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Simulation of Man-Automobile Systems by the Driver's Steering Model with Predictability*

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To investigate the driving dynamics of automobiles it is necessary to study the control characteristics of drivers.

In this paper, a driver's steering model is proposed to represent the action of a driver who is driving an automobile along a curved course, assuming that he can predict the future position of the automobile based upon the present position, the yawing angle and the yawing velocity, and that the predicted future deviation is fed back to the steering force. With this model the dynamics of such man-automobile systems is investigated making use of simulation by a digital computer.

From these simulation experiments, the adequateness of the model is confirmed.

1. Introduction

To investigate the driving dynamics of automobiles it is necessary to study the control characteristics of drivers, because automobiles cannot run along roads without the driver's control. Many investigators have already proposed various driver's steering models and have explained qualitatively the driving stability employing their models^{(1)~(8)}, but their models are not accurate enough to investigate the tracking characteristics of automobiles. Hence a simple, useful model which characterizes the human driver satisfactorily is desired.

In this paper, a new driver's steering model is proposed, assuming that the driver can predict the future position of the automobile from the present position, the yawing angle and the yawing velocity, and that the future deviation is fed back to the steering force. This model is called the predictable correction model hereafter. With this model the dynamics of such man-automobile systems running at various speeds along curved courses or under cross-wind disturbance conditions is investigated making use of the simulation by a digital computer. From these simulation experiments, the adequateness and usefulness of this model are confirmed.

Kondo⁽¹⁾ has proposed the preview correction model, assuming that the driver can predict linearly the future position of the vehicle from the present position and the yawing angle as shown in Fig. 1 (a) and he handles the steering wheel in proportion to the future deviation ϵ . The future deviation ϵ is

predicted by comparing the observed course with the linear extrapolation of the vehicle position. As this ϵ exists even if the vehicle is right on the course, the motion of the vehicle along the curved course becomes possible by turning the steering wheel proportionally to the predicted error ϵ . In this case, however, the adequate proportional gain and the adequate eye-fixation L are in accordance with each curved course and with each vehicle speed, and it is difficult to satisfy the conditions for the stability and for the tracking accuracy simultaneously. While a large L is desirable from the stability point of view in the high speed driving simulation, the offset

(b) The predetermined correction model
Fig. 1 The driver's steering models

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from the course will become larger for the larger L .

Ohno⁽⁷⁾ has proposed the predetermined correction model, assuming that the driver can determine the steering program beforehand based on the known characteristics of the vehicle and the pattern of the previewed course, and that he steers not only by the program signal but also by the feedback signals consisting of the lateral position error and the yawing angle error as caused by the program steering. As shown in Fig. 1 (b) the output of the model driver proposed by Ohno is the steering angle proportional to the future deviation ε which is equal to ε_0 plus ε' , where ε_0 is caused by the curved course and is proportional to the program signal, while ε' is the error which cannot be compensated by the programmed steering and is proportional to the feedback signal. With this driver's model we must determine the accurate program of steering in accordance with each curved course, and in particular, for higher speed the steering program must be determined based on the kinetic characteristics of the vehicle, and an inaccurate program may result in a large deviation from the course.

Thus, the driver's models proposed in the past seem to have certain shortcomings when they are applied to the steering action of the skilled drivers.

2. The steering model based on the second order predictability

The new model proposed in this paper comprises the following three assumptions:

(1) The detection of the future deviation based on the second order prediction The driver can perceive not only the present position and the yawing velocity through the turning of the sight or the feeling of the lateral acceleration. As shown in Fig. 2 the driver is able to predict the future position (x^*, y^*) which the vehicle will reach after τ sec. and to detect the future deviation ε from the right course. This τ will be called the prediction time.

(2) The steerage by the steering torque

Fig. 2 The driver's steering model with predictability

The driver's output is not the steering displacement but the steering torque, because it is difficult to drive the vehicle with the reactionless handle wheel. Therefore, the influence of the inertia, viscosity, elasticity and reaction of the steering system must be taken into account in this simulation.

(3) Discrete time correction In a periodic way the driver detects the future deviation and corrects the steering torque. If the sampling period is T sec., the conventional human response time is nearly equal to $T/2$ sec. But for such a previewed system as automobile driving this discrete model seems to be more practical.

Figure 3 shows the block-diagram of the driver's steering model, where H , T and Σ express the proportional gain, the sampling period and the summing operation, respectively.

3. The equations of the dynamics of automobile

In the following discussions, only the horizontal motion of automobile is considered, the running resistance and thrust are neglected, and the equations of motion are linearized.

The following notations will be used (see Fig. 4, 5).

- g : Acceleration of gravity
- c.g.: Center of gravity of vehicle
- c.p.: Center of pressure
- (x, y) : Coordinate of c.g.
- (x^*, y^*) : Predicted coordinate of c.g. after τ sec.
- V : Speed of vehicle along c.g. path
- φ : Course angle
- θ : Yawing angle of vehicle
- $\omega = d\theta/dt$: Yawing velocity of vehicle
- α : Steering angle of front wheels
- $\beta = \varphi - \theta$: Side-slip angle of vehicle
- l : Wheelbase
- l_f, l_r : Distance between c.g. and front and rear wheel center, respectively
- M : Total mass of vehicle
- J : Moment of inertia of vehicle around normal axis through c.g.
- k_f, k_r : Cornering stiffness of front and rear tires, respectively
- f : Steering force of steering wheel

Fig. 3 The block-diagram of the driver's steering model with the second order predictability

A : Steering angle of steering wheel

r : Radius of steering wheel

n : Steering-gear ratio

$I = J_h/n^2 + J_s$: Equivalent moment of inertia of steering system with respect to front wheels

$C = C_h/n + C_s$: Equivalent damping coefficient of steering system with respect to front wheels

k_s : Equivalent spring constant of steering system with respect to front wheels

k_e : Equivalent self-aligning moment coefficient of front wheels

F_w : Lateral force owing to cross wind

l_w : Distance between c.g. and c.p.

$l_w > 0$ if c. p. before c. g.

$s.m. = (k_r l_r - k_f l_f) / l(k_f + k_r)$: Static margin

$s.m. > 0$ means the understeering characteristics

The lateral motion of the vehicle is expressed as

$$MV \frac{d\beta}{dt} + MV\omega = -2k_f \left(\beta + \frac{l_f}{V}\omega - \alpha \right) - 2k_r \left(\beta - \frac{l_r}{V}\omega \right) + F_w \quad (1)$$

The rotational motion of the vehicle around the c.g. is expressed as

$$J \frac{d\omega}{dt} = -2k_f \left(\beta + \frac{l_f}{V}\omega - \alpha \right) l_f + 2k_r \left(\beta - \frac{l_r}{V}\omega \right) l_r + F_w l_w \quad (2)$$

Considering the moment balance around the steering gear, the equation of motion of the steering system can be represented approximately as follows:

$$fr - J_h \frac{d^2 A}{dt^2} - C_h \frac{dA}{dt} = n \left\{ J_s \frac{d^2(nA)}{dt^2} + C_s \frac{d(nA)}{dt} - k_s(nA - \alpha) \right\}$$

By putting $C_h/n + C_s = C$ and $J_h/n^2 + J_s = I$, the above equation is reduced to

$$In \frac{d^2 A}{dt^2} + Cn \frac{dA}{dt} + k_s(nA - \alpha) = \frac{fr}{n} \quad (3)$$

The relation between the steering angle of the steering wheel A and that of the front wheels α can be obtained from the moment balance around the king pin as

$$k_s(nA - \alpha) = -k_e \left(\beta + \frac{l_f}{V}\omega - \alpha \right)$$

From this equation the steering angle at the front wheels α is given in the form:

$$\alpha = \frac{k_e}{k_s + k_e} \left(\beta + \frac{l_f}{V}\omega \right) + \frac{k_s}{k_s + k_e} nA \quad (4)$$

The coordinates of the c. g. of the vehicle are given by

$$\left. \begin{aligned} x &= x_0 + \int_0^t V \sin \varphi dt \\ y &= y_0 + \int_0^t V \cos \varphi dt \end{aligned} \right\} \quad (5)$$

The coordinates of the predicted position of the c. g. after τ sec. can be expressed as follows:

$$\left. \begin{aligned} x^* &= x + \int_0^\tau V \sin(\varphi + \dot{\varphi}t) dt \\ y^* &= y + \int_0^\tau V \cos(\varphi + \dot{\varphi}t) dt \end{aligned} \right\} \quad (6)$$

4. Simulation experiments

Combining the driver's steering model with the equation of motion of the vehicle, we shall proceed to the simulation of the man-automobile systems by a digital computer making use of the Runge-Kutta-Gill Method. As an example a passenger car of middle size is taken, the data of which are as follows:

$$l = 2.690 \text{ m} \quad M = 169 \text{ kg sec}^2/\text{m}$$

$$J = 200 \text{ kg m sec}^2 \quad r = 0.2 \text{ m}$$

$$I = 2.0 \text{ kg m sec}^2/\text{rad} \quad C = 90 \text{ kg m sec}$$

$$k_s = 880 \text{ kg m/rad} \quad k_e = 290 \text{ kg m/rad}$$

$$F_w = 200 \text{ kg (where } V = 20 \text{ m/sec, cross-wind velocity } v = 18 \text{ m/sec)}$$

where k_f , k_r , l_f and l_r are determined properly in

Fig. 4 Axis system for simplified automobile

Fig. 5 The dynamic model of the simplified steering system

accordance with the change of s , m .

4.1 Tracking of curved courses

We shall compare various driving models regarding to the accuracy of tracking of U-shaped courses, each of which consists of a semi-circular path and two straight paths. To investigate the influences of the vehicle speed and the curvature of the course, two simulation experiments have been carried out. In one case the curvature of the path is changed keeping the vehicle speed constant, and in another case the vehicle speed and the curvature of the path are changed so that the maximum lateral accelera-

tion takes a certain constant value.

Figure 6 shows the trajectories of c. g. of the vehicle in the former case, where the vehicle is driven at 15 m/sec (54 km/hr) along the U-shaped courses whose radii of the curvature are 45 m and 90 m, and the maximum lateral accelerations are $0.5g$ and $0.25g$, respectively. From this figure we can see the following facts. As to the preview correction model we must choose a proper value of the eye-fixation distance L in accordance with the curvature. This model also shows the shortcoming that the feedback of the present position error of the c.g. becomes insufficient at higher speeds. With the predetermined correction model, we need not adjust the parameters of the feedback action, but we must determine the accurate program beforehand according to the curvature of the course. In the simulations with the above driving models it is difficult to exclude the steady deviations from the right courses, even when the parameters of the model are adjusted to be optimum. On the contrary, making use of the predicted correction model the accurate tracking results are obtained without adjusting the parameters of the model driver to the change of the course and this fact suggests us that this model may represent actions of a skilled driver considerably well.

Figure 7 shows the maximum deviation from the standard path in the latter case, where the vehicle speed and the curvature of the course are changed to compare the three models concerning the tracking accuracy under the same maximum lateral acceleration of $0.5g$. Consequently, with the predicted correction model we can realize the most accurate and the most practical tracking results in spite of

- Trajectories with the preview correction model
- Trajectories with the predetermined correction model
- · - Trajectories with the predicted correction model
- Standard course

Vehicle speed 15 m/sec (54 km/hr)
(note) Trajectories with the preview correction model and the predetermined correction model are taken from Ohno's report⁽⁷⁾

Fig. 6 Trajectories of c.g. when the vehicle is driven along the U-curved course

Fig. 7 The simulation results of tracking of the U-shaped courses with the various steering models—The maximum deviation of c. g. from the standard path under the same maximum acceleration of $0.5g$

S-shaped course

Fig. 8 Tracking of an S-shaped course

Fig. 9 Saturation characteristics of cornering force of tires

Fig. 10 The spin caused by the saturation characteristics of tires (The direction of the arrow shows the direction of the center-line of the vehicle)

Fig. 11 The behaviors of the man-automobile system when the vehicle is disturbed by a cross-gust while it is driven along a straight course

the fact that the parameters of the model driver are kept at constant values (i.e. $\tau = 2.5$ sec., $H = 0.35$ kg/m and $T = 0.6$ sec.) regardless of the change of the vehicle speed and the curvature of the right course.

Now, with the predicted correction model to represent the steering action of the standard driver, we will investigate the influence of the dynamic characteristics of the vehicle on the tracking results. Figure 8 illustrates the variations of the steering force, of the steering angle and of the displacement of the c. g. for two different values of $s. m.$ Here, the standard course is an S-shaped course which comprises two semi-circular paths and two straight paths. This figure suggests that a slightly oversteering vehicle responds more quickly to the steering action and has better tracking accuracy to the curved course.

4.2 Influence of the saturation characteristics of the cornering force

In this simulation experiment, as the broken line in Fig. 9 shows, the tires are assumed to have the saturation characteristics such that the cornering force takes a constant value when the slip angle of the tire is beyond 0.11 rad. Figure 10 shows the tracking results to a sharply curved course. The influence of saturation appears from the point marked with the broken line. It is generally said that the oversteering vehicles have good tracking characteristics though they have the tendency to fall into a spin.

4.3 Influence of a cross-wind disturbance

Figure 11 shows the behavior of a man-automobile system when it is disturbed by a cross-gust while it is driven along a straight course. From this figure it is found that the change of the steering angle caused by the handle reaction owing to the cross-gust can be simulated. According to this example, if the driver continues the same control action when the vehicle is disturbed by the cross-gust, we can say that the vehicle whose c. p. is slightly behind c. g. will deviate less from the standard path and be more stable.

5. Discussion of the predicted correction model

This discrete model may be approximated by the continuous model shown in Fig. 12. If we consider that the input to the vehicle is the steering force and the output is the lateral displacement, the gain of the vehicle will become almost constant regardless of the

change of the vehicle speed. In order to keep the characteristics of the man-automobile system constant under the change of the vehicle speed, it will be desirable to fix the product of the steering model gain H/T and the coefficient of the second derivative $\tau^2/2$ which are the most important terms in the predict operation, thus adjusting the driver's gain H proportional to T/τ^2 .

The sampling period T may be more than 0.6 sec., because the human response time is usually about 0.3 sec.

As for the prediction time τ , in these simulation experiments, the prediction time $\tau=2.2\sim 3.5$ sec. gives good tracking results. Figure 13 shows the braking distance and the eye-fixation distance which is equal to the product of the prediction time and the vehicle speed. It is found that the adequate eye-fixation distance is a little larger than the braking distance and thus the prediction time employed here may be considered reasonable from a practical viewpoint. Measuring the eye-fixation distance in the steady-state cornering tests, Kondo and Ajimine have also reported that the so-called spare time, which is the value of the eye-fixation distance divid-

ed by the vehicle speed, is $2\sim 4$ sec⁽⁹⁾.

6. Conclusions

In this paper a new driver's steering model with the second order predictability has been proposed, and with this model the dynamics of such man-automobile systems has been investigated making use of the simulation by a digital computer. From these simulation experiments, the adequateness and practicability of the model and of the simulation method are confirmed.

In the near future if we carry out more simulation experiments and more driving tests of such man-automobile systems and accumulate sufficient reliable data, it will become possible to estimate not only the tracking characteristics but also the driving feeling with the knowledge of the structural specifications of the vehicle. This simulation method will be also interesting from the safety engineering point of view and may be useful for the training of drivers and for the synthesis of highways. For the purpose of the development of this method, it may be also necessary to develop the criterion for evaluation of the behavior of man-automobile systems.

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Fig. 12 The approximation of the predictable correction model by the continuous model (s : Laplace's operator)

Fig. 13 Comparison between the eye-fixation distance and the braking distance