A comparison study of disturbance rejection property of motion controllers

Dept. Of EECS., Yeungnam Univ./241-1, Daeong, Gyongsan, Kyungbuk, 712-749, KOREA

ABSTRACT

This paper presents a comparison study of performance of three motion controllers such as conventional PID controller, LQR(Linear Quadratic Regulator) and RIC(Robust Internal-loop Compensator). All of comparison studies were verified by using a Industrial Servo Emulator(Model 220) made by ECP Corp. To evaluate the disturbance rejection feature, the various disturbances are applied into a motion system.

Keyword list: Disturbance observer, robust internal-loop compensator, motion control, industrial emulator

1. INTRODUCTION

The motion control system is used to regulate mechanical motions in terms of position, velocity and acceleration. Much effort has been devoted to enhance the dynamic performance of motion control system. The primary issue for the precision positioning systems such as factory automation, high-tech computer hard disk drives, and semiconductor chip mounter, and so on is how to achieve the high-speed/high-accuracy performance.

In designing a robust controller for a system in the presence of uncertainty and disturbance, the requirements can be classified into two major kinds. The first corresponds to the robustness property on the uncertainty including external disturbance, variation of the system parameters, modeling uncertainty, and etc., and the second corresponds to the performance specifications for given tasks. The various advanced controller design methods after the observer was proposed by Luenberger in 1966 have been studied to meet these desired specifications. The Disturbance Observer(DOB)\textsuperscript{2,3}, Adaptive Robust Control(ARC)\textsuperscript{4}, Robust Internal-loop Compensator(RIC)\textsuperscript{5,6} are good examples. These methods commonly require the design of 2-loop structures. One is to design the internal-loop compensator for robustness, the other is to design the external-loop controller for desired performance specifications. In such schemes, the internal-loop compensator should generate corrective control inputs to reject equivalent disturbance so that the actual system become a given nominal model as much as possible. The equivalent disturbance is defined as sum of external disturbance and all of possible components due to the differences between actual plant and nominal model such as modeling uncertainty and parameter variation. Thus, the actual plant with such an internal-loop compensator can be regarded as a nominal model if the internal-loop compensator works well. This is the concept of Internal Model Control(IMC)\textsuperscript{7}. On the other hand, the external-loop controller is designed for the nominal model to enhance overall system performance.

This paper provides the property comparison study of motion controllers such as PID, LQR, RIC. In Section 2, these motion control theories are reviewed. In Section 3, the Industrial Servo Emulator which is used to evaluate the performance of each motion controller is described. In Section 4, the motion control experiments using PID controller, RIC and LQR are performed. Finally, conclusions will be followed.

2. HIGH-PRECISION MOTION CONTROLLER

In this Section, the motion control theories will be described. Since PID is the most common feedback controller, it is not described in here. The system to be controlled can be represented as Eq. (1) and the optimal LQR(Linear Quadratic Regulator) control input is given by Eq. (2). $K$ is the gain matrix, which is chosen so that the following performance index shown in Eq. (3) is minimized.

\[ x = Ax(t) + Bu(t) \quad (1) \]
\[ u(t) = -Kx(t) \quad (2) \]
\[ J = \int_0^{t_f} (x^T Q x + u^T R u) dt \quad (3) \]

$Q, R$ are a positive-definite matrices. The optimal control input $u(t) = -Kx(t) = -R^{-1}B^T Sx(t)$ is calculated using the following Riccati matrix differential equation.
\[ A^T S + SA - SBR^{-1}B^T S + Q = 0 \]  

(4)

Recently the robust internal-Loop compensator (RIC) which is shown in Fig. 1 was proposed\[5,6\]. DOR estimates the disturbance and the estimated signal is utilized as a disturbance compensation. Hence, the DOR makes it possible to robust the system behavior between control input and plant output in the presence of uncertainty and disturbance. Fig. 2 shows the structure of the conventional DOB. Comparing Fig. 1 with Fig. 2, we can obtain Eq. (5). As a result, it can be seen that RIC is equivalent to DOB structure.

\[ Q(s) = \frac{P_m(s)K(s)}{1 + P_m(s)K(s)} \quad K(s) = \frac{Q}{P_m(s)(1 - Q)} \]  

(5)

Figure 1: A generalized framework of RIC.  
Figure 2: Conventional disturbance observer (DOB).

The disturbance attenuation characteristics of the designed system can be easily analyzed based on \( Q(s) \). From the Fig. 1, the input output relationship from the external disturbance to the plant output can be expressed as:

\[ G_D = \frac{P(s)P_m(s)(1 - Q)}{P_m(s) + (P(s) - P_m(s))Q} \]  

(6)

Below the cutoff frequency of \( Q(s) \), \( Q = 1 \) is achieved. Hence \( G_D = 0 \) is obtained. This indicates that low frequency disturbance is attenuated and the mismatch between plant and reference model can be compensated in the low frequency range. As a result, unlike the optimization procedure for \( Q(s) \) in the conventional DOR design, the systematic design of \( Q \) function is possible thanks to the parameterization of \( P_m \) and \( K \).

In order to design a robust motion controller in the RIC framework, let’s consider the following reference model.

\[ J_m \ddot{y} + c_m \dot{y} = u \]  

(7)

where \( J_m \) and \( c_m \) are the reference values of \( J \) and \( c \), respectively. Hence, from the definition of the equivalent disturbance, Eq. (7) can be rewritten in terms of \( J_m \) and \( c_m \).

\[ \dot{J}_m \dot{y} + c_m \dot{y} = u + d_{eq}, \quad d_{eq} = (J_m - J) \ddot{y} + (c_m - c) \dot{y} - F_r(y) + d_{fr} \]  

(8)

where \( F_r(y) \) is the friction term and \( d_{eq} \) is the uncertain external disturbance. And the reference control input that can stabilize the reference model given by Eq.(7) can be chosen as \( u = J_m \dot{y} + c_m \dot{y} \).

The reference model \( P_m \) is described by

\[ P_m = \frac{1}{J_m s^2 + c_m s} \]  

(10)

Hence, various \( Q \) can be designed by \( K \) using Eq. (5). For example, if the controller is chosen as \( K = (J_m s + c_m)D \) then, \( Q \) function has the form of

\[ Q = \frac{D}{s + D} \]  

(11)

Therefore, it can be roughly said that the disturbances can be attenuated below the cutoff frequency \( \omega_c = D \text{ [rad/s]} \) of Eq. (11). Figure 3 shows the overall control structure with RIC and feedback controller.
3. INDUSTRIAL EMULATOR

Industrial Servo Emulator (Model 220) used in motion control is the practical motion control equipment including spindle drives, turntables, belts, and automated assembly machines.

3.1 System structure

An Industrial Servo Emulator shown in Fig. 4 was designed to provide insight to control system principles through hands-on demonstration and experimentation. Each consists of an electromechanical plant and a full complement of control hardware and software. The industrial emulator consists of a multi-mass system connecting a drive inertia and load inertia via a drive belt through an assembly gear.

In the Fig. 4, $m_{el}$ and $m_{ld}$ are the weights of each brass element loaded on each plate, and $r_{el}$ and $r_{ld}$ are the distances from the center of the plates. $n_{el}$ and $n_{ld}$ are the number of teeth in the assembly gear. The inertia is determined by $m_{el}$ and $r_{el}$, as well as $m_{ld}$ and $r_{ld}$. The order of the transmission characteristics varies depending on the drive vector. In the experiment, $m_{el}$: 0.5 kg and $m_{ld}$: 0.2 kg were used, with $r_{el}$: 10 cm and $r_{ld}$: 5 cm. The assembly gear had $n_{el}$: 36 and $n_{ld}$: 24. Table 1 shows mass parameters.

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<thead>
<tr>
<th>Table 1. Mass parameters</th>
<th>Radius from center of plate (m)</th>
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<tr>
<td>Mass of brass weights (kg)</td>
<td>On drive disk: 0.8kg</td>
</tr>
<tr>
<td></td>
<td>On load disk: 2.0kg</td>
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</tbody>
</table>

A disturbance motor connects to the load disk via a 4:1 speed reduction and is used to emulate viscous friction and disturbances at the plant output. A brake below the load disk may be used to introduce Coulomb friction.

![Industrial emulator (ECP System model 220).](image)

The drive inertia and load inertia each have a 16,000(counts/rev) encoder attached, and the drive inertia is directly connected to a brushless DC motor. A computer via a DSP board controls the industrial emulator. In other words, the model and each controller are built into the DSP board. The input/output data to be observed are the drive torque [N-m] and the motor angular velocity [rad/s] input by the brushless DC motor.

3.2 System identification

In this section, the inertia, gain, and damping ratio of the Industrial Emulator are found indirectly by measuring their system characteristics. The corresponding block diagram which has a PD control is shown in Fig. 5 and the output/input transfer function and typical standard $2^{nd}$ order dynamic equation are given by
\[ c(s) = \frac{\theta(s)}{r(s)} = \frac{k_p k_m}{s^2 + (c + k_d k_m / J) \pi + k_p k_m / J} \tag{12} \]

Comparing Eq. (12) with Eq. (13), the natural frequency \( \omega_n \) and damping ratio \( \zeta \) become as follows:

\[ \omega_n = \sqrt{\frac{k_p k_m}{J}} \quad \zeta = \frac{1}{2 \omega_n} \left( \frac{c + k_d k_m}{J} \right) = \frac{k_d k_m}{2 \omega_n \sqrt{k_p k_m}} \tag{14} \]

![Figure 5: Control block diagram to measure parameters of Industrial Emulator using a PD controller.](image)

The hardware gain \( k_{hw} \) of the system consists of the product:

\[ k_{hw} = k_c k_v k_s k_m = 5.75 \tag{15} \]

where: \( k_c \) (DAC gain) = 10V / 32,768 DAC counts, \( k_v \) (Servo Amp gain) = approx. 2 [amp/V], \( k_s \) (Servo Motor Torque constant) = approx. 0.1 [N-m/amp], \( k_m \) (Encoder gain) = 16,000 pulses (counts) / 2 radians, \( k_c \) (Controller Software gain) = 32 [controller input counts / encoder or ref. input counts]

**4. EXPERIMENTAL RESULTS**

The plant parameters which were described in previous chapter are used in designing controller. PID, LQ and RIC structure will be used to compare performance of disturbance attenuation. The experimental setup shown in Fig. 4.

**4.1 Motion control**

All of experiments use same input reference command and same disturbance. In RIC structure, \( K(s) \) is chosen as

\[ K(s) = \frac{1}{p_m(s)} D(s) = \frac{1}{s (J_m s + c_m)} D(s) \tag{16} \]

The external loop PD controller gains are \( k_p = 0.4142, k_d = 0.0208 \). The practical plant parameters were calculated and the nominal model is chosen as follows:

\[ p(s) = \frac{k_{hw}}{J s^2 + cs} = \frac{5.75}{0.0042s^2 + 0.0071s}, \quad p_m(s) = \frac{J_{hw}}{J_m s^2 + c_s s} = \frac{5.75}{0.005s^2 + 0.008s} \tag{17} \]

The initial parameters of three kinds of controllers are shown in Table 2. Two kinds of disturbance signal, step or sinusoidal form, as shown in Fig. 6 and Fig. 7 are applied load disk. Maximum disturbance is 1 volt (= approx. 0.2 N-m). Two kinds of reference command, step or trapezoidal motion profile, are used. Maximum reference command angle is 4000 counts(90 degrees).

**4.2 Experimental results**

Figure 8 and Fig. 9 depict the step responses and trapezoidal responses of each controller when the disturbance is not applied. The trapezoidal responses are better performance than step responses. The response of RIC controller has the best performance. Figure 10 through Fig. 12 show the disturbance attenuation property of step responses and trapezoidal responses of each controller in the presence of step disturbance. For step command, RIC has the best performance, even though larger overshoot occurs in the transient response. In case of trapezoidal responses, RIC has much better
performance than LQR, and PID. Figure 12 shows only the tracking errors of each controller. In case of RIC the disturbance is attenuated very quickly. In case of PID, the error decreases very slowly.

<table>
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<th>Table 2. Initial parameter values</th>
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<tr>
<td><strong>PID</strong></td>
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<tr>
<td>$k_p = 0.4143$</td>
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<tr>
<td>$k_i = 0.4$</td>
</tr>
<tr>
<td>$k_d = 0.0208$</td>
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Figure 6. Step disturbance

Figure 7. Sinusoidal disturbance

Figure 8: Step responses of each controller without disturbance

Figure 9: Trapezoidal responses of each controllers without disturbance

Figure 10: Step responses for each controller in the presence of step disturbance.

Figure 11: Trapezoidal responses in the presence of disturbance for each controller

Figure 12: Tracking errors of each controller with step disturbance for trapezoidal command

Figure 13 shows the tracking errors of trapezoidal command for various D when the sinusoidal disturbance is applied. If D is increased from 1 to 10, the tracking error is reduced from 6 degrees to 0.6 degrees. Consequently, it can be easily seen that the performance is depends on the gain D.

Now the experiment to compensate the friction is performed to consider affection by friction. The experiment for Coulomb friction was performed. 0.5 N-m of friction torque was applied to the load shaft. Figure 14 shows the reference command in case of Coulomb friction attenuation experiment. Distance=4000 counts, Velocity=8000 counts/sec, Dwell time=4 sec. Figure 15 shows the tracking errors for each controller in the presence of Coulomb friction. RIC has the best property to attenuate tracking error by friction.
5. CONCLUSIONS

Through the motion control experiment using Industrial Emulator (Model 220 by ECP), the performance comparison of three kinds of controllers such as PID, RIC, and LQR was carried out. It was shown that RIC has best performance in the presence of disturbances such as step disturbance and sinusoidal one and Coulomb friction. The design process of PID controller is the most simple, unlike the design process of the advanced modern LQR in the state space is the most complex. RIC lies between PID and LQR in view of complexity of design. To apply LQR, the additional state observer should be used in order to estimate disturbance. So, we suggest RIC as a high precision controller in motion control field in view of complexity and performance.

REFERENCES


*jpark@yu.ac.kr; phone 82 53 810-2498; fax 82 53 810-4629; http://yu.ac.kr/~jpark/