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A Method to Improve Stability of Adaptive Steering Driver-Vehicle Systems

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Abstract: We have developed an adaptive steering controller achieving good tracking performance. However, we have designed an ideal vehicle model in disregard of the variations of the driver properties. For the variations of the driver properties, if an adequate ideal vehicle model can be designed, the better handling stability of the adaptive driver-combined-vehicle systems can be realized. In this paper, we propose a scheme to design an ideal vehicle model adequate for the variations of the driver properties. Finally, it is shown by carrying out numerical simulations that the designed ideal vehicle model is very effective.

Keywords: Steering Control, Adaptive Control, Model Following Control

I. INTRODUCTION

To improve handling stability of combined vehicles, the three states, the combination angle, the lateral velocity and yaw rate of tractor, must be controlled with three independent control inputs. Recently, the researches of steer-by-wire, are conducted actively [1]. Using the technology of steer-by-wire, the three steering angles such as the steering angle of the trailer, the front and rear steering angle of the tractor can be used as the independent control inputs possibly. Based on the notion, control schemes have been proposed [2,3] in which three independent steering control inputs are used. However, these proposed controllers require all accurate knowledge of combined vehicle parameter.

To straggle with the problem stated above, the authors have proposed robust steering control schemes [4,5]. In the schemes, an ideal vehicle model is designed, and then, the steering controller is developed so that the actual vehicle tracks the ideal vehicle model even if the large parameter variation occurs in the vehicle dynamics. In the proposed schemes, however, an ideal vehicle model is designed in disregard of the variations of the driver properties. Therefore, the authors have proposed a design scheme for ideal vehicle models [6,7] adequate for the variations of the deriver properties. In the scheme, several ideal vehicle models are designed against the variation of the driver properties and the ideal vehicle model has to be changed according to the variation. For this reason, the scheme proposed in [6,7] require accurate information of the variation of the driver properties. In addition, there exists the problem that the construction of controller becomes too complex.

In this paper, we propose a new design scheme for an ideal vehicle model. A design method based on a cost function is proposed and only one ideal vehicle model is designed. In the designed ideal vehicle model, the rough information of the variation of the driver properties is only required. Carrying out numerical simulations, for the variation of the driver properties, it is shown that the adaptive steering controller using the proposed ideal vehicle model has good effectiveness.

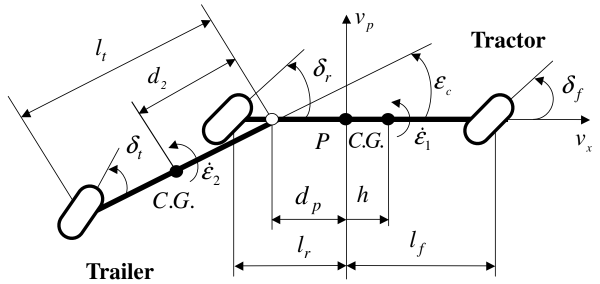


Fig.1 Tractor-semitrailer model

Ⅱ. INTRODUCTION

In this paper, a simplified bicycle model of tractor-semitrailers shown in Figure 1 is used to design an adaptive steering controller. In Figure 1, the point C.G. is the center of gravity, and the point P is the reference point. Definitions for parameters of combined vehicles are shown in Table 1.

The following assumptions are made to develop a controller for the combined vehicles shown in Fig.1.

A1 Lateral velocity of tractor $v_p(t)$, yaw rate of tractor $\gamma_1(t)$, combination yaw rate $\dot{\varepsilon}_c(t)$ and combination angle $\varepsilon_c(t)$ can be measured.

A2 The length d_p, l_f, l_r and the length of trailer l_t are known. The other parameters include uncertainties.

A3 Longitudinal velocity v_x is a bounded constant and available.

The dynamic equation of combined vehicle can be described as follows.

Table 1 Notation of tractor-semitrailer model

v_x	longitudinal velocity of tractor
v_p, v_c	lateral velocity of tractor at P and C.G.
$\varepsilon_1, \varepsilon_2$	yaw angles of tractor and trailer
ε_c	combination angle
γ_1	yaw rate of tractor
$\delta_f, \delta_r, \delta_t$	steering angles
m_1, m_2	mass of the tractor and trailer
J_{z1}, J_{z2}	tractor's moment of inertia and trailer's moment of inertia
l_f, l_r	distances from P to front and rear wheel axle of tractor
l_t	length of trailer
d_p, d_2	distances from the connector to P and C.G.
h	distance from P to C.G.
c_f, c_r, c_t	cornering stiffness

$$\dot{\boldsymbol{z}}(t) = A_z \boldsymbol{z}(t) + B_z(\boldsymbol{u}(t) - \boldsymbol{b}\varepsilon_c(t)) \tag{1}$$

$$\left.\begin{aligned}
&\boldsymbol{z}(t) = H_p^T \begin{bmatrix} v_p(t) & \gamma_1(t) & \dot{\varepsilon}_c(t) \end{bmatrix}^T, \ \boldsymbol{u}(t)^T = \begin{bmatrix} \delta_f(t) & \delta_r(t) & \delta_t(t) \end{bmatrix} \\
&A_z = -v_x D_z - v_x^{-1} B_z, \ B_z = QK, \ \boldsymbol{b}^T = \begin{bmatrix} 0 & 0 & 1 \end{bmatrix} \\
&Q = H_p^T (T^T M_c T)^{-1} H_p, \ K = \mathrm{diag}\begin{bmatrix} c_f & c_r & c_t \end{bmatrix}
\end{aligned}\right\} \tag{2}$$

$$\left.\begin{aligned}
&M_c = \begin{bmatrix} m_1 + m_2 & -m_2 d_3 & m_2 d_2 \\ -m_2 d_3 & J_{z3} + m_2 d_3^2 & -J_{z2} - m_2 d_2 d_3 \\ m_2 d_2 & -J_{z2} - m_2 d_2 d_3 & J_{z2} + m_2 d_2^2 \end{bmatrix} \\
&D_z = H_p^T \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} (H_p^T)^{-1}, \ T = \begin{bmatrix} 1 & h & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \\
&H_p^T = \begin{bmatrix} 1 & 1 & 1 \\ l_f & -l_r & -(d_p + l_t) \\ 0 & 0 & l_t \end{bmatrix}
\end{aligned}\right\} \tag{3}$$

Where $d_1 = d_p + h, d_3 = d_1 + d_2$ and $J_{z3} = J_{z1} + J_{z2}$. M_c is a positive definite matrix.

The nominal values of combined vehicle are shown in Table 2.

Ⅲ. IDEAL VEHICLE MODEL

1. Driver Model

In this paper, a linear preview driver model [8] is employed to describe the properties of drivers. In Fig.2, the solid lines represent the state of the combined vehicles, and the dashed lines represent the ideal state of the combined vehicles tracking the target lane. $y_r(t)$ is the lateral distance between P and the target lane, $\varepsilon_r(t) = \varepsilon_1(t) - \varepsilon_d(t)$ is the relative yaw angle between the vehicle and the target lane. $\varepsilon_d(t)$ is the yaw angle of the ideal state, $\rho(t)$ is the curvature of target lane. The predicted deviation $\tilde{y}_r(t)$ is defined by

$$\tilde{y}(t) = y_r(t) + T_p \dot{y}_r(t) \tag{4}$$

where T_p is the preview time constant, v_x is the longitudinal velocity of the combined vehicles.

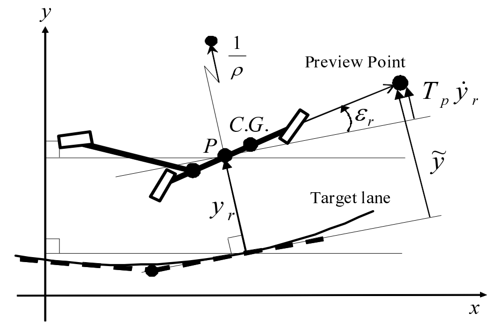


Fig.2 Driver-Vehicle-System

Therefore, the equation of the driver steering angle $\delta_c(t)$ is given by

$$T_s \ddot{\delta}_c(t) + \dot{\delta}_c(t) = -g_p \dot{\tilde{y}}(t) - g_i \tilde{y}(t) \tag{5}$$

where T_s is a steering time constant and denotes the reaction time delay of drives and mechanical systems, g_p, g_i is the steering gain. In this paper, we add an integrator to the driver model of [8] so that $y_r(t)$ can converge to zero when the curvature of target lane is not zero.

2. The Form of Ideal Vehicle Model

In the following explanation, s is Laplace variable, the symbols $\mathcal{L}, \mathcal{L}^{-1}$ denote Laplace transform and inverse Laplace transform.

Based on the proposed ideal vehicle model proposed in [5], the form of ideal vehicle modes is proposed as

$$\left.\begin{aligned}
&v_{pd}(t) = \mathcal{L}^{-1}\left[\frac{g_v \omega_n^2 s}{A_M(s)} \mathcal{L}[v_x \delta_c(t)]\right] \\
&\gamma_{1d}(t) = \mathcal{L}^{-1}\left[\frac{g_\gamma \omega_n^2 (\eta s + 1)}{A_M(s)} \mathcal{L}[v_x \delta_c(t)]\right] \\
&A_M(s) = s^2 + 2\omega_n \zeta s + \omega_n^2 \\
&g_v = \bar{g}_v \frac{\bar{g}_p s + \bar{g}_i}{\hat{g}_p s + \hat{g}_i} \frac{\bar{T}_p s + 1}{\hat{T}_p s + 1}, g_\gamma = \bar{g}_\gamma(v_x) \frac{\bar{g}_p s + \bar{g}_i}{\hat{g}_p s + \hat{g}_i} \frac{\bar{T}_p s + 1}{\hat{T}_p s + 1}
\end{aligned}\right\} \tag{6}$$

$$\mathcal{L}[\varepsilon_{cd}(t)] = \frac{\omega_n^2}{A_M(s)} \mathcal{L}[\delta_{ci}(t)], \varepsilon_{ci}(t) = \frac{2 d_p v_x \gamma_{1d}(t)}{v_x^2 - (d_p \gamma_{1d}(t))^2} \tag{7}$$

where $\omega_n, \zeta, \eta, \bar{g}_v, \bar{g}_\gamma(v_x), \bar{g}_p, \bar{g}_i, \bar{T}_p$ are positive design parameters to be determined later. $v_{pd}(t)$ is the ideal lateral velocity of the tractor, $\gamma_{1d}(t)$ is the ideal combination yaw angle, and $\varepsilon_{cd}(t)$ is the ideal combination yaw angle. In (6), $\hat{g}_p, \hat{g}_i, \hat{T}_p$ are the estimated values of the steering gains g_p, g_i and the preview time constant T_p . In the case of a constant driver steering angle, the desired lateral velocity $v_{pd}(t)$ of the tractor converges to zero and the desired yaw rate $\gamma_{1d}(t)$ of the tractor tends to a constant value $\lim_{t \to \infty} \gamma_{1d}(t) = \bar{\gamma}_{1d}$. Then, the tractor becomes tangent to the circular lane with the radius $v_x / \bar{\gamma}_{1d}$ and travels.

In the actual combined vehicles using the adaptive steering controller, the property of the actual vehicles becomes as same as that of the ideal vehicle model. In this case, the following property can be obtained.

Table 2 Nominal values of parameters

m_1	1060	kg	J_{z1}	1507	kgm^2
l_f	1.13	m	l_r	1.34	m
d_p	2.27	m	h	0	m
d_2	2.26	m			
c_f	29700	N/rad	c_r	41460	N/rad
m_2	436	kg	J_{z2}	805	kgm^2
l_t	2.48	m	c_t	100000	N/rad

$$\left.\begin{aligned} Y_r(s) &= \frac{-s(T_s s+1)A_M(s)v_x^2}{s^3(T_s s+1)A_M(s)+C(s)v_x\omega_n^2 B_M(s)}\rho(s) \\ &\quad + \frac{s^2(T_s s+1)A_M(s)+D(s)T_p v_x \omega_n^2 B_M(s)}{s^3(T_s s+1)A_M(s)+C(s)v_x\omega_n^2 B_M(s)}Y_r(0) \\ B_M(s) &= \bar{g}_v s^2 + v_x \bar{g}_\gamma \eta s + v_x \bar{g}_\gamma \\ C(s) &= (\bar{g}_p s + \bar{g}_i)(\bar{T}_p s+1)\frac{g_p s + g_i}{\hat{g}_p s + \hat{g}_i}\frac{T_p s+1}{\hat{T}_p s+1} \\ D(s) &= (\bar{g}_p s + \bar{g}_i)\frac{g_p s + g_i}{\hat{g}_p s + \hat{g}_i}\frac{\bar{T}_p s+1}{\hat{T}_p s+1} \end{aligned}\right\} \tag{8}$$

From (8), it can be seen easily that if the estimated values $\hat{g}_p, \hat{g}_i, \hat{T}_p$ are almost equal to the parameters g_p, g_i, T_p of actual vehicle, the variation of the property shown in (8) becomes small.

3. Determination of Design Parameters

In the passive combined vehicles, the nominal values of vehicle parameters (see Table 2) are used. In the following explanation, the determination method of design parameters $\omega_n, \zeta, \eta, \bar{g}_v, \bar{g}_\gamma(v_x), \bar{g}_p, \bar{g}_i, \bar{T}_p$ is shown.

Step 1 In order that oscillation cannot occur in the desired trajectories of a ideal vehicle model, ζ is set as $\zeta = 1$.

Step 2 Using the average values of the driver parameters, the driver parameters g_p, g_i, T_p are set as $g_p = 0.008, g_i = 0.0012, T_p = 2.2$. In addition, it is assumed that $g_p = \bar{g}_p = \hat{g}_p, g_i = \bar{g}_i = \hat{g}_i, T_p = \bar{T}_p = \hat{T}_p$.

Step 3 For a constant longitudinal velocity v_x, we can obtain the yaw rate gain as $\bar{g}_\gamma(v_x) = \frac{3.06\times10^2}{7.59\times10^2+2.24\times v_x(t)^2}$. In order to make it easy for the ideal vehicle model to drive a corner, the design parameter $\bar{g}_\gamma(v_x)$ is determined as

$$\bar{g}_\gamma(v_x) = \frac{1.8\times3.06\times10^2}{7.59\times10^2 + 2.24\times v_x(t)^2} \tag{9}$$

Step 4 To design a vehicle model with good handling stability, an evaluation function is introduced by using the trail and error approach.

$$\begin{aligned} J &= \int_0^t (2y_r(t)^2 + 2y_{rt}(t)^2 + 4\ddot{y}_r(t)^2 + 4\ddot{\varepsilon}_c(t)^2)dt \\ &\quad + 3\max\ddot{y}_r(t) + 3\max\ddot{\varepsilon}_c(t) \end{aligned} \tag{10}$$

Where T is the simulation time. Based on the evaluation function, the remaining design parameters $\bar{g}_v, \eta, \omega_n$ are determined so that the following conditions can be satisfied.

(a) The imaginary part of the eigenvalues of $B_M(s)$ is minimized preferably.

(b) The evaluation function is minimized preferably in lane change and cornering maneuver.

Step 5 If the designed ideal vehicle model does not have good handling stability, we return to Step 3~4.

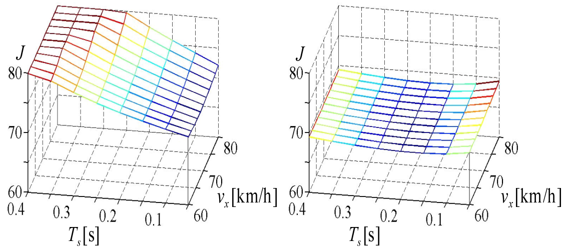
As result, $\bar{g}_v, \eta, \omega_n$ are set as $\bar{g}_v = 0.25, \eta = 0.39, \omega_n = 5.4$. In the following part, using numerical simulations, the property of designed the ideal vehicle model is shown. In the lane change maneuver, the curvature of target lane is $\rho(t) = 0$[m]. The initial value of $y_r(t)$ is set as $y_r(0) = -3.0$[m]. In the cornering maneuver, the curvature of target lane is $\rho(t) = 1/500$[m]. The initial value of $y_r(t)$ is set as $y_r(0) = -1.0$[m]. Until now, from many research results, the value of T_s is obtained as $0.05 \sim 0.4$[s]. For this reason, in the following numerical simulations, we check controlled performance in $0.05 \le T_s \le 0.4$[s].

At first, the effectiveness of the proposed ideal vehicle model is shown in the lane change maneuver. In the case of varying v_x and T_s, the variation of J is shown in Figure 3. Fig.3 (a) shows the values of J for the passive combined vehicle, Fig.3 (b) shows the values of J for the adaptive steering driver vehicle system using the proposed ideal vehicle model. Hereafter, it is called the adaptive vehicle simply. In Fig.3, in case where J exceeds 80, the values of J are shown as 80. As shown Fig.3 (a), there exists a region in which J exceed 80. In the region, the handling performance of the passive vehicle becomes unstable. However, as shown in Fig.3 (b), for the adaptive vehicle, the values of J are less than 70 and good handling performance can be achieved.

Next, the effectiveness of the proposed ideal vehicle model is shown in the cornering maneuver. In the case of varying v_x and T_s, the variation of J is shown in Figure 4. Fig.4 (a) shows the values of J for the passive combined vehicle, Fig.4 (b) shows the values of J for the adaptive vehicle. In Fig.4, in case where J exceeds 35, the values of J are shown as 35. As shown Fig.4 (a), there exists a region in which J exceeds 35. In the region, the handling performance of the passive vehicle becomes unstable. However, as shown in Fig.4 (b), for the adaptive vehicle, the values of J are less than 21 and good handling performance can be achieved.

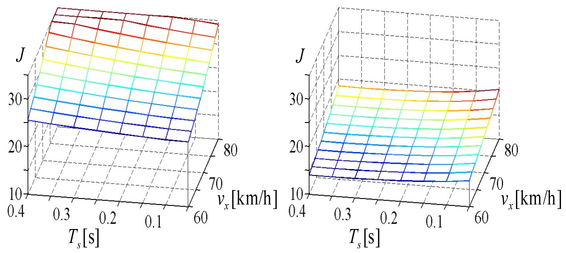
Moreover, we show the effectiveness of the proposed ideal vehicle in the presence of the estimated errors in $\hat{g}_p, \hat{g}_i, \hat{T}_p$. In case where the values of $g_p / \hat{g}_p$ and $g_i / \hat{g}_i$ are not equal to one, we can obtain the similar graphs to Fig.3 (b) and Fig.4 (b). For each values of $g_p / \hat{g}_p$ and $g_i / \hat{g}_i$, the maximum value of Fig.3 (b) is plotted in Fig.5 (a) and the maximum value of Fig.4 (b) is plotted in Fig.5 (b). Similarly, for each values of $T_p / \hat{T}_p$, the maximum values of Fig.3 (b), Fig.4 (b) are plotted in Fig.6.

As shown in Fig.5, in the region of $0.6 \le g_p / \hat{g}_p \le 1.4$ and $0.8 \le g_i / \hat{g}_i \le 1.6$, the values of J became less than 75 in the lane change maneuver and



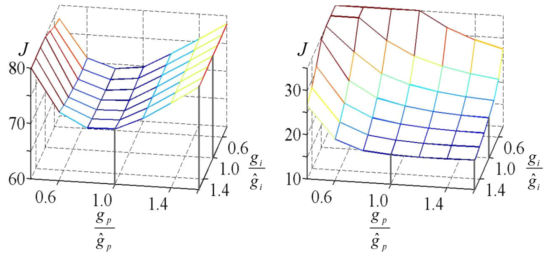
(a) passive (b) ideal vehicle model

Fig.3 Variation of J for various T_s and v_x (lane change)



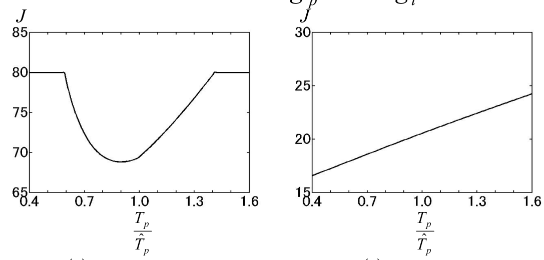
(a) passive (b) ideal vehicle model

Fig.4 Variation of J for various T_s and v_x (cornering)



(a) Lane Change (b) Cornering

Fig.5 Variation maximum value of J for various g_p and g_i



(a) Lane Change (b) Cornering

Fig.6 Variation maximum value of J for various T_p

the values of J became less than 30 in the cornering maneuver. As shown in Fig.6, in the region of $0.7 \le T_p / \hat{T}_p \le 1.3$, the values of J became less than 75 in the lane change maneuver and the values of J became less than 25 in the cornering maneuver. From the facts, it can be concluded that the proposed ideal vehicle model is very effective when the estimated values of the driver parameters are within the regions $0.6 \le g_p / \hat{g}_p \le 1.4, 0.8 \le g_i / \hat{g}_i \le 1.6$ and $0.7 \le T_p / \hat{T}_p \le 1.3$. Namely, it can be seen that accurate values of the driver parameters are not required.

IV. CONCLUSION

In this paper, we propose a design method to design not several ideal vehicle models but only one ideal vehicle model against variations of the driver properties. Carrying out numerical simulations, it is shown that the handling performance in the adaptive driver vehicle system is far better than that of passive combined vehicle. Moreover, it is also shown that in the design scheme of an ideal vehicle model, rough information of the driver properties are only required to maintain good handling performance.

REFERENCES

(1) P. Pisu, A. Serrani, S. You, and L. Jalics, Adaptive Threshold Based Diagnostics for Steer-by-wire Systems, Trans. ASME. Journal of Dynamic Systems, Measurement, and Control, vol.128, (2006), pp.428-435.

(2) B.A. Jujnovich, A.M.C. Odhams, R.L. Roebuck and D. Cebon, Active Rear Steering Control of a Tractor-Semitrailer, Proceedings of International Symposium on Advanced Vehicle Control, Aachen, 1996, pp.395-408.

(3) R. Hayama, S. Maeda, S. Kawahara, S.Nakano and H. Kumamoto, Bilateral Control of Human Machine Interface for Steer-By-Wire System, Proceedings of International Symposium on Advanced Vehicle Control, Kobe, 2008, pp.752-757.

(4) Qiang Wang, Masahiro Oya, and Toshihiro Kobayashi, Robust Integral Control to Improve Stability of Tractor-Semitrailer Combination Vehicles, Proc. 2005 International Symposium on Advanced Control of Industrial Processes, Seoul, Korea, (2005), pp. 121-126.

(5) Q.Wang, M.Oya, N.Takagi, Y.Taira and H.Ota, Adaptive Steering Controller to Improve Handling Stability for Driver-Combined-Vehicle System, Proc. of the 8th IEEE International Symposium on Computational Intelligence in Robotics and Automation, Daejoen, Koria, December 15-18, (2009), pp.409-414.

(6) Qiang Wang, Masahiro Oya, Natsuki Takagi, On Design of Ideal Vehicle Models in Adaptive Steering Driver-Vehicle Systems, Proc. of the Second International Conference of Modeling Identification and Control, Okayama, Japan, July 17-19, pp.203-208, 2010.

(7) Qiang Wang, Singo Tamaru, Hideki Wada, Masahiro Oya, One Design Method for Ideal Vehicle Models In Adaptive Driver-Combined Vehicle Systems, Proc. of the SICE Annual Conference 2010, Taipei, Taiwan, August 18-21, pp.156-161, 2010.

(8) J.Ishio, H.Ichikawa, Y.Kano, M.Abe, Vehicle-handling quality evaliation through model-based driver steering behaviour, Vehicle System Dynamics, vol.46, 2008, pp.549-560.