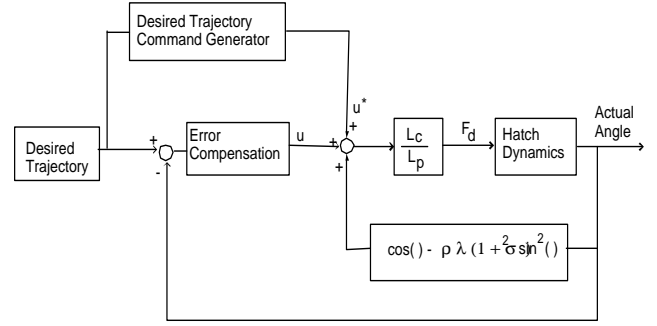
Practical implementation of the controller will require measurement of the hatch position and speed. Because of the specific geometry of the hatch control mechanism, a simple measurement of the shaft extension and its speed would suffice to estimate the hatch position and speed. The hatch position signal is required for implementation of the linearizing controller as well as error compensation block. Additional measurement of force on the hatch at the hinge could be taken to estimate the wind load and ship motion.

IV.   SIMULATION RESULTS

Control design was carried out using a desired hatch opening time of 10 seconds.   The gains of the error compensation block were taken as $k_p = 11$, $k_d = 7$, and $k_i = 2$. The random process ψ in equation (13) was simulated as a zero mean Gaussian white noise.   Initial simulations were done for a nominal wind load of 10% of the weight of the hatch, and variance of 0.05. The wind direction was assumed to be aiding the hatch opening. Figures 7 and 8 show the control of hatch-opening trajectory.
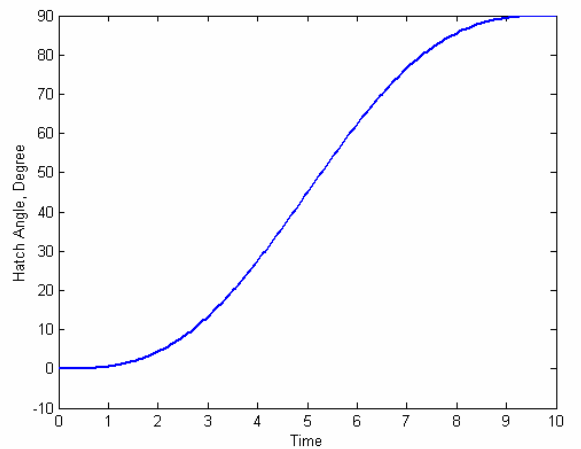

Figure 7   Opening of Hatch as a function of Time
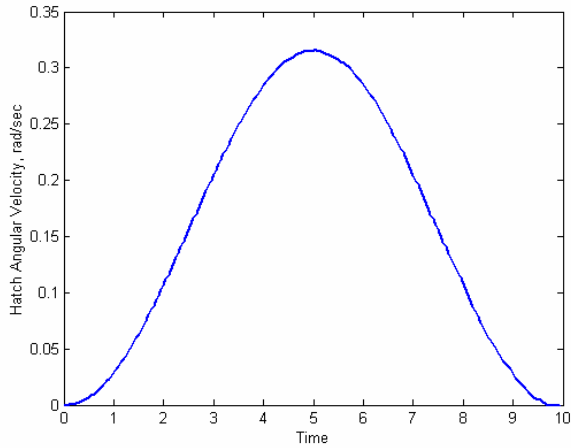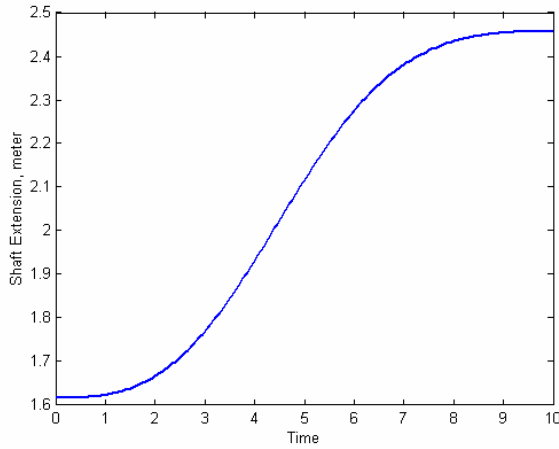
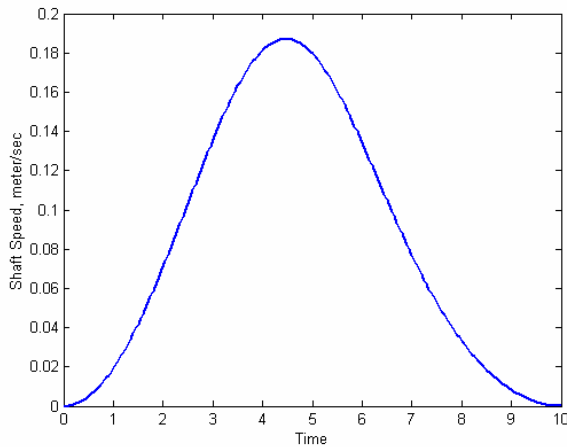Figure 8   Hatch Angular Speed

Figure 9   Shaft Extension

Figure 10 Shaft Speed

The hatch clearly followed the desired profile, and moved exactly along the desired trajectory to the full open position of 90 degrees. Figure 9 shows the shaft extension or the actuator length, AB (Figure 3), that initiates the hatch opening and the corresponding shaft velocity shown in Figure 10. Note that the drive shaft is in the retracted position when the hatch is closed, and is fully extended when the hatch is opened.

The effective load exerted on the actuator varies with angular position of the hatch. In particular, the load due to hatch weight is maximum when the hatch is closed or horizontal, and is at the minimum when the hatch is fully open. As such, we observe from Figure 11 that as the hatch opens, the actuating force reduces. In this figure, the drive force is normalized with respect to the weight of the hatch. Also note that when the hatch is near the open position, the actuating force is negative. Recall that the wind direction has been assumed to be assisting the opening the hatch, and secondly it is desired that a specific hatch position trajectory be followed. This means that a breaking force has to be applied if following a specific trajectory is required.
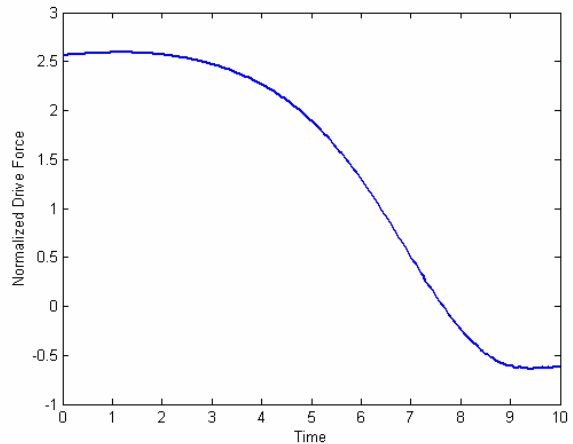

Figure 11   Drive Force

Simulations were repeated with wind direction to be obstructive to hatch opening. In Figure 12, we observe that a higher drive force is required if the wind load obstructs the hatch opening. The initial profiles of drive force in Figure 11 and Figure 12 are about the same, but in Figure 12 we observe that a higher drive force is required when the hatch is close to the full open position. This is due to the fact that wind load increases as the hatch opens. The hatch angle trajectory is similar to that of Figure 7.
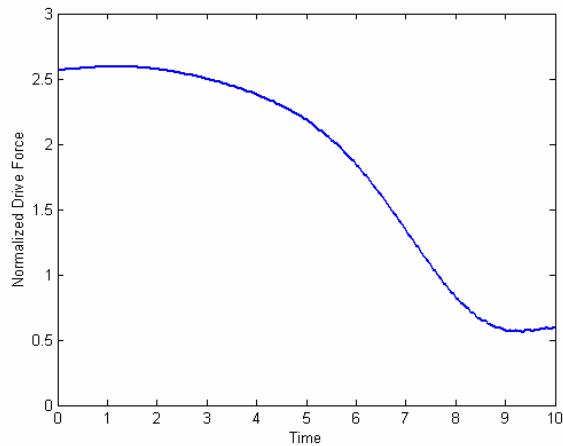
Figure 12 Drive Force for obstructive wind direction

Figure 13 shows the drive force for a higher wind load of 20% of the weight of the hatch with the wind direction obstructing the hatch opening. It is expected that the drive force must be higher than that shown in Figure 12. Also notice that drive force requirement at the beginning of the hatch opening is about the same, but a substantially higher drive force is required as the hatch opens.
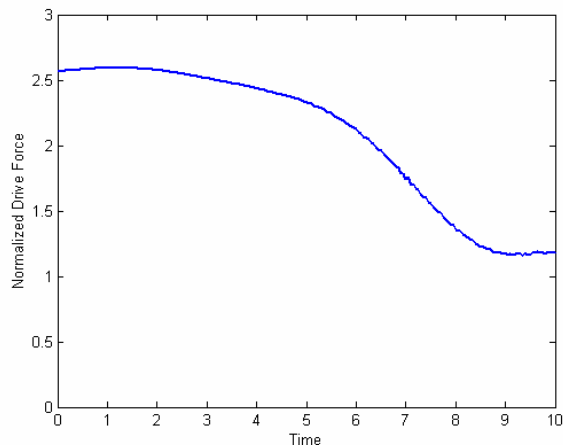

Figure 13   Drive force for higher wind speed.

Energy imparted on the hatch by the wind is proportional to the noise covariance. As such, it is expected that the drive force required to open the hatch for higher noise covariance will be higher. Figure 14 shows the drive force for a covariance of 0.1, but with the same mean speed direction as in Figure 13.
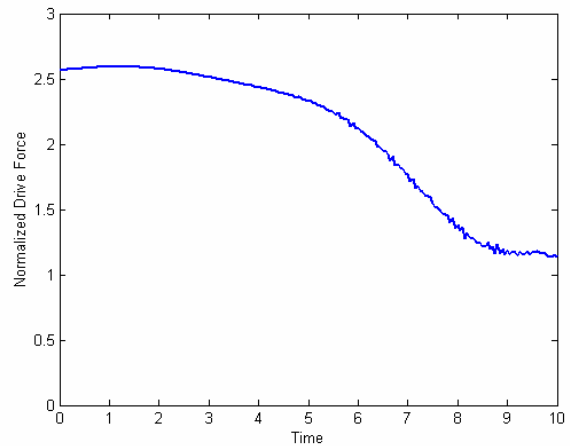

Figure 14   Hatch Angle for higher noise covariance

V.   CONCLUSIONS

A design of a stochastic linearizing controller for the control of a Hybrid Linear Actuator for the CV/CVN type flush deck-hatch is outlined. The control system consists of a simple feedback linearizing loop and a trajectory command generator. An error correction loop minimizes the effects of uncertainties due to randomness of wind and sea states. The controller requires sensing of only the hatch position and velocity. Simulation studies show that the hybrid linear actuator is capable of generating the required force, and drive the hatch according to a desired trajectory.

VI.   REFERENCES

[1]   Sendaula M., Teter J. P., and Crawford E.J., "Magnetostrictively Actuated Power Screw", Proceedings *of 2000 Naval Symposium on Electric Machines*, December 2000.

[2] Teter J.P., Sendaula M, Vranish J., and Crawford E.J., "Magnetostrictive Linear Motor Development", IEEE *Transactions on Magnetics,* Vol. 34, pp. 2081-2083, 1998.

[3] Boldea I., and Nasar S.A., *Linear Actuators and Generators*, Cambridge University Press, 1997.

[4] Biswas S.K., "Robust Stabilization of Linear Systems in the Presence of Gaussian Perturbation of Parameters", *Optimal Control Applications and Methods,* Vol. 19, pp. 271-286, 1998.

[5] Ahmed N.U., *"Elements of Finite Dimensional Systems and Control Theory",* Longman Publishers, 1988.