PERFORMANCE ANALYSIS OF LATERAL GUIDANCE SYSTEM FOR DUAL MODE TRUCK

Hitoshi TSUNASHIMA and Tetsuya KANEKO

ABSTRACT
This paper describes a computer simulation study and field test on performance of lateral guidance system for Dual Mode Truck. A stability limit of vehicle lateral motion is analyzed by using 9 DOF vehicle dynamic model. Relations between steering parameters and stability limit are shown. Dynamics of power steering device in the lateral guidance system are described. Experiments with actual Dual Mode Truck is carried out to show the effectiveness of the simulation study. Both the simulation study and experiment show that lateral guidance with one side guide rail causes unstable vehicle motion. The results have highlighted that power steering device has large influence on the vehicle running stability. It is shown that the unstable motion can be suppressed by cutting off the power steering equipment in the guideway.

INTRODUCTION
Truck transport is dominant in freight transport in Japan and is increasing as frequent services are required to meet consumer's demand. However it causes several social problems such as traffic congestion, environmental issues and shortage of drivers. Ministry of Construction (MOC) of Japan proposed and developed new freight transport system (Fig. 1) to solve these problems since late 1980's.

The Dual Mode Truck (DMT) system (Fig. 2) is a new type of freight transport system that can be operated both in exclusive guideway and conventional road by electric power. [1] Full automatic unmanned operation with very short headway (1 - 3 sec.) is planned for the guideway operation. Manual operation is provided for conventional road operation. The vehicle consists of lateral guidance system to guide the vehicle inside the guideway, sensor for detecting spacing to preceding vehicle and automatic vehicle operation system.

In the guideway, the scheduled vehicle speed is about 60 km/h for intercity system and is 100 km/h for intercity system. As the lateral guidance system, mechanical guidance system with guide rail and guide wheel is selected for prototype system based on some experience of Automated Guideway Transit system.

The purpose of this study is to improve running stability of the vehicle with mechanical guidance system coupled with conventional steering system. Firstly, we model the dynamics of DMT with mechanical guidance system coupled with steering wheel. Dynamics of power steering device are also discussed. Secondly, simulation study is
carried out to show the vehicle performance in realistic situation. The relation between vehicle stability limit and steering parameters is analyzed in this simulation study. We carried out field test to confirm the results of the simulation study. Based on the simulation study and the field test with prototype DMT, improvements of steering system are proposed.

**VEHICLE DYNAMIC MODEL**

_Nomenclature_

\( \varphi \): yaw angle  \( \beta \): side slip angle  
\( \phi \): roll angle of vehicle body  \( \phi_F \): roll angle of front axle  
\( \phi_R \): roll angle of rear axle  \( \delta \): front wheel steering angle  
\( \theta \): steering wheel angle  \( V \): vehicle forward velocity  
\( m \): vehicle total mass  \( m_s \): sprung mass  
\( m_G \): guide wheel mass  
\( W_F, W_R \): weight on front and rear axle  
\( I_\theta \): roll moment of inertia (body)  
\( I_{\phi F} \): roll moment of inertia (front axle)  
\( I_{\phi R} \): roll moment of inertia (rear axle)  
\( I_Z \): yaw moment of inertia (body)  
\( I_{ZF} \): lumped yaw moment of inertia of wheels and steering link  
\( I_{SW} \): moment of inertia of steering wheel  
\( K_S \): guide bar stiffness  \( K_{\phi F} \): roll stiffness of front suspension  
\( K_{\phi R} \): roll stiffness of rear suspension  \( K_{GW} \): guide wheel radial stiffness  
\( K_Z \): tire vertical stiffness  \( K_{SW} \): torsional stiffness of steering shaft  
\( C_S \): guide bar damping coefficient  \( C_{\phi F} \): front roll damping coefficient  
\( C_{\phi R} \): rear roll damping coefficient  \( C_{SW} \): steering shaft damping coefficient  
\( G_F \): steering gain  \( C_C \): cornering coefficient  
\( n \): steering gear ratio  \( t_P \): pneumatic trail  
\( t_C \): caster trail  \( t_d \): front wheel tread  
\( t_{d0} \): rear wheel outer tread  \( t_{dI} \): rear wheel inner tread  
\( \delta_0 \): toe-in angle  
\( h_S \): distance from roll axis to c.g  
\( h_F, h_R \): front and rear roll center height  
\( M_B \): bias moment of steering system  
\( M_{PRI} \): friction at king pin  
\( M_K \): self aligning moment of steering stabilizer  
\( M_0 \): max. self aligning moment of steering stabilizer  
\( y_{GR}, y_{GL} \): lateral displacement of guide wheel  
\( L_f \): distance from vehicle c.g. to front axle  
\( L_r \): distance from vehicle c.g. to rear axle  
\( D \): position of the guide wheel
\( \varepsilon \): clearance between guide wheel and guideway

\( k_t \): torsion bar stiffness

\( C_B \): steering system damping coefficient

\( A_p \): effective area of ball nut

\( r_c \): pitch circle radius of sector gear

\( p_e \): pressure of power cylinder

\( k_s \): coefficient of power steering fluid

\( g_m \): coefficient of flow rate

\( r_p \): coefficient of control valve

**Vehicle dynamic model without power steering model**

The vehicle model for Dual Mode Truck is shown in Fig. 3. A mechanical guidance system of the vehicle consists of a steering link, guide bar and two guide wheels which contact to wayside guide rails. Front wheels connected with the mechanical guidance system are allowed to rotate about king pins and a steering wheel connected to the steering link through a modeled spring is also allowed to rotate about its axis. Further 2-degree-of-freedom rotational motion of the steering system and lateral motions of the guide wheels relative to the guide bar are assumed. [2]

1) **Vehicle side slip:**

\[
mV(\ddot{\beta} + \phi) - m_h \dot{h} = F_F + F_R + F_G,
\]

(1)

where the cornering force and the force acting on guide bar are

\[
F_F = C_C W_F (\delta - \beta - L_f \phi / V) + C_K K_2 L_y \delta \phi_F,
\]

(2)

\[
F_R = C_C W_R (-\beta + L_f \phi / V),
\]

(3)

\[
F_G = K_S (y_{GL} + y_{GR}) + C_S (\dot{y}_{GL} + \dot{y}_{GR}).
\]

(4)

2) **Rotational motion of front wheel:**

\[
I_{ZF} (\ddot{\delta} + \phi) = F_G / G_F - (t_p + t_e) F_F - M_B - M_K - M_{FRI} + n K_{SW} (\theta - n \delta),
\]

(5)

where \( M_K \) is self aligning moment of front wheel stabilizer given by

\[
M_K = M_0 \text{sign}(\delta).
\]

(6)

3) **Yawing motion of vehicle body:**

\[
I_{Z} \dot{\phi} = L_f (F_G + F_F) - L_f F_R + M_B + M_K + M_{FRI} + C_{SW} \dot{\theta}.
\]

(7)
4) Rolling motion of vehicle body:

\[ I_{\phi} \ddot{\phi} = -K_{\phi F} (\phi - \phi_F) - K_{\phi R} (\phi - \phi_R) - C_{\phi F} (\dot{\phi} - \dot{\phi}_F) - C_{\phi R} (\dot{\phi} - \dot{\phi}_R) + m_s g h S \phi + m_s h V (\ddot{\theta} + \dot{\phi}) . \]  

(8)

5) Rolling motion of unsprung mass:

\[ I_{\phi F} \ddot{\phi}_F = -K_{\phi F} (\phi_F - \phi) - C_{\phi F} (\dot{\phi}_F - \dot{\phi}) - K_Z (t_{20}^2 / 2) \dot{\phi}_F + h_F F_R , \]  

(9)

\[ I_{\phi R} \ddot{\phi}_R = -K_{\phi R} (\phi_R - \phi) - C_{\phi R} (\dot{\phi}_R - \dot{\phi}) - K_Z ((t_{20}^2 / 2) + (t_{21}^2 / 2)) \dot{\phi}_R + h_R F_R . \]  

(10)

6) Motion of guide wheel:

\[ m_G \ddot{y}_{GL} = -C_S \dot{y}_{GL} - K_S y_{GL} - [K_S K_{GW} \varepsilon / (K_s + K_{GW})] \varepsilon + G F_{GL} - m_G (\dot{y}_F - \dot{y}) / G_F - m_G V (\ddot{\theta} + \dot{\phi}) - m_G L_J \dot{\phi} , \]  

(11)

\[ m_G \ddot{y}_{GR} = -C_S \dot{y}_{GR} - K_S y_{GR} - [K_S K_{GW} \varepsilon / (K_s + K_{GW})] \varepsilon + G F_{GR} - m_G (\dot{y}_F - \dot{y}) / G_F - m_G V (\ddot{\theta} + \dot{\phi}) - m_G L_J \dot{\phi} . \]  

(12)

The * terms are considered for pre-loading (\varepsilon < 0). Moreover, \( GF_{GL} \) and \( GF_{GR} \) are the force between the guide wheel and the guide rail.

7) Motion of steering wheel:

\[ I_{SW} (\ddot{\theta} + \dot{\phi}) = -K_{\phi W} (\theta - n \delta) - C_{\phi W} \dot{\phi} . \]  

(13)

The vehicle motion can be simulated by obtaining the solution of these nonlinear differential equation by using the Runge-Kutta-Gill method.

**Dynamics of power steering system**

We modeled the interacting effect of conventional steering system and the mechanical guidance system by using the mechanical coupling as shown in Fig. 3. However, it is expected that a power steering device, which is installed for DMT vehicle to control the vehicle in conventional road by the driver, has large effect on the motion of the steering system. The simplified dynamics of the power steering system can be described by following equations.
\[ I_{SW} \ddot{\theta} = -k_i(\theta - n\delta) - C_{SW} \dot{\theta}, \]
\[ (14) \]
\[ I_{ZP} \ddot{\delta} = nk_i(\theta - n\delta) - C_{ZP} \dot{\delta} + A_p \dot{r}_p p_c + T_s, \]
\[ (15) \]
where \( T_s \) is external torque acting at king pin.

\[ p_c = -k_v r_c n \dot{\theta} + k_v g m(\theta - n\delta) - \frac{k_v}{r_p} p_c, \]
\[ (16) \]
The relation between the power steering system and the DMT vehicle is shown in Fig.4.

**SIMULATION RESULTS**

The simulation was done based on the baseline vehicle configuration shown in Table.1 and guideway model shown in Fig.5. Here the effects of design parameters on the stability limit of the vehicle was evaluated.

A steering gain (\( G_F \)) determined by steering lever length of the mechanical guidance system is one of the important steering parameters having a great influence upon the vehicle stability. It is a ratio at which front wheels are steered depending on a unit displacement of the guide wheel. The smaller the steering gain is, the greater lateral displacement relative to the guideway is necessary for the vehicle to pass through at the curved guideway, consequently, the wider guideway is required, however, the larger steering gain reduces the vehicle stability limit.

The DMT vehicle has the simple guidance system where the steering gain is determined by the position of the guide wheel. The running stability of the vehicle changes largely because the steering gain changes by the sensing point. Figure 6 (a) shows the effect of sensing point on the stability limit. It is shown that the longer guide arm can improve the vehicle stability, however, consideration for curved guideway is also necessary to determine the optimum. Figure 6 (a) also shows the influence of clearance between guide wheel and guideway. The stability limit of the vehicle can be improved by the small clearance or pre-load of guide wheel. However, the clearance should be limited from respect of guide rail precision.

The DMT vehicle has steering wheel because the manual operation is required at conventional load. When running on the exclusive guideway, the steering wheel is forced to rotate by the mechanical guidance system. Figure 6 (b) shows the effect of steering wheel on the stability limit. It is shown that the steering wheel deteriorate the stability of

**Table 1 Baseline vehicle configuration**

<table>
<thead>
<tr>
<th>( m )</th>
<th>6450.0 [kg]</th>
<th>( C_{OF} \cdot C_{OR} )</th>
<th>9.8 [kNms / rad]</th>
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<tr>
<td>( L_f )</td>
<td>2.23 [m]</td>
<td>( h_s )</td>
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</table>


**FIELD TEST**

**The prototype DMT vehicle and test track**

To examine the running stability of a real DMT vehicle, the running characteristic of the prototype DMT test vehicle was measured on the DMT field test course in Public Works Research Institute, Ministry of Construction ([Fig. 7](#)). The test course (Total; length 760m) is composed of the curve section (Section 1 and Section 3: radius 65 m), the straight section (Section 3: grade 6%), the switching section (Section 4 and 5). The design maximum speed in the guideway is about 40 km/h.

**Unstable motion in the field test**

**Figure 8** shows the data of speed, yaw rate, and steering wheel angle of the DMT experiment vehicle on the guideway running. In Fig. 8, the motion of the vehicle is unstable in the latter half in curve section (Section 1) and switching section (Section 4). On the other hand, in the curve section with the same radius (section 3), unstable motion is not appeared under similar running condition. In the straight section (section 2), the unstable motion is not appeared when the running speed is higher than the curve section.

**Single-side guidance simulation**

The steering stabilizer by the self aligning moment is equipped in the prototype DMT vehicle. At the curved guideway, vehicle is guided by outside guide rail with the steering stabilizer as shown in [Fig. 9](#). It is expected that the unstable motion is caused by single-side guidance. To confirm the unstable motion in the field test by the simulation, a running simulation with single side guide rail is carried out. Here, the single side guidance is realized by bias steering moment as shown in [Fig. 9](#). The simulation results are shown in [Fig. 10](#). In Fig. 10, we can see that the unstable motion is caused by single side guidance depending on the disturbance from the guideway. It is also shown that the damping factor in steering shaft is effective for avoiding the unstable motion caused by single-side guidance.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
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<td>[m]</td>
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<tr>
<td>$D$</td>
<td>0.5</td>
<td>[m]</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>0.005</td>
<td>[m]</td>
</tr>
<tr>
<td>$n$</td>
<td>25.0</td>
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</tr>
<tr>
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<td>[kgm^2]</td>
</tr>
<tr>
<td>$I_{ZF}$</td>
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<td>[kgm^2]</td>
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<tr>
<td>$I_{SW}$</td>
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<td>[kgm^2]</td>
</tr>
<tr>
<td>$I_q$</td>
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<td>[kgm^2]</td>
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<td>$I_{qF}$</td>
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<td>[kgm]</td>
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<td>[kg]</td>
</tr>
<tr>
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<td>[Ns / m]</td>
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<td>[m]</td>
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<td>1.12</td>
<td>[m]</td>
</tr>
<tr>
<td>$\rho$</td>
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<td>[m]</td>
</tr>
</tbody>
</table>

the DMT vehicle. It is necessary to control the motion of the steering wheel to achieve a high-speed running of the vehicle in the guideway from this and not effect the steering system as much as possible.
Field test results without power steering

The power steering device is installed in the prototype DMT vehicle as shown in Fig.4. It is expected that the power steering device has large influence on the motion of mechanical guidance system. We carried out the field test without power steering device in the guideway. Figure 11 shows the experiment results without the power steering device. We can see that the unstable motion of the vehicle in the guideway disappeared. We can conclude that improvement of the vehicle stability in the single guidance situation can be realized by cutting off the power steering device in the guideway operation.

CONCLUSIONS

The conclusions of this study are summarized as follows.

1) Avoiding the coupling effect between steering wheel and mechanical guidance system is the most effective for high speed vehicle operation.

2) The stability limit of the vehicle improves largely by moving the position of the guide wheel forward.

3) The simulation study and the filed test shows that the single side guidance situation causes the unstable motion of the vehicle even in low speed operation.

4) Cutting of the power steering device in guideway operation is effective for avoiding the unstable motion caused by single side guidance.
REFERENCES


AUTHOR BIOGRAPHIES

Hitoshi TSUNASHIMA, Dr. Eng.
Assistant Professor,
Department of Mechanical Engineering, College of Industrial Technology Nihon University
1-2-1 Iumi-cho,Narashino-shi,Chiba 275,Japan
e-mail : tsuna@me.cit.nihon-u.ac.jp

Born in 1959
1993-ME, University of Osaka Prefecture, Joined Kobe Steel Ltd.
1993-Visiting Researcher, The University of Tokyo
1995-Dr. Eng., The University of Tokyo
1996-Assistant Professor of Nihon University
Member of The Japan Society of Mechanical Engineers, International Association for Vehicle System Dynamics, The Japan Society of Automotive Engineers

Tetsuya KANEKO
Department of Mechanical Engineering, College of Industrial Technology Nihon University
e-mail : c71080@ccu.cit.nihon-u.ac.jp

Born in 1973
1997-BE, Nihon University
Graduate school student, Nihon University
Member of The Japan Society of Automotive Engineers
Figure 1  New freight transport system
Figure 2 Dual mode truck
Figure 3  Vehicle model with mechanical guidance system
Figure 4  Vehicle model with power steering system
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Figure 6  Simulated unstable motion in single-side guidance

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Figure 6  Simulated unstable motion in single-side guidance
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