System Modeling and Control of a Clutch Actuator System for Dual Clutch Transmissions

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Abstract : In this paper, a clutch actuator system for DCTs is developed by integrating a brushless DC motor with mechanical subsystems. The system modeling is performed with experimental validations. However, since the developed model is based on the linear system, it may include modeling uncertainties and external disturbances. To overcome this difficulty, a cascade control structure is proposed. Therefore, the position controller is composed of two parts. One is a proportional–derivative controller in the outer-loop, and the other is a disturbance observer (DOB) based on the nominal system model in the inner-loop.

Key words : Clutch Actuator, Position Actuator, Transmission, Position Control, Modeling

Nomenclature

J: moment of inertia of rotor, kg /s^2
g : viscous coefficient,
θ : angular position, rad
ω : angular velocity, rad/s
ξ dynamic system
k: motor torque constant, Nm/A
r: radius, m
u: control input , A
x: linear stroke of the actuator carrier, m
y: system output
F: force, N
K: control gain
T : torque, Nm
P: plant
Q: Q-filter (low-pass filter)

Subscripts

c: carrier
l : load
m : BLDC motor
n : nominal model
p: pinion gear
cam: cam
fk: fork
md: desired trajectory of a BLDC motor

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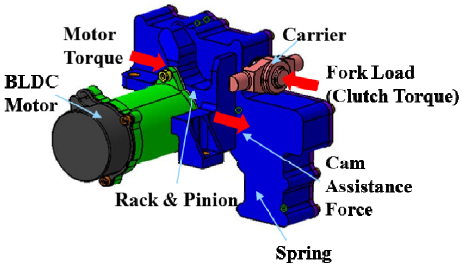
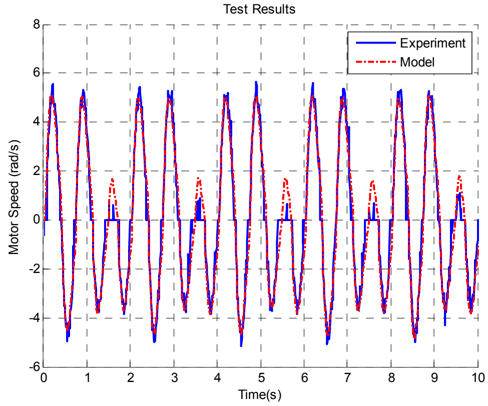
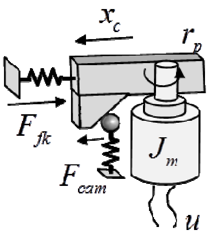
1. INTRODUCTION

As the progress of advanced mechatronic technology, there have been servo-controlled automotive subsystems for many parts. In this context, a clutch system within transmissions is also automated by using electronic or hydraulic actuators. In particular, dual clutch transmissions (DCTs) can improve the performance of driver's convenience and fuel efficiency at the same time. Consequently, the DCT has received considerable attention in automotive industry[1-7]. Since DCT have two power flow paths without a torque converter, control problems are quite challenging. Moreover, the use of dry-clutches makes a control more difficult in spite of high efficiency. Since the clutch engagement torque is proportional to the stroke of







the actuator, the position control quality of the clutch servo system determines the overall performance of DCTs. If there is a mismatch between the command from the transmission control unit and the actual position signal, the resulting clutch torque will be inaccurate. Thus, the position control of the clutch actuator should be as accurate as possible.

From this background, this paper considers the clutch positioning system for DCTs that plays the role in actuating the clutch disk by an electric motor. The dynamic system is modeled with the experimental validation. With this model, the control development uses a disturbance observer-based control to eliminate the effect of unmodeled dynamics. Simulation results show the effectiveness of this approach.

2. SYSTEM MODELING

2.1 System Description

Overall clutch actuator system consists of the BLDC motor and the mechanical subsystem shown in Fig. 1.

The motor torque generated by electrical energy is converted into the linear force through a rack-and-pinion gear. This force moves the carrier toward the direction in the clutch disk along with the actuator mounting frame. The linear motion of the carrier x_c in Fig. 2 is controlled such that the required clutch torque is generated. From this formulation, the clutch torque is considered as a variable load applied in the electrical motor.

Note that the cam is equipped with the spring in order to obtain the assistance force to react to the fork load shown in Fig. 1. Due to the high stiffness of the diaphragm spring, such additional components should be required. Without the assistance mechanisms, the required motor power needs to be increased significantly.

2.2 Dynamic Model

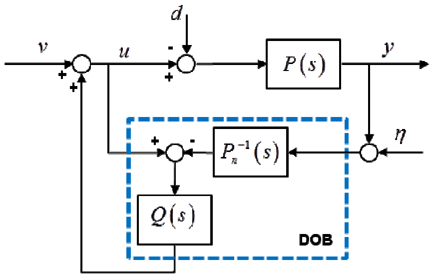
Since the electrical motor is driven by a current controlled amplifier, it can be described by the following model:

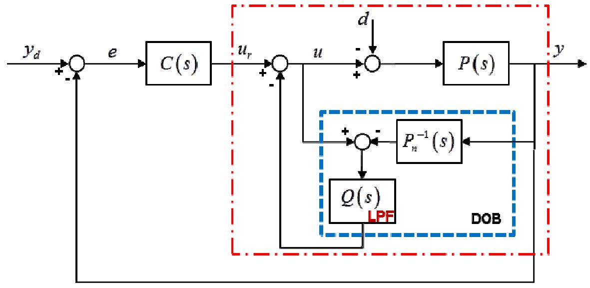
$$J_m\dot{\omega}_m + g_m\omega_m + T_l = k_m u \qquad (1)$$

The load torque T_l includes the clutch force F_{fk} during the clutch engagement phase and the cam assistance force F_{cam}. Therefore, the load torque model is represented as

$$T_l = r_p F_{fk}\left(x_c\right) - r_p F_{cam}\left(x_c\right) \qquad (2)$$

Where, r_p is the rack and pinion gear radius and x_c is the linear stroke of the carrier of mechanical subsystems through the rack and pinion gear, .e.g. $v_c = r_p\omega_m$, $\dot{x}_c = v_c$.





Combining equations (1) and (2) yields

$$J_m\dot{\omega}_m + g_m\omega_m + r_n F_{fk} = k_m u + r_n F_{cam}\,. \qquad (3)$$

Note that F_{cam} includes some nonlinearities due to the inherent design of the cam assistance force. Since the clutch torque is affected by a diaphragm spring and damper springs built in the disk, F_{fk} is also nonlinear. Although these loads are nonlinear as mentioned before, they could be approximated by a linear model in the operating range for a controller development. Thus, the cam assistance force and the fork load on the actuator are linearly parameterized as

$$\overline{F}_{cam} = \overline{k}_{cam}x_c = \overline{k}_{cam}r_p\theta_m\,. \qquad (4)$$

$$\overline{F}_{fk} = F_{fk0} + \overline{k}_{fk}x_c = F_{fk0} + \overline{k}_{fk}r_n\theta_m \qquad (5)$$

Where, F_{fk0} is the preload of the clutch disk, and $\overline{k}_{cam}$ and $\overline{k}_{fk}$ are the equivalent stiffness of the cam and the fork, respectively. Replacing the corresponding terms with (4) and (5) yields the following linear model

$$J_m\dot{\omega}_m + g_m\omega_m + r_n\overline{F}_{fk} + d = k_m u + r_n\overline{F}_{cam}\,. \qquad (6)$$

where, d denotes unknown disturbances including linearization error, nonlinear friction effect, and external disturbances.

2.3 System Identification

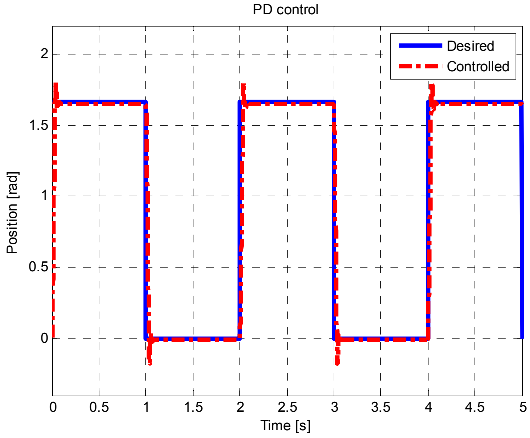
To validate the model developed in the previous subsection, system identification is performed by experiments. The input and the output are chosen as the current u and the motor speed ω_m respectively. This problem is taken into consideration as the following optimization problem:

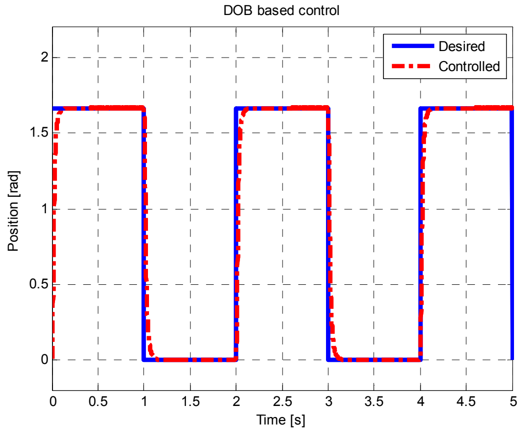
$$\min_{J_m,k_m,g_m} \frac{1}{N}\sum_{k=1}^{N}\left(\xi_{\text{model,k}} - \xi_{\text{meas,k}}\right)^2 . \qquad (7)$$

where $\xi_{\text{model,k}}$ and $\xi_{\text{meas,k}}$ denote outputs of the nominal model and the actual plant, respectively. And, k is the sequence of experiments, N is the total measurements. The input is chosen as a pseudo random binary signal having sufficiently large amplitude to overcome static friction nonlinearity and then a sinusoidal signal is applied to verify the identified result. The motor parameters are first considered to solve this optimization problem. Therefore, the parameters identified from this procedure are J_m g_m and k_m A least square method is applicable to determine these unknown parameters. The identification result is shown in Fig. 3. The identified model could fit the nominal model well.

3. CONTROLLER DESIGN

Based upon the developed model, a controller is designed such that the motor position follows the desired one precisely. First, simple proportional-derivative (PD) controller will be developed as a reference controller. Then, a disturbance observer (DOB) based controller will be proposed to compensate the nonlinearities.





3.1 PD Controller

The first controller considered is a PD type controller whose structure is given by

$$u=-K_n\left(\theta_m-\theta_{md}\right)-K_d\left(\omega_m-\omega_{md}\right) \qquad (8)$$

where, the position measurement θ_m is available from the BLDC motor drive and the rotational velocity signal ω_m can be obtained from the numerical differentiation of two consecutive position measurements.

3.2 Disturbance Observer Based Controller

The Second control method is based on the disturbance observer (DOB). The main role of the DOB is to estimate unknown disturbance in the framework of a linear system. The estimated disturbance by the DOB can be fed back into the control loop to compensate the variation of the modeling uncertainties[8) 9)]

The concept is depicted in Fig. 4. The equivalent disturbance estimate can be calculated by the difference between the control input u and the inverse of the nominal plant model $P_n^{-1}(s)$ Then it is filtered out by a low-pass filter *Q(s)* over specified frequency range. The DOB performance depends on the selection of *Q(s)*. The bandwidth of *Q(s)* is related with the natural frequency of a given system. For an implementation purpose, *Q(s)* has to be selected such that $P_n^{-1}(s)Q(s)$ becomes proper.

In our case, the nominal plant model $P_n^{-1}(s)$ is composed of the motor parameters as follows.

$$P_n(s)=\frac{k_m}{J_m s+g_m} \qquad (9)$$

Under the assumption that the DOB works well in the inner-loop, the actual plant, which is uncertain, can be viewed as a nominal model in (9). Therefore, the outer-loop controller is simply designed for a nominal model without uncertainties to achieve the desired control objective. This control architecture is shown in Fig. 5 where the inner-loop controller is represented by a dash-dot box. In Fig. 5, the output y is written as

$$y=\frac{PP_n}{P_n+(P-P_n)Q}u_r+\frac{PP_n(1-Q)}{P_n+(P-P_n)Q}d \qquad (10)$$

where, u_r is the output of the outer-loop controller *F(s)*. When the low-pass filter is set by *Q(s)=1*, the above equation becomes $y=P_n u_r$ which shows that the disturbance d disappears It implies that disturbance rejection is only valid in a certain frequency range.

The outer-loop controller is constructed as the following form

$$u_r=u_{pdo}+u_{FF} \qquad (11)$$

where,

$$u_{pdo}:=-K_{po}\left(\theta_m-\theta_{md}\right)-K_{do}\left(\omega_m-\omega_{md}\right)$$
$$u_{FF}:=-\bar{F}_{kf0}-\bar{k}_{fk}r_p\theta_m-\bar{k}_{cam}r_p\theta_m$$

In (11), u_{pdo} is the PD controller for the nominal plant, and

u_{FF} is the feed-forward input which is obtained from the experiments.

4. SIMULATION RESULTS

This section show the simulation results for the evaluation of the developed controller discussed in the previous section.

Fig. 6 shows the simulation result of the PD control. The PD controller does not work properly in a transient period. The transient response in Fig. 6 has some oscillation. In clutch operations, the ultimate goal of control is to ensure a fast but smooth engagement as fast as possible. Overshoot in a clutch positioning makes the engagement operation uncomfortable. But, such a behavior cannot be eliminated in spite of repeated gain tuning process. Moreover, the steady-state error does not converge to zero. This behavior is due to the preload from the cam assistance force that is nonlinear. From this result, it is concluded that the PD control cannot guarantee the robustness under the presence of uncertainties.

On the other hand, the DOB based approach proposed in Section 3.2 can overcome this problem. Fig. 7 shows that the transient performance is improved significantly compared with the result of the PD control case. The steady-state error also converges to zero. The DOB as an inner-loop controller can successively eliminate modeling uncertainties and external disturbances. It can be concluded that the DOB based controller is superier than the PD controller.

4. CONCLUSION

A clutch positioning control system is developed. The BLDC motor is connected with the carrier mass through a mechanical linkage. The carrier position is controlled to ensure the clutch engagement by a disturbance observer based controller. The system model is validated experimentally. Comparable study for the proposed DOB based controller with simple PD controller is performed by simulations. The result shows the effectiveness of the proposed DOB based control method.

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