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Article

Low Cost Position Controller for EGR Valve System

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Abstract: This paper proposes a position control method for low cost EGR valve system in automotive application. Generally, position control system using in automotive application has many restrictions such as cost and space, the mechanical structure of actuator implies high friction and large difference between static friction and coulomb friction. This large friction difference occurs the vibrated position control result when the controller uses conventional linear controller such as P, PI. In this paper, low cost position control method which can apply under the condition of high difference friction mechanical system. Proposed method is verified by comparing conventional control result of experiments.

Keywords: Position control, Static friction, EGR valve system, Automotive application.

1. Introduction

Recently, many mechanical components using in vehicle has been replaced by electrical components for high efficiency. Not even hybrid electric vehicle or electric vehicle, MDPS(Motor Driven Power Steering) and ISG(Integrated Starter and Generator) are generally adopt in gasoline or diesel vehicle, these electric automotive components make increasing the drive efficiency and fossil fuel reduction. This change spreads into the transmission system and engine valve system. Among them, exhaust gas recirculation(EGR) valve is the target mechanical component to replace of small DC motor.[1]-[5] However, general mechanical systems using in the valve, the allowable cost is very low and the space for implement is very narrow, therefore, the electrical system including actuator should have cost effective with small size. To achieve this, mechanical actuating system can not avoid to be roughly designed which implies high friction force. More worse, the difference between coulomb friction and static friction is very large, correct and fast response of position control is almost impossible to solve by conventional linear control system such as P, PI, or PID.

To regulate the position control against this friction torque, some research has been proposed.[6]-[10] In [6], H infinite control and impulse control are combined for fast control response. Robust control is achieved by disturbance observer is proposed in [7]. Fuzzy controller [8] and neural network controller [9] are proposed to overcome this problem. In [10], adaptive control method for friction compensation is proposed. These methods can dramatically reduce the effect of the friction, however, the parameters have to be set are too much, and the processing burden for realization is also complex in low cost drive system.

This paper proposes the position control method for low cost system. General position control method for this low cost system is P-PI control method which is described in [11]. Aforementioned, the correct and fast control can not be achieved under the mechanical system which has high friction condition with this linear controller. Generally, in this case, feedforward compensation is adopt for improving the control performance.[11]-[13] However, this feedforward data is incorrect because of the aging the mechanical system, environmental change affection such as temperature and humidity. Moreover, feedforward compensation can improve the dynamics of the controller, however, it can not be the solution for instable control performance which caused from the difference of static and coulomb friction torque. In this paper, to achieve the stable position control, firstly analyze the EGR valve mechanical model, define the cause of occurring vibration, and then, a proposed novel and

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simple algorithm which can possibly adapt in low cost system to solve this problem will be illustrated. Finally, comparing the performance of conventional method and proposed method to verify its superiority by experiment.

2. Mechanical model of EGR valve and torque measurement

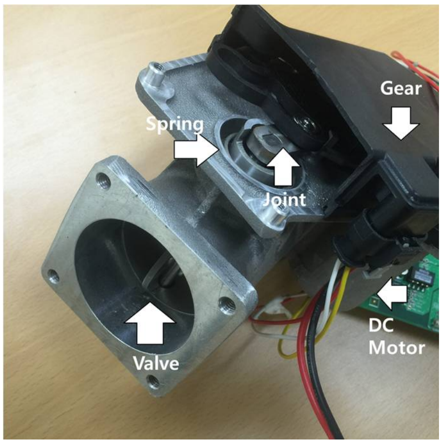

Figure 1. Mechanical composition of EGR valve.

2.1. Model analysis of EGR valve

Figure 1 shows the mechanical composition of EGR valve. General EGR valve is composed of spring for recover the valve initial position, joint and gear for transforming the power from rotation to translation, DC motor and throttle valve which are the source and actuator of mechanical system, respectively.

First, same as the general structure of EGR valve, the motor to operate this system is DC motor. Therefore, the generated torque from the motor is modelled as below.

$$T_e = k_t i_a \tag{1}$$

where, k_t is torque constant, i_a is armature current of DC motor.

The mechanical equation of the valve system shown in Figure. 1 can be described as following general equation.

$$T_e = J\frac{d^2\theta_r}{dt^2} + T_{fric} + T_{spring} + T_L \tag{2}$$

where, J is inertia, θ_r is rotating angle, T_{fric} is friction torque, T_{spring} is spring torque, T_L is load torque.

On the other hands, this rotating angle is transferred to linear position by mechanical joint and gear, linear position x can be expressed as below.

$$x = r\{\cos(\theta_{L0}) - \cos(\theta_L + \theta_{L0})\} \tag{3}$$

$$\theta_L = \frac{\theta_r}{n} \tag{4}$$

where, n is ratio of the gear, r is joint distance, θ_L is joint angle, θ_{L0} is initial joint angle.

The spring force according to the linear position is described as below equation.

$$F_{spring} = k_{spr}(x + x_0) \tag{5}$$

where, k_{spr} is constant of spring character, x_0 is initial linear position.

This spring force can be transferred to the torque on the load side as below equation.

$$T_{springL} = \frac{rk_{spr}}{n}(x + x_0)\sin(\theta_L + \theta_{L0}) \tag{6}$$

And then, transfer this spring torque on the load side to motor side, (6) can be changed as (7).

$$T_{spring} = \frac{rk_{spr}}{n}\sin(\theta_L + \theta_{L0})[r\{\cos(\theta_{L0}) - \cos(\theta_L + \theta_{L0})\} + x_0] \tag{7}$$

Equation (7) describes spring torque is only affected by the spring position. However, in practical, spring torque is not only affected by the position but also the speed direction. To apply this on (7), we defines spring coulomb friction torque as (8).

$$f_{spr\_c} = F_{spr\_col}\,\mathrm{sgn}(\frac{dx}{dt}) \tag{8}$$

As shown in (8), spring coulomb friction is negative when the motor speed is reverse. As a result, spring torque can be modelled as (10).

$$T_{spr\_col} = \frac{rF_{spr\_col}}{n} \tag{9}$$

$$T_{spring} = \frac{rk_{spr}}{n}\sin(\theta_L + \theta_{L0})[r\{\cos(\theta_{L0}) - \cos(\theta_L + \theta_{L0})\} + x_0] + T_{spr\_col}\sin(\theta_L + \theta_{L0})\,\mathrm{sgn}(\omega_r) \tag{10}$$

On the other hands, EGR valve mechanical system is not only affected by the spring but also affected by joint and gear. Due to the low cost gear and joint occurs the friction like lead-screw which emphasizes nonlinear static friction. In this paper, LuGre friction model described on [10] is derived.

$$T_{fric} = [T_{ge\_col} + (T_{ge\_sta} - T_{ge\_col})e^{-(\omega_r/\omega_s)^2}]\,\mathrm{sgn}(\omega_r) \tag{11}$$

where, T_{ge_col} is coulomb friction torque on gear and joint, T_{ge_sta} is static friction torque on gear and joint, ω_s is Stribeck velocity.

In this paper, these modeled load torques is measured experimentally in order to implement feedforward controller as same as [12]. This feedforward compensation can reduce the burden of feedback controller, it can help to enhance the control performance when the nonlinear load has to be controlled by linear controller.

2.2. Measurement procedures of spring and friction torque

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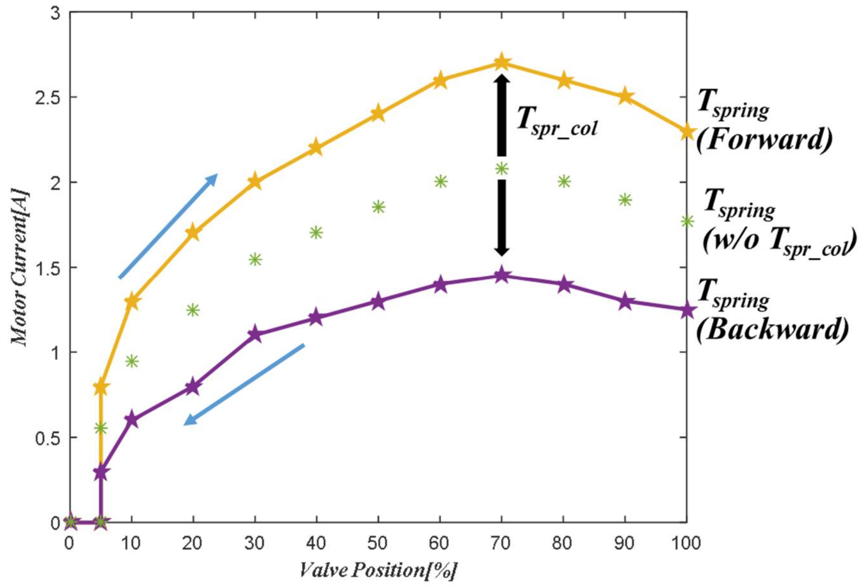

Figure 2. Measured spring torque.

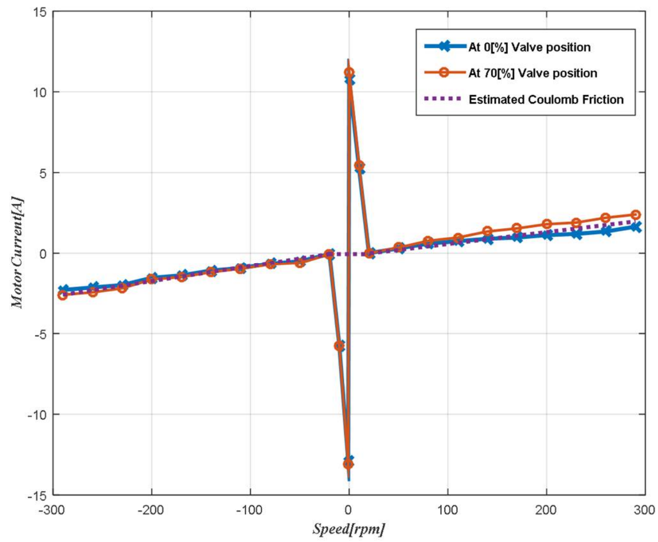

Figure 3. Measured friction torque.

Fig. 2 and Fig.3 show the measured spring torque and friction torque of tested EGR valve, respectively. First of all, electric torque from the motor is proportional to the DC motor current based on (1). Therefore, this paper assume that the current waveform can indirectly describe the generated torque. To measure spring torque, follow the below steps for identifying them.

1. Perform the speed control started on initial EGR valve position.
2. Apply the speed reference from 10[rpm] to 300[rpm].
3. Measure averaged current. The speed that the lowest averaged current is observed is Stribeck velocity. Repeated experiment is necessary for gathering data.
4. Control the motor on Stribeck velocity. Measured the instantaneous current on steady state is spring torque with the assumption that the friction torque at Stribeck velocity can be ignored.

With the obtained spring torque, we can measure the static friction torque with following steps.

1. Perform the speed control started on each EGR valve position.
2. Set up the small gain on current controller on order to apply the ramp increasing current reference.
3. Sudden current change occurs due to the position movement, measure the peak current point.
4. Subtract the spring torque amount from the measurement of procedure 3. Remain value is static friction torque.

The coulomb friction torque can be obtained with following steps.

1. Perform the speed control started on initial EGR valve position.
2. Apply the speed reference from 100[rpm] and 300[rpm].
3. Measure the instantaneous current on steady state of each speed.
4. Subtract instantaneous current at 100[rpm] from the current at 300[rpm].
5. Divide 200[rpm] from the result of procedure 4 for removing spring torque component.
6. Multiply the speed from the result of procedure 5. This is the coulomb friction torque.

As shown in the Fig. 2, different spring torque is occurred according to the valve position direction. If the valve position direction is to open the valve, the spring torque is increased due to the coulomb friction in spring torque which depicted on (8). Reversely, if the valve position direction is to close the valve, the spring torque is decreased. On the other hands, Fig. 3 indicates that the friction torque at each position has almost same static friction torque. Also, the static friction in reverse direction has different value from positive direction value. Coulomb friction torque calculation is based on simple principle. First, torque equation at 300[rpm] can be described as bellow.

$$T_e(\omega_{r2}) = J\frac{d^2\theta_r}{dt^2} + T_{fric}(\omega_{r2}) + T_{spring}(\omega_{r2}) \tag{12}$$

where, ω_{r2} is angular speed of 300[rpm].

If the steady state condition is only effective to identify the coulomb friction torque, inertia term can be neglected. With (12), the torque difference which represents at 4 procedure can be calculated as follow.

$$\begin{aligned} &T_e(\omega_{r2}) - T_e(\omega_{r1}) \\ &= (T_{fric}(\omega_{r2}) + T_{spring}(\omega_{r2})) - (T_{fric}(\omega_{r1}) + T_{spring}(\omega_{r1})) \end{aligned} \tag{13}$$

where, ω_{r1} is angular speed of 100[rpm].

As described on (5), if the position is coincide, spring torque is not affected by the speed. Static friction torque does not interfere during constant speed operation, (13) can be simply described as below.

$$T_e(\omega_{r2}) - T_e(\omega_{r1}) = T_{ge\_col}(\omega_{r2}) - T_{ge\_col}(\omega_{r1}) \tag{14}$$

If the coulomb friction is proportional to speed, it can be expressed by coulomb friction gain and speed. Assumption that this gain is almost same as all over the position, in order to simplify the coulomb friction, (14) can be transformed to (15).

$$T_e(\omega_{r2}) - T_e(\omega_{r1}) = B(\omega_{r2} - \omega_{r1}) \tag{15}$$

where, B is coulomb friction gain.

In this paper, assuming that coulomb friction occurs over the Stribeck velocity.

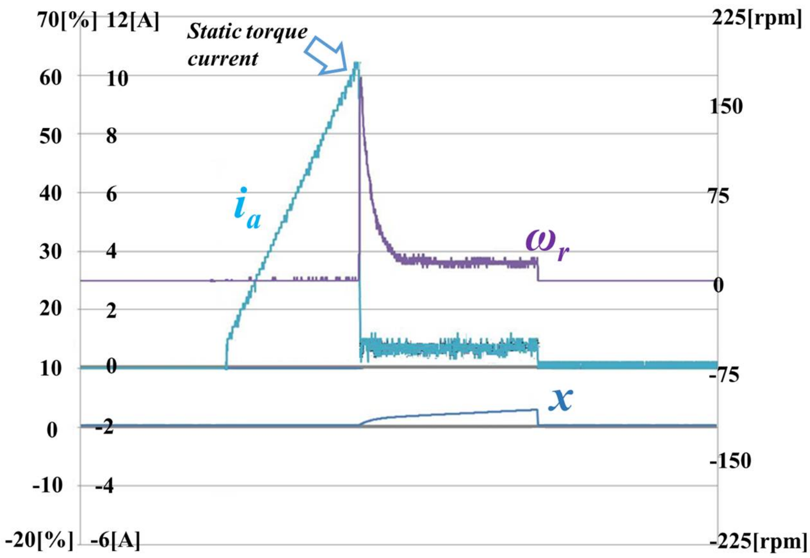

Figure 4. Static friction torque measurement at zero valve position (green : current, purple : speed, blue : position).

Fig. 4 illustrates the measurement of static friction torque. As shown on the figure, current is increasing to overcome the static friction force. However, the valve position does not move. If the current reaches to the point described on the figure, valve position starts to move due to the generated motor torque is over the static torque. At this time, speed is increased radically when static friction torque and coulomb friction torque has a large difference. Note that controlled speed is 20[rpm] which is Stribeck velocity, in this case, coulomb friction torque current is 0.8[A]. However, static friction torque is 10.4[A] which means that static friction torque is over ten times of coulomb friction torque.

3. Proposed position controller

Fig. 5 shows the conventional P-PI controller and proposed control system.[11] As shown on the figure, proposed control system does not implement speed controller. The main reason is motor position detection sensor is absence in practical product to reduce the cost. Although the speed information can be obtained from the linear position sensor implemented for detecting valve position, however, sensing dynamic of linear position sensor is not enough to calculate motor speed. Moreover, the speed information is the derivative component of position information, it is essential to use the filter to mitigate the noise. This restricts the bandwidth of the controller more worse which is already restricted because of slow dynamic of the linear position sensor.

Due to the same reason, D controller can not be adopt because the effective derivative component of the position error is difficult to obtain. Moreover, it can amplify the noise of the position information signal. Therefore, the position controller is selected as PI controller in this system. Actually, this position control system has a problem in the performance. First, proposed position control transfer function shown on Fig.5 can be described as below.

$$\frac{\theta_r}{\theta_r^*} = \frac{G_p(s)G_c(s)G_m(s)}{1+G_p(s)G_c(s)G_m(s)} \tag{16}$$

The control dynamic of the current controller is much faster than position controller, the transfer function of current controller $G_c(s)$ can be approximated as 1 in the position control view. Assuming

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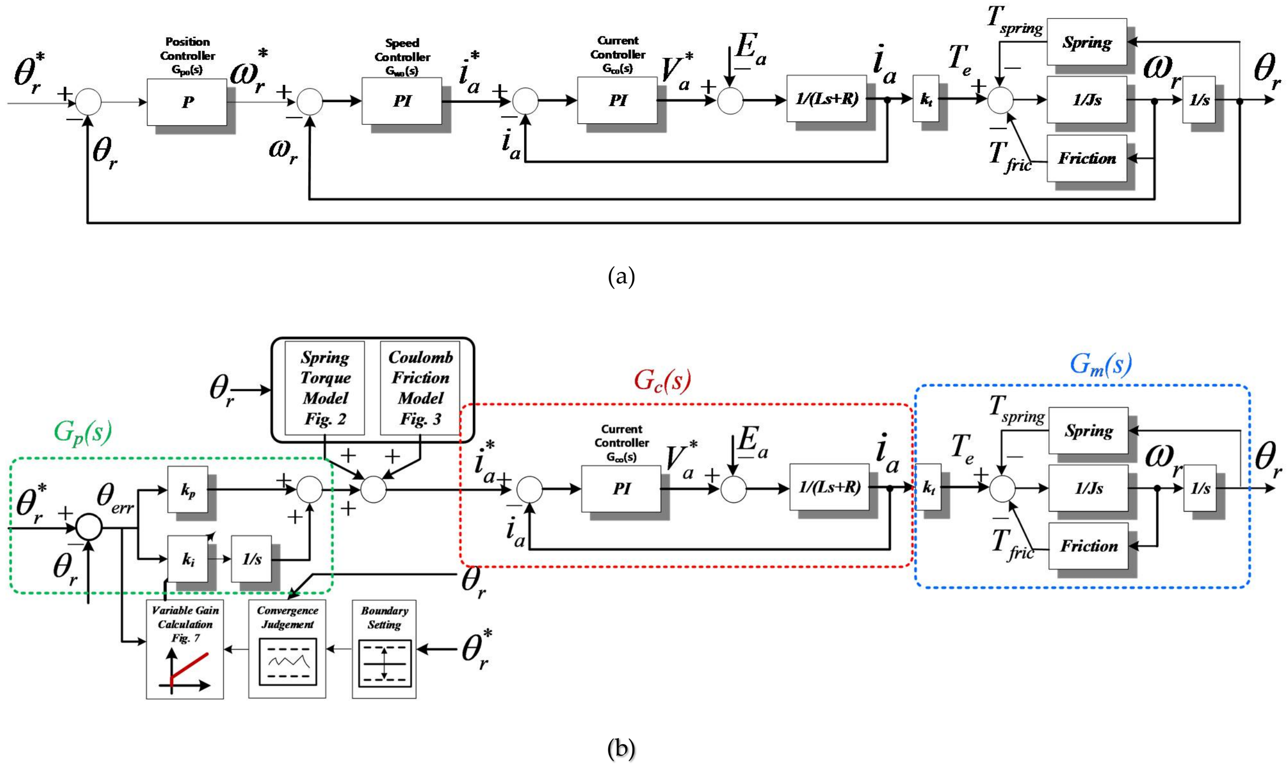

Figure 5. Position control method (a) P-PI controller[11] (b) Proposed position controller

that spring load torque is fully compensated by feedforward path, the transfer function can be changed as follow.

$$\frac{\theta_r}{\theta_r^*} = \frac{k_p k_m s + k_i k_m}{s^3 + k_p k_m s + k_i k_m} \tag{17}$$

where, k_m is k_t/J.

Insert this transfer function to final value theorem, the error of step response can be obtained as below.

$$e_\infty = \lim_{s \to 0} \frac{1}{1 + G_p(s) G_m(s)} = 1 \tag{18}$$

From the above equation, PI controller for position control has an error in steady state. To solve this problem, proposed control method is derived from hysteresis control. Proposed control sets the allowable boundary to perceive that practical position follows the reference. If the sensed linear position is going to inside of the boundary, the timer is activated to observe that the controlled position is stably located in the boundary or it is just transient operation. In this paper, the time to perceive that controlled position is in the steady state is 200[ms].

Fig. 6 shows the problem of conventional PI controller. If the sensed position gradually reaches the position reference, controller output also reduced. It also makes to reduce generated current and motor speed. In advance, if the speed is reduced below the Stribeck velocity, static friction torque majorly affects the entire load torque. As a result, motor is stopped as shown on the figure when the motor current does not overcome the static friction torque. Next, I controller integrates the position error when the sensed position does not exactly follow the reference. This integrated error gradually increases the current reference. If it reaches specific current value that generated motor torque is above the static friction torque, it causes sudden speed variation shown on Fig. 4, which makes the position vibration.

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To solve this controller, variable I controller gain is adopt according to the position error. When the position error reaches to the boundary, reduce I controller gain to minimum accordance with the position error as shown on Fig. 7. As aforementioned, I controller is the root of position control vibration, makes this controller inactive since the valve position locates inside the allowable range. In this case, only P controller affects the current reference generation. As a result, position vibration does not occur with proposed position control method. In this paper, this allowable range is 5% of the position reference.

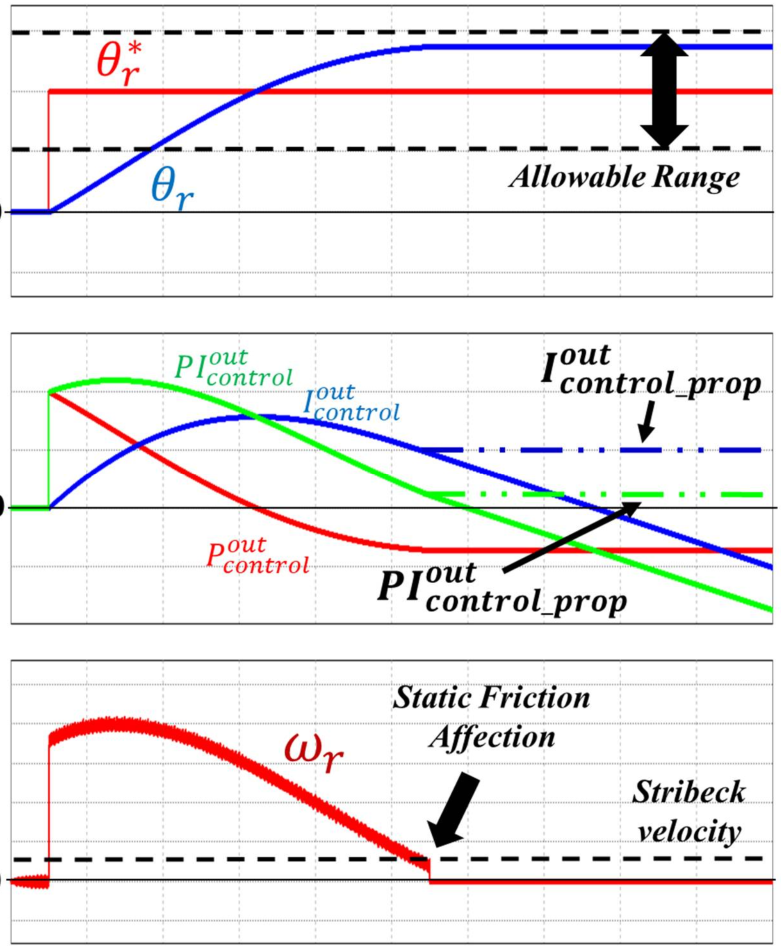

Figure 6. The operation of conventional and proposed PI controller

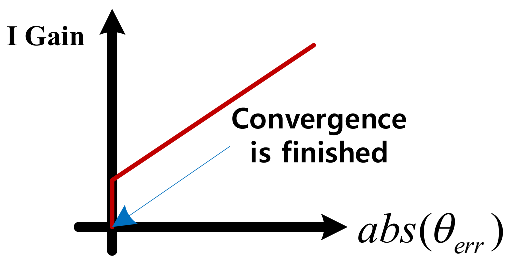

Figure 7. Variable I controller gain adaption according to the position error

4. Experimental Result

Fig. 8 shows the experimental test setup. To compare conventional control method, high performance DSP TMS320F28335 is used, and speed sensor is also instantaneously implemented in

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mechanical system. Sampling frequency of current controller and switching frequency are the same as 20[kHz]. Position control frequency is 2[kHz]. The motor parameters are shown in Table 1.

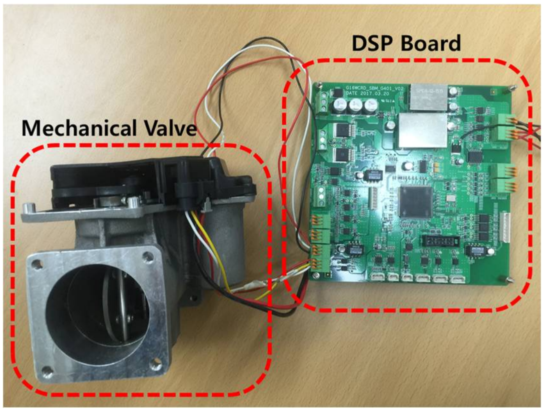
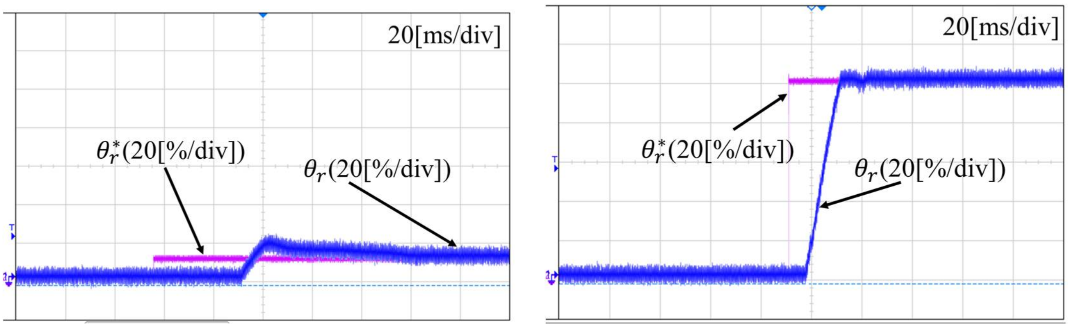


Figure 8. Experiment setup

Table 1. Motor parameters

Parameters	Value	Unit
Rated Power	200	*[W]*
Input voltage	12	*[V]*
Max. Current	20	*[A]*
Rated Speed	500	*[rpm]*



(a) (b)

Figure 9. Position dynamic response (a) 10% valve reference is applied (b) 100% valve reference is applied

Fig. 9 shows the position dynamic responses of 10% valve reference and 100% valve reference. Due to the high static friction torque, if the position error is small, I controller needs some time to generate output for suitable torque against static friction torque. Therefore, the gain has to be tuned considering maximum allowable control response time when the smallest position reference is applied. As shown on the figure, when the 10% reference is applied, the control response time is much longer than the result 100% reference is applied because of static friction torque. On the other hands, this control gain can not be increased infinitely because of the overshoot restriction. Therefore, control gain tuning has to be trade-off by considering two aspects, response time and overshoot.

Fig. 10 shows the comparison experimental results using conventional P-PI controller[11] and proposed controller. As shown on the figure, position controlled by conventional method is vibrated due to the large difference between static and coulomb friction torque. Until the current reaches 2.5[A] for forward direction, the position movement is very little because the static friction torque resists the movement, however, since the current is above the 10[A], position is radically moved forward because of the sudden friction change to coulomb friction torque. The reverse direction operation also

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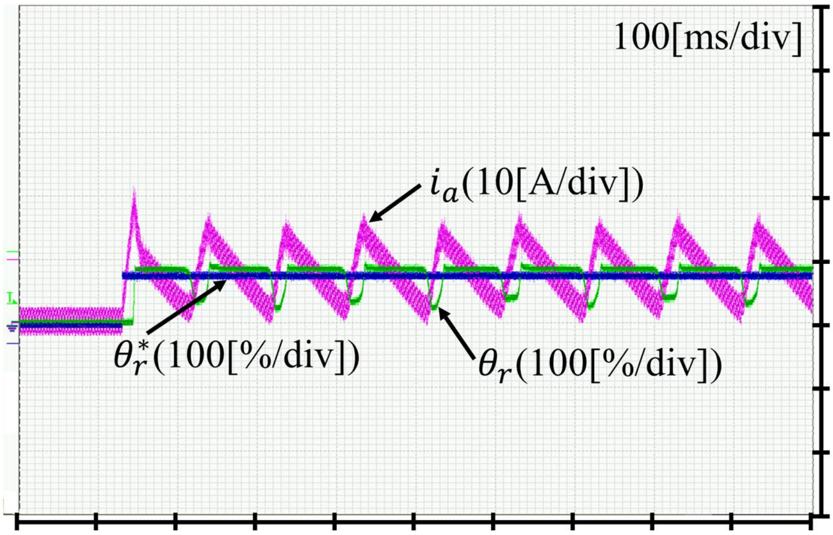
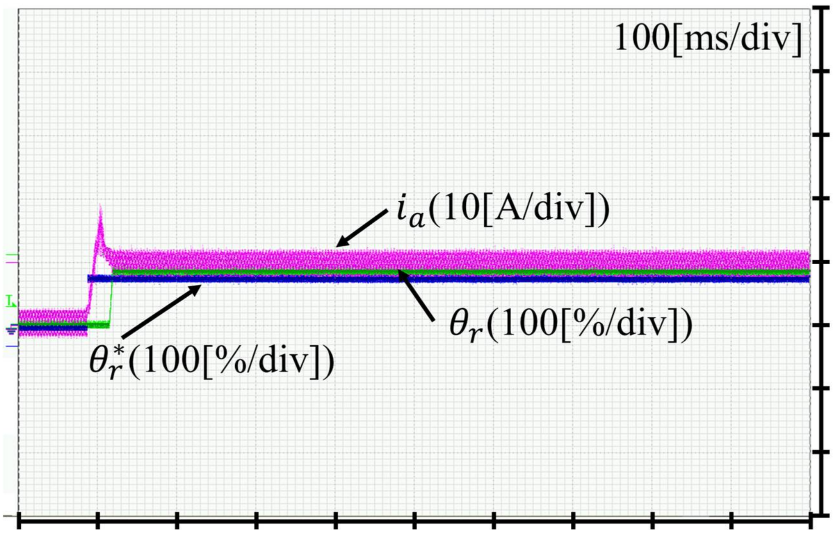
occurs the vibration, as same as forward direction operation. As aforementioned, I controller integrates the small position error, and if the current reference is over the limit, the difference of static friction and coulomb friction torque causes the vibration.



(a)



(b)

Figure 10. Comparison result between conventional and proposed position control (80% reference) (a) Conventional position control (b) Proposed position control

However, position vibration is no more existed with the proposed controller, because the position error is not integrated if it is located inside of the allowable reference range. Stably controlled valve position is 83.4[%] which means 3.4[%] error from the reference. As a result, this experimental result indicates that proposed control method can be effectively applicable in the low cost mechanical drive which has a large difference between static and coulomb friction torque.

5. Conclusion

This paper proposes the position control method for cost effective and fast response which can be used in vehicle valve system. Because the low cost mechanical system has the high difference of static friction and coulomb friction, the control performance is deteriorated with conventional linear controller. However, with the proposed control method, allowable boundary and selectable operation of I controller can achieves the proper control performance which has an acceptable error. Proposed method is verified by comparing conventional method in experiment.

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Author Contributions: Jung-Hyo Lee designed the experiment; Jung-Hyo Lee performed the experiment; Jung-Hyo Lee analyzed the theory. Jung-Hyo Lee wrote the manuscript. Jung-Hyo Lee and Habib Bhuiyan participated in research plan development and revised the manuscript. All authors have contributed to the manuscript.

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