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PERFORMANCE ANALYSIS OF LATERAL GUIDANCE SYSTEM FOR DUAL MODE TRUCK

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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 descussed. 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

φ : yaw angle
β : side slip angle
ϕ : roll angle of vehicle body
ϕ_F : roll angle of front axle
ϕ_R : roll angle of rear axle
δ : front wheel steering angle
θ : 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_ϕ : 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
δ_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_{FRI} : 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

ε : 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_v : 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:

$$mV(\dot{\beta} + \dot{\varphi}) - m_s h_s \ddot{\phi} = F_F + F_R + F_G , \tag{1}$$

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

$$F_F = C_C W_F (\delta - \beta - L_f \dot{\varphi} / V) + C_C K_Z t_d \delta_0 \phi_F , \tag{2}$$

$$F_R = C_C W_R (-\beta + L_r \dot{\varphi} / V) , \tag{3}$$

$$F_G = K_S (y_{GL} + y_{GR}) + C_S (\dot{y}_{GL} + \dot{y}_{GR}) . \tag{4}$$

2) Rotational motion of front wheel:

$$I_{ZF} (\ddot{\delta} + \ddot{\varphi}) = F_G / G_F - (t_p + t_c) F_F - M_B - M_K - M_{FRI} + nK_{SW} (\theta - n\delta) , \tag{5}$$

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

$$M_K = M_0 sign(\delta) . \tag{6}$$

3) Yawing motion of vehicle body:

$$I_Z \ddot{\varphi} = L_f (F_G + F_F) - L_r F_R + M_B + M_K + M_{FRI} + C_{SW} \dot{\theta} . \tag{7}$$

4) Rolling motion of vehicle body:

$$I_\phi \ddot{\phi} = -K_{\phi F}(\phi - \phi_F) - K_{\phi R}(\phi - \phi_R) - C_{\phi F}(\dot{\phi} - \dot{\phi}_F) - C_{\phi R}(\dot{\phi} - \dot{\phi}_R) + m_S g h_S \phi + m_S h_S V(\dot{\beta} + \dot{\varphi}) \,. \tag{8}$$

5) Rolling motion of unsprung mass:

$$I_{\phi F}\ddot{\phi}_F = -K_{\phi F}(\phi_F - \phi) - C_{\phi F}(\dot{\phi}_F - \dot{\phi}) - K_Z(t_d^2/2)\phi_F + h_F F_R \,, \tag{9}$$

$$I_{\phi R}\ddot{\phi}_R = -K_{\phi R}(\phi_R - \phi) - C_{\phi R}(\dot{\phi}_R - \dot{\phi}) - K_Z\{(t_{d0}^2/2) + (t_{dI}^2/2)\}\phi_R + h_R F_R \,. \tag{10}$$

6) Motion of guide wheel:

$$m_G \ddot{y}_{GL} = -C_S \dot{y}_{GL} - K_S y_{GL} - [K_S K_{GW}\varepsilon/(K_S + K_{GW})]_* + GF_{GL} - m_G(\ddot{\varphi}_F - \ddot{\varphi})/G_F - m_G V(\dot{\beta} + \dot{\varphi}) - m_G L_f \ddot{\varphi} \,, \tag{11}$$

$$m_G \ddot{y}_{GR} = -C_S \dot{y}_{GR} - K_S y_{GR} - [K_S K_{GW}\varepsilon/(K_S + K_{GW})]_* + GF_{GR} - m_G(\ddot{\varphi}_F - \ddot{\varphi})/G_F - m_G V(\dot{\beta} + \dot{\varphi}) - m_G L_f \ddot{\varphi} \,. \tag{12}$$

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:

$$I_{SW}(\ddot{\theta} + \ddot{\varphi}) = -K_{SW}(\theta - n\delta) - C_{SW}\dot{\theta} \,. \tag{13}$$

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.

$$I_{SW}\ddot{\theta} = -k_t(\theta - n\delta) - C_{SW}\dot{\theta}\ , \tag{14}$$

$$I_{ZF}\ddot{\delta} = nk_t(\theta - n\delta) - C_B\dot{\delta} + A_p r_C p_e + T_S\ , \tag{15}$$

where T_S is external torque acting at king pin.

$$\dot{p}_e = -k_v \frac{r_C}{n} A_p n\dot{\delta} + k_v gm(\theta - n\delta) - \frac{k_v}{r_p} p_e\ , \tag{16}$$

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 configration 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

m	6450.0	[kg]	$C_{\phi F}, C_{\phi R}$	9.8	[kNms / rad]
L_f	2.23	[m]	h_S	0.7	[m]

L_r	1.12	[m]	h_F, h_R	0.6	[m]
D	0.5	[m]	W_F	21.1	[kN]
ε	0.005	[m]	W_R	42.1	[kN]
n	25.0		C_C	4.22	[1 / rad]
I_Z	7460.0	[kgm^2]	K_{GW}	294.0	[kN / m]
I_{ZF}	9.8	[kgm^2]	m_G	0.5	[kg]
I_{SW}	0.061	[kgm^2]	C_S	9.8	[Ns / m]
I_ϕ	2940.0	[kgm^2]	K_S	294.0	[kN / m]
$I_{\phi F}$	58.8	[kgms^2]	t_d, t_{d0}	1.63	[m]
$I_{\phi R}$	294.0	[kgm^2]	t_{dI}	1.12	[m]
$K_{\phi F}, K_{\phi R}$	98.0	[kNm / rad]	t_p	0.042	[m]
K_Z	196.0	[kN / m]			

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

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 Pubic 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.

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.

Introduction Content

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[2] Abe, M. et al (1987) "A Study on the Performance of Guideway Bus Steering Control System", Proc. of SAE 4th Int. Conf. on Automotive Engineering, pp. 231.1 - 231.8.

Introduction Content

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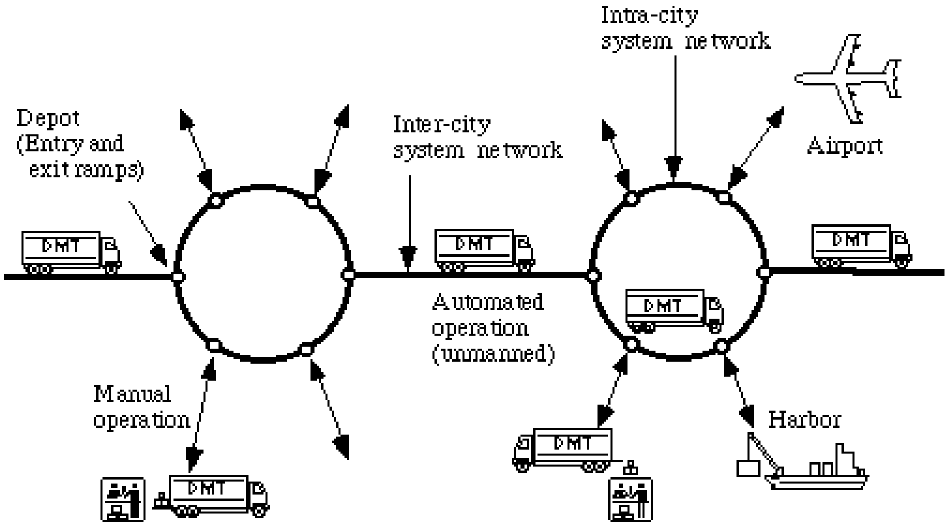

Figure 1 New freight transport system

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Figure 2 Dual mode truck

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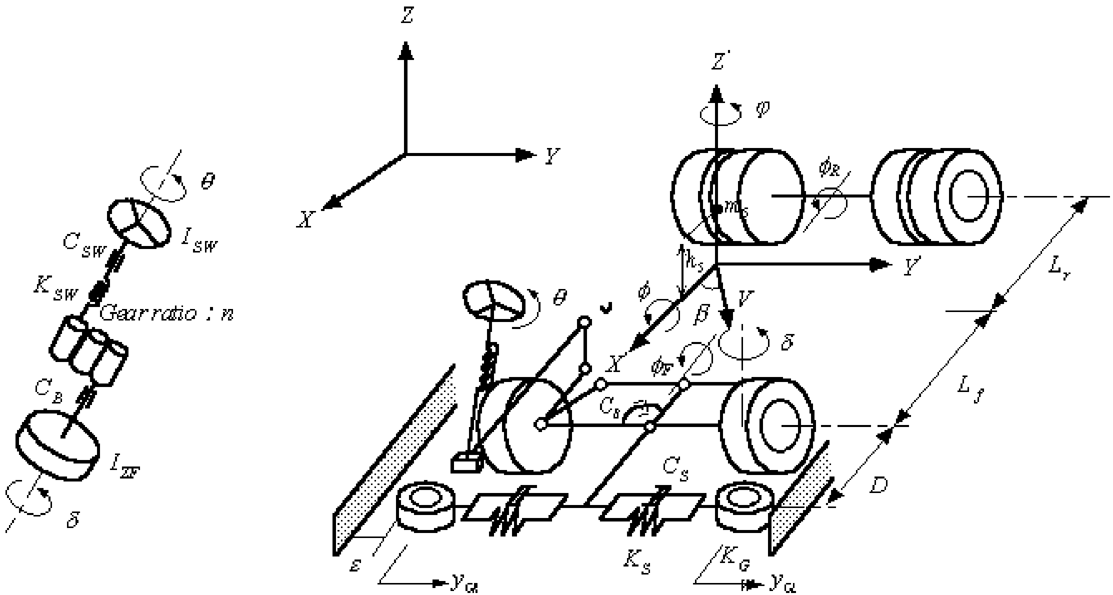

Figure 3 Vehicle model with mechanical guidance system

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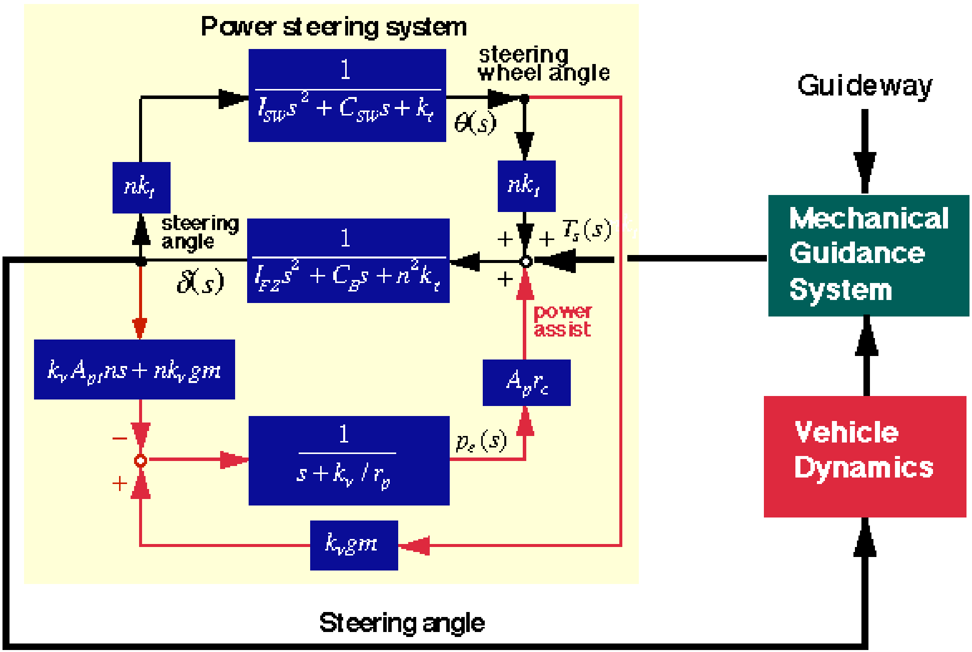

Figure 4 Vehicle model with power steering system

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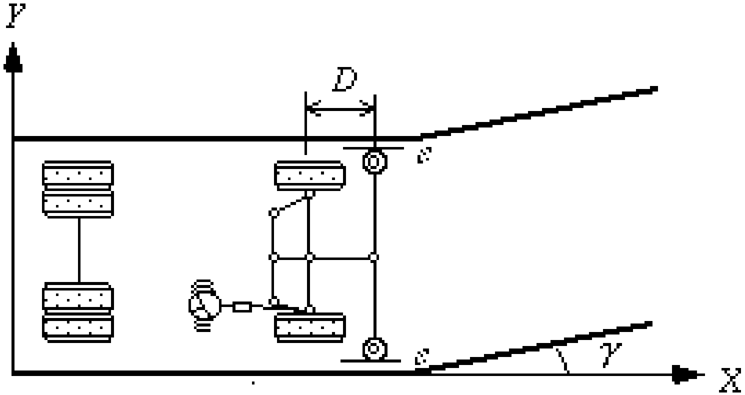

Figure 5 Guideway model

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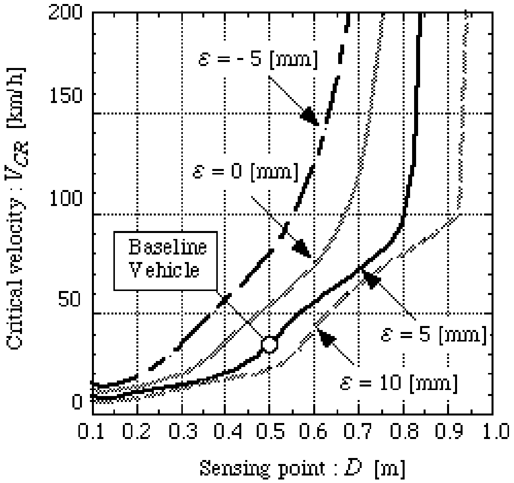

(a) Effect of sensing point

Figure 6 Simulated unstable motion in single-side guidance

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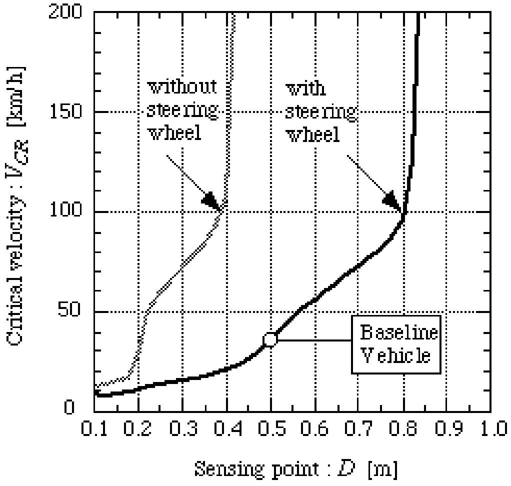

(b) Effect of steering wheel

Figure 6 Simulated unstable motion in single-side guidance

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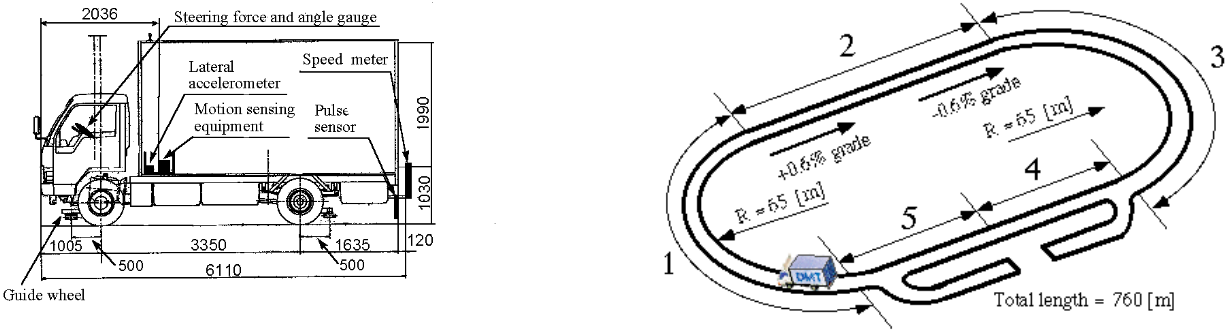

Figure 7 The prototype DMT test vehicle and test track

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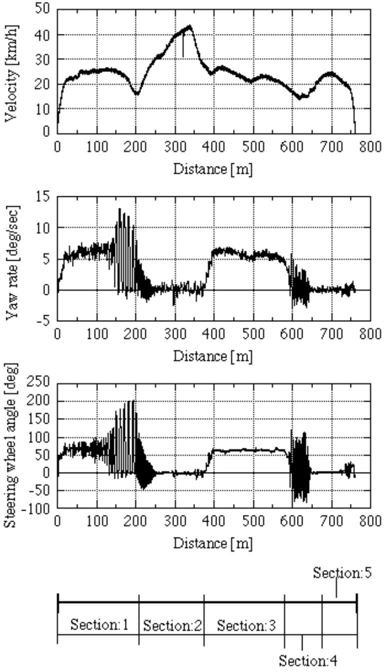

Figure 8 Results of field test (with power steering)

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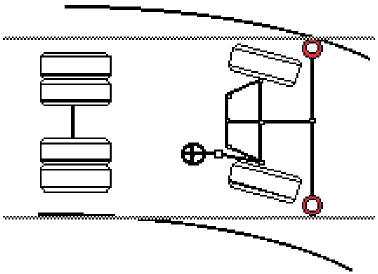
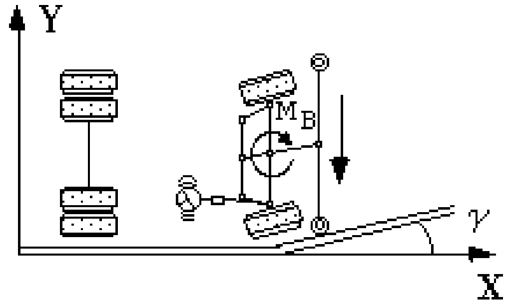
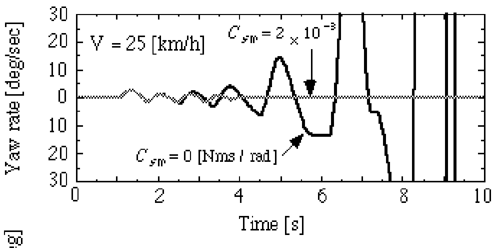
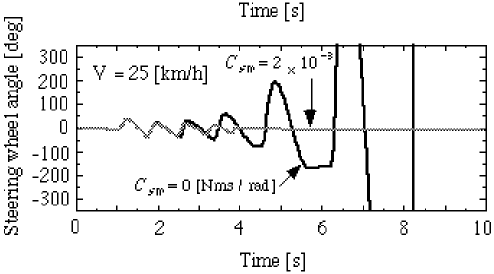






(a) Effect of guideway disturbance



(b)Effect of damping factor around

Figure 9 Single-side guidance at curved guideway and a model for single-side guidance

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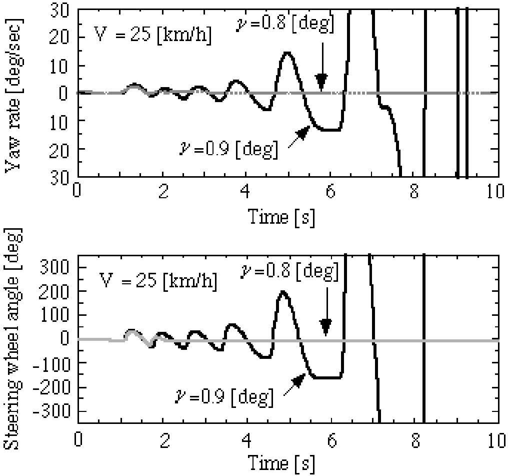

Figure 10 Simulated unstable motion in single-side guidance

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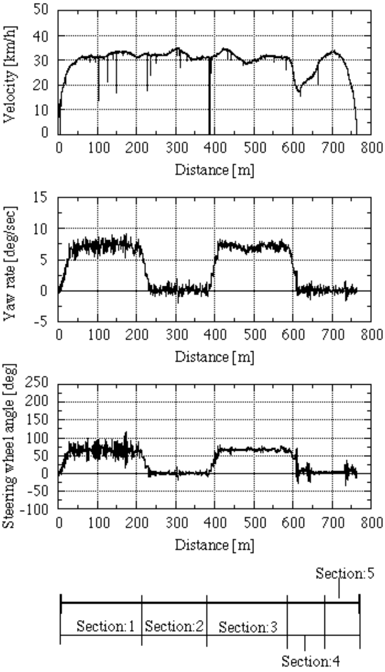

Figure 11 Result of field test (without power steering)

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