

Effects of Steering Torque Control Using a Driving Diagram

Shigeru YAMAWAKI*

Yasuo SHIMIZU*

ABSTRACT

This paper proposes a driving diagram, sets out a new steering torque control method derived from the diagram, and explains the effectiveness of this method. The driving diagram uses the actual steering angle δ of the front road wheels and the difference between the front tire slip angle and the rear tire slip angle β_{fr} to express the vehicle behavior resulting from driver operation. The new steering torque control applies steering reaction torque to the driver in order to elicit the actual steering angle of the front road wheel that will bring about the most stable state of the vehicle. In this way, when the vehicle enters a spinning state, the driver can know the direction in which to steer in order to help stabilize the vehicle.

Along with a discussion of the effectiveness of this driving diagram, the paper also provides the results of driving tests conducted using a vehicle equipped with Electric Power Steering (EPS) with steering torque control. These tests confirmed the effectiveness of this method for the vehicle in both under-steer and over-steer states.

1. Introduction

Steering reaction torque plays an important role in driver-vehicle systems. For example, if the self-aligning torque is increased with the vehicle in the over-steer state, the lag by which the driver returns the steering wheel can be reduced when recovering from spinning. In addition, there are also recent reports that the vehicle yaw response can be improved and vehicle fluctuation relative to disturbances can be suppressed by controlling the steering reaction torque according to the vehicle yaw rate, lateral acceleration and other factors^{(1) - (3)}. On the other hand, it is also known that the vehicle under-steer and over-steer states can be described using the difference between the front tire slip angle and the rear tire slip angle ($\beta_f - \beta_r$), etc⁽⁴⁾. However, there are as yet no effective proposals for avoiding unexpected driver situations such as drift-out or spinning by applying steering reaction torque according to under-steer, over-steer or other driving conditions.

This paper proposes a driving diagram as a method for understanding the under-steer and over-steer states using the

actual steering angle of the front road wheel and the difference between the front tire slip angle and the rear tire slip angle, and sets out steering reaction torque characteristics based on this diagram⁽⁵⁾. These characteristics are then used to derive a steering torque control method for recovering from drift-out in the under-steer state or spinning in the over-steer state. Tests were conducted under low road surface friction coefficient conditions using a vehicle equipped with an Electric Power Steering (EPS) system that uses this control method, and this paper reports the results of these tests which confirmed an improvement in performance.

2. Driving Diagram ($\delta - \beta_{fr}$ diagram)

2.1 Driving Diagram

The vehicle conditions when a driver turns the steering wheel are expressed using the actual steering angle of the front road wheel δ and the difference between the front tire slip angle and the rear tire slip angle β_{fr} ($= \beta_f - \beta_r$) as

* Tochigi R&D Center

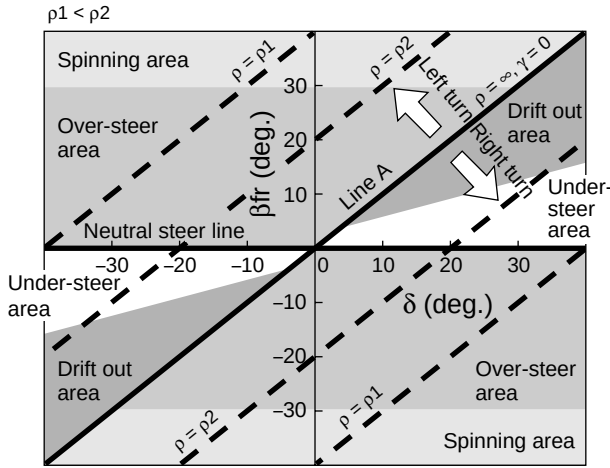


Fig. 1 Driving diagram

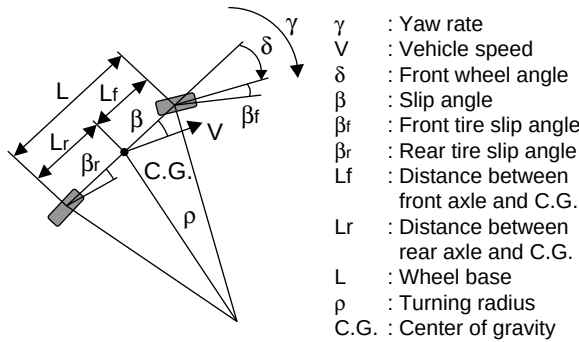


Fig. 2 Vehicle model and nomenclature

shown in Fig. 1, and this is called the driving diagram. This diagram is explained using the bicycle model shown in Fig. 2. The tire slip angles β_f and β_r can be expressed by the following equations. Note that the slip angle is taken into account so that right turn is the first quadrant and left turn is the third quadrant in the driving diagram, and the opposite of the normal direction is positive.

$$\beta_f = -\beta - L_f \cdot \gamma / V + \delta \quad (1)$$

$$\beta_r = -\beta + L_r \cdot \gamma / V \quad (2)$$

Here, β is eliminated from equations (1) and (2) and β_{fr} is obtained to express the vehicle steering characteristic conditions as follows.

$$\beta_{fr} = \beta_f - \beta_r = -L \cdot \gamma / V + \delta \quad (3)$$

Also, ignoring β as a small value compared to γ at this condition and setting the turning radius to $\rho = V/\gamma$ yields the following equations.

$$\beta_{fr} = -L / \rho + \delta \quad (4)$$

$$\rho = L / (\delta - \beta_{fr}) \quad (4')$$

Equation (4') indicates that for example when $\delta > 0$ in steady state cornering, ρ becomes larger (under-steer area) when $\delta > \beta_{fr} > 0$, and ρ becomes smaller (over-steer area) when $\beta_{fr} < 0$. In addition, even when $\delta < 0$, ρ becomes even smaller when $\beta_{fr} < 0$ and $\delta - \beta_{fr} > 0$, so this area is also the over-steer area.

Line A in Fig. 1 shows the relationship between δ and β_{fr} in straight ahead driving ($\gamma = 0$). This indicates that $\beta_{fr} = \delta$ when $\gamma = 0$ according to equation (3). In addition, line A shifts over the β_{fr} axis with ρ as the parameter as shown by the dotted lines according to equation (4). This means that ρ can be obtained from δ and β_{fr} .

2.2 Driving Locus by Actual Running Tests

Next, the driving locus was investigated to determine how best to express δ which shows the driver operation and β_{fr} which shows the vehicle slip condition on the driving diagram. Figs. 3 to 6 show the driving diagrams when running in an actual vehicle. The vehicle used was an FF vehicle.

2.2.1 Under-steer area (Lane change)

When the vehicle speed is constant and assuming the yaw rate gain $G_y (= \gamma/\delta)$ is fairly constant, equation (3) can be modified as follows.

$$\beta_{fr} = \delta (1 - L \cdot G_y / V) \quad (5)$$

From this equation (5), β_{fr} has the gradient $(1 - L \cdot G_y / V)$ and is proportional to δ . However, in actual running a phase lag occurs when δ begins to turn, so the driving locus has hysteresis as shown in Fig. 3.

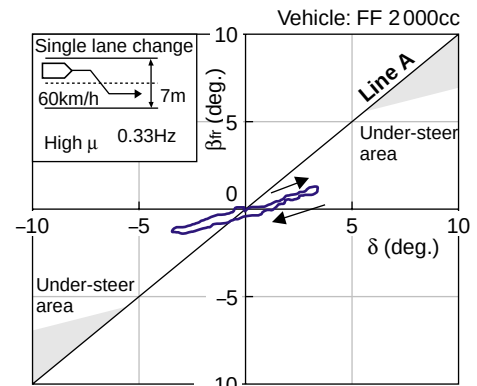


Fig. 3 Single lane change (Under-steer)

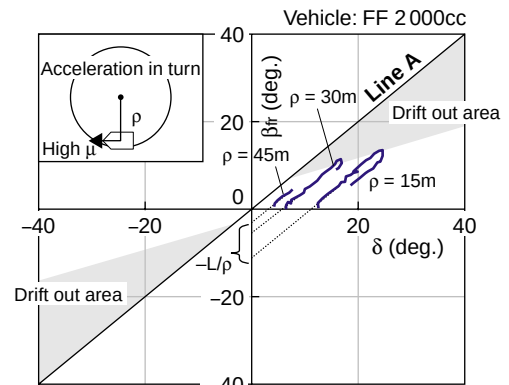


Fig. 4 Acceleration in turn (Under-steer)

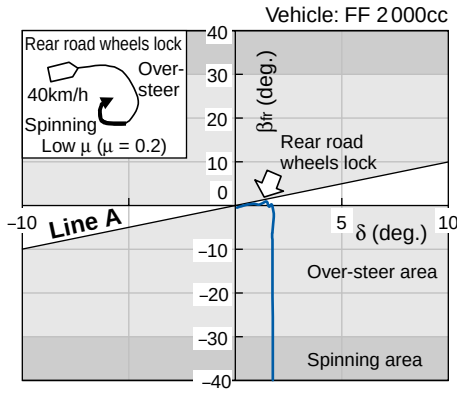


Fig. 5 Spinning (Over-steer)

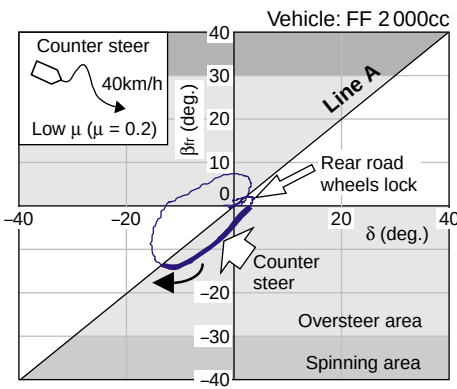


Fig. 6 Spinning recovery (Over-steer)

2.2.2 Drift-out area (Acceleration in turn)

As shown in Fig. 4, δ is turned at a constant angle to the right in the stopped condition so as to trace a circle, and then the vehicle starts to accelerate to make a circular turn while increasing δ . Here, ρ is a constant, so according to equation (4), the driving locus moves in a straight line parallel to line A which intersects the β_{fr} axis at $(-L/\rho)$. If δ or the vehicle speed V is increased further, β_{fr} increases and reaches the drift-out area. The vehicle β_{fr} has a limit, and due to the relationship with δ a drift-out area exists in which the turning radius does not decrease even if δ is increased. Fig. 4 shows the experimentally obtained drift-out area.

2.2.3 Over-steer area (Spinning)

When δ is fixed and the rear road wheels are forcibly locked using the brakes when starting to turn as shown in Fig. 5, the vehicle spins and $|\beta_{fr}|$ increases rapidly. This occurs even at the same δ because equation (4) shows that ρ becomes smaller. The vehicle $|\beta_{fr}|$ has a limit, and a spinning area exists in which the turning radius cannot be increased even if δ is returned. Fig. 5 shows the experimentally obtained spinning area.

Fig. 6 shows counter-steer when recovering from spinning. After entering the over-steer state, it was found that spinning can be avoided by turning δ in the opposite direction and performing operations that suppress the increase in $|\beta_{fr}|$ and bring γ to 0 (approach line A).

3. Investigation of New Steering Torque Control

3.1 Steering Reaction Torque Characteristics

Fig. 4 shows that in order to avoid entering the drift-out area, steering torque control in the under-steer state should provide heavy steering controlled reaction torque which makes it difficult to increase δ so that β_{fr} does not increase as it nears the drift-out area. In addition, Figs. 5 and 6 show that counter-steer in the over-steer state should be assisted by applying steering controlled reaction torque in the steering direction to return β_{fr} and γ to zero. Therefore, equation (3) can be modified as follows.

$$\delta = \beta_{fr} + L \cdot \gamma / V \quad (3')$$

In order to reduce β_{fr} and γ toward zero, the controlled reaction torque $Tr1$ can be obtained as follows from equation (3') as a steering reaction torque characteristic.

$$Tr1 = G1 \cdot \beta_{fr} + G2 \cdot \gamma \quad (6)$$

($G1$ and $G2$ are coefficients.)

3.2 Control Method for Stabilizing the Vehicle

Specific control algorithms were investigated using Fig. 1 and equation (6). The vehicle under-steer and over-steer states can be determined as shown in the equations below from the sign of β_{fr} and the sign of γ used when obtaining β_{fr} . The under-steer state is given by equation (7), and the over-steer state by equation (9).

Note that the signs of $G1$ and $G2$ were given as follows in order to obtain the above-mentioned steering reaction torque.

$$\beta_{fr} \cdot \gamma > 0 \quad \text{Under-steer} \quad G1 > 0, G2 = 0 \quad (7)$$

$$\beta_{fr} = 0 \quad \text{Neutral steer} \quad (8)$$

$$\beta_{fr} \cdot \gamma < 0 \quad \text{Over-steer} \quad G1 < 0, G2 > 0 \quad (9)$$

In addition, the steering rate generally tends to become higher in these areas, so the controlled reaction torque $Tr2$ is provided as a supplement to further increase the vehicle convergence.

$$Tr2 = C1 \cdot d\delta / dt \quad (10)$$

($C1$ is a coefficient.)

Therefore, the EPS control assist torque Tt is provided by subtracting equations (6) and (10) from the assist torque Th that is necessary to reduce the steering torque when stopped. Therefore, labeling the load torque as Tl , the steering torque Ts is obtained as follows.

$$Ts = Tl - Tt = Tl - (Th - Tr1 - Tr2) \quad (11)$$

Equations (6), (7), (10) and (11) show that when the β_{fr} is large during drift-out, $Tr1 > 0$ and the driver's steering torque Ts becomes heavier.

In addition, equations (6), (9), (10) and (11) show that γ and $|\beta_{fr}|$ increase and $Tr1 > 0$ in the over-steer state. At this time the steering torque Ts returns in the Tl direction as it becomes heavier, so the amount by which the self-aligning torque is attenuated by the assist becomes smaller. Furthermore, when that γ and $|\beta_{fr}|$ increase and $Tr1 > Th$, Tt acts in the direction to increase the self-aligning torque, so the driver's counter-steer is assisted.

4. Investigation when Applied to Under-steer

Vehicle drift-out occurring in the under-steer state was investigated. Bicycle model simulations were performed using nonlinear tire characteristics, and Fig. 7 shows the calculation results of the vehicle lateral distance when the amplitude θ (sine wave) is varied with the frequency of the input steering angle θ as the parameter while running at a constant vehicle speed (40 km/h) under low road surface friction coefficient conditions (friction coefficient $\mu = 0.2$).

In Fig. 7, when the input frequency is changed from 0.2 to 0.25 Hz, the lateral distance decreases in the area where β_{fr} exceeds 6 degrees, that is to say the area where the steering angle θ exceeds 130 degrees. Therefore, the driver performs operations to prevent β_{fr} from increasing. This area is the drift-out area, and operations such as the following are necessary as methods to prevent β_{fr} from increasing.

- (1) Preventing the steering angle θ from increasing
- (2) Preventing the acceleration force and braking force from increasing

In this study, the former method (1) was investigated.

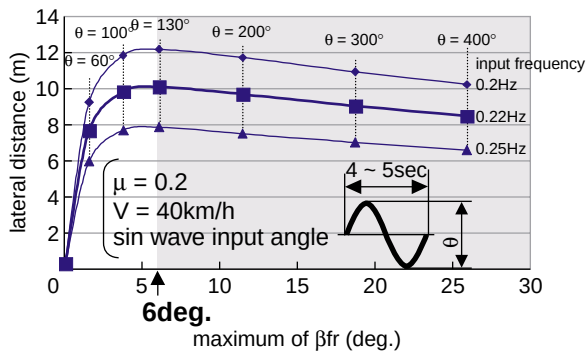


Fig. 7 Lateral distance in drift out area

4.1 Investigation of Controlled Reaction Torque Tr_1 and Tr_2 Using a Simple Simulator

As previously mentioned, investigations were conducted as to whether the driver could be prevented from increasing the steering angle by making the steering torque T_s heavier when β_{fr} increases in order to prevent the vehicle from entering the drift-out area. The effects of steering torque control for tasks similar to low road surface friction coefficient conditions were estimated using the simulator⁽⁶⁾ shown in Fig. 8 to investigate the minimum required controlled reaction torque Tr_1 and Tr_2 .

This simulator is comprised of two horizontal rows of LED indicators (LED1, LED2) that indicate the target vehicle position used as the operation target and the current position of the driver's own vehicle, and a reaction component consisting of a steering wheel coupled to an EPS gear box that applies the actual reaction force. The actual vehicle position display (LED2) reflected the primary lag

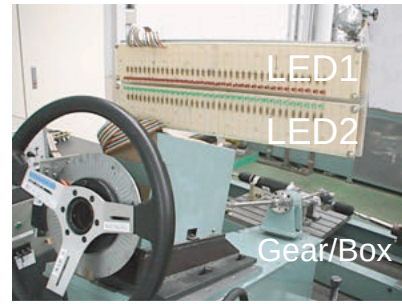
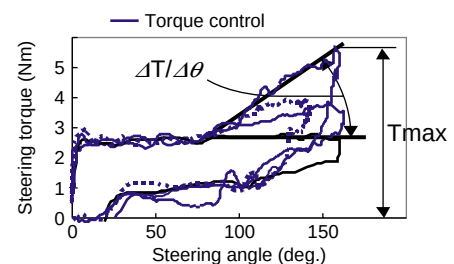


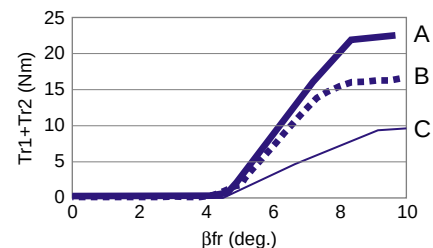
Fig. 8 Driving simulator

of the vehicle, and was set to move horizontally according to the steering operations of the test subject.

The test subjects were two normal drivers and one expert driver as the target. The controlled reaction torque ($Tr_1 + Tr_2$) was varied according to β_{fr} as shown in Fig. 9 (b), and multiple controlled reaction torque gradients ($\Delta T/\Delta \theta$) were applied as shown in Fig. 9 (a). The maximum steering torque and maximum steering rate of the driver was surveyed at this time. This is because the steering rate generally tends to become faster under low road surface friction coefficient conditions, so it was thought that reducing the maximum steering rate would allow driving without increasing the steering angle. Fig. 10 shows the effects on the controlled reaction torque ($Tr_1 + Tr_2$). These results showed that at a controlled reaction torque gradient ($\Delta T/\Delta \theta$) of 0.25 Nm/deg. and a maximum steering torque (T_{max}) of 8.3 Nm, the maximum steering rate could be reduced to the same level as an expert driver.



(a) Characteristics of torque



(b) (Tr_1+Tr_2) curves

Fig. 9 Steering torque on simulation

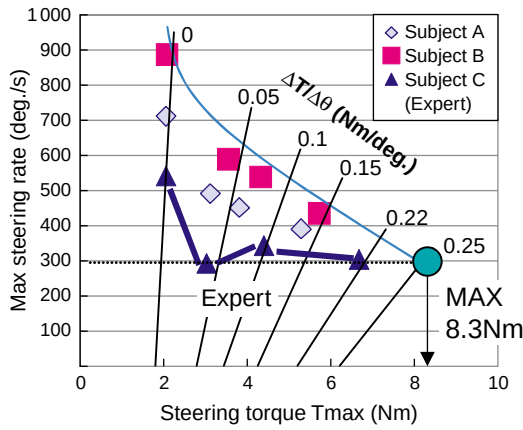


Fig. 10 Test result

4.2 Confirmation of Effects by Actual Running Tests

A test course was set as shown in Fig. 11 so that running was performed using the drift-out area under low road surface friction coefficient conditions in order to confirm the effects of steering torque control in the under-steer state.

There were six test subjects including one woman, and the test results with and without steering torque control were compared. The controlled reaction torque ($Tr1 + Tr2$) used the value obtained using the above-mentioned simulator.

Fig. 12 shows the test results. In the vehicle with steering torque control, five of the six subjects (83%) successfully navigated the course. In contrast, only one of the six subjects (17%) successfully navigated the course in the vehicle without steering torque control.

At this point, the steering operations of subject *a* who was unable to successfully navigate the course in the

vehicle with steering torque control were investigated by comparing with subject *b* who successfully navigated the course. Fig. 13 shows this comparison.

These results show that subject *a* who was unable to successfully navigate the course, continued the steering operation even when the steering torque increased due to steering torque control, and the torque produced by steering torque control was insufficient to suppress the steering operation. As shown in Fig. 13, subject *a* input a large steering angle ($\theta = 500$ degrees, $\delta = 33$ degrees) in order to avoid the pylon at the 40 m mark in the drift-out area,

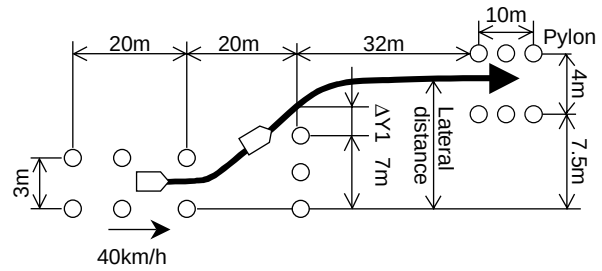


Fig. 11 Test course

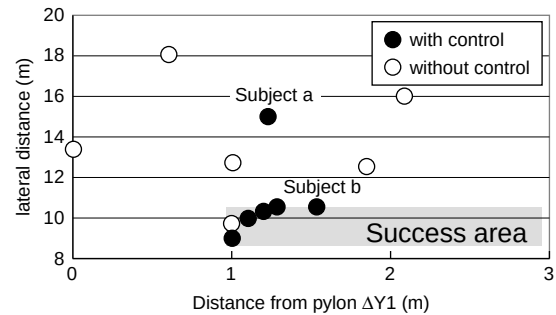
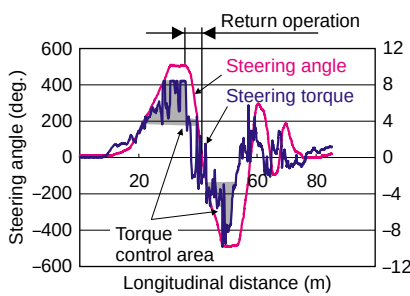
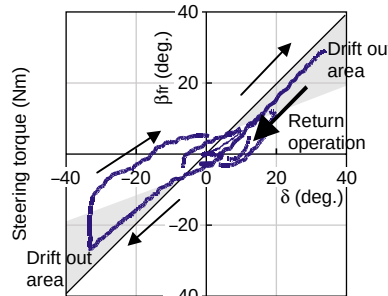


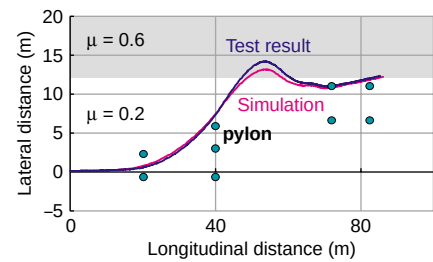
Fig. 12 Effect of steering torque control



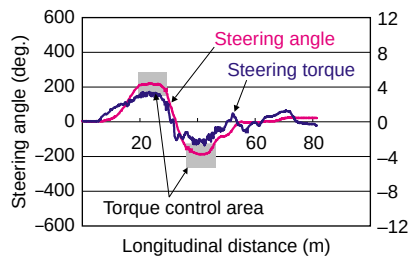
Steering operation (subject a)



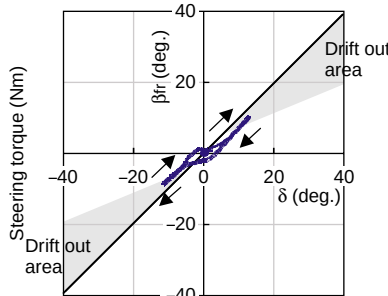
Driving diagram (subject a)



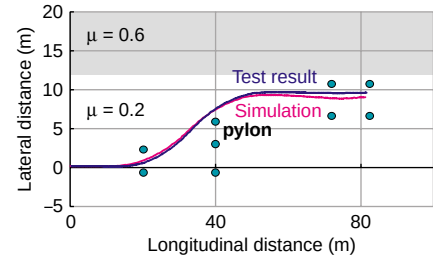
Locus of vehicle (subject a)



Steering operation (subject b)



Driving diagram (subject b)



Locus of vehicle (subject b)

Fig. 13 Test results

and as a result the steering return operation was delayed and the vehicle deviated from the course. On the other hand, subject *b* began to steer at approximately the same timing as subject *a*, but performed steering operations as shown in the driving diagram to prevent δ from increasing in the drift-out area, and thus successfully navigated the course. This confirmed the validity of steering control torque in the under-steer area. In addition, Fig. 13 shows both the results obtained using the above-mentioned bicycle model simulation and the actual running test results.

5. Investigation when Applied to Over-steer

The amount of control required to recover from spinning that occurs in the over-steer state was investigated.

5.1 Investigation of Controlled Reaction Torque Tr1 and Tr2

It has been reported that there is a relationship between the required steering rate and the reaction time lag of the driver after the vehicle γ increases until the start of steering operations when recovering from spinning.⁽⁷⁾

Therefore, actual running tests were conducted to investigate the degree of the actual driver steering lag in the over-steer area. The tests were conducted with four subjects as shown Table 1. The rear road wheels of the vehicle were locked during lane change under the low road surface friction coefficient conditions shown in Fig. 14 causing the over-steer state, and then the driver steering lag time, vehicle behavior and other factors were measured at that time.

The driver steering lag time (reaction time) was defined as shown in Fig. 15 as the response time from the point

Table 1 Subjects list

Classification		Numbers
age	sex	
20's ~ 30's	male	2
40's ~ 50's	male	2

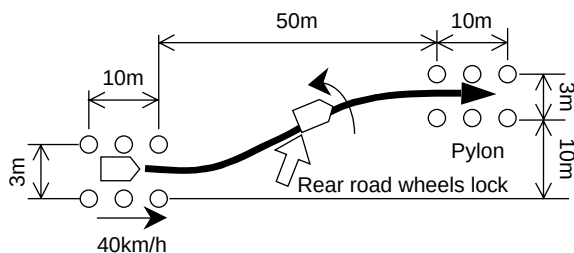


Fig. 14 Test course

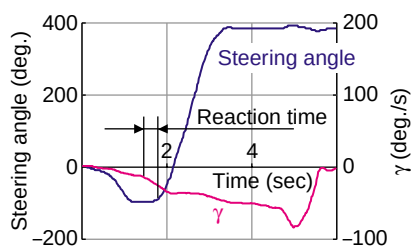


Fig. 15 Reaction time

when the rear road wheels lock and γ starts to increase until counter-steer starts.

Fig. 16 shows the test results. Here, the vehicles driven by three of the four subjects spun. The driving diagrams in Fig. 16 show that subject D was unable to prevent γ from rising with respect to δ , and that while subjects A and C were able to prevent γ from rising, the β_{fr} was too large so the spinning could not be stopped. Therefore, although there is some variance depending on the test conditions, vehicle conditions and other characteristics, a large steering rate must be input to avoid entering the spinning area.

Fig. 17 shows the steering lag time and steering rate obtained by the actual running tests in Fig. 16 plotted over the line L obtained by investigating spin recovery performance when spinning is produced by locking the rear road wheels in the bicycle model simulation. Here, the subjects indicated with an $\times$ could not avoid spinning, and the subject indicated with an O could avoid spinning. This shows that spinning occurs when steering input equal to or greater than the steering rate indicated by line L cannot be applied.

Here, it is thought that spin recovery can be improved if assistance can be provided to add controlled reaction torque so that the reaction time is shorter and the steering rate increases. As shown in Fig. 17, subjects A, C and D were able to input the steering rate indicated by line Lc or greater. Therefore, G1, G2 and C1 were set so that a steering rate of approximately 200 deg./s can be added by the above-mentioned controlled reaction torque (Tr1 + Tr2) to enable spin recovery.

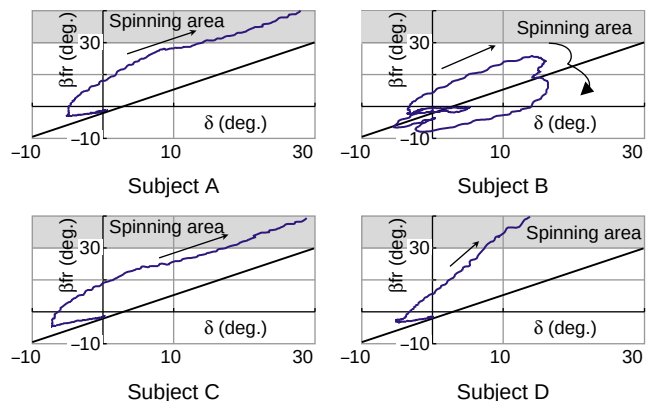


Fig. 16 Driving diagrams

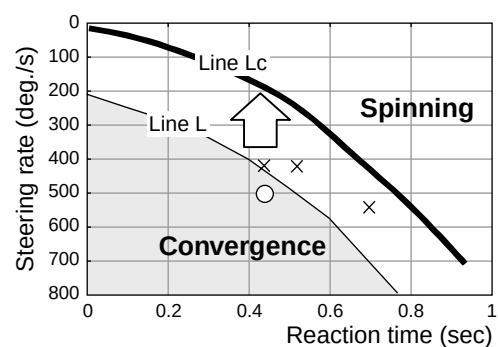


Fig. 17 Characteristics of steering wheel input to stop spinning

5.2 Confirmation of Effects by Actual Running Tests

The effects of steering torque control were confirmed on the test course shown in Fig. 14.

Four test subjects were selected from the same age group to prevent test experience from affecting the test results. As shown by the time series data in Figs. 18 to 20, controlled reaction torque was generated at the same time as over-steer occurred to assist the driver's steering operations. These results show that when γ increases, it is suppressed by the controlled reaction torque $Tr1$ (γ term), and when γ approaches 0, the control torque $Tr1$ (β_{fr} term) assists in the direction to reduce β_{fr} . Furthermore, Fig. 19 shows that convergence is improved by $Tr2$ which is a viscous term, and Fig. 20 shows the composite control torque amount.

Figs. 21 to 24 show the test results. None of the vehicles driven by the four subjects spun.

These results show that even subject H was able to suppress the rise in β_{fr} from a δ of 12 degrees as shown in the Fig. 21 driving diagram.

In addition, Fig. 22 shows that compared to the results of subjects in vehicles without steering torque control indicated by the white circles, the controlled reaction torque increased the steering rate for vehicles with steering torque control indicated by the black circles. Furthermore, the steering lag time was also shortened as shown in Fig. 23.

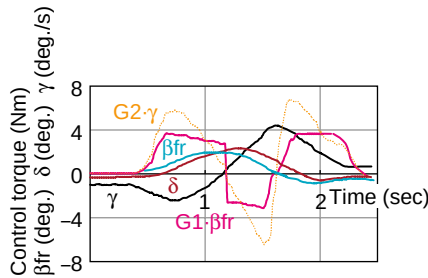


Fig. 18 Main control time chart

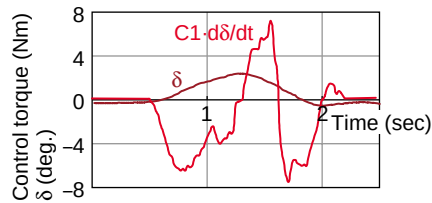


Fig. 19 Sub control time chart

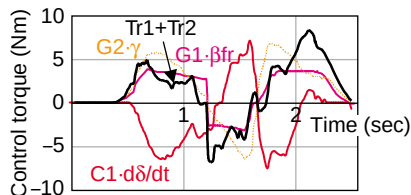
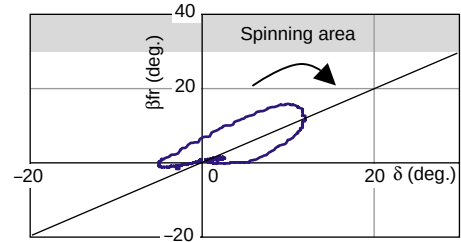
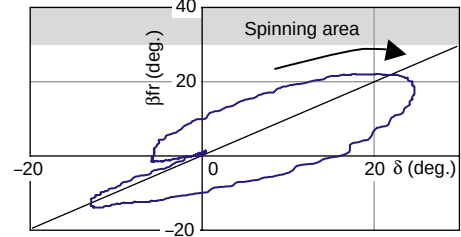


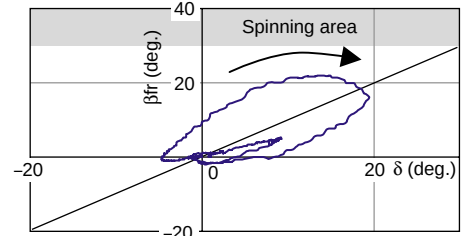
Fig. 20 (Tr1+Tr2) time chart



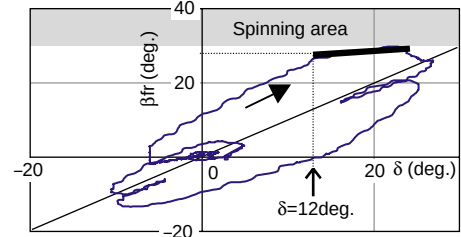
Subject E



Subject F



Subject G



Subject H

Fig. 21 Driving diagrams

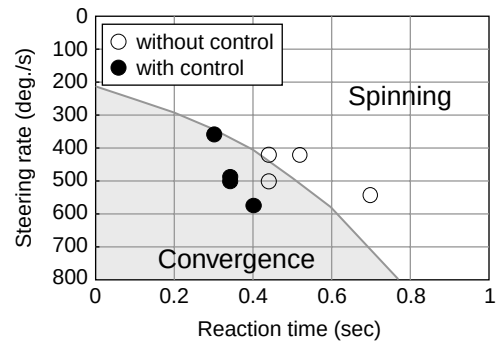


Fig. 22 Effect of steering torque control

Fig. 24 also shows that the yaw rate peak with respect to the steering lag time also decreased.

These results confirmed the developed method effectively improved recovery performance when spinning occurred.

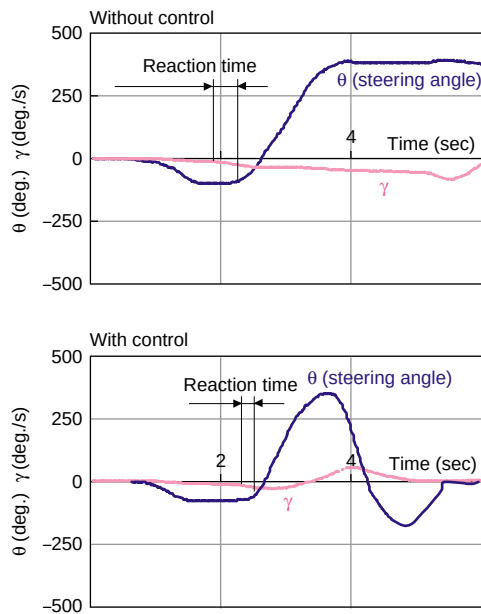


Fig. 23 Spinning recovery time chart

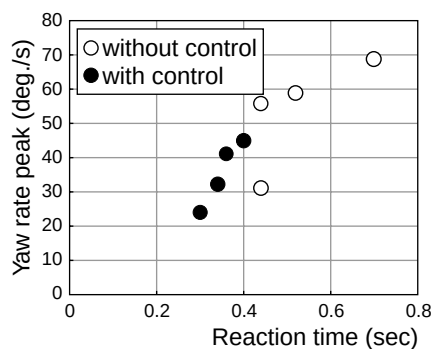


Fig. 24 Reaction time - yaw rate peak

6. Conclusion

This study proposed a driving diagram as a method for understanding the under-steer and over-steer states using the actual steering angle of the front road wheel and the difference between the front tire slip angle and the rear tire slip angle, and devises a steering torque control method based on this diagram. Tests were then conducted under low road surface friction coefficient conditions using a vehicle equipped with an Electric Power Steering (EPS) system that uses this steering torque control method, and the results confirmed the following.

- (1) The under-steer and over-steer areas which are driving states when the driver is operating the vehicle can be described by a driving diagram using $\delta - \beta_{fr}$.
- (2) Based on this driving diagram, a new steering torque control was proposed which provides assistance to operate δ in the direction that β_{fr} and γ approach zero.
- (3) A method was established for detecting under-steer, over-steer and other driving states using the driving diagram.

- (4) The following items were understood as a result of applying this method to the under-steer area.

- 1) Drift-out tests were conducted using a vehicle with the controlled reaction torque gradient set to 0.25 Nm/deg. and the maximum torque set to 8.3 Nm based on the above-mentioned steering torque control. As a result, five of the six test subjects (83%) improved their test results.

- 2) The one test subject whose test results did not improve was investigated, and the control torque set by this study was found to be insufficient.

- (5) Spin recovery tests were conducted using a vehicle with the control torque set to allow addition of a steering rate of approximately 200 deg./s based on the above-mentioned steering torque control applied to the over-steer area. In this case as well, the control was confirmed to improve the test results.

These observations confirmed both the validity of the driving diagram and the effectiveness of steering torque control based on this driving diagram.

References

- (1) M. Yamamoto, Active Control Strategies for Improved Handling and Stability, Automotive Technology, Vol. 45, No. 3, p. 43 – 51 (1991) (in Japanese)
- (2) N. Yuhara, S. Horiuchi, N. Asanuma, Y. Nishi, T. Kohata: Improvement of Vehicle Handling Quality Through Active Control of Steering Reaction Torque, Proc. of International Symposium AVEC, Yokohama, Japan, p. 407 – 412 (1992)
- (3) Y. Amano, S. Doi, H. Kawaguchi: Improvement of active safety by active front steering angle and torque control, JSAE Proceedings, Vol. 30, No. 3, p. 87 – 92 (1998)
- (4) M. Abe: Vehicle Dynamics and Control, Tokyo, Sankaido, p. 249 (1992) (in Japanese)
- (5) S. Yamawaki, Y. Shimizu: A Study on Steering Torque Control Using driving Diagram at the Oversteer, Japan Mechanical Association, Traffic Distribution Division Conference, No. 99 – 59, p. 144 – 158 (1999) (in Japanese)
- (6) A. Yoneda, Y. Shimizu: A Study on Control Performance of the Quick Steering Vehicle, Japan Mechanical Association, Traffic Distribution Division Conference, No. 98 – 37, p. 239 – 242 (1998) (in Japanese)
- (7) Y. Shimizu, T. Kawai, J. Yuzuriha, S. Takimoto: A Study of a New Steering System with Variable Gear-ratio According to Vehicle Speed and Steering Wheel Angle. HONDA R&D Technical Review, Vol. 11, No. 1, p. 107 – 118 (1999) (in Japanese)

■ Authors ■



Shigeru YAMAWAKI



Yasuo SHIMIZU