Crossing a step of a robust power assist cart taking tumble prevention of conveyed objects into account

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Abstract
Owing to the aging of societies, the research and development of power-assist systems that increase human-operational forces have become popular. However, a power-assist system is a typical human-machine system. Therefore, the design and use of such a system should take into account the safety of both the operator and the conveyed objects. Some of the authors have already studied on power-assist systems, using an assist-control system design consisting of an impedance controller, a disturbance-accommodating optimal controller, a disturbance observer, and a reaction-force controller. However, unintentional input forces acting on a power-assist system may cause reckless motion and severe accidents in actual use. Therefore, we discuss the use of a robust control method to prevent this abrupt motion by substituting a frequency-shaped disturbance-accommodating optimal controller (FSDAOC) for the original optimal controller. The FSDAOC enables us to eliminate high-frequency-range motion by its frequency-shaping characteristic. The objective of this study is to assess the effectiveness of the FSDAOC-based method in enabling a power-assist cart to cope with the unexpected input forces caused by crossing a step without its load tumbling. We verified the effectiveness experimentally and found that our method had a tumble-prevention rate of 80%, which was an improvement over the 23% of the previous method.

Key words: Power assist, Impedance control, Shock, Optimal control, Frequency-shaping, Tumble prevention

1. Introduction

Owing to the aging of societies, the research and development of power-assist systems that increase human operational forces have become popular for industrial applications. Some of the authors studied robust control-system design for power-assist carts (Hara, et al., 2010, 2012, 2013). In Hara, et al., (2013), the authors observed that conveyed objects tumble easily; therefore, two actuators for realizing the power assist and smooth handling of the objects were applied. Hara, et al., (2013) adopted an impedance control-based power-assist control method. The impedance characteristic of the power-assisted cart was regarded as the disturbance dynamics for the smooth-handling control part. The handling-control algorithm was accomplished using a disturbance-accommodating optimal-control method (Hara, et al., 2010). Moreover, a robust assist-control method that took into account the influences of structured uncertainties and vibratory reaction forces to the operators was also proposed in Hara, et al., (2010).

Generally, the safety of both the operators and conveyed objects is one of the most important issues in the actual use of power-assist carts. Therefore, some of the authors investigated the reduction of the abrupt handling motion to prevent conveyed objects from tumbling, and adopted the frequency-shaped disturbance-accommodating optimal control (FSDAOC) as the handling-control algorithm in Hara, et al., (2013). During the actual use of power-assist carts, preventing conveyed objects from tumbling when crossing a step is one of the most common and important obstacles. This study focuses on the influences of the disturbance forces to the carts when crossing steps (Ioi, et al., 2013, Tashiro and Murakami, 2007, Yokota, et al., 2012). The effectiveness of the FSDAOC-based method for such a disturbance has not been addressed yet. Therefore, the objective of this study is to verify the effectiveness experimentally.
2. Controlled object

The controlled object and its mathematical model in this study are shown in Figs. 1 and 2. The width, length, and loading-plate height from the ground of the cart are 0.6, 0.9, and 0.3 m, respectively. In the same manner as Hara, et al., (2013), a linear actuator (DC voice coil motor; stroke constraint: ±0.05 m) is installed on the cart. The conveyed objects are set on the slider of the actuator. We adopt an aluminum alloy cylinder whose diameter, length, and thickness are 30, 300, and 2 mm, respectively, as one of the objects. The minimum acceleration at which it begins tumbling was determined as 0.98 m/s$^2$ by Takagi, (1990).

![Fig. 1 Photo of the controlled object example in this study. A power assist cart. The controlled object is an aluminum alloy cylinder.](image)

![Fig. 2 Controlled object model. The model is derived from the actual experimental system in Fig. 1.](image)

Table 1  Mass parameters of the controlled object.

| Parameter                        | Value  |
|----------------------------------|--------|
| $m_b$ Equivalent mass of the cart | 131.9 kg |
| $m_a$ Equivalent mass of the linear actuator slider and the conveyed object | 1.96 kg |

3. Control system design

The relative displacement between the cart and the slider should be controlled to realize the conveyance because the slider is connected with the cart by an actuator only in the model in Fig. 2 (Hara, et al., 2010). Moreover, as mentioned in Section 1, the frequency-shaped disturbance-accommodating optimal control (FSDAOC) method in Hara, et al., (2013) is applied to this study. In this section, to suppress the high-frequency-range motion of the relative motion between the cart and the slider, the following frequency-shaping weight is adopted on the relative displacement:
where \( \omega_q \) and \( \zeta_q \) are the cut-off angular frequency and the damping ratio, respectively. To solve the linear-quadratic (LQ) optimal-control problem, including the weight in Eq. (1), the frequency-shaping characteristic is realized as a state-space formula, as follows:

\[
\begin{align*}
\dot{x}_q(t) &= A_q x_q(t) + b_q [x_s(t) - x_q(t)] , \\
x_q(t) &= [x_q(t) \quad \dot{x}_q(t)]^T , \\
A_q &= \begin{bmatrix}
0 & 1 \\
-\omega_q^2 & -2\zeta_q \omega_q \\
\end{bmatrix}, \\
b_q &= \begin{bmatrix}
0 \\
\omega_q^2 \\
\end{bmatrix}, \\
c_q &= [1 \ 0],
\end{align*}
\]

where \( x_q(t) \) is the state-variable vector regarding the dynamics in Eq. (1), \( x_s(t) - x_q(t) \) is the relative displacement between the slider \( x_s(t) \) and the cart \( x_q(t) \), and \( z_q(t) \) is a controlled variable. By combining the above frequency-shaping dynamics and the dynamics consisting of the slider and the cart in Hara, et al., (2012), the augmented system is obtained as follows (Hara, 2004):

\[
\begin{bmatrix}
\dot{x}_s(t) \\
\dot{x}_q(t) \\
\dot{x}_s(t) \\
\dot{x}_q(t) \\
\dot{x}_q(t) \\
\omega_q^2 \\
\end{bmatrix} = \begin{bmatrix}
0 & 1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 \\
0 & 0 & 0 & -c_{id} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 \\
0 & 0 & 0 & 0 & 0 & 0 \\
\omega_q^2 & 0 & -\omega_q^2 & 0 & -\omega_q^2 & -2\zeta_q \omega_q \\
\end{bmatrix} \begin{bmatrix}
x_s(t) \\
x_q(t) \\
x_s(t) \\
x_q(t) \\
x_q(t) \\
\end{bmatrix} + \begin{bmatrix}
0 \\
K_e \\
0 \\
0 \\
0 \\
F_s(t) \\
\end{bmatrix} + \begin{bmatrix}
c_e(t) \\
0 \\
0 \\
0 \\
0 \\
\end{bmatrix}, \quad (\text{3})
\]

where \( m_{id} \) and \( c_{id} \) are the “impedance parameters;” they are the virtual mass and viscous damping coefficient to realize the impedance characteristics, respectively. \( m_s, K_e, c_e(t) \), and \( F_s(t) \) are the mass of the slider and the conveyed objects, the thrust constant of the linear actuator, the control input voltage to the linear actuator, and the operator’s force, respectively. The state-space vector of the augmented system is defined as follows:

\[
x(t) = [x_s(t) \quad \dot{x}_s(t) \quad x_q(t) \quad \dot{x}_q(t) \quad x_q(t) \quad \dot{x}_q(t)]^T . \quad (\text{4})
\]

Let us apply the LQ optimal-control theory to the augmented system. Generally, its criterion function is defined as follows:

\[
J = \int_0^T \left[ x_s^T(t) Q x_s(t) + r c_e^2(t) \right] dt , \quad (\text{5})
\]

where \( r \) is the weight of the control input. Its value is set to 1.0 in this study. The weights on the state variables \( Q \) are selected as

\[
Q = \text{diag}[0 \ 0 \ 0 \ q^2_e \ 0] , \quad (\text{6})
\]

where \( q^2_e \) is a weight coefficient for a state variable in the frequency-shaping dynamics. In this problem, other weights...
on no frequency-shaped state variables should be zeros for reducing high-frequency-range motion. Hence, Eq. (5) can be rewritten concretely as follows:

$$J = \int_0^\infty \left( (q(t)z_0(t))^2 + e_a^2(t) \right) dt.$$  

(7)

The FSDAOC input is obtained as the optimal-control input $e_a(t)$ in Eq. (7), by solving the above LQ optimal-control problem.

4. Crossing a step (transition strip)

This section provides some experimental results. In the experiments, we consider that the cart crosses a step (transition strip) under a door, as shown in Fig. 3, over about five seconds. The strip’s height and width are 5 and 50 mm, respectively.

First, the FSDAOC-based controller in Hara, et al., (2013) is applied. The LQ control weight $q_i^2$ on the square value of the relative displacement between the cart and slider is set to $3.1 \times 10^4$. $w_i$ is set to $6.28 \times 10^{-2}$ rad/s. $q_i^2$ and $w_i$ are tuned by simulations based on an example of the operational force to move the cart in Fig. 4. $c_i$ is tuned as 0.707 to obtain the Butterworth characteristic. Its gain is shown in Fig. 5. The mass of the slider and controlled objects $m_a$ is 1.96 kg, as summarized in Table 1. The thrust constant of the linear actuator $K_a$ is 2.51 N/V. The impedance parameters $m_d$ and $c_d$ are tuned as 30.0 kg and 10.0 N·s/m, respectively, by referring to the conventional experiments (Hara, et al., 2012). If we set $m_d$ to too small value, the power assisting effect is enlarged, but the impedance control system may become unstable. Then, the designer should choose $m_d$ appropriately. Simultaneously, $c_d$ is tuned such that the cart can be stopped easily by the viscous damping effect. As shown in Fig. 4,
at approximately 3 s, the front wheels cross the strip. The rear wheels cross the strip at approximately 5.5 s.

The response example of the proposed FSDAOC-based method is shown in Fig. 6. Figure 6 shows the displacement of the cart, the acceleration of the linear actuator slider, and the relative displacement of the slider, respectively. The maximum slider acceleration in Fig. 6(b) is 0.51 m/s², in this case. This case shows 12 no-tumbling responses in 15 trials. Therefore, the tumble-prevention rate corresponds to 80% (12/15). On the other hand, when we applied the no frequency-shaped controller (the disturbance-accommodating optimal control in Hara, et al., (2010)) whose \( q^2 \) is the same as that in the previous case, the no-tumbling responses are limited to 4 times in 15 trials. The tumble-prevention rate is 23% (4/15). Its example response is shown in Fig. 7. In this case, the maximum slider acceleration in Fig. 7(b) is 0.83 m/s². Therefore, the effectiveness of FSDAOC is verified experimentally by comparing the two cases. Of course, these rates depend on the specifications of steps, moving speed, etc. The controller design parameters should be tuned appropriately taking actual specifications into account.

5. Conclusion

This technical note focused on the influence of disturbance forces on power-assist carts when crossing steps in industrial environments. This is one of the most common and important obstacles for such a system, and the tumbling of the conveyed objects should be prevented. The effectiveness of the FSDAOC-based method to realize tumble prevention was verified experimentally.

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Fig. 6 Example response of the proposed control system when crossing a transition strip. The acceleration of the linear actuator slider (b) is effectively reduced by the low-pass characteristic.

Fig. 7 Example response of the case without frequency-shaping when crossing a transition strip. The maximum value of the acceleration of the linear actuator slider in (b) is larger than that in the proposed method in Fig. 6 (b).