Actuator compensated UDE based robust Roll Autopilot design

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***Abstract*—Design of robust autopilots along with actuator dynamics compensation continues to be a challenging task. Few works are available in the literature to address this issue. In this work an attempt has been made to design a robust roll autopilot using the technique of uncertainty and disturbance estimator, assuming ideal actuator. Further, considering a second order actuator, three methods have been proposed for actuator compensation. Numerical simulations have also been carried out to ascertain the efficacy of the proposed methods.**

I. Introduction

Design of autopilots for aerospace vehicles has been in vogue for the past few decades. Analysis of motion of aerospace vehicles, particularly at high speeds and angle of attacks, is a complex task, due to the inherent cross-coupling between pitch and yaw channels due to inadvertent rolling of the vehicle. Hence the design of autopilots is also a challenging activity. Roll autopilots are primarily designed to ensure the desired roll stabilization to prevent cross-coupling.

Plenty of research has gone into the design of robust autopilots, broadly, assuming linear and nonlinear mathematical models. To mention a few works available in literature, [1] discusses a new approach in motion modeling and subsequent design of autopilots for skid-to-turn missiles. Control based on H_∞ synthesis has been employed in [2] for the design of integrated Roll-Pitch-Yaw robust autopilot. Towards robust roll autopilot, two design strategies based on disturbance estimation have been proposed in [3]. Authors in [4] have proposed a new reaching law based on the conventional exponential reaching law while employing variable structure control strategy. The widely popular disturbance observer - based control (DOBC), in its nonlinear form can be seen in [5]. Authors in [6] have proposed a novel roll autopilot to reduce the cross coupling in skid-to-turn missiles and employed the concept of uncertainty and disturbance estimator (UDE) to improve its robustness.

Generally, while designing autopilots, researchers tend to assume ideal actuators and concentrate on the robustness. However it has been realized in practical situations, actuators play a major role in deciding the stability [7]. Compensation for the actuator dynamics (first as well as second order) by means of a compensator designed using back-stepping technique has been discussed in [8]. In [9], the authors had designed a robust control law based on sliding mode technique considering second order actuator dynamics. A robust roll autopilot using model following concept and at the same time employing a second order actuator has been presented in [10]. A linear ESO based design for robust roll autopilot used in tactical missiles can be found in [11]. Recently, in [12], authors have proposed a robust roll autopilot design employing a higher order sliding mode using super twisting algorithm while considering a first order actuator.

The technique of uncertainty and disturbance estimator (UDE) [13] utilizes a first order filter of sufficient bandwidth encompassing the uncertainties and disturbances; estimates them in an integrated manner and enables the estimate to be used in the control law to mitigate their effects. Unlike other robust control laws, this technique does not require the magnitude of uncertainties and disturbances except their bandwidth.

Since UDE has established itself to be a viable robust control strategy, it is felt to examine its efficacy in dealing with actuator compensation. In this work, the same has been examined and evaluated. Further, two more actuator compensation strategies are proposed and tested for their performance. This paper is organized as follows. Section II deals with mathematical modeling, design of UDE based control law and performance study of the proposed control law for ideal as well as second order actuator. Actuator compensation by adopting three different strategies and their performance evaluation are the topics of discussion in Section III. Finally, Section IV concludes this work.

II. Design of UDE based robust control law and its performance study

The rigid body dynamics for a tactical missile as referred from [11] can be represented through the following transfer function

$$\frac{\phi(s)}{\delta(s)} = \frac{K_\delta}{s(s+\omega_{RR})} \tag{1}$$

where ϕ is the roll angle, δ is the aileron deflection (control effort), K_δ is the fin effectiveness and ω_{RR} is the roll rate bandwidth. The schematic representation of the system defined above is depicted in Fig. 1. Here, the plant is assumed to be driven through an ideal actuator, thus for this system $\delta = \delta_c$, where δ_c is the commanded aileron deflection.

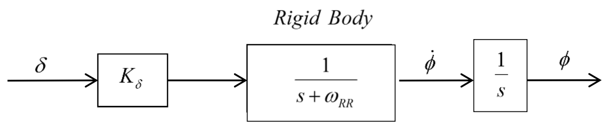


Fig. 1. Block Diagram Schematic of roll dynamics

A. Design of UDE based control law

As stated above, for an ideal actuator $\delta = \delta_c$. Now, defining the states x_1 and x_2 to be ϕ and p, respectively, the state space representation for (1) is as follows;

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= -\omega_{RR}x_2 + K_\delta \delta \end{aligned} \tag{2}$$

Taking into account, the uncertainties in ω_{RR} and K_δ and external disturbance (d_{ext}) acting on the system, we can express the actual value of roll rate bandwidth as $\omega_{RR} = \omega_{RRn} + \Delta\omega_{RR}$ and actual fin effectiveness as $K_\delta = K_{\delta n} + \Delta K_\delta$, where $K_{\delta n}$ and ω_{RRn} are the nominal values, with the uncertainties being expressed by ΔK_δ and $\Delta\omega_{RR}$. The dynamics given in (2) can thus be re-written as

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= -(\omega_{RRn} + \Delta\omega_{RR})x_2 + (K_{\delta n} + \Delta K_\delta)\delta + d_{ext} \end{aligned} \tag{3}$$

Clubbing the uncertainties and the external disturbance terms in (3), defining $d = -\Delta\omega_{RR}x_2 + \Delta K_\delta \delta + d_{ext}$; the dynamics of the plant are now given by

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= -\omega_{RRn}x_2 + K_{\delta n}\delta + d \end{aligned} \tag{4}$$

With δ as the control input (u), the control law is proposed as

$$u = \frac{1}{K_{\delta n}}\left[u_a + u_d + \nu\right] \tag{5}$$

where $u_a = \omega_{RRn}x_2$ and ν is that part of the control law which is responsible for driving the system to the desired dynamics. ν is defined as;

$$\nu = \ddot{\phi}_{ref} - m_1(\dot{\phi} - \dot{\phi}_{ref}) - m_2(\phi - \phi_{ref}) \tag{6}$$

where ϕ_{ref} is the desired output or the reference value, with ϕ being the actual roll angle value. m_1 and m_2 are feedback gains whose values can be chosen as per the desired characteristics. With u_a and ν having been defined, we now need to design u_d to cater to the effects of d. As referred in [13], we define $\hat{d}$ to be the estimate of d and relate the two as follows;

$$\hat{d} = \left(\frac{1}{1+s\tau}\right)d \tag{7}$$

where τ is the time constant of the first order filter. τ must be chosen to have a sufficient bandwidth, such that it nullifies the effect of the lumped disturbance d. From (4), d can be defined as;

$$d = \dot{x}_2 + \omega_{RRn}x_2 - K_{\delta n}u \tag{8}$$

Using (7) and (8), we express $\hat{d}$ as

$$\hat{d} = \left(\frac{1}{1+s\tau}\right)(\dot{x}_2 + \omega_{RRn}x_2 - K_{\delta n}u) \tag{9}$$

Taking $u_d = -\hat{d}$, after few algebraic manipulations, the closed form expression for u_d from (9) can be derived as;

$$u_d = -\frac{1}{\tau}\left[x_2 - \int \nu dt\right] \tag{10}$$

The final UDE based robust control law using (5), (6) and (10) can be written as

$$u = \frac{1}{K_{\delta n}}\left[\omega_{RRn}x_2 - \frac{1}{\tau}\left[x_2 - \int \nu dt\right] + \nu\right] \tag{11}$$

Since the stability analysis for the UDE based control law for similar systems is available in the literature, it has been omitted.

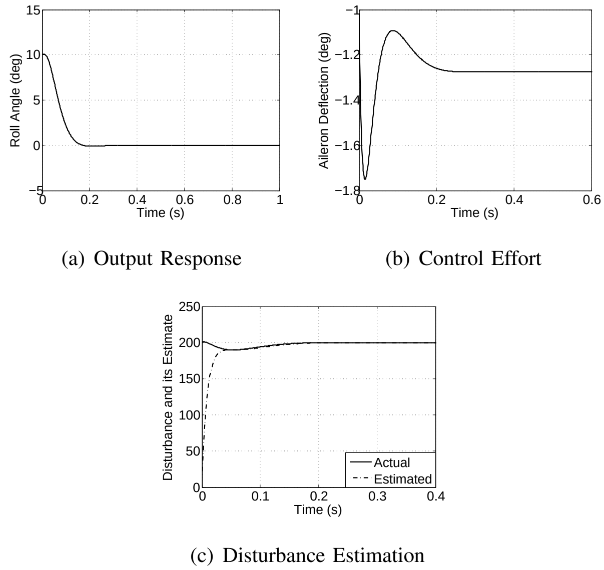
B. Performance analysis of UDE based controller

To analyze the performance of the plant with an ideal actuator in accordance with the UDE based control law given in (11), the simulation parameters for the considered tactical missile were chosen using Table. 1 of [11]. τ was chosen to be 0.01. With a desired settling time of 180 ms and damping factor of 0.8, the feedback gains m_1 and m_2 were evaluated to be 42.45 and 771.13, respectively. Simulations were carried out with ϕ having an initial condition of 10 deg. To ascertain the stabilization capability of control law in (11) in presence of the external disturbance d_{ext} and an uncertainty in ω_{RR} simulations were carried out. Taking ω_{RR} to be -3 rad/s , the simulation results are presented in Fig. 2. The results confirm the satisfactory performance of the proposed control law (11) with a phase margin of 69 deg. A similar performance was also seen when $\omega_{RR} = +3$ rad/s.

TABLE I
PERFORMANCE SPECIFICATIONS

ω_{RR}	roll rate bandwidth	2 rad/s
K_δ	fin effectiveness	9000 $1/s^2$
ω_A	actuator bandwidth	100 rad/s
ζ_A	actuator damping	0.65
ϕ_{max}	maximum desired roll angle	10 deg
$\dot{\phi}_{max}$	maximum desired roll rate	300 deg/s
d_{ext}	external disturbance	200 rad/s^2
δ_{cmax}	maximum desired fin deflection	30 deg

UDE based control law thus exhibits a satisfactory performance in the presence of external disturbances as well as uncertainties, assuming ideal actuator.



(a) Output Response (b) Control Effort

(c) Disturbance Estimation

Fig. 2. Performance of UDE based controller with ideal actuator in presence of disturbances and uncertainties

C. Performance of UDE based controller with second order actuator in the loop

An actuator modeled as a second order transfer function, as referred in [11], of the form $\frac{\delta(s)}{\delta_c(s)} = \frac{\omega_A^2}{s^2+2\zeta_A\omega_A s+\omega_A^2}$ is introduced in the plant, as illustrated in Fig. 3. Here ω_A is the actuator bandwidth and ζ_A is the actuator damping ratio. Simulations were carried out with the parameter values as specified in Table. 1, using the control law defined in (11), with τ as 0.01. In this case only disturbance of d_{ext} has been considered. Since actuator dynamics included here were unaccounted for in the design of control law in (11), the performance of the system does not remain the same as in previous case, as can be seen in Fig. 4.

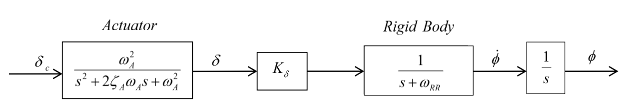
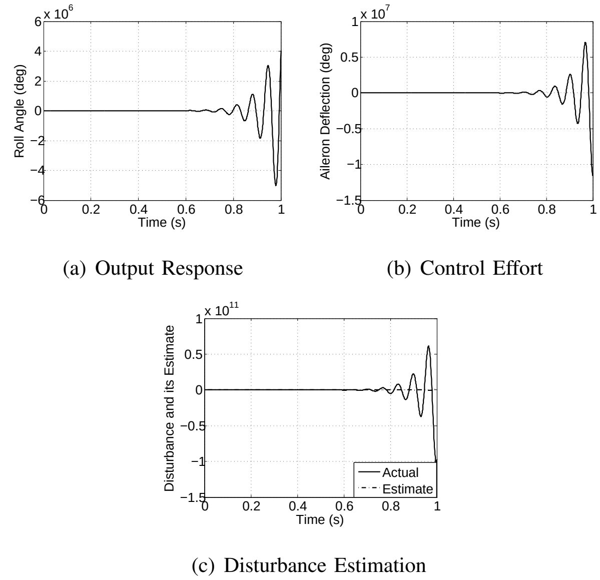


Fig. 3. Schematic of roll dynamics with second order actuator

III. Actuator compensated UDE based control law

It has been noted that the addition of a second order actuator to the plant has rendered the system unstable. The reason for this instability is an additional lag introduced by the actuator. Hence there is a need to compensate the effects of the actuator by proper tuning the UDE based robust control law or design a compensator for the actuator. The attempts thus made are explained in the subsequent discussion.



(a) Output Response (b) Control Effort

(c) Disturbance Estimation

Fig. 4. Performance of system with second order actuator

A. Method 1: Tuning of τ

As inferred from Fig. 3, the relation between δ and δ_c is as follows

$$\delta = \delta_c - \left[\frac{1}{\omega_A^2}\left(\ddot{\delta} + 2\zeta_A\omega_A\dot{\delta}\right)\right] \tag{12}$$

As δ_c is the only control available for design purposes, the plant as expressed in (4) can be represented as

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= -\omega_{RRn}x_2 + K_{\delta n}\left[\delta_c - \frac{1}{\omega_A^2}\left(\ddot{\delta} + 2\zeta_A\omega_A\dot{\delta}\right)\right] \\ &\quad +d \end{aligned} \tag{13}$$

Since $\frac{-K_{\delta}n}{\omega_A^2}\left(\ddot{\delta} + 2\zeta_A\omega_A\dot{\delta}\right)$ is unknown, it can be treated as an unknown disturbance and clubbed with d and denoted as d_1.

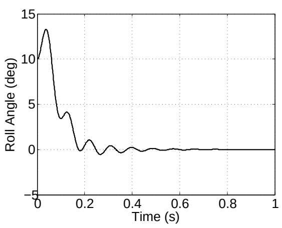
$$d_1 = \frac{-K_{\delta}n}{\omega_A^2}\left(\ddot{\delta} + 2\zeta_A\omega_A\dot{\delta}\right) + d \tag{14}$$

The resultant plant dynamics thus become;

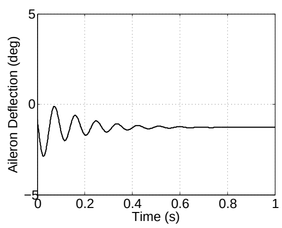
$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= -\omega_{RRn}x_2 + K_{\delta n}\delta_c + d_1 \end{aligned} \tag{15}$$

For the system (15), the control law in (5) remains the same, with $u_a = \omega_{RRn}x_2$, $\nu = \ddot{\phi}_{ref} - m_1(\dot{\phi} - \dot{\phi}_{ref}) - m_2(\phi - \phi_{ref})$ and $u_d = -\frac{1}{\tau}\left[x_2 - \int \nu dt\right]$. The feedback gains m_1 and m_2 also remain the same. It is to be noted that the plant continues to be driven through δ, in keeping with the physical structure of the system. Unlike previous discussions, the control term u_d now caters to the additional disturbance term $\frac{-K_{\delta}n}{\omega_A^2}\left(\ddot{\delta} + 2\zeta_A\omega_A\dot{\delta}\right)$ in addition to d. This calls for careful choice of τ to take care of d_1.

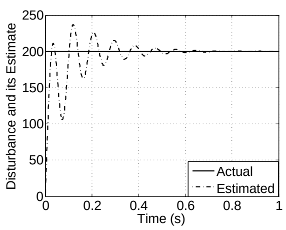
1) Performance analysis of UDE based controller with actuator compensation using Method 1: Simulations were carried out with the data given in Table. 1, along with an external disturbance d_{ext}. The time constant τ when chosen to be 0.01, rendered the system unstable. Hence it was varied to achieve the desired performance. It was observed that as the value of τ increases from 0.01 till 0.025, the system remained unstable, as in Fig. 4. On further increasing the value of τ, the system gradually began to stabilize. The value of τ which gave the best possible performance was $\tau = 0.04$, the results of which can be seen in Fig. 5.



(a) Output Response



(b) Control Effort



(c) Disturbance Estimation

Fig. 5. Performance with Method 1

From Fig. 5a, it can be seen that the roll angle does not show a smooth descent to equilibrium. An overshoot from its initial condition of 10 deg was also observed, which was dependent on the magnitude of the disturbance. Disturbance estimation was found to be satisfactory after 500 ms and the control effort was found to be oscillatory, though within acceptable limits. It can thus be inferred, that tuning of τ did help the system in attaining stability but not to the desired extent, as was shown in Fig. 2.

B. Method 2: Using Compensator Form 1

Due to the presence of a second order actuator in the loop, there is a lag between δ and δ_c. The primary aim is to ensure that δ tracks δ_c in a sufficiently small time interval which would require a compensator, as inferred from [8]. In this direction, an attempt has been made to design a compensator, cascaded with the actuator, to reduce the delay. The schematic is shown in Fig. 6 The transfer function of the actuator, as given in Fig. 3 can be written as;

$$\ddot{\delta} = -2\zeta_A\omega_A\dot{\delta} - \omega_A^2\delta + \omega_A^2\delta_c \tag{16}$$

Unlike the previous approach where the tuning of τ was done in the control input u or δ_c, here δ_c derived in (11), is taken

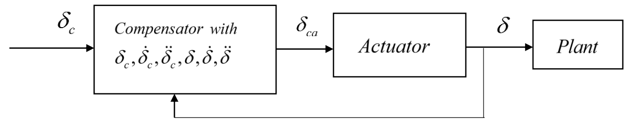


Fig. 6. Block Diagram Schematic for Method 2

as a reference signal which δ is required to track before it is fed to the plant, as depicted in Fig. 6. Denoting δ_{ca} to be the output of the compensator, (16) can be re-written as;

$$\ddot{\delta} = -2\zeta_A\omega_A\dot{\delta} - \omega_A^2\delta + \omega_A^2\delta_{ca} \tag{17}$$

where δ_{ca} is the compensated control to the actuator and is designed as follows;

$$\delta_{ca} = \frac{1}{\omega_A^2}(\nu_a + u_{na}) \tag{18}$$

Here u_{na} is chosen to be equal to $2\zeta_A\omega_A\dot{\delta} + \omega_A^2\delta$. Substituting this in (17) we get;

$$\ddot{\delta} = \nu_a \tag{19}$$

In order to ensure that δ tracks δ_c, ν_a is defined as follows;

$$\nu_a = \ddot{\delta}_c - k_{a1}\dot{\delta}_e - k_{a2}\delta_e \tag{20}$$

Taking $\delta_e = \delta_c - \delta$, we can infer that $\dot{\delta}_e = \dot{\delta}_c - \dot{\delta}$ and $\ddot{\delta}_e = \ddot{\delta}_c - \ddot{\delta}$. Application of (20) into (19), results in a tracking error dynamics of the form;

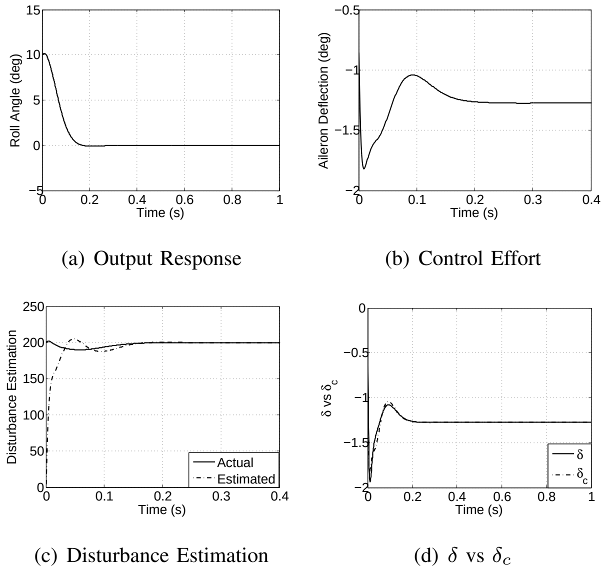
$$\ddot{\delta}_e + k_{a1}\dot{\delta}_e + k_{a2}\delta_e = 0 \tag{21}$$

By proper choice of k_{a1} and k_{a2}, stability of (21) can be ensured. To implement the control ν_a, we need $\dot{\delta}$, $\ddot{\delta}$, $\dot{\delta}_c$ and $\ddot{\delta}_c$ in addition to δ and δ_c and they are assumed to be available.

1) Performance Analysis of UDE based controller with actuator compensation using Method 2: Referring to Table. 1 for the actuator parameters ω_A and ζ_A, it can be inferred that the actuator has a settling time of 50 ms. Keeping the desired settling time of error dynamics as 50 ms with a damping ratio of 0.8, k_{a1} and k_{a2} are evaluated to be 160 and 10000, respectively. τ was retained as 0.01. Other simulation parameters were according to Table. 1. In addition to the disturbance d_{ext}, an uncertainty was introduced such that $\omega_{RR} = -3$ rad/s. The results are shown in Fig. 7, which are quite similar to the performance in Fig. 2. It can be observed in Fig. 7d, that the compensator (18) has been successful in its attempt to make δ track δ_c. While this approach satisfies all criteria of the desired performance, its main drawback lies in its dependence on the availability of the derivatives of δ and δ_c. While the requirement for their availability was met with, solely for simulation purposes, their availability cannot be assured in real-life situations.

C. Method 3: Using Compensator Form 2

To eliminate the need for the derivatives of δ and δ_c, another strategy for designing an actuator compensator is proposed, which not only compensates the actuator lag, but also reduces



(a) Output Response (b) Control Effort

(c) Disturbance Estimation (d) δ vs δ_c

Fig. 7. Performance with Method 2

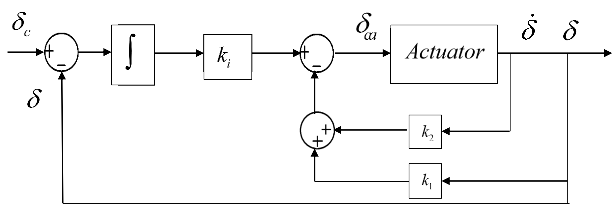


Fig. 8. Block Diagram Schematic for Method 3

the number of derivatives required. As referred from [14], the block diagram schematic is as shown in Fig. 8. From Fig. 8, the expression for compensated actuator input δ_{ca} can be expressed as ;

$$\delta_{ca} = -k_1\delta - k_2\dot{\delta} + k_i\int[\delta_c - \delta]dt \tag{22}$$

The transfer function of the second order actuator, as shown in Fig. 3, can be written as;

$$\frac{\delta(s)}{\delta_c(s)} = \frac{b}{s^2 + as + b} \tag{23}$$

with $b = \omega_A^2$ and $a = 2\zeta_A\omega_A$. The actuator dynamics can be re-written in the state space model as follows;

$$\begin{aligned} \dot{x} &= Ax + Bu \\ y &= Cx \end{aligned} \tag{24}$$

where A, B and C are the matrices of the system and are given by

$$A = \begin{bmatrix} 0 & 1 \\ -b & -a \end{bmatrix}; B = \begin{bmatrix} 0 \\ b \end{bmatrix}; C = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}$$

With the state variables x_1 and x_2 being δ and $\dot{\delta}$, respectively, following [14], the control input u is defined as follows;

$$u = \delta_{ca} = -k_1\delta - k_2\dot{\delta} + k_i\int(\delta_c - \delta)dt \tag{25}$$

To determine the values of user defined gains k_1, k_2 and k_i, we need to determine the desired and the actual characteristic equations for the closed loop system comprising of the actuator and its compensator. The desired characteristic equation for this system is of the form $(s^2 + 2\zeta_d\omega_d s + \omega_d^2)(s + \zeta_d\omega_d) = 0$ for a desired settling time and damping factor. To derive the actual characteristic equation for this system, following [14], we first need to define matrices $\hat{A}$, $\hat{B}$, $\hat{K}$;

$$\hat{A} = \begin{bmatrix} A & 0 \\ -C & 0 \end{bmatrix}; \hat{B} = \begin{bmatrix} B \\ 0 \end{bmatrix}; \hat{K} = \begin{bmatrix} k_1 & k_2 & k_i \end{bmatrix}$$

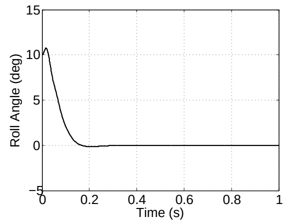
Computing $sI - (\hat{A} - \hat{B}\hat{K}) = 0$ results in the actual characteristic equation of the form;

$$s^3 + s^2(a + bk_2) + s(b + bk_1) + bk_i = 0 \tag{26}$$

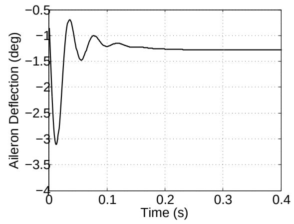
From the desired and the actual characteristic equation, we can evaluate the values of k_1, k_2 and k_i .

To determine the desired characteristic equation, a damping factor of 0.8 is chosen. The desired settling time for the actuator compensator-actuator system now needs to be determined, the choice of which bears a close relationship with the actuator parameters. From the knowledge of the parameters of actuator: $\omega_A = 100$ rad/s and $\zeta_A = 0.8$, the settling time for the actuator is 50 ms. Since the compensator dynamics need to be faster than the actuator dynamics, by rule of thumb we choose a desired settling time to be $1/4$ times the actuator settling time, i.e. 12.5 ms, For these values of settling time and damping factor, the desired characteristic equation is derived and compared with the actual characteristic equation to give values of k_1, k_2 and k_i, as discussed subsequently.

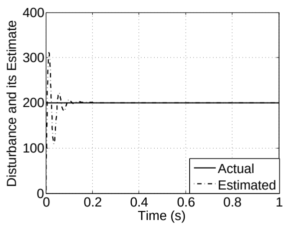
1) Performance Analysis of UDE based controller with actuator compensation using Method 3: Simulation is carried out with simulation parameters as per Table. 1 and an external disturbance equal to d_{ext}. As stated before, simulations were carried out for a desired settling time of 12.5 ms for the actuator compensator-actuator system, retaining the value of τ as 0.01. For this value of desired settling time, with damping ratio of 0.8, k_1, k_2 and k_i were calculated to be 35.48, 0.083, 5120, respectively. Simulation results are shown in Fig. 9. From Fig. 9a, it can be observed that the roll angle ϕ has settled down to zero, as desired. The output response shows a small overshoot. It can be seen from Fig. 9d, that δ tracks δ_c as desired. It can thus be concluded that by proper choice of k_1, k_2 and k_i, the delay created by the actuator can be compensated. When a desired settling time of 25 ms was chosen with corresponding values k_1, k_2 and k_i, the performance was observed to be similar to Method 1 (Fig. 5 refers). Few more trials were carried out to explore the relation between the choice of τ and desired settling time of the actuator compensator-actuator system. It was observed that for a given damping ratio, say 0.8, the relation between τ and settling time



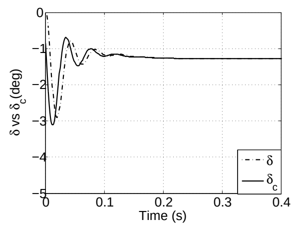
(a) Output Response



(b) Control Effort



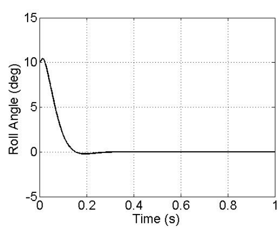
(c) Disturbance Estimation



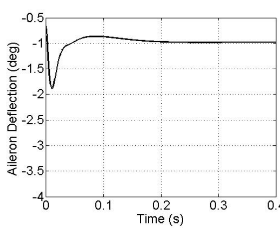
(d) δ vs δ_c

Fig. 9. Performance with Method 3 ($k_1 = 35.48, k_2 = 0.083, k_i = 5120$)

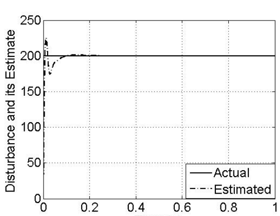
for the error dynamics (t_s) can be expressed as $\tau = 1.25t_s$. Following this analogy, keeping $\tau = 0.01$, the desired settling time was computed to be 7.96 ms. k_1, k_2 and k_i were then calculated to be 88.89, 0.137 and 19800, respectively. Further, an uncertainty of -30 % in ω_{RR} and +30 % in K_δ along with d_{ext} were introduced. Simulation results for this choice, are shown in Fig. 10, which indicate satisfactory performance. This approach when compared to the previous two approaches,



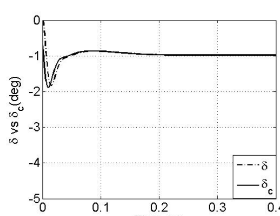
(a) Output Response



(b) Control Effort



(c) Disturbance Estimation



(d) δ vs δ_c

Fig. 10. Performance with Method 3 ($k_1 = 88.89, k_2 = 0.137, k_i = 19800$)

yields better results, while eliminating the need for $\ddot{\delta}$, $\ddot{\delta}_c$ and even $\dot{\delta}_c$. To demonstrate the validity of the proposed Method 3 in practical situations, constraints were imposed on δ and $\dot{\delta}$ to the tune of +/- 24 deg and +/- 250 deg/s, respectively. While simulating Fig.10, it was found that the obtained values were within specified limits. Effects of dead zone, backlash of a practical actuator and measurement noise on the performance, have been assigned for future study.

IV. Conclusion

In this work, the strategy of UDE was employed in the design of a robust roll autopilot with actuator compensation. Design assuming ideal actuator followed by the effects of second order actuator in the performance have been presented. By tuning the filter time constant, the performance of UDE based controller for robustness and actuator compensation was found to be acceptable, though required improvement. To achieve this, two forms of compensator were proposed and evaluated. It was found that Compensator Form 2 gave the desired results at the same time reducing the requirement of derivatives of the control effort. Our future work would involve further investigation into the role of filter time constant (τ) in actuator compensation. Design of UDE based control law to cater to time varying uncertainties and disturbances along with actuator compensation, is another area to be focused in future. Applicability of the proposed methods for non-linear roll dynamics would also be explored in our future work.

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