Suppression of Low-frequency Lateral Vibration in Tilting Vehicle Controlled by Pneumatic Power

A. Kazato, S.Kamoshita
Railway Technical Research Institute, Tokyo, Japan

1. Introduction
The authors performed a study intended for an improvement on the ride comfort of tilting vehicles. In Japan, a tilting system with pneumatic power, which tilts the vehicle body (Fig.1). However, the tilting system has a certain problem area causing a possible motion sickness as caused by tilting delay in transition curve tracks and low-frequency rolling motions on straight tracks [1].

The authors developed a new tilting system with electro-hydraulic power [2]. However, the system has the following two problem areas: One is that the electro-hydraulic system is more expensive than the pneumatic system, and other is that the high stiffness increases lateral vibration of frequency exceeding the 1-Hz [3]. To avoid these problems, the authors studied how to improve the performance of the current tilting system with pneumatic power. First, we constructed a numerical simulation model of the pneumatic servo control system, which is based on the state equation of air. The validity and the effectiveness of the proposed system were demonstrated by comparison with results of experiments. Secondly, we analyzed the ride comfort of a full vehicle model by multi-body dynamics simulation. We calculated motion sickness dose value for lateral motion (as abbreviated MSDV_y) [1] and ride comfort level (L_T) from the acceleration observed on the floor of the vehicle body. Consequently, we have indicated that the proposed system with a flow control valve was able to suppress the low-frequency lateral vibration, which causes motion sickness.

2. Tilt control system with pneumatic power
2.1 Current system
Figure 2 shows the current pneumatic servo system for tilt control with a pressure control valve. The target tilt angle is calculated from the curvature and the cant of the track and the running speed of the vehicle. The command voltage, proportional to the difference between the target tilt angle and the actual tilt angle by displacement of the cylinder, drives the servo valve via the current driver. The servo valve is a proportional pressure control valve. The internal pressure of the cylinder is fed back to the
spool displacement of the valve. In Fig.2, when the spool moves left, the source air is charged to the left chamber of the cylinder. At the same time, the air of the right chamber of the cylinder is discharged into the atmosphere. Then the cylinder displaces.

The target tilt angle is generated following a proportional control law with a step gain, called "Mode CA". The Mode CA has developed for the improving of a poor response of the current system. A detailed description of the Mode CA can be found in section 2.3.

2.2 Proposed system

Figure 3 shows the proposed system. This system has a flow control valve substituting the pressure control valve of the current system. The flow control valve doesn’t have channels for pressure feedback as the pressure control valve. Therefore, the aperture size of the flow control valve is proportional to the flow late. The flow characteristic is improved by use of the flow control valve. And improving the response and the power of the actuator are expected.

The generate law of the target tilt angle is called "JTM pattern". The JTM pattern is calculated based on an ergonomic evaluation functions. A detailed discussion of JTM pattern can be found in section 2.3.
2.3 Target tilt angle

(1) Mode CA

Figure 4 shows an example of shape of the Mode CA. The lead time \( t_0 \) and step gain \( S \) are given for compensating operation delay of the pneumatic actuator. The length of a transition curve is apparently extended for preventing increase of the roll angular velocity of the vehicle.

(2) JTM pattern

Figure 5 shows an example of shape of the JTM pattern. The JTM pattern was developed for next-generation tilt control system by RTRI [2]. On this system, the target angle \( \phi(t) \) is calculated with following evaluation functions \( f_{JTM}(\phi(t)) \). (Eq. (1) and Eq. (2))

\[
f_{JTM}(\phi(t)) = (1 - \alpha) \cdot f_{JT}(\phi(t)) + \alpha \max\{y_f(\phi(t))\}_{i=0}^{T_i}
\]
Challenge D: A world of services for passengers

Here,

\[ f_{JT}(\phi(t)) = 0.6 \max \left[ y_p(\phi(t)) \right]_{r_0}^{r_f} + 0.3 \max \left[ y_j(\phi(t)) \right]_{r_0}^{r_f} + 0.03 \max \left[ \theta_p(\phi(t)) \right]_{r_0}^{r_f} + 0.12 \max \left[ \theta_j(\phi(t)) \right]_{r_0}^{r_f} \]

\( f_{JT} \) was made from the \( TC_T \) (Transition Curve Total index), and consists of maximum values of four state quantities observed on the car body: lateral acceleration \( y_p \), lateral jerk \( y_j \), roll angular velocity \( \theta_p \), and roll angular acceleration \( \theta_j \). These state quantities are supposed from running speed, curvature and cant of the track. Furthermore, \( f_{JTM} \) consists of \( f_{JT} \) and low-frequency lateral acceleration \( y_f \) (applied a bandpass filter with a focus on 0.3Hz to \( y_p \)) which cause motion sickness. The target tilt angle \( \phi(t) \) is calculated as minimize \( f_{JTM} \) with quadratic programming. To achieve tilting with the JTM pattern, a high response actuator is needed.

\[ \text{Fig.5 Example of shape of the JTM pattern} \]

3. Model construction of pneumatic servo system

The simulation model of the pneumatic servo system is constructed for investigating the performance of the proposed system. In this model, the input variable is the target displacement of the cylinder, and the output variable is the cylinder force.

3.1 Symbols

- \( a_i \) Aperture area of port of servo valve [m\(^2\)]
- \( A_i \) Effective pressure area of cylinder [m\(^2\)]
- \( A_{sp} \) Effective pressure area of back side of spool [m\(^2\)]
- \( b \) Critical pressure ratio
- \( C_i \) Sonic conductance of aperture area of port of servo valve [m\(^3\)/(s·Pa)]
- \( C_v \) Specific heat at constant volume [m\(^3\)/(s\(^2\)·K)]
- \( G_i \) Mass flow rate through port of servo valve [kg/s]
- \( h_i \) Heat transfer coefficient between air and inner surface in chamber [W/(m\(^2\)·K)]
- \( k_{sp} \) Spring constant of return-to-neutral spring [N/m]
- \( P_e \) Atmospheric pressure [Pa, abs.]
- \( P_i \) Air pressure in chamber [Pa, abs.]
- \( P_s \) Air pressure of source [Pa, abs.]
3.2 Servo Valve

Figure 6 and 7 shows balance of forces acting on the spool of the pressure control valve and the flow control valve. The spool is subjected to the following forces: the solenoid force generated by current from the current driver, the return-to-neutral spring force, and pressure feedback forces (only if the pressure control valve). These forces decide the displacement of the spool and the aperture area of opening ports.

The driving current \( i \) is obtained by the Eq. (3).

\[
i = K (x_{pc} - x_{p}) \quad (3)
\]

Here, \( x_{pc} \): Target cylinder stroke (=0.0272 \( \phi \)) [m]

In case of the pressure control valve, the total force \( F_{sp} \) of the solenoid force and pressure feedback forces, which subjects to the spool, is obtained by Eq. (4). In case of the flow control valve, \( F_{sp} \) is obtained by Eq. (5).

\[
F_{sp} = k_{s}i - A_{sp}(P_{1} - P_{2}) \quad (4)
\]

\[
F_{sp} = k_{s}i \quad (5)
\]

The displacement of the spool \( x_{sp} \) is obtained by Eq. (6).

\[
x_{sp} = F_{sp}/k_{sp} \quad (6)
\]

Then the aperture area of ports \( a_{i} \) is obtained by Eq.(7).
Challenge D: A world of services for passengers

\[
a_i = \begin{cases} 
0, & |x_{sp}| \leq \varepsilon \\
W_p \left( |x_{sp}| - \varepsilon \right), & |x_{sp}| > \varepsilon
\end{cases}
\] (7)

Here, \( w_p \): Circumferential length of a port [m]
\( \varepsilon \): Length of overlap [m]

When the sign of \( x_{sp} \) is positive, high-pressure air leads to the chamber 1, and atmospheric air leads to the chamber 2. When the sign of \( x_{sp} \) is negative, high-pressure air leads to the chamber 2, and atmospheric air leads to the chamber 1.

3.3 Cylinder force

Getting time variation of the internal pressures of chambers precisely is important for acquiring the cylinder force. Therefore, the internal pressures of chambers are calculated from the equation of state of the air (Eq. (8)).

\[
P_i = \frac{W_iRT_i}{V_i} \quad (8)
\]

The derivative with respect to time of Eq. (8) is obtained by Eq. (9).

\[
\frac{dP_i}{dt} = -\frac{P_i dV_i}{V_i dt} + \frac{W_iR dT_i}{V_i dt} + \frac{RT_i G_i}{V_i} \quad (9)
\]

As will be noted from the Eq. (9), the time variations of the pressures are the sums of the following time variation terms: the volume, the temperature and the mass of the air.

The volumes of the chambers and its derivatives with respect to time are obtained by the Eq. (10) and Eq. (11).

\[
V_i = V_{i0} \pm A_i x_{sp} \quad (10)
\]

\[
\frac{dV_i}{dt} = \pm A_i \frac{dx_{sp}}{dt} \quad (11)
\]

When the time variations of the internal temperatures of the chambers are calculated, the transfers of the heat quantities \( Q_i \) (Eq. (12)) between the internal air and the inner surface are considered [4].
Challenge D: A world of services for passengers

\[ Q_i = hS_i(T_a - T_i) \] (12)

The time variations of the internal temperatures are obtained by the Eq. (13).

\[
\frac{dT_i}{dt} = \begin{cases} 
\frac{1}{C_v W_i} \left[ G_i C_v (T_a - T_i) + RT_i G_i + Q_i \right] & \quad (G_i \geq 0) \\
\frac{1}{C_v W_i} \left[ RT_i G_i + Q_i \right] & \quad (G_i < 0)
\end{cases}
\] (13)

The mass flow rates \( G_i \) are obtained by the Eq. (14) [5].

\[
G_i = \begin{cases} 
C_i \rho N P_a \frac{T_N}{T_a} & \quad (P_d / P_u \leq b) \\
C_i \rho N P_a \frac{T_N}{T_a} \sqrt{1 - \left( \frac{P_d / P_u - b}{1 - b} \right)^2} & \quad (P_d / P_u > b)
\end{cases}
\] (14)

The sonic conductance \( C_i \) are obtained by the Eq. (15).

\[
C_i = \frac{\alpha_{sp} a_i \sqrt{\kappa}}{\rho N \sqrt{RT_N}} \left( \frac{2}{\kappa + 1} \right)^{\frac{k+1}{2(k-1)}}
\] (15)

The internal pressures \( P_i \) are obtained by substituting the equations (10) - (15) into the Eq. (9). The cylinder force is calculated by the Eq. (16).

\[
F_p = A_1 P_1 - A_2 P_2 - (A_1 - A_2) P_e
\] (16)

3.4 Validation of servo system model

Numerical simulations and experiments with a half vehicle model were conducted for validation of the servo system model. Figure 8 shows the experimental facility, and Fig. 9 shows the simulation model of the half vehicle model. Table 1 shows parameter values of pneumatic servo system.

Figure 10 shows the result of the current system with the pressure control valve. Figure 11 shows the result of the proposed system with the flow control valve. In both cases, simulation results showed good agreement with experimental results. As to cylinder stroke, the operation delay time of the flow control valve was shorter than the pressure control valve, and the maximum displacement of the flow control valve was greater than the pressure control valve. As mentioned above, the servo system model was considered valid. Furthermore, it was demonstrated that the flow control valve has high performance characteristics than the pressure control valve.
Table 1 Parameter values of pneumatic servo system

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Value</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_1$</td>
<td>$1.227 \times 10^{-2}$ $m^2$</td>
<td>$S_{k10}$</td>
<td>$8.345 \times 10^{-2}$ $m^2$</td>
</tr>
<tr>
<td>$A_2$</td>
<td>$1.131 \times 10^{-2}$ $m^2$</td>
<td>$S_{k20}$</td>
<td>$9.802 \times 10^{-2}$ $m^2$</td>
</tr>
<tr>
<td>$A_p$</td>
<td>$0.63 \times 10^{-4}$ $m^2$</td>
<td>$T_0$</td>
<td>293.15 K</td>
</tr>
<tr>
<td>$b$</td>
<td>0.5</td>
<td>$T_a$</td>
<td>293.15 K</td>
</tr>
<tr>
<td>$C_r$</td>
<td>$717$ $m^2/(s^2 \cdot K)$</td>
<td>$T_N$</td>
<td>293.15 K</td>
</tr>
<tr>
<td>$h_1$</td>
<td>13.66 $W/(m^2 \cdot K)$</td>
<td>$T_f$</td>
<td>293.15 K</td>
</tr>
<tr>
<td>$h_2$</td>
<td>25.67 $W/(m^2 \cdot K)$</td>
<td>$V_{10}$</td>
<td>$1.841 \times 10^{-3}$ $m^3$</td>
</tr>
<tr>
<td>$P_0$</td>
<td>401.3 kPa, abs.</td>
<td>$V_{20}$</td>
<td>$1.697 \times 10^{-3}$ $m^3$</td>
</tr>
<tr>
<td>$P_e$</td>
<td>101.3 kPa, abs.</td>
<td>$\alpha_{sp}$</td>
<td>0.6</td>
</tr>
<tr>
<td>$\rho$</td>
<td>601.3 kPa, abs.</td>
<td>$\kappa$</td>
<td>1.4</td>
</tr>
<tr>
<td>$R$</td>
<td>$287$ $m^2/(s^2 \cdot K)$</td>
<td>$\rho_s$</td>
<td>$1.185 \times 10^{-3}$ $kg/dm^3$</td>
</tr>
</tbody>
</table>
4. Improvement effects of ride comfort of a full vehicle by calculation

4.1 Vehicle model

Figure 12 shows the dynamic simulation model of a tilting vehicle. This model was built with SIMPACK [6]. Table 2 shows major specifications.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Value</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$BC$</td>
<td>14 m</td>
<td>$I_{bl}$</td>
<td>$2.13 \times 10^4$ kg·m$^2$</td>
</tr>
<tr>
<td>$c_f$</td>
<td>20 kN·s/m</td>
<td>$M_b$</td>
<td>25.2 t</td>
</tr>
<tr>
<td>$h_b$</td>
<td>1.56 m</td>
<td>$WB$</td>
<td>1.125 m</td>
</tr>
<tr>
<td>$h_f$</td>
<td>2.27 m</td>
<td>(Gauge)</td>
<td>1.067 m</td>
</tr>
</tbody>
</table>

![Table 2 Major specifications of full vehicle model](image)

Cross section

Top view

Fig.12 Dynamic simulation model of tilting vehicle (Full vehicle model)

4.2 Conditions of calculation

Conditions of the calculation are as follows.

- Running speed: 80km/h (constant)
- Track geometry: 4km length, seven sharp curves (300m in radius)
- Track irregularity: line alignment, cross level

4.3 Conditions of tilt control

Table 3 shows conditions of tilt control. The target tilt angle was calculated off-line in advance, and it was entered into the servo system model during simulation.
4.4 Results of simulation

(1) Motion sickness

Evaluation of motion sickness was carried out with motion sickness dose value for lateral motion (abbreviated as $MSDV_y$) [1]. The $MSDV_y$ is obtained by Eq. (17). If the evaluating time is less than 30 minutes, the calculated value are translated into the equivalent of 30 minutes.

$$MSDV_y = \left( \int_0^T a_w^2(t) dt \right)^{1/2} \quad (17)$$

$a_w$: Lateral acceleration applied $W_f$ filter (Fig. 13) [m/s²]

$T$: Evaluating time [s]

Figure 14 shows the comparison of the $MSDV_y$ for the car body. The proposed system was able to decrease the $MSDV_y$ to 50% as compared to the condition of without control, to 75% compared to the condition of the current system. Then, Figure 15 shows time history comparison of ride comfort between current system and proposed system. Here, the short-term $MSDV_y$ is the value obtained over a duration of 20 seconds. Figure 15 shows that the low-frequency lateral acceleration and the short-term $MSDV_y$ were larger while travelling in transition curve sections.
(2) Ride comfort

Although the high responsibility of the tilt actuator reduces low-frequency vibration, it often increases high-frequency vibration. The proposed system needs to be investigated whether to increase such vibration. Ride comfort was evaluated using an index \( L_T \), which is a representative index for evaluating the ride comfort of railway vehicles in Japan. This level is employed by Japanese National Railways on the basis of ISO-2631 and is given by Eq. (18) under the assumption that the frequency of vibration is \( f \), the car body vibration acceleration power spectral density (PSD) is \( P_\alpha \), and the weight of the ride comfort filter is \( W_{LT} \) (Fig.16).

\[
L_T = 10 \log_{10} \frac{1}{\alpha_{ref}} \int_{f_1}^{f_f} W_{LT}^2(f) P_\alpha(f) df \quad (18)
\]

Figure 17 shows the comparison of the \( L_T \) value for the car body. Although the proposed system showed a slight increase the \( L_T \) value, the difference between that system and the current system was less than 1dB. And in the bottom graph in Fig.15, the short-term \( L_T \) is shown. The short-term \( L_T \) is the value obtained over a duration of 20 seconds. In any section or with any control condition, there was no significant difference each other.
5. Conclusions

Consequently, we have indicated the following conclusion.

- The proposed model of pneumatic servo control system enabled to represent the behavior of the tilt actuator.
- The proposed system with a flow control valve has a higher level of response and more power to compare with the current system with a pressure control valve.
- The proposed system decreases the $MSDV_{y}$ to 50% as compared to the condition of without control, to 75% compared to the condition of the current system.
- The proposed system has not worsened ride comfort in vibration.

The pneumatic system, which is essentially considered poor response and power, is able to suppress the low-frequency lateral vibration, which causes motion sickness. Then this system is not only limited to low cost and low environmental impact but also provided a satisfactory ride comfort.

Acknowledgement

Deep gratitude is expressed to the relevant personnel at the Pneumatic Servo Controls, Ltd.

References


