2013-01-2587
Published 10/14/2013

doi:10.4271/2013-01-2587
saepcmech.saejournals.org

# Robust Feedback Tracking Controller Design for Self-Energizing Clutch Actuator of Automated Manual Transmission

Jiwon Oh
KAIST

Jinsung Kim
Hyundai Motors

Seibum Choi
KAIST

**ABSTRACT**

This study mainly focuses on developing an accurate tracking controller for the self-energizing clutch actuator (SECA) system consisting of a DC motor with an encoder applied on the automated manual transmission (AMT). In the position-based actuation of the SECA, abruptly increasing torque near the clutch kissing point during the clutch engagement process induces control input saturation and jerky response when a conventional feedback controller is applied. The proposed work resolves such issue and significantly increases the control accuracy of the actuator through the development of an effective H-infinity controller. The control performance is shown to be more effective than a simple PID controller via simulation and experiments using an AMT test bench equipped with SECA aided by d-SPACE and Matlab/Simulink.

**CITATION:** Oh, J., Kim, J., and Choi, S., "Robust Feedback Tracking Controller Design for Self-Energizing Clutch Actuator of Automated Manual Transmission," *SAE Int. J. Passeng. Cars - Mech. Syst.* 6(3):2013, doi: 10.4271/2013-01-2587.

## INTRODUCTION

Vehicles are surely vital part of our lives, but they are also major sources of environmental issues. To improve the vehicle emission quality, researchers of various fields have attempted to develop more efficient engines, hybridization, weight reduction, etc. Among these, one of the technologies that can bring the most prominent improvement in vehicle energy efficiency and thus emission relates to power transmission.

Two of the most widely accepted transmission types are automatic transmission with a torque converter and traditional manual transmission. Both types involve distinct strengths and weaknesses.

The manual transmission has the advantage of light weight and high energy transmission efficiency. The power source and the wheels are mechanically linked through the clutch disks. However, it has the disadvantage of inconvenience coming from having to frequently maneuver the clutch pedal and gear selectors as the driver drives the vehicle.

On the other hand, the automatic transmission is known for its smooth gear shifting and convenience. Through the use of torque converter, it can operate without having to operate the clutch. However, such use of torque converter induces energy loss, since slipping is allowed in the driveline (despite the previous works to effectively control the lock-up clutch [1, 2, 3, 4]). Also, the unit cost is relatively higher than that of traditional manual transmission.

Such trade-offs that exist in the two transmission types inspired the researchers to develop a novel type of transmission called automated manual transmission (AMT) which can provide both energy efficiency and convenience [5, 6, 7]. Here, the clutch actuator and gear shifting mechanisms actuate the gear shifts to form an automated system.

Compared to the conventional AMT system, the AMT equipped with SECA [8, 9] uses smaller-size actuator. This is because, in the conventional system, the actuator alone must overcome the normal force great enough to prevent the mission from slipping when high amount of torque is applied to the driveline, whereas in the AMT with SECA, the self-energizing mechanism of the clutch hardware assists the motor when positive torque is applied. As a result, SECA enables cost and weight reduction and improves actuation efficiency, but involves higher sensitivity to actuator stroke.

Thus, to ensure robust actuator control performance, accurate actuation even under the influence of uncertainties is crucial, and it cannot be achieved with conventional PID control. The proposed study finds its resolution in h-infinity control method. This paper is structured as follows. Section I introduces the hardware and modeling of the AMT system equipped with SECA. Section II deals with formation of the test bench and how the empirical model used for h-infinity controller synthesis is induced. Section III shows the loop shaping process for h-infinity controller synthesis and analysis. Section IV displays the simulation/experimental results obtained by applying the suggested controller onto the test bench with SECA, and compares it with the result obtained by using the conventional PID control.

## AUTOMATED MANUAL TRANSMISSION WITH SELF-ENERGIZING CLUTCH ACTUATOR

One of the major weaknesses of the conventional AMT/DCT is in the actuator. To ensure sufficient normal force between the clutch disks for engagement even during high amount of torque flow, stiff diaphragm or coil springs are used. The proper actuation of the clutch system with such nature demands a high-power motor, which increases the required amount of control effort and weight. Here, the self-energizing clutch actuator system can be effective in eliminating such issues. SECA replaces the diaphragm spring of the conventional system, by designing the hardware in which the torque flow can work in a favorable way to increase the normal force in the clutch. In other words, torque applied onto the clutch induces the self-energizing clutch disk to telescope toward the direction of increasing normal force, so that the clutch disks can be engaged even with a small amount of actuation force. Such mechanism allows the amount of required actuation energy to be significantly reduced, which can lead to the optimization of the transmission both in its efficiency and weight, and at the same time securing the dramatic fuel efficiency improvement.

Design principles of the SECA is shown in fig. 1. It is designed so that a motor pushes the actuation lever (i), which in turn presses onto the actuator (ii) which is then turned into a rotational telescoping action (iii) that causes the clutch plate to engage (iv) according to the inclined gear shown in (b).

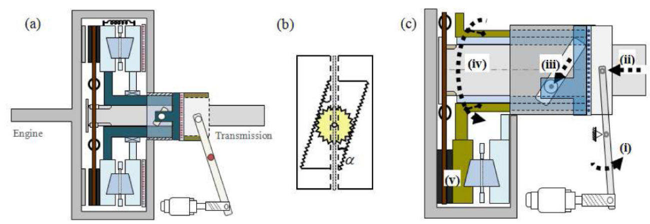


***Figure 1. (a) Sectional drawing of the self-energizing clutch actuator system, (b) implementation of the self-energizing mechanism by installing the pinion guide gears between the actuation and fixed plate, (c) operating schematics of the clutch actuation for engagement.***

Further details and specification of the SECA system can be found in [8].

A comprehensive model of the AMT with SECA is required to perform the simulation to test the controller performance.

$$\dot{i}_m = -\frac{R_m}{L_m} i_m - \frac{k_m}{L_m}\omega_m + u \tag{1}$$

$$\dot{\omega}_m = \frac{k_t}{J_m} i_m - \frac{1}{J_m N_g} T_a - T_{fm}(\omega_m) \tag{2}$$

Equation (1) and (2) indicate the dynamic model of the DC motor used in the actuator. Here, the friction $T_{fm}$ $(\omega_m)$ within the actuator is modeled using the static coulomb and viscous effects. This is shown in the following.

$$T_{fm}(\omega_m) = \begin{cases} T_{cm+} + b_{m+}\omega_m, & \omega_m > \varepsilon_m \\ T_{cm-} + b_{m-}\omega_m, & \omega_m < -\varepsilon_m \\ T_m, & |\omega_m| < \varepsilon_m \text{ and } |T_{fm}| < T_{fsm} \\ T_{fsm}\,\mathrm{sgn}(T_m), & |\omega_m| < \varepsilon_m \text{ and } |T_{fm}| > T_{fsm} \end{cases} \tag{3}$$

Here, $T_{fsm}$ and $T_{cm}$ are the static and coulomb frictional torques, respectively, $b_m$ is viscous damping coefficient, and $\varepsilon_m$ is a threshold between the static and coulomb friction.

An illustration of the mechanical part of the SECA is shown in fig. 2.

When the clutch is completely disengaged, the following dynamics applies to the system.

$$\dot{\omega}_a = \frac{1}{J_a} T_a - \frac{1}{J_a} T_{fa}(\omega_a) \tag{4}$$

where no external torque other than the actuation torque and friction torque is involved. Here, the actuation torque can be expressed as follows.

$$T_a = k_a\left(\frac{\theta_m}{i_g} - \theta_a\right) \tag{5}$$

Where $i_g$ and $k_a$ are the gear ratio of motor to the actuation lever, and the equivalent actuation lever stiffness, respectively. Also, similar to how the motor friction is modeled, the actuator friction is modeled as the following.

$$T_{fa}(\omega_a) = \begin{cases} T_{ca+} + b_{a+}\omega_a, & \omega_a > \varepsilon_a \\ T_{ca-} + b_{a-}\omega_a, & \omega_a < -\varepsilon_a \\ T_a, & |\omega_a| < \varepsilon_a \text{ and } |T_{fa}| < T_{fsa} \\ T_{fsa}\,\mathrm{sgn}(T_a), & |\omega_a| < \varepsilon_a \text{ and } |T_{fa}| > T_{fsa} \end{cases} \tag{6}$$

Again, the terms are named analogously to those of the motor friction.

Now, as the clutch begins to engage, another term must be included to express the transmitted clutch torque.

$$\dot{\omega}_a = \frac{1}{J_a}T_a + \frac{1}{J_a}T_c - \frac{1}{J_a}T_s - \frac{1}{J_a}T_{fa}(\omega_a) \tag{7}$$

The added terms $T_c$ and $T_s$ each represent the clutch friction torque and the self-energizing torque. Apart from the friction torque as a loss, the clutch friction torque is the torque that is actually transmitted to the engaged clutch. The self-energizing torque arises from the structural design of the SECA which works in the direction of locking the clutches together. Both are functions of the clutch normal force, and thus they can be expressed as follows.

$$T_c = \mu R_c F_n \tag{8}$$

where $R_c$ is the effective clutch radius, and

$$T_s = 2 r_p F_p \sin\alpha \tag{9}$$

Here, since the clutch normal force and the self-energizing reaction force can be expressed as

$$F_n = k_p x_p = 2 k_p r_p \theta_a \tan\alpha \tag{10}$$

$$F_p = \frac{F_n}{\cos\alpha} \tag{11}$$

equation (7) can be rewritten as the following.

$$\dot{\omega}_a = \frac{1}{J_a}T_a + \beta\theta_a - \frac{1}{J_a}T_{fa}(\omega_a),$$
$$\text{where } \beta = \frac{2\mu R_c k_p r_p \tan\alpha - 4 r_p k_p r_p \tan^2\alpha}{J_a} \tag{12}$$

Notice how $\beta$ can simply be zero to reach (4) in case the clutch is disengaged.

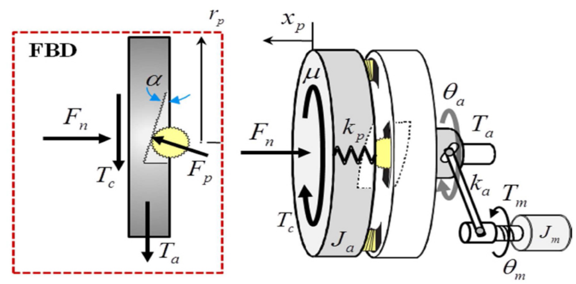


*Figure 2. Illustration of the self-energizing clutch actuator mechanism*

# TEST BENCH FREQUENCY ANALYSIS

The test bench for control performance test as well as the model for simulation is prepared. As shown in fig. 3, the test bench is realized by using an AC motor which performs like an engine, and the fixed-gear transmission with a clutch equipped with the self-energizing actuator.

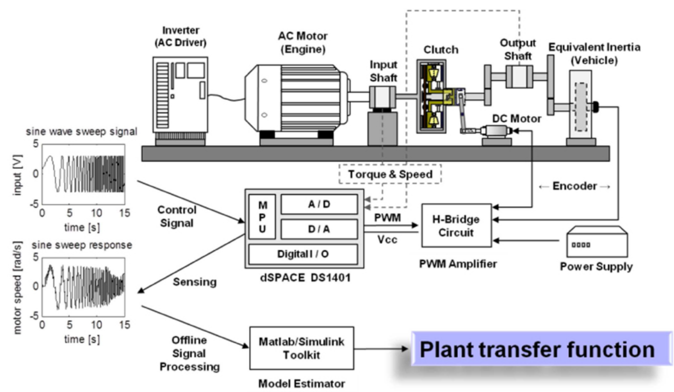


*Figure 3. Illustration of the test bench components and the frequency characteristic identification procedures*

The procedural diagram in fig. 3 also shows the process of identifying the actuator characteristics on frequency domain. With a sine wave sweep signal applied onto the controller as an open-loop voltage input to the DC motor, the motor speed as a response is recorded using an encoder.

Mean frequency response of the repeated analysis is taken as the empirical model. As shown in fig. 4, fine lines represent the results of the repeated frequency analysis of the driveline with SECA, and the bold line represents the mean model obtained from the previous analysis.

The model estimation was performed using Matlab/ Simulink tool, and the model order was decided to be sufficiently high to cover all major dynamics of the actuator.

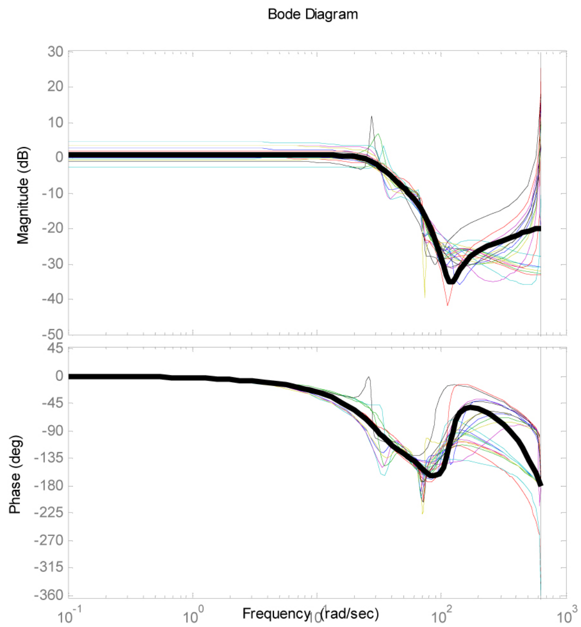


***Figure 4. Plots of frequency analysis results***

# H-INFINITY CONTROLLER DESIGN

The H-infinity controller is chosen for this system, since the friction property in the clutch can be extremely tedious to develop an accurate model due to the effects of long-term wear or temperature variation. By using the robust controller, accurate tracking control performance can be maintained even under the friction uncertainty induced by such nature. The block diagram that describes the basic structure of the H-infinity controller used in the suggested work is shown in fig. 5. Here, the shaping filters $W_1$ and $W_2$ are carefully selected and applied to ensure the robustness of the feedback system.

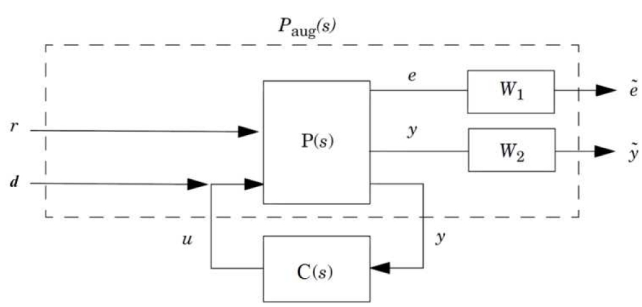


***Figure 5. Basic structure of the H-infinity controller system***

For the loop shaping purpose, the following functions are defined.

Open loop transfer fuction:

Open loop transfer fuction: $$L = PC \tag{13}$$

Sensitivity function: $$S = \frac{1}{1+PC} \tag{14}$$

Complementary sensitivity function: $$T = \frac{PC}{1+PC} \tag{15}$$

Now we let the performance shaping factor $W_1 = S_{ref}^{-1}$, where $S_{ref}(j\omega)$ is the target performance (error upperbound). Also, we let the actual plant with uncertainty be described by the multiplicative uncertainty, namely,

$$M(P_0, W_2) = \left\{ \begin{array}{l} P_0(1+W_2\Delta) : \Delta \in \mathbb{R}, \sup_{\omega\in\mathbb{R}} |\Delta(j\omega)| < 1, \\ \#\text{ of RHP pole}(P_0) = \#\text{ of RHP pole}(P_0(1+W_2\Delta)) \end{array} \right\} \tag{16}$$

where $W_2$ $(j\omega)$ is the uncertainty weighting function. Then, given the four transfer functions $P_0$, $C$, $W_1$, and $W_2$ with a $C$ stabilizing $P_0$, the robust stability is ensured if and only if

$$\left\| \frac{W_2 PC}{1+PC} \right\|_\infty \le 1 \tag{17}$$

and the robust performance is ensured, i.e.,

$$\sup_{P\in M(P_0,W_2)} \left\| \frac{W_1}{1+PC} \right\|_\infty \le 1 \tag{18}$$

if and only if

$$\sup_{\omega\in\mathbb{R}} \left[ \left| \frac{W_1}{1+PC}(j\omega) \right| + \left| \frac{W_2 PC}{1+PC}(j\omega) \right| \right] \le 1 \tag{19}$$

This can be graphically shown using a nyquist plot in fig. 6.

Taking the system characteristics into consideration, the shaping filters are designed using second order functions. both the performance sensitivity function and the uncertainty weighting are designed to be strict up to the frequency of 10 to 100 rad/sec, but they are loosened in the higher frequency range. Such criteria fit effectively with the actual clutch control bandwidth. The shaping filters are plotted in fig. 7.

The actual H-infinity controller synthesis is formulated using the Matlab commad - hinflmi - based on the augmented system. Fig. 8 shows that the robust stability and performance criterion stated in (17) and (19) are met indeed, for all frequency domain.

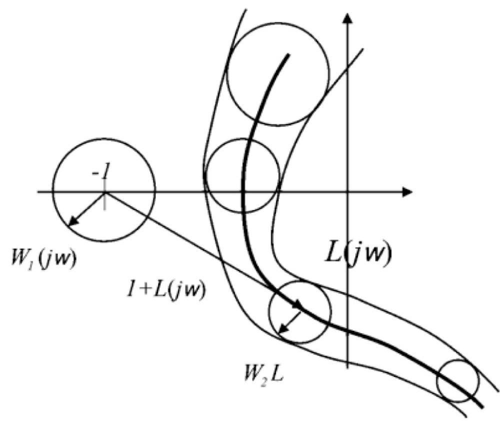


***Figure 6. Nyquist plot showing the graphic representations of the loop transfer function and the shaping filters of a stable system***

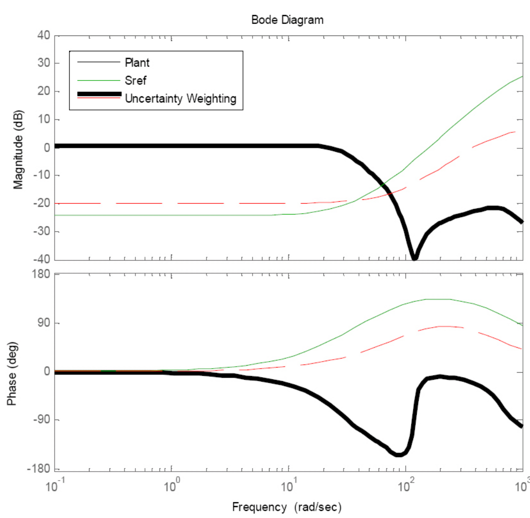


***Figure 7. Bode diagrams of the plants and the shaping filters consisting the augmented system***

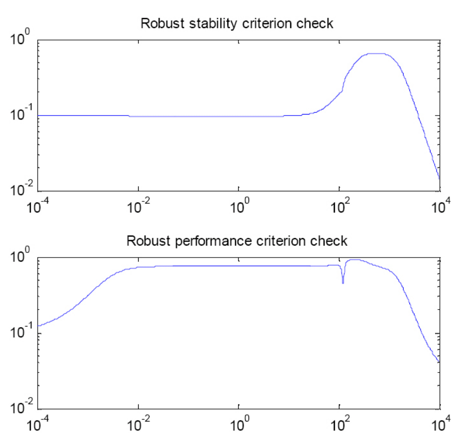


***Figure 8. Plots of the robust stability and performance criterion indices***

# CONTROLLER PERFORMANCE VALIDATION

## Simulation Results

The H-infinity controller is applied to the model developed in the previous section in order to test the controller performance by simulation. To simulate the actual system, measurement noises are intentional added to the signals.

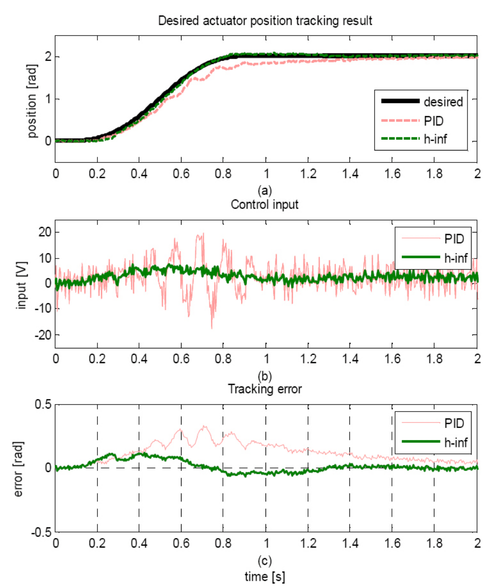


***Figure 9. Step reference (with rate limiter and low pass filter applied) by simulation: (a) tracking control results, (b) control inputs, (c) tracking errors***

In both step and sine wave reference inputs shown in fig. 9 and 11, the suggested H-infinity controller much more effectively tracks the desired inputs. Here, not only the tracking error of the suggested controller is smaller than that of the PID controller, but the amount of control input magnitude and oscillation of the suggested work is smaller

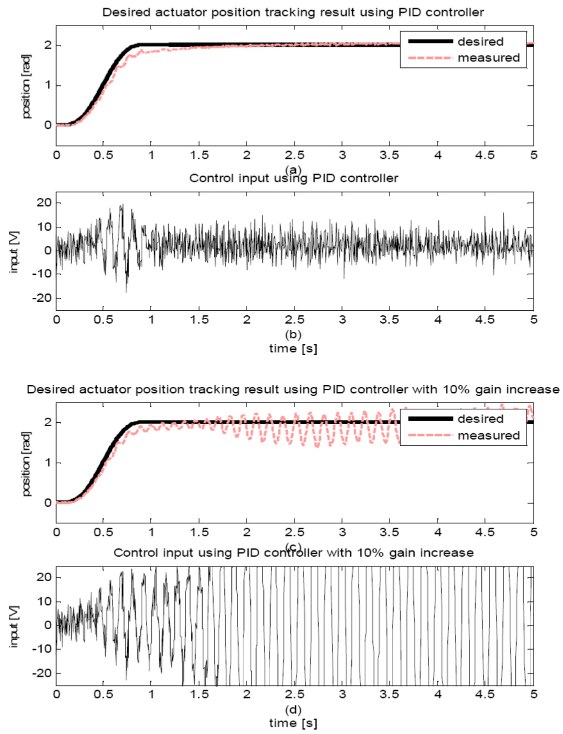


***Figure 10. PID controller performance for step reference by simulation: (a) tracking control result, (b) control input, (c) tracking control result with 10% increased controller gain, (d) control input with 10% increased controller gain***

than that of the conventional PID controller. Here, the gains of the conventional PID controller are tuned so that the performance level achieved by this controller can be maximized. As evidence, fig. 10 plot (c) and (d) show how the feedback system fails to stabilize itself with the PID controller gains just 10% greater than those of the cases shown in fig. 10 plot (a) and (b).

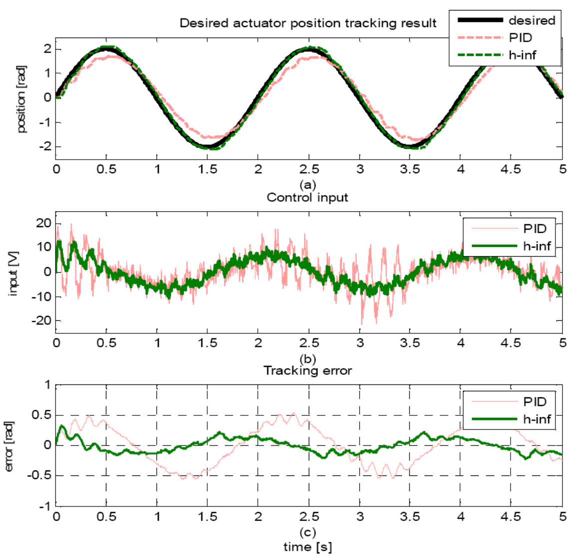


***Figure 11. Sine wave reference by simulation: (a) tracking control results, (b) control inputs, (c) tracking errors***

# Experiment Results

In addition to the simulation, actual experiments using the test bench with SECA are performed to practically and thoroughly validate the controller performance.

As done for the tests by simulation, PID gains are again tuned to maximize the conventional controller's performance level. The results of the PID controller tuning are shown in fig. 12, where only 10% increase in the PID gains causes the system to be unstable.

Fig. 13 and 14 compare the tracking test results of the conventional PID controller and the suggested controller with a step clutch engagement command. This includes the case of clutch engagement, which involves the model uncertainty of friction. With the suggested controller, the actuator follows after the desired step input as swift as the actuator physical limit permits, whereas with the conventional controller, the tracking speed is relatively slow. Hence the robust tracking performance of the suggested H-infinity controller is shown. Also, it can be seen that the suggested controller eliminates the jerky response exhibited by the conventional controller, especially near the clutch engagement disturbance that exists around 7s. Lastly, fig. 15 shows the smooth tracking ability of the suggested controller when an actual clutch actuation profile is given to swiftly engage and disengage the clutch.

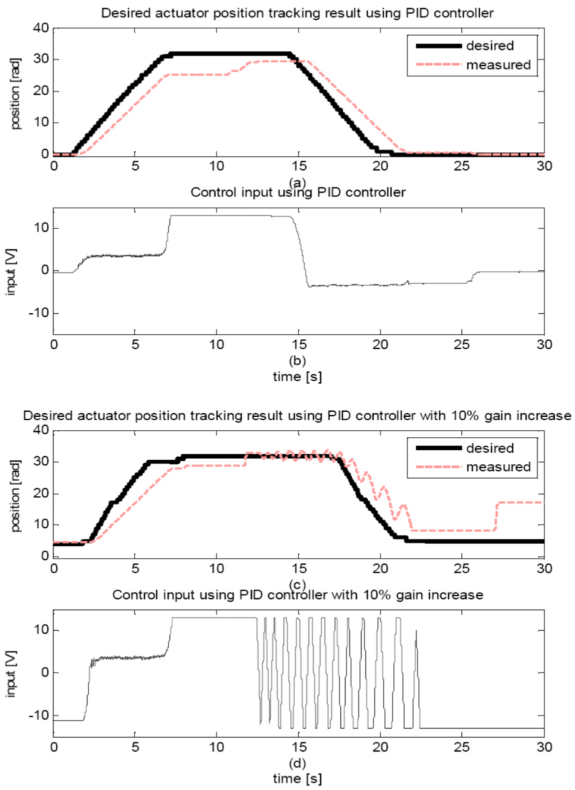


***Figure 12. PID controller performance for ramp reference by experiment: (a) tracking control result, (b) control input, (c) tracking control result with 10% increased controller gain, (d) control input with 10% increased controller gain***

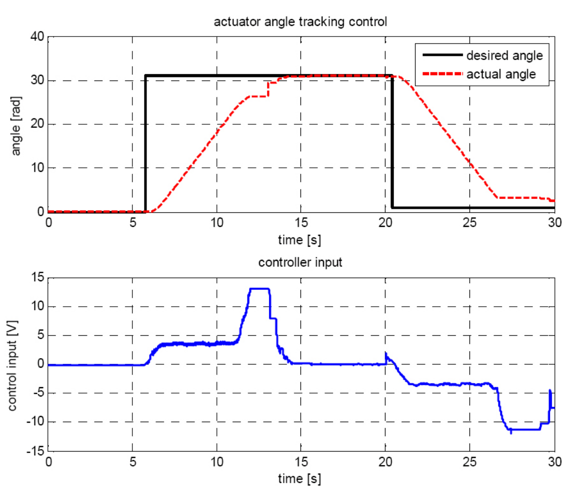


***Figure 13. PID controller performance for step clutch engagement command by experiment***

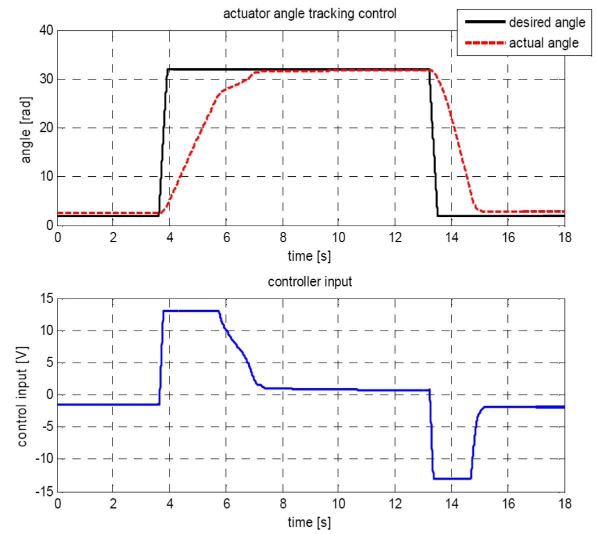


***Figure 14. Suggested controller performance for step clutch engagement command by experiment***

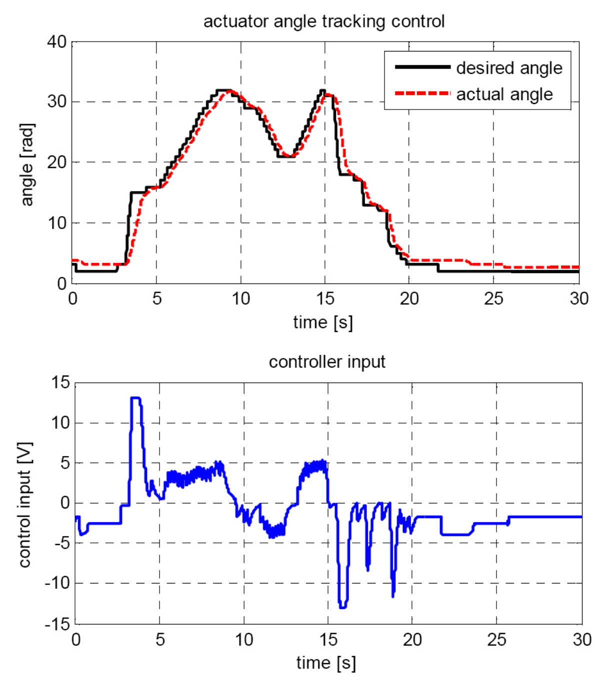


***Figure 15. Suggested controller performance for actual clutch engagement command by experiment***

# CONCLUSION

This study has proposed an effective controller for the motor-driven self-energizing clutch actuator, which shows robust performance even under high amount of disturbance which comes from the sudden engagement or disengagement of the clutch. Through empirically analyzing the actuator system in frequency domain, actuator response is used to synthesize a robust H-infinity controller. Summarizing the paper, noteworthy contributions of the work are the following: improvement of the actuator tracking control accuracy in general, reduction of the amount of control voltage fluctuation, effective rejection of the sudden disturbance, and reduction of the jerky response of the clutch actuator. The proposed controller has been verified to be effective both through simulation using the modeled driveline, and experiments using an actual test bench with SECA.

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## ACKNOWLEDGMENTS

This work was supported by the National Research Foundation of Korea (NRF) grant funded by the Korea government (MSIP) (No. 2010-0028680), Global Ph. D. Fellowship Program (NRF-2012H1A2A1048938), and CITRC(Convergence Information Technology Research Center) support program (NIPA-2013-H0401-13-1008) supervised by the NIPA(National IT Industry Promotion Agency).