

# Effect of Three Controls (Camber Angle Control, Derivative Steering Assistance Control, and Inside-Outside Wheel Braking Force and Driving Force Control in Body Slip Angle Area

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## ABSTRACT

In this research, we examine the three controls inside-outside wheel braking force and driving force, camber angle, and the derivative steering assistance to determine how angle differences affect cornering performance and controllability. This is accomplished by comparing body slip angle area differences in a closed loop examination of the grip to drift area using a driving simulator. The results show that inside-outside wheel braking force and driving force control in the area just before critical cornering occurs has a significant effect on vehicle stability. We also clarified that controlling the camber angle enhances grip-cornering force, and confirmed that the sideslip limit could be improved in the vicinity of the critical cornering area. Additionally, when the counter steer response was improved by the use of derivative steering assistance control in the drift area exceeding the critical cornering limit, corrective steering became easier. Moreover, the effect could be achieved by using camber angle and derivative steering assistance controls in combination over a wide area. Based on the above, we conclude that it is possible to control wide-ranging body slip angle areas by combining the three abovementioned controls.

**CITATION:** Yamaguchi, R. and Nozaki, H., "Effect of Three Controls (Camber Angle Control, Derivative Steering Assistance Control, and Inside-Outside Wheel Braking Force and Driving Force Control in Body Slip Angle Area," *SAE Int. J. Passeng. Cars - Mech. Syst.* 9(1):2016, doi:10.4271/2016-01-1666.

## INTRODUCTION

Rear-wheel active steering has been reported to have a significant effect on emergency evasion performance [1]. On the other hand, a new steering system incorporating a proportional derivative based on the action of the front wheels of a vehicle has recently been proposed by Hirao [2, 3]. However, while this latter technique is effective in the lower lateral acceleration range (where the cornering force of the front and rear wheels is stable), front- and rear-wheel cornering force declines in the higher lateral acceleration range, and the desired effects cannot be obtained. Thus, it can be said that technologies for improving maneuvering stability when driving at the limits of controllability in the body slip angle area that extends from the grip driving area to the drift driving area have not yet been fully researched. Furthermore, with the spread of electric vehicles (EVs) in recent years, in-wheel motor designs that allow all four wheels to be independently powered and controlled have been produced. Such designs can offer improved steering and increased driving stability, thereby making it possible to exert more precise control of braking and driving force to all four wheels. It should also be noted that the addition of camber angle control, which was difficult to achieve in previous vehicles, has been made possible by simplifications to the mechanisms around the suspension and increases to the available degree of freedom within the layout. For further information on the camber angle control mechanism, refer to our previous paper [4]. Moreover, because counter steering delay can be controlled by

derivative steering assistance, very effective improvements to drift area technique have become possible. In this research, the cornering performance difference of the inside-outside wheel braking force and driving force control in the body slip angle area, along with camber corner control and a derivative steering assistance system, are explored by using a driving simulator in a wide-ranging area where body slip angle is controlled.

## EXPERIMENTAL APPARATUS

### *Driving Simulator to Simulate Drift Cornering*

Since the use of a driving simulator facilitates experiments involving sudden collision avoidance situations, we used a drift cornering driving simulator (Figure 1) jointly developed by Nozaki and Mitsubishi Heavy Industries, Ltd. This device can simulate the bodily sensations of drift cornering due to its ability to produce (1) large lateral acceleration behaviors and (2) large yawing behaviors. The yawing mechanism can be rotated infinitely and the effects of normal lateral acceleration can be reproduced using the roll function of the simulator. The simulator can reproduce a total maximum lateral acceleration of  $\pm 0.7$  G, which is equivalent to that experienced by an actual driver in a collision avoidance situation, and thus facilitates the examination of advanced steering control technologies and methods for drivers.

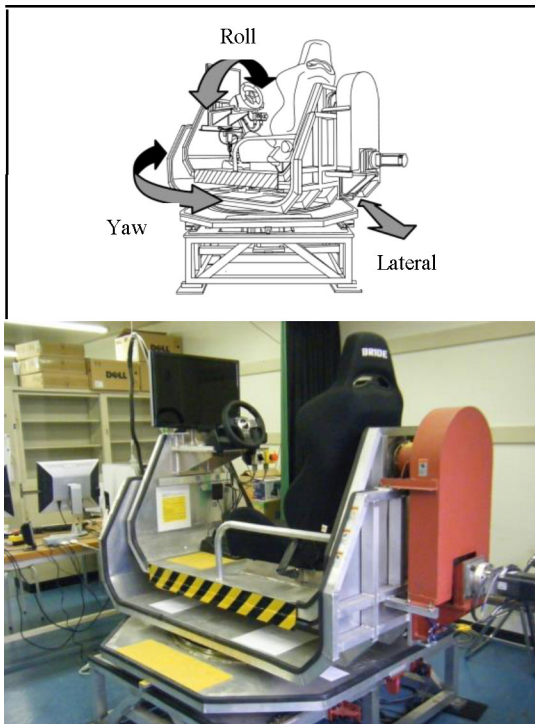


Figure 1. Driving simulator

### Vehicle Model

For the driving simulator vehicle, CarSim, which is a full-vehicle simulator developed by the Mechanical Simulation Corporation (MSC) of the USA, was used. Table 1 lists the main vehicle components and their degrees of freedom, while Table 2 shows the vehicle model parameters used in our experiments. The vehicle model considered herein has a front engine mount and rear drive (FR) layout that easily enters the drift state. For tire cornering force characteristics, we used the general tire characteristics shown in Figure 2. The maximum cornering force occurs at a slip angle of about 10 deg, and the acceleration and deceleration characteristics are as shown in Figure 3. When added simultaneously, slip angle and ratio are calculated using the friction circle concept based on CarSim's combined characteristics calculations. This concept produces simulated driving behaviors that match actual vehicle conditions when limitations are exceeded. Thus, when accelerating during the critical cornering limits, the rear wheels' maximum lateral force decreases as rear-wheel drive force increases and the vehicle, which is now unable to withstand the centrifugal force, begins to spin out of control. However, skillful control of the direction of travel of the front part of the vehicle body through a counter steer maneuver can put the vehicle into a drift-cornering maneuver. The final behavior in this study demonstrates a rear-wheel skid. In this paper, the grip limit is defined as the point where the rear wheels generate the maximum cornering force.

Table 1. Vehicle model components and degrees of freedom

| 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                   | 3  |
| Sprung body rotation (yaw, pitch, roll)   | 3  |
| Suspension stroke                         | 4  |
| Wheel spin                                | 4  |
| Powertrain (engine crankshaft)            | 1  |
| Tire delayed slip (lateral, longitudinal) | 8  |
| Brake fluid pressure                      | 4  |
| Total                                     | 27 |

Table 2. Calculation parameters

|                                                              |        |
|--------------------------------------------------------------|--------|
| Width of vehicle (mm)                                        | 1988   |
| Wheelbase (mm)                                               | 2690   |
| Distance from center of front axle to center of gravity (mm) | 1014   |
| Height from ground to center of gravity (mm)                 | 542    |
| Vehicle mass (kg)                                            | 1527   |
| Roll moment of inertia ( $\text{kg} \cdot \text{m}^2$ )      | 606.1  |
| Pitch moment of inertia ( $\text{kg} \cdot \text{m}^2$ )     | 2741.9 |
| Yaw moment of inertia ( $\text{kg} \cdot \text{m}^2$ )       | 2741.9 |

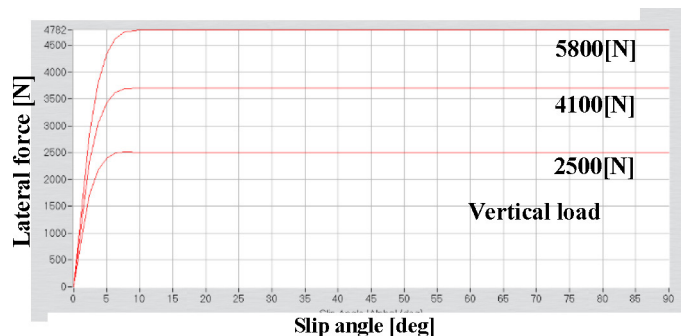


Figure 2. Tire cornering force characteristics

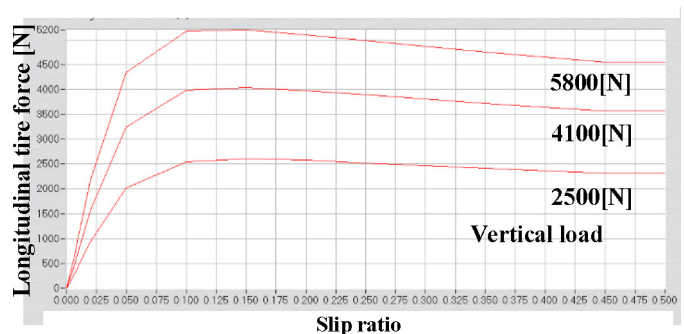


Figure 3. Longitudinal tire force characteristics

## OUTLINE OF EACH CONTROL

### Inside-Outside Wheel Braking Force and Driving Force Control

When a tire with idealized characteristics is examined, lateral force continues to increase as the slip angle increases. Thus, utilizing the difference between inside-outside wheel braking and driving force can control yaw moments. The yaw moment produced by the idealized tire model is calculated by [equation \(1\)](#). The yaw moment of the driving vehicle is requested from the inertia yaw moment  $I_Z$  of the vehicle and the differentiation value of yaw rate  $\dot{\gamma}$ . [Equation \(2\)](#) gives a direct vehicle the control by right and left driving force distributing moment  $M$  of the difference share. A comparison chart for idealized tire and an actual tire models at a vertical load of  $F_z = 4100$  N per wheel is shown in [Figure 4](#).

$$M_e = -F_{ye_{R1}} * L_f - F_{ye_{L1}} * L_f + F_{ye_{R2}} * L_r + F_{ye_{L2}} * L_r \quad (1)$$

$$M = M_e - I_Z \dot{\gamma} \quad (2)$$

( $M_e$ : Yaw moment of idealized tire;  $F_{yei}$ : Lateral force of each idealized tire;  $L_f, L_r$ : Length of front axle (rear axle)-center of gravity)

Driving force generating margins of each tire is calculated the wheel loads and the lateral force applied to each of the tire while driving and the road surface friction  $\mu$  by the [equation \(3\)](#). The difference between the moment  $M$  be allocated by 50% in the right front and rear wheels and the left front and rear wheels. Longitudinal force control amount applied to each tire are determined based on the driving force generation allowance determined in [equation \(3\)](#). To illustrate this relationship equation is as shown in [equation \(4\)](#).

$$X_i = \sqrt{\mu F_{z_i}^2 - F_{y_i}^2} \quad (3)$$

$$\left. \begin{aligned} f_{x_{L1}} &= \frac{X_{L1}}{X_{L1} + X_{L2}} \frac{M}{2 \cdot (tr/2)} \\ f_{x_{L2}} &= \frac{X_{L2}}{X_{L1} + X_{L2}} \frac{M}{2 \cdot (tr/2)} \\ f_{x_{R1}} &= \frac{X_{R1}}{X_{R1} + X_{R2}} \frac{M}{2 \cdot (tr/2)} \\ f_{x_{R2}} &= \frac{X_{R2}}{X_{R1} + X_{R2}} \frac{M}{2 \cdot (tr/2)} \end{aligned} \right\} \quad (4)$$

( $X_i$ : Braking and driving force generating margin of each tire;  $F_{z_i}$ : Wheel load of each tire;  $F_{y_i}$ : Lateral force of each tire;  $f_{x_i}$ : Driving force control amount of the idealized tire)

Idealized tire control consists chiefly of (1) the idealized yaw moment calculation:  $M_e$ ; (2) the difference calculation of the actual yaw the idealized yaw moment:  $M$ ; (3) the room driving force ratio that is calculated in consideration of vertical load movement per wheel

(front and rear, left and right); and (4) the driving force that could actually be demonstrated. This is calculated according to friction circle model and it is controlled by the braking force and driving force control, as can be seen by the flowchart shown in [Figure 5](#).

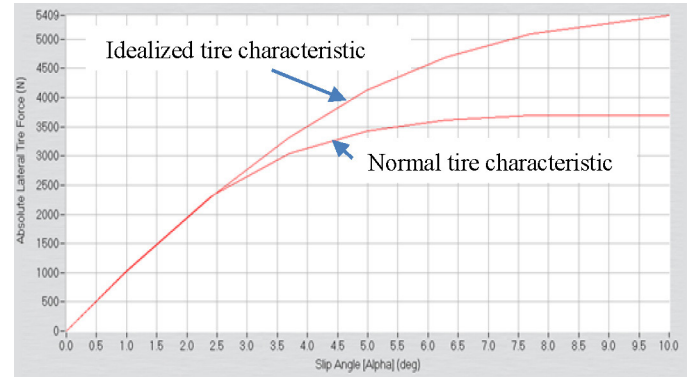


Figure 4. Tire characteristic comparison at a vertical load of 4100 N per wheel

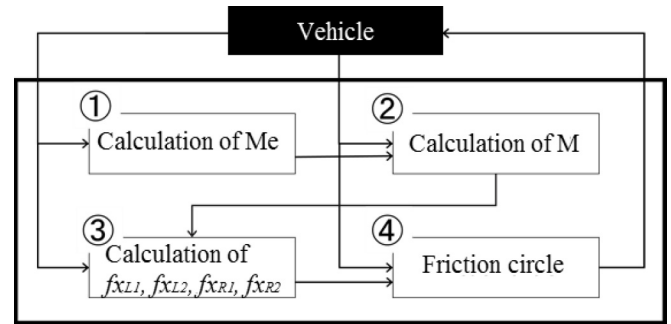


Figure 5. Idealized tire control flowchart

### Camber Angle Control

In camber angle control, the steer angle proportion of the negative camber angle where the tire is inclined in the direction of the turn center is controlled as shown in [Figure 6](#). Critical cornering performance is improved by the effect of the camber thrust on the critical cornering. The side force characteristic (calculation value obtained by the 'magic-formula') in the negative camber angle is shown in [Figure 7](#).

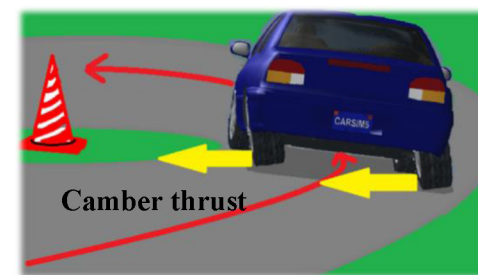


Figure 6. Negative camber angle where the tire is inclined in the turn center direction

In the driving simulation used in the present study, a negative camber angle of approximately 20 deg (corresponding to 1.2 times the cornering force) is added to the rear wheel when the body slip angle value exceeds the maximum cornering force. The camber angle control flowchart is shown in [Figure 8](#).