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CONSIDERATIONS CONCERNING IMPROVEMENT OF EMERGENCY EVASION PERFORMANCE

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ABSTRACT−When emergency evasion during running is required, a driver sometimes causes a vehicle to drift, that is, a condition in which the rear wheels skid due to rapid steering. Under such conditions, the vehicle enters a very unstable state and often becomes uncontrollable. An unstable state of the vehicle induced by rapid steering was simulated and the effect of differential steering assistance was examined. Results indicate that, in emergency evasion while cornering and during which the vehicle begins to drift, unstable behavior like spins can be avoided by differential steering assistance and both the stability and control of the vehicle is improved remarkably. In addition, reduction of overshoot during spin evasion by the differential steering assistance has been shown to enable the vehicle to return to a state of stability in a short time in emergency evasion during straight-line running. Moreover, the effectiveness of differential steering assistance during emergency evasion was confirmed using a driving simulator.

KEY WORDS : Motion control, Automobile, Vehicle dynamics, Steering system, Maneuverability, Stability, Simulation, Driver model

1. INTRODUCTION

Rapid steering during emergency evasion may cause the rear wheels to skid and the vehicle may begin to drift. Under such conditions, the vehicle becomes unstable and running becomes impossible. In this research, a vehicle made unstable by rapid steering was simulated, and the effect of differential steering assistance was examined. A steering system involving a "differentiation steering wheel" was investigated by Hirao (1969); Hirao and Yamada (1966) and the optimum range of differential terms was investigated by Nakaya (1994); Miyamori and Nakaya (1997). These studies revealed the effect of grip area. The grip area is the area over which the rear wheel does not exceed the maximum cornering force. In addition, the four-wheel steering (4WS) system is thought to be an effective means of stability improvement in the grip area compared with the "differentiation steering wheel", and the 4WS system has already been implemented in commercially available vehicles. However, the 4WS system has not been demonstrate to be effective for countering drift. In contrast, since differential steering assistance improves the delay of counter steering, differential steering assistance may be effective for countering drift. Therefore, the effectiveness of a system that has front-wheel steer angle assist that considers the steer angle velocity with respect to normal front-wheel steer angles was examined as a drift running performance improvement technique. Delay in steering by the driver can be improved by steering assistance proportional to the steer angle velocity. Here, the drift area is the area over which the rear wheel exceeds the maximum cornering force, and the area where counter steer is required. And, the grip area is a comparatively small area in the tire slip angle of front and rear wheel, and the area where the maximum cornering force is not exceeded (Figure 1).

And, when the degree of the differentiation steer assistance is large, the over shot of the front wheel steer

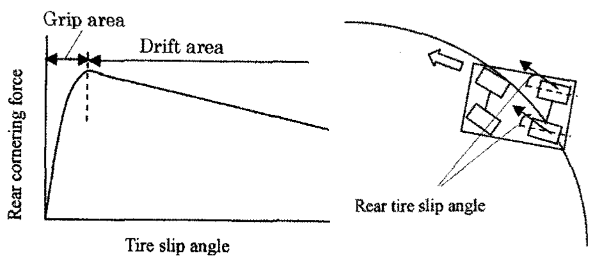

Figure 1. Use area of rear wheel tire in the drift area and the grip area in the vehicle.

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angle might be generated in the grip area for the step steer as demerit of the differentiation steer assistance. Therefore, though too big differentiation steer assistance is not preferable in the accuracy of the steer. But, there is little influence if the assistance ratio is small. On the other hand, though the delay of counter steer becomes a problem in the drift area, it has been understood that this delay is remarkably improved even if it is small differentiation steer assistance. Therefore, when the drift running with counter steer, it has been understood that the effect of the differentiation steer assistance was large.

2. OUTLINE OF THE SIMULATION

2.1. Outline of the Vehicle Model

The CarSim (Version 5.16) simulation model (MSC Co.: USA) was used as a vehicle model for full vehicle movement. Figure 2 shows a schematic diagram of the vehicle model. Table 1 shows a number of the primary components of the vehicle and indicates the degrees of freedom. For example, the rear axle is rigid and has two freedom degrees: vertical movement and rotation of the axle (Figure 3). Details are provided in Reference (Watanabe and Sayers, 2002).

The vehicle model has an FR layout (front-mounted engine, rear-wheel drive), which is particularly susceptible to drift. Table 2 shows the specifications of the parameters of the vehicle model used in the experiment.

The tire cornering force characteristic used for the experiment is the general tire characteristic, as shown in Figure 4.

In addition, the tire characteristic during driving and braking is assumed to be as shown in Figure 5. By adding

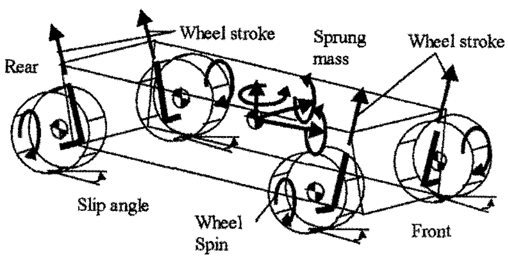


Figure 2. Primary components and degrees of freedom of the vehicle model.

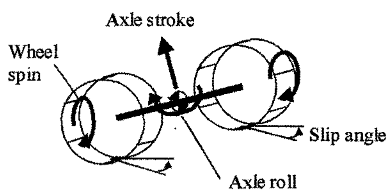


Figure 3. Solid-axle movement.

Table 1. Components and degrees of freedom of the vehicle model.

Bodies	
Sprung mass body	1
Unsprung mass bodies (wheel carriers)	4
Rotating wheels	4
Engine crankshaft	1
Total	10

Degrees of freedom	
Sprung body translation (X, Y, Z)	3
Sprung body rotation (yaw, pitch, roll)	3
Suspension stroke	4
Wheel spin	4
Powertrain (Engine crank shaft)	1
Tire delayed slip (lateral, longitudinal)	8
Brake fluid pressure	4
Total	27

Table 2. Specifications of vehicle.

Width of vehicle (mm)	1500
Wheelbase (mm)	2690
Distance from center of front axle to rear end of vehicle (mm)	3800
Distance from center of front axle to center of gravity (mm)	1014
Height from ground to center of gravity (mm)	542
Vehicle mass (kg)	527
Roll moment of inertia (kg·m^2)	606.1
Pitch moment of inertia (kg·m^2)	2741.9
Yaw moment of inertia (kg·m^2)	2741.9

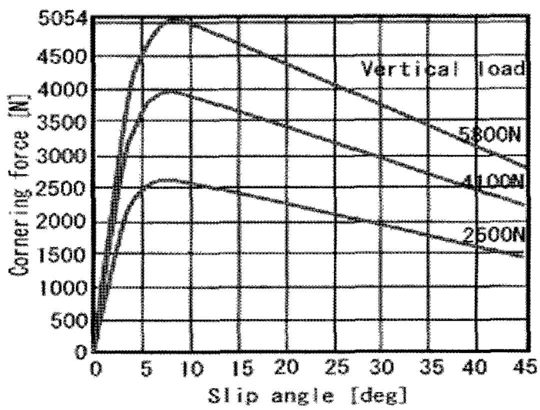


Figure 4. Tire cornering force characteristic.

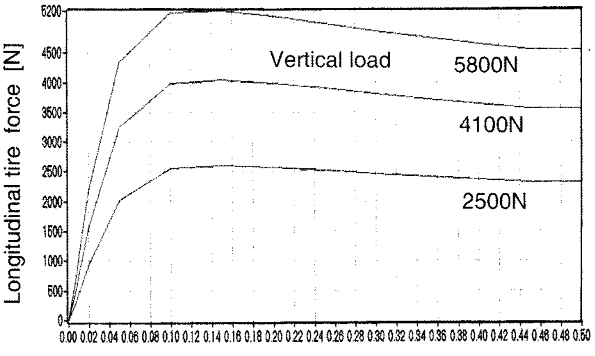


Figure 5. Longitudinal tire force characteristic.

the slip angle and the slip ratio for a specific time, the concept of the friction circle based on the calculation of the combined characteristic of CarSim is applied. Therefore, the vehicle tends to enter spins because the lateral force of the rear wheel decreases when accelerating during critical cornering. Thus, drift cornering can be achieved by counter steering, and a real car running under conditions in which the maximum cornering force is exceeded can be simulated.

2.2. Driver Model

In this driver model (CarSim), the application car model of the optimum control theory of the MacAdam proposal is used. Details are provided in Reference (MacAdam, 1980; 1981). In the driver model, the driver perceives the target course ahead of the vehicle, as shown in Figure 6. The perception time is obtained by dividing the perceived distance by the velocity of the vehicle. The driver recognizes the target course and steers a presumed orbit (presumed course) based on the present state of the vehicle and the perception time. This presumed course is

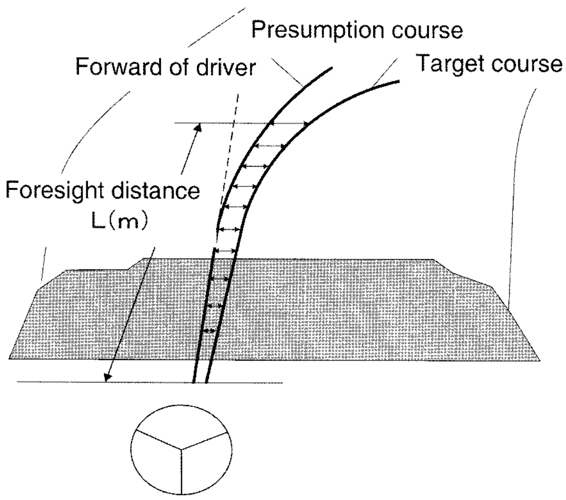


Figure 6. Outline of the driver model.

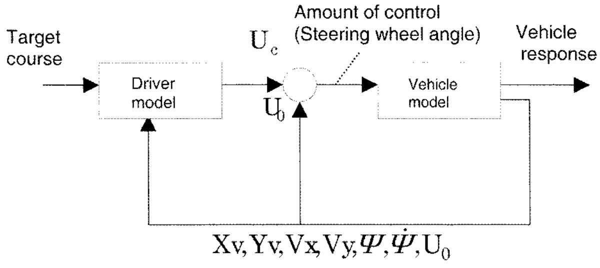


Figure 7. Algorithm of the driver model.

thought to be predicted based on the state of the vehicle, as perceived by the driver, and the flow of forward view. In addition, in order to minimize the error between the target course and the presumed course in the perception time, steering control is performed. Figure 7 shows the algorithm of the driver model. The target course is given to the driver, and the response of the vehicle is fed back to the driver. The response includes the position (X_V, Y_V) of the center of front axle of the vehicle in the absolute coordinates, the yaw angle and rate $(\psi, \dot{\psi})$, the longitudinal and lateral velocities (V_X, V_Y), and the compliance steering U_0.

The driver performs the following actions based on the following information:

① The movement tracks of the vehicle in the perception time are presumed from the present state of the vehicle (course presumption).

② A The driver steers so as to minimize the deflection between the target course and the presumed course. In this case, the compliance steer is added to the feedback steer of the driver.

③ B The steering of the driver considers the reactive delay time of the driver.

This driver model is assumed to be a steer model by which compliance steer is quantitatively added to feedback. The purpose of the reason is to delete of the error caused by compliance steer. Though a general driver might not be able to do the feedback steer by compliance steer's feeling, the expert driver might be able to steer feedback by compliance steer's feeling. Anyway, to improve the follow of the course, it is assumed the steer model by compliance steer feedback. Next, the outline of the equation of the driver model is shown. In the calculation of the presumption course, the following state variable procession equations can be expressed.

$$\begin{cases} x=Ax+Bu \\ y=Cx \end{cases} \tag{1}$$

Here, u is an input of the steer angle from the driver, and state vector x is shown as follows.

x_1: In the driver X coordinate system taken centering on forward of the driver, Y coordinates at position of vehicle

center of gravity which will be forecast in the future when position of present driver is made starting point. coordinates at a present vehicle center of gravity position are assumed to be 0.
x_2: In the driver coordinate system, posture angle of the vehicle which will be forecast in the future yaw angle. The posture angle of a present vehicle is assumed to be 0.
x_3: Lateral velocity of vehicle
x_4: Yaw rate of vehicle

Output y shows Y coordinates of the driver which will be forecast in the future in the driver coordinate system. In the driver coordinate system where the position of a present driver is a starting point, and the direction ahead of the driver is X axis, Y coordinates at the position of the driver which moves between 0 and foresight time T are requested by the Euler integration of equation (1) (Figure 8). In the calculation of the target course, distance (S) along from the starting point to the course was defined assuming that other than the course data of an absolute coordinate system. The way of the target course in the foresight when the vehicle runs at vehicle velocity V_x in present way S becomes the next equation.

$$S_{t\arg,i} = S + \frac{iV_xT}{m} \tag{2}$$

Here, $i = 1 \cdots m(=10)$

If the way in target course $S_{t\arg,i}$ is understood, This is compared with data (S, X, Y), and target course ($X(S_{t\arg})$, $Y(S_{t\arg})$) in the absolute coordinate system is obtained. Therefore, the amount of the gap of the target course in the driver coordinate system becomes the next equation

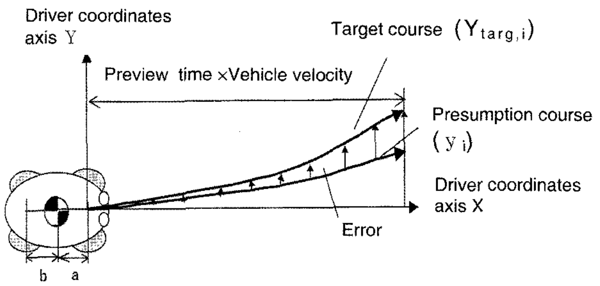


Figure 8. Error between target course and presumption course in driver coordinates axis.

sidewise by using yaw angle ψ of a present vehicle (The reference is Figure 9).

$$Y_{t\arg,i} = [Y(Y_{t\arg,i}) - Y_V]\cos(\psi) - [X(S_{t\arg,i}) - X_V]\sin(\psi) \tag{3}$$

This driver model is optimal controlled, and steered like minimizing the error between the target course and the presumption course in the foresight time.

$$J = \frac{1}{m}\sum_{n=1}^{m} W_i(Y_{t\arg,i} - y_i)^2 \tag{4}$$

Here, W_i: The function which puts arbitrary weight.

2.3. Vehicle Steering Method (differential steering assistance)

Next, the steering method of the differential steering assistance is given as

$$\delta_f = \delta_H/N + P \cdot \dot{\delta}_H \tag{5}$$

and the block diagram is shown in Figure 9.

Here, δ_f is the front-wheel steer angle, δ_H is the steering wheel angle, N is the steering wheel gear ratio (= 12), P is the assistance constant, and $\dot{\delta}_H$ is the steer angle velocity. The effect of differential steering assistance on the vehicle was simulated in three cases (vehicle A, $P = 0$ (no assistance): vehicle B, $P = 0.005$; and vehicle C, $P = 0.007$) for a drifting vehicle. Figure 10 shows the course used for the simulation. The lane change while following a curve shown in Figure 10(a) was simulated at vehicle velocities of 100 km/h and 110 km/h, and the double lane change shown in Figure 10(b) was simulated at vehicle velocities of 130 km/h and 140 km/h.

3. SIMULATION RESULTS

The simulation result of the lane change on a curve is shown for the case of the vehicle velocity of 110 km/h if Figures 11 to 14. Based on the vehicle tracks of vehicle A, which had no differential steering assistance, the driver was unable to control the vehicle, which fell into a state of drift during rapid steering and entered a spin (Figure 11). However, vehicles B and C, which had

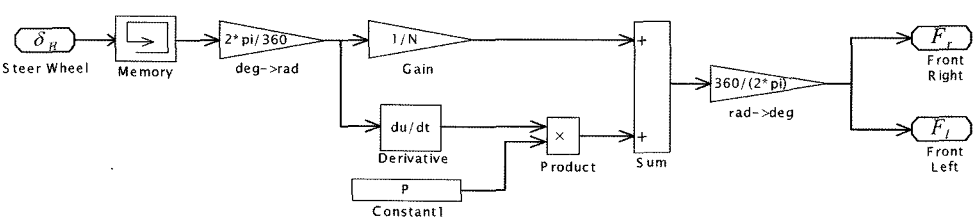
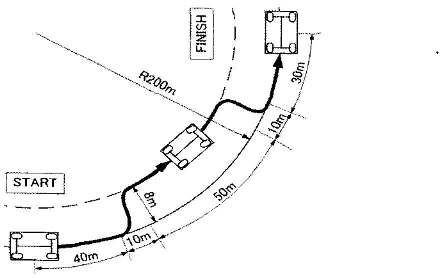
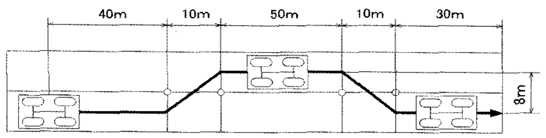


Figure 9. Block diagram of differential steering assistance.



(a) Turning course



(b) Double lane change course

Figure 10. Running course.

differential steering assistance, were able to avoid entering a spin and returned to the course. This indicates the effectiveness of the counter steering. Compared with the steer angle generated by rapid steering of vehicle A, those of vehicles B and C were preferable (Figure 12). The frequency of the repetition of counter steer generated by vehicle C was less than that generated by vehicle B. Although the vehicle is entered spin because the driver of vehicle A cannot control the vehicle, which falls into a

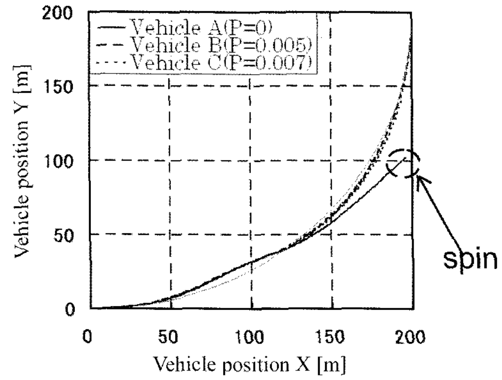


Figure 11. Running trajectory.

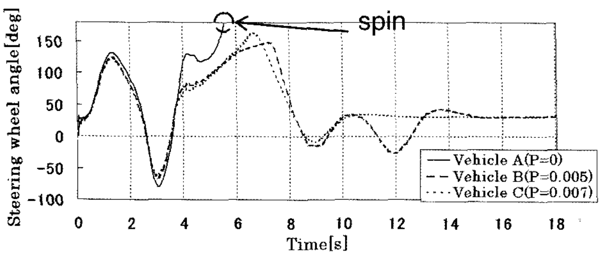


Figure 12. Steering wheel angle.

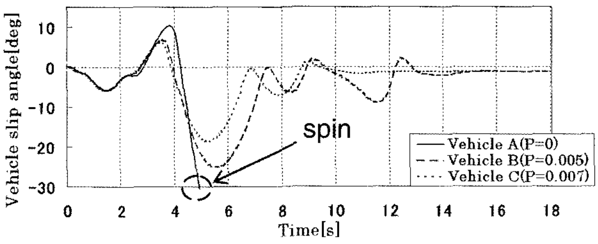


Figure 13. Vehicle body slip angle.

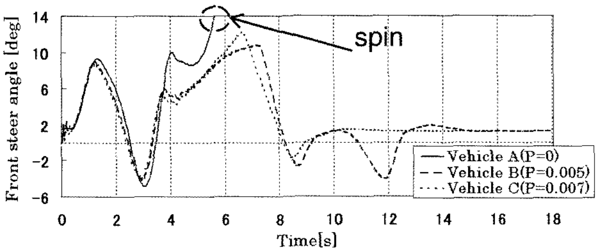


Figure 14. Front-wheel steer angle.

state of drift when the body slip angle exceeds 10 deg, vehicle B and vehicle C, which fall into a state of controlled drift, become stable (Figure 13). The phase of the front-wheel steer angle increases for vehicles B and C, which had differential steering assistance, compared vehicle A, which had no differential steering assistance for the front-wheel steer angle (Figure 14). Figures 12 and 14 indicate that the front-wheel steer angle required for controlling the vehicle decreases as the phase of the front-wheel steer angle increases during drifting.

The simulation results for the double lane change are shown for the vehicle velocity of 140 km/h in Figures 15 to 18. Vehicle A, which had no differential steering assistance, and vehicle B encountered spins, whereas vehicle C did not enter a spin. As a result, the effect of the differential steering assistance when running at high velocity was clarified. Based on the graph of the front-wheel steer angle in Figure 18, for vehicles B and C, the phase of the front-wheel steer angle increased by the differential steering assistance compared to vehicle A. Moreover, as shown in Figure 17, the effect on the body

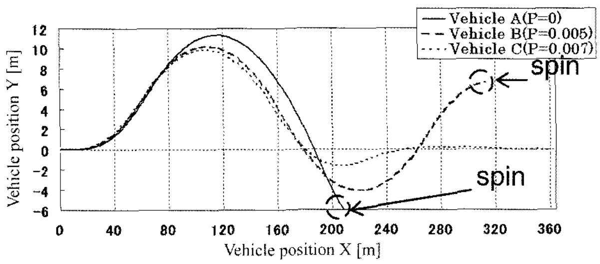


Figure 15. Running trajectory.

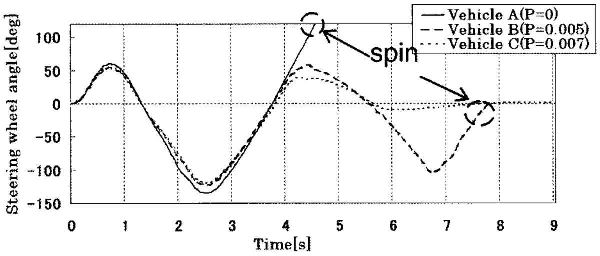


Figure 16. Steering wheel angle.

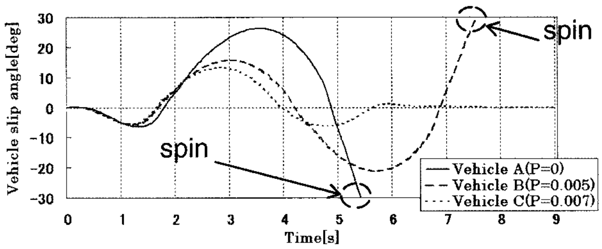


Figure 17. Vehicle body slip angle.

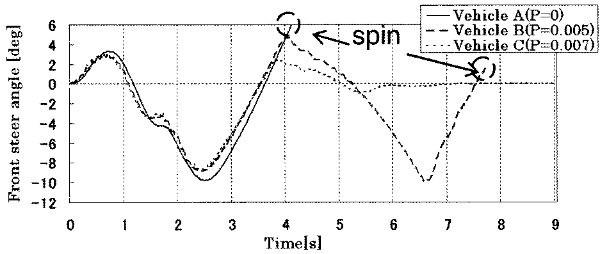


Figure 18. Front-wheel steer angle.

slip angle of the slight difference in phase of the front-wheel steer angle was remarkable. The body slip angles of vehicles A and B were increased by delaying the timing of the application and release of sufficient counter steering, making the vehicle impossible to control and causing the vehicle to enter a spin. For the vehicle tracks shown in Figure 15, compared to vehicle A, the stability of vehicle C increases as the overshoot is reduced.

4. EXAMINATION RESULTS (CONFIRMATION BY THE DRIVING SIMULATOR)

4.1. Outline of Experimental Apparatus (driving simulator)
Figures 19 and 20 show the driving simulator used in the present study. The CarSim (Version 5.16) simulation model (MSC Co.: USA) for full vehicle movement was used as the vehicle model for the driving simulator as well as the above-mentioned simulation model. Regarding the simulator, a view reflection system and sound generation system to simulate the engine sound of a running vehicle were used to reproduce the actual driving

Figure 19. Driving simulator.

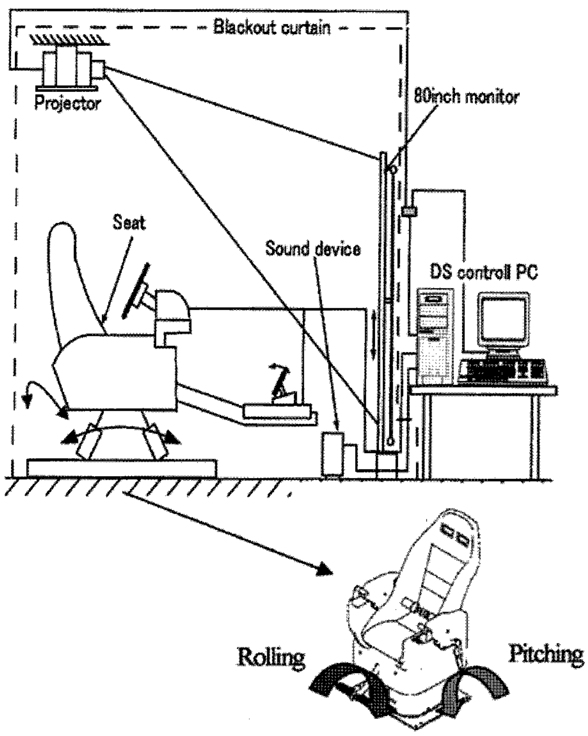


Figure 20. Outline of driving simulator.

conditions, and a motion device allowing two degrees of freedom (roll and pitch) was built into the model. Rolling of the motion device is operated by inputting the roll angle signal of the movement model into the vehicle. Rolling of the motion device is not performed by the lateral acceleration signal. The reason for the driver is that the problem of getting drunk with the simulator is caused according to the difference with an actual vehicle with a simulator to the reproduction of high lateral accelerations. Therefore, the rolling of the movement device is simulated by the volume control of extent by which the roll of an actual running can be imitated. Moreover, the following items were measured.

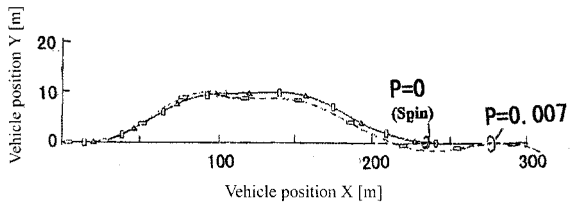


Figure 21. Experimental results for running trajectory with driving simulator.

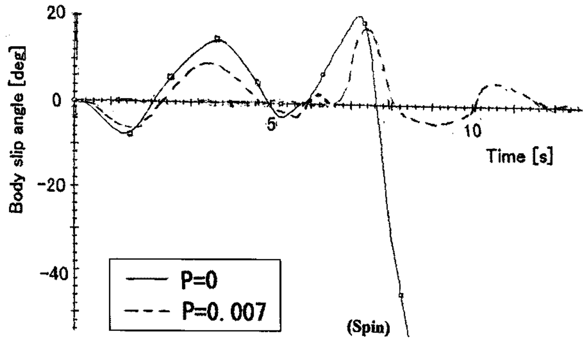


Figure 22. Experimental results for vehicle body slip angle with driving simulator.

The vehicle drive state data include steer angle, steer torque, vehicle velocity, running track, body slip angle, yaw rate, yaw angle, roll angle, pitch angle, lateral acceleration, and the steer angle of each wheel.

4.2. Experimental Results

For the double lane change, in which drifting is assumed, for the cases of no differential steering assistance ($P = 0$) and a differential steering assistance of $P = 0.007$, running examination using the driving simulator was performed. Figures 21 and 22 show the experimental results (for a vehicle velocity of approximately 100 km/h). The side-slip behavior of the rear wheels when drifting was controlled became stable for a differential steering assistance of $P = 0.007$ compared with $P = 0$ (no differential steering assistance). Moreover, at $P = 0.007$, the counter steer operation improved the vehicle stability. In contrast, counter steering for $P = 0$ was insufficient, and the vehicle entered a spin. Therefore, differential steering assistance was confirmed to be effective for performance improvement during emergency evasion involving drifting.

5. CONCLUSIONS

Improvement of the emergency evasion performance of vehicles was considered and the following conclusions were obtained:

(1) When the driver performs an emergency evasion during cornering and the vehicle enters a state of drifting, unstable spin behavior can be avoided through the use of differential steering assistance, which improve both stability and control remarkably.

(2) Differential steering assistance can be used to avoid entering into a spin and can return an unstable vehicle to a state of stability in a short time for the case of emergency evasion in the straight line. Moreover, a running experiment involving a driving simulator confirmed the effectiveness of differential steering assistance during emergency evasion.

In the future, more effective methods of steering assistance will be examined.

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