12.1 PREFACE

A road vehicle is called an automobile for the first time when the vehicle is controlled by a human driver on board. The vehicle dynamics controlled by the human driver are dealt with in Chapter 10, and the relation of handling quality evaluation by the driver to the vehicle handling dynamics is discussed in Chapter 11. However, no further discussion has been extended so far regarding the relation of the handling quality evaluation to the vehicle dynamics controlled by human drivers. This is because there is no fundamental established yet for reasoning the vehicle handling quality from the vehicle handling dynamics.

Though many investigations have been done, of course, handling quality evaluation by human drivers and its relation to the chassis design variables of the vehicles and the chassis control parameters have still been under ongoing debates, and a consistent, general way of understanding or explaining them has not been established yet. On the other hand, because it has recently been regarded that active vehicle motion controls are promising tools to improve vehicle handling and stability, it becomes more important than ever to have a general way of estimating or predicting the vehicle handling quality, which is possible to be controlled by the motion controls, evaluated by human drivers subjectively.

As shown in Chapter 11, there are many studies on the correlations between the subjective evaluations and objective measurements of the vehicle response or vehicle response parameters. However, the results are not necessarily generalized and directly applicable to the chassis design process. A primary reason for that seems to be the driver’s subjective evaluation based on feeling that can’t avoid psychological and mental effects, especially for ordinary drivers. The results are qualitatively reliable; yet, they are not necessarily quantitatively reliable and, thus, do not result in a generalized method.

In this chapter, some trials of applying a driver model and its parameter identification to dealing with the driver-vehicle system behavior will be introduced to generally and consistently understand what, how, and why the vehicle handling dynamics reflect the handling quality of the vehicle. This must be possible to be an effective and general method of realizing and estimating a relation of the handling quality evaluation with the characteristics of vehicle handling dynamics.

12.2 DRIVER MODEL AND PARAMETER IDENTIFICATION [1]

A driver is considered to determine the steering parameters not only depending on his/her inherent characteristics but also on adapting to the vehicle handling characteristics, as is discussed in Section 10.4, by changing the parameters during vehicle motion to keep the system...
response almost always the same, even with wide variations of the vehicle handling characteristics. Therefore, the vehicle performance controlled by the driver is unchanged even when the vehicle handling characteristics change. This is one of the primary reasons why the vehicle response itself does not always directly reflect the handling characteristics of the vehicle, and it is difficult to understand the handling quality from the objective measurement of the vehicle responses.

On the other hand, because a driver adapts his/her characteristics to the vehicle handling characteristics, the parameters in a driver model must directly reflect not only his/her inherent characteristics but also the vehicle handling characteristics. So, the idea arises that if the parameters are reasonably identified, it must be possible to estimate the evaluation of the vehicle handling characteristics through the driver parameters identified. This section intends to show the driver model for this specific purpose and the method of how to identify the driver parameters by using experimental data.

12.2.1 DRIVER MODEL

The driver behavior during his/her motion control of the vehicle is dealt with in Chapter 10. As the transfer function of the driver steering behavior is described by Eqn (10.2) and the course error detected by the driver is expressed by (10.3), the steering angle of the driver is described as follows:

\[ \delta(s) = \frac{-h(1 + \tau_D s)}{1 + sLs} \begin{bmatrix} y(s) + L\theta(s) - y_{OL}(s) \end{bmatrix} \]  

(12.1)

To simplify the model and reduce the number of parameters in the driver model, it is assumed that \( \frac{dy}{dt} \approx V\theta \), as shown in Section 10.3.3, and the driver performs the equivalent derivative control action by looking ahead—a preview behavior. As a result, the driver derivative time, \( \tau_D \), is regarded to be almost zero, and Eqn (12.1) can be rewritten as follows:

\[ \delta(s) = -he^{-\tau_s} \begin{bmatrix} \left(1 + \frac{L}{V}s\right)y(s) - y_{OL}(s) \end{bmatrix} \]

Moreover, assuming that \( \tau_L \) is small and \( e^{-\tau_L s} \approx 1/(1 + \tau_L s) \), in which \( \tau_L \) is regarded to represent all the response delay of the driver, the following simplified driver model is used that describes the steering angle determined by the driver:

\[ \delta(s) = \frac{-h}{1 + \tau_L s} \begin{bmatrix} \left(1 + \tau_L s\right)y(s) - y_{OL}(s) \end{bmatrix} \]  

(12.2)

Assuming that \( \tau_L \) represents the effects of the look ahead (preview) behavior and includes the effect of the derivative control action of the driver, if any derivative control action is negligible, this is regarded to be the preview time, \( L/V \). The block diagram of the simplified driver-vehicle model is shown in Figure 12.1, and the driver steering characteristics can be represented by the three parameters \( h, \tau_L, \) and \( \tau_L \). Here, it is important to note that this model is applicable only to the driver making a sudden lane change on a straight road with constant lane width, \( y_{OL} \).

12.2.2 PARAMETER IDENTIFICATION

Next is to study the experimental identification of the three parameters. The handling parameters in the driver model are identified here by using experimentally logged data of driver steering angle and vehicle trajectory. As a typical behavior of the driver-vehicle system can be observed
during a lane change on a straight road, the parameters can be identified from an experimentally measured time history of the driver’s steering angle, $\delta^*$, and lateral displacement, $y^*$, during the lane change on a straight road.

The error is defined between the measured steering angle, $d^*$, and the driver’s steering angle response to the lateral displacement, $y^*$, and the lane change width, $y_{OL}$, calculated by the driver model, Eqn (12.2):

$$e(s) = \left(1 + \tau_LS\right)\left[\delta^*(s) + \frac{h}{1 + \tau_LS}\left\{\left(1 + \tau_hs\right)y^*(s) - y_{OL}(s)\right\}\right]$$

(12.3)

The square integral of the weighted sum of the error and error rate is defined as the evaluation function:

$$J = \int_0^T e^2dt = \int_0^T \left[\delta^* + \tau_L\frac{d\delta^*}{dt} + h\left\{y^* + \tau_h\frac{dy^*}{dt} - y_{OL}\right\}\right]^2 dt$$

(12.4)

where $T$ is the time period long enough for the driver to finish the lane change. It is possible for us to find the parameters, $h$, $\tau_h$, and $\tau_L$ that minimize $J$ by solving the following equations:

$$\frac{\partial J}{\partial h} = 0, \quad \frac{\partial J}{\partial \tau_L} = 0, \quad \frac{\partial J}{\partial (h\tau_h)} = 0$$

(12.5)

These equations are first-order linear algebraic equations of $h$, $\tau_L$, and $h\tau_h$, and they can easily be solved for $h$, $\tau_L$ and $\tau_h$. The solved parameters are the identified driver parameters. With these, the driver steering angle can be described by the identified model and is as close as possible to the real driver steering angle measured.

12.3 DRIVER PARAMETERS REFLECTING DRIVER CHARACTERISTICS [1]

It seems that one of the typical examples of the dependence of driver handling behavior upon his/her inherent characteristics is the dependence on the age of the driver. For confirming this, there is a result of the parameter identification of young and aged drivers during a lane change with a small-size personal vehicle.

Figure 12.2 is a typical result of the driver steering behavior during the lane change, in which the time history of the measured steering angle is compared with that calculated by the driver
model with the identified parameters for the young and the aged drivers, respectively. They are in very good agreement with each other, showing that it is reasonable to describe the driver behavior by the simplified driver model introduced in Section 12.2.1 with the parameters identified by the method shown in Section 12.2.2.

Figure 12.3 shows how the driver parameters identified are distributed on a parameter plane depending on the age. The parameters of the aged drivers are distributed on relatively larger $\tau_h$, larger $\tau_L$, and smaller $h$ on the parameter planes compared with those of the young drivers.

It is understood from Figure 12.3 that as smaller $\tau_L$ causes higher tension and more severe workload, especially on aged drivers, he/she adopts relatively larger $\tau_L$ and compensates for the delay of the driver response and stabilizes a less stable vehicle motion due to larger $\tau_L$ by adopting larger $\tau_h$. This is equivalent to a preview behavior of the driver and is easy for drivers to change by changing the look ahead distance, $L$, with almost no workload for the driver to make a larger value of required $\tau_h$, though its value would be limited by the lane change distance. Figure 12.3 also shows that the aged drivers adopt smaller gain, $h$, compared with the young drivers. This is supposed to be influenced by adopting $\tau_h$ and $\tau_L$ as described previously and is contributory to stabilizing the vehicle motion.

It is concluded from the preceding that it is possible to characterize the driver with the driver parameters identified in Eqn (12.2).

12.4 DRIVER PARAMETERS REFLECTING VEHICLE HANDLING CHARACTERISTICS

We have already understood in Chapter 3 that vehicle speed has significant effects on vehicle handling characteristics, and the tire cornering characteristics also have predominant effects as well. In this section, the effects of the vehicle speed and tire cornering stiffness on the identified
12.4 DRIVER PARAMETERS REFLECTING VEHICLE HANDLING

12.4.1 DRIVER PARAMETERS TO VEHICLE SPEED [2]

As the vehicle speed has a significant effect on vehicle dynamics, the driver is assumed to adapt his (or her) characteristics to the change of vehicle speed, which is discussed in Section 10.4. To confirm this, the way that vehicle speed influences the identified driver parameters will be looked at in this section.

There is a result of applying the identification method in Section 12.2.2 to the experimental data obtained in the lane change test on a proving ground. The lane change test course is shown in Figure 12.4. The lane change width, $d_C$, is 3.0 m, and the lane change lengths, $L_C$, are 15, 22.5, and 30 m for vehicle speeds 40, 60, and 80 km/h, respectively.

In Figure 12.5, the calculated steering angle by the model, Eqn (12.2), is compared with the real driver steering angle measured. They agree well with each other, which means that the driver steering behavior is adequately described by Eqn (12.2).

The parameters identified are shown in Figure 12.6 and illustrate how the driver parameters change according to the increase in vehicle speed. It is understood that the proportional gain, $h$, significantly decreases with the increase of vehicle speed. In contrast, $\tau_h$ is always around...
0.8 ~ 1.0 s and does not change with increasing vehicle speed. So, if $\tau_h$ is approximated as the preview time, $L/V$, the preview distance (looking ahead distance), is almost proportional to the vehicle speed and equal to $0.8 \text{ V} ~ 1.0 \text{ V}$. This view corresponds with the driver adaptation to vehicle characteristics discussed in Section 10.4. The response delay, $\tau_L$, decreases with the vehicle speed, which suggests that the increasing vehicle speed pushes the driver to achieve more stressful workloads on the steering control.
The vehicle handling characteristics strongly depend on tire cornering stiffness. We have already seen in Chapter 3 that the vehicle response parameters such as stability factor, yaw rate gain, natural frequency, damping ratio, yaw rate lead time, and response time are described as follows, and the considerable dependence of them to the tire is obvious:

\[ A = -\frac{m}{2l^2} \frac{l_f K_f - l_r K_r}{K_f K_r} \]

\[ G(0) = \frac{1}{1 + AV^2} \frac{V}{l} \]

\[ \omega_n = \frac{2l}{V} \sqrt{\frac{K_f K_r}{mI}} \sqrt{1 + AV^2} \]

\[ \zeta = \frac{m \left( l_f^2 K_f + l_r^2 K_r \right) + I(K_f + K_r)}{2l \sqrt{m K_f K_r (1 + AV^2)}} \]

FIGURE 12.6
Change of driver parameters with increase of vehicle speed.

12.4.2 DRIVER PARAMETERS TO TIRE CHARACTERISTICS [3]

The vehicle handling characteristics strongly depend on tire cornering stiffness. We have already seen in Chapter 3 that the vehicle response parameters such as stability factor, yaw rate gain, natural frequency, damping ratio, yaw rate lead time, and response time are described as follows, and the considerable dependence of them to the tire is obvious:
To experimentally confirm how the change of vehicle handling characteristics with changing the tire characteristics influences the driver parameters identified, the four different handling characteristic vehicles are provided by changing the front and rear tire cornering stiffness, as shown in Table 12.1. The driver steering behavior and vehicle motions are measured in the lane change tests on the same lane change course as shown in Figure 12.4 with the vehicle speed of 80 km/h. Some results are shown in Figure 12.7. As the drivers adapt to the change of vehicle response characteristics corresponding with the tire characteristics supposedly by changing their driver parameters, the driver-vehicle system behaviors are almost similar.

The driver parameters are identified using the measured date during the lane changes with four vehicles of different handling characteristics, respectively. Figure 12.8 shows a good agreement of the driver steering time history calculated by Eqn (12.2) using the parameters identified with the measured time history. This supports that it is reasonable to describe the driver behavior by Eqn (12.2).

Though the preview time, $\tau_{h}$, of the drivers is almost unchanged from that observed in Section 12.4.1, we can see, in Figure 12.9, a distribution of the driver parameters, $h$ and $\tau_{L}$, depending on the respective vehicle handling characteristics controlled by the tire cornering stiffness. It is easy to see how the drivers vary their parameters according to the change of the vehicle handling characteristics, and the drivers are forced to change the time constant of the response delay, $\tau_{L}$, widely with the variations of the handling characteristics.

As decreasing $\tau_{L}$ in the steering tasks makes the driver experience a more severe workload and if a larger $\tau_{L}$ is allowable for the driver to control the vehicle, the more relaxed the driver behaves during his (or her) maneuvering. In addition, changing $h$ and $\tau_{h}$, more or less, is easier for the driver, and the handling quality evaluation is assumed to correspond predominantly with $\tau_{L}$. Figure 12.10 shows the correlation of the subjective handling quality evaluation by the drivers in the lane change with the identified response delay of the driver, $\tau_{L}$. It seems reasonable that if a driver can behave with larger $\tau_{L}$, then the driver feels it is easier to control the vehicle, which means a higher handling quality evaluation.

### Table 12.1 Four cases of vehicle response characteristics at vehicle speed $V = 80$ km/h

<table>
<thead>
<tr>
<th>Vehicles</th>
<th>$K_f$ (kN/rad)</th>
<th>$K_r$ (kN/rad)</th>
<th>$A$</th>
<th>$G_\delta^f$ (0) (1/s)</th>
<th>$\omega_n$ (rad/s)</th>
<th>$\zeta$</th>
<th>$\tau_R$ (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline</td>
<td>66.1</td>
<td>81.9</td>
<td>0.00109</td>
<td>5.58</td>
<td>10.63</td>
<td>0.835</td>
<td>0.113</td>
</tr>
<tr>
<td>A</td>
<td>40.7</td>
<td>47.5</td>
<td>0.00162</td>
<td>4.78</td>
<td>6.86</td>
<td>0.767</td>
<td>0.190</td>
</tr>
<tr>
<td>B</td>
<td>40.7</td>
<td>81.9</td>
<td>0.00275</td>
<td>3.65</td>
<td>10.32</td>
<td>0.737</td>
<td>0.131</td>
</tr>
<tr>
<td>C</td>
<td>66.1</td>
<td>47.5</td>
<td>−0.000037</td>
<td>8.76</td>
<td>6.46</td>
<td>1.009</td>
<td>0.177</td>
</tr>
</tbody>
</table>
12.4 DRIVER PARAMETERS REFLECTING VEHICLE HANDLING

**FIGURE 12.7**
Driver steering behavior and vehicle motion during lane change.

**FIGURE 12.8**
Comparison of measured and calculated driver steering behavior.
FIGURE 12.9
Relation of driver parameters to vehicle response parameters.
12.5 HANDLING QUALITY EVALUATION BASED ON DRIVER PARAMETER

As the handling quality evaluation seems strongly related to one of the identified driver parameters, $\tau_L$, we will look at several examples of the handling quality evaluation by the parameter $\tau_L$ in this section.

12.5.1 HANDLING QUALITY TO NATURAL FREQUENCY AND DAMPING RATIO [4]

The natural frequency, $\omega_n$, and the damping ratio, $\zeta$, in the denominator of the transfer functions of side-slip and yaw rate responses to steering input described by Eqns (3.77)' and (3.78)' are important response parameters to dominate over the vehicle handling characteristics. They are described in Section 12.4.2, and it is difficult to independently change their values by changing specific vehicle parameters and chassis parameters such as tires and suspension. Therefore, a problem lies in experimentally examining the independent effects of $\omega_n$ and $\zeta$ on the handling quality evaluation by the driver.

A feed-forward type of front-wheel active steering control can equivalently solve this problem. From Eqns (3.77)' and (3.78)', the response of the vehicle with the active front-wheel steering control to steering wheel input is described as follows:

\[
\beta(s) = G_\beta'\left(0\right) \frac{1 + T_\beta s}{1 + \frac{2\zeta s}{\omega_n} + \frac{s^2}{\omega_n^2}} \frac{\delta}{\delta h(s)} \quad (12.6)
\]

\[
r(s) = G_r\left(0\right) \frac{1 + T_r s}{1 + \frac{2\zeta s}{\omega_n} + \frac{s^2}{\omega_n^2}} \frac{\delta}{\delta h(s)} \quad (12.7)
\]

where $\delta/\delta h(s)$ represents a transfer function of front-wheel steering angle to steering wheel angle input for the active steering control.

If the active control transfer function is set as follows:

\[
\frac{\delta}{\delta h(s)} = \frac{1 + \frac{2\zeta s}{\omega_n} + \frac{s^2}{\omega_n^2}}{1 + \frac{2\zeta s}{\omega_n} + \frac{s^2}{\omega_n^2}} \quad (12.8)
\]
then putting Eqn (12.8) into Eqns (12.6) and (12.7) brings us the response of the active steering vehicle to steering wheel angle input as follows:

\[
\frac{\beta(s)}{\delta_h(s)} = G_\beta(0) \frac{1 + T_{\beta}s}{1 + \frac{2s^2}{\omega_n^2} + \frac{s^2}{\omega_z^2}} \tag{12.9}
\]

\[
\frac{r(s)}{\delta(s)} = G_r(0) \frac{1 + T_r s}{1 + \frac{2s^2}{\omega_n^2} + \frac{s^2}{\omega_z^2}} \tag{12.10}
\]

where

\[
\omega_n = \alpha_N \omega_n, \quad \zeta = \alpha_D \zeta \tag{12.11}
\]

Here, \(\alpha_N\) and \(\alpha_D\) are the adjustment parameters, and the natural frequency, \(\omega_n\), and the damping ratio, \(\zeta\), are changed independently by changing the adjustment parameters appropriately around 1.0, respectively. If both parameters \(\alpha_N\) and \(\alpha_D\) are set equal to 1.0, the front-wheel active steering is eliminated, and the vehicle response becomes like that of the normal vehicle—the baseline vehicle.

Figure 12.11 shows the results of applying the previous idea to the driving simulator with a full dynamics model of the vehicle motion. The yaw rate frequency responses of the driving simulator vehicle with the front active steering control are compared in this figure to the vehicle yaw rate responses calculated by Eqn (12.10) for various increase and decrease of \(\omega_n\) and \(\zeta\). It is confirmed that the natural frequency and the damping ratio are equivalently changed independently by the active front-wheel steering control.

Next is to experimentally examine the effects of the natural frequency and the damping ratio on the driver parameters, especially on the response delay time constant, \(\tau_L\), which correlates well with the handling quality evaluation. Using the driving simulator mentioned above, the driver model parameters during the same lane change as in Section 12.4.2 are identified by the method learned in Section 12.2.2. Figure 12.12 shows the measured time histories of lateral acceleration and yaw rate of the driving simulator motion-base compared with those calculated by the simulator software in the lane change. This guarantees fidelity of the driving simulator in terms of vehicle motion to be sensed by the drivers.

Figure 12.13 is the result of the identification. The identified response delay time constant of four drivers during the lane change with the vehicles of various combinations of the natural frequency, \(\omega_n\), and the damping ratio, \(\zeta\), is shown on the \(\omega_n-\zeta\) plane. The driver parameter, the time constant, \(\tau_L\), on the \(\omega_n-\zeta\) plane tells us in this figure that there is a combination of \(\omega_n\) and \(\zeta\) that gives us a peak value of \(\tau_L\) for each driver. This implicates that there is an optimum value of \(\omega_n\) and \(\zeta\) for the vehicle handling characteristics that raises the vehicle handling quality evaluation to the peak. This agrees well with our normal experiences.

### 12.5.2 Handling Quality to Steering Torque [5]

Steering torque has a big effect on the handling quality evaluation. In this section, we will look at how the various steering torque characteristics to steering angle, shown in Figure 12.14, influence the parameter of the driver response delay, \(\tau_L\), identified, which eventually influences the handling quality evaluation of the driver during a lane change. Six cases of the steering reaction torque characteristics, as shown in Figure 12.14, are provided in the steering system of the same
driving simulator as was used in Section 12.5.1. The driver parameters are identified in the same way as the previous cases of the identifications during the lane change with the simulator vehicles equipped with the six various steering torque characteristics, respectively. The lane change course used is the same one as shown in Figure 12.4.

A result, shown in Figure 12.15, is the identified time constant, $\tau_L$, of the 10 drivers for the six various steering torque characteristics. The time constants of the 10 drivers change with the variations of the steering reaction torque characteristics in a similar aspect, as
FIGURE 12.12
Calculated and measured motions of driving simulator in lane change.

FIGURE 12.13
Identified driver parameter, \( \tau_L \), on \( \omega_N \zeta \) plane (\( V = 80 \) km/h, \( L_C = 45 \) m).
12.5 Handling Quality Evaluation Based on Driver Parameter

**FIGURE 12.14**

Reactive torque to steering angle characteristics.

**FIGURE 12.15**

Identified time constant, $\tau_L$, of 10 drivers for six various steering torque characteristics at $V = 100$ km/h and $L_C = 55$ m.
shown. Figure 12.16 shows the increase rate of the averaged time constant of the 10 drivers according to the steering torque characteristics for the different vehicle speeds and lane change lengths. The average handling quality rated by the 10 drivers is also shown in the figure. The drivers adopted a higher time constant of the response delay, $\tau_L$, during the lane change with the steering reaction torque of spring + damping and spring + friction + damping, and they gave a higher handling quality evaluation to them. This figure confirms that the identified time constant of the response delay, $\tau_L$, strongly correlates with the handling quality evaluation.

The interesting point is that the steering torque has an effect on the driver parameters similar to vehicle response characteristics to steering angle input changes, even though there is no change in the vehicle response to steering angle input itself. Actually, it is clear that the steering torque has nothing to do with the open-loop transfer function in the control block diagram of steering angle control for the driver-vehicle system shown in Figure 12.1. Nonetheless, the drivers change their parameters according to the steering reactive torque as though the vehicle response characteristics to steering angle input was changed, and if they feel it is easy to handle, then they adopt a higher $\tau_L$ and give a good handling evaluation.
A road vehicle is normally operated under a wide variation in the number of passengers and/or various load conditions, which result in fairly large changes of the vehicle weight. The rise in the vehicle weight bring us the increase in vehicle specifications of mass and moment of inertia; meanwhile, the change of the vehicle weight causes variations in the vertical loads of the front and rear tires. Eventually, considerable variations in the cornering stiffness occur that depend on the vertical load. Thus, the vehicle handling characteristics will be changed considerably by the variations of the vehicle weight.

At the upper part of Figure 12.17, there is a calculated result of the change of one of the response parameters, the natural frequency, depending on five different tire sets in the dependence of cornering stiffness to vertical load, as shown in Figure 12.18. Also, the change of the natural frequency resulting from the rise in the weight of the vehicles with a respective tire set is shown.

As the vehicle weight as well as the tire characteristics have significant effects on the vehicle response characteristics, their corresponding effects on the driver’s handling quality evaluation must emerge. To confirm this, there is a result of the identification of the driver time constant, $\tau_L$, during a lane change using the previous driving simulator vehicles with various tires and vehicle weights. Figure 12.17, at the middle, shows the change of the identified $\tau_L$ averaged by four drivers for the vehicles with the five different tire sets and the different weights.

The increase of the time constant, $\tau_L$, resulting from the change of the tire characteristics corresponds with the increase in the vehicle natural frequency shown in Figure 12.17. It seems reasonable from the study results in Sections 12.4.2 and 12.5.1, however, that there is a contradiction. Figure 12.17, at the top, also shows us that the time constant, $\tau_L$, increases with the rise in the vehicle weight, even though the weight increment of the vehicle with any of the five tire characteristics results in the reduction of the natural frequency.

Because the natural frequency only is not necessarily enough to consistently explain these results, we will pay attention to a parameter other than the response parameters—a sensitivity parameter to the disturbance. As is discussed in the latter part of Section 4.2.3, the disturbance sensitivity parameter of the vehicle is defined here as the yaw angle gain to yaw moment disturbance input, and it is described as the following:

$$\theta_m = \frac{(K_f + K_r)V}{2l^2K_fK_r\left\{1 - \frac{m(l_jK_r - l_jK_f)}{2l^2K_fK_r}V^2\right\}}$$

From the preceding equation, it is obvious that the greater the cornering stiffness, the less sensitive it is to the disturbance. In the bottom of Figure 12.17, we can see the calculated results of the change of disturbance sensitivity corresponding to the rise in the weight of the respective vehicles with five sets of the tire characteristics. A significant decrease in the sensitivity parameter with the increase in the vehicle weight corresponds well to the increase of $\tau_L$ with the weight increase shown in Figure 12.17. It seems that the driver feels a good handling quality strongly depending on the decrease in the vehicle’s sensitivity parameter value, and the contradiction mentioned is now solved.

The natural frequencies of the vehicles equipped with tire-2 and tire-4 decrease less with the rise in the vehicle weight, as shown in Figure 12.17, because of more linear dependence of the
FIGURE 12.17
Vehicle parameters and driver parameter, $\tau_L$, with respect to tire and vehicle weight.

FIGURE 12.18
Five cases of dependence of tire cornering stiffness on vertical load.
cornering stiffness of the tires to the vertical load. Therefore, it is reasonable that the increase of the time constant, $\tau_L$, with the rise in the vehicle weight is more considerable, especially for the vehicles with tire-2 and tire-4 in Figure 12.17 due to the decrease in the sensitivity parameter with the weight increase.

The bottom of Figure 12.19 shows the calculated result of the variation in the vehicle response parameters and the sensitivity parameter in more detail to the rise in the vehicle weight for the vehicle with tire-2. While all the response parameters change less, the sensitivity parameters decreases significantly with an increase in the vehicle weight, which coincides well with the increase in the identified $\tau_L$ of four drivers according to the vehicle weight increase, as shown at the top of the same figure. This supports the view that the vehicle handling quality depends not only on the natural frequency of the vehicle but also strongly on the sensitivity parameter, and the handling quality can be evaluated by the identified driver parameter, $\tau_L$.

The sensitivity parameter itself is not the response parameter to steering angle input, and it has no explicit effect on the closed-loop characteristics of the driver-vehicle system of steering angle control input. Even so, it is interesting to see that the driver changes his/her control parameter, $\tau_L$, of steering angle supposedly according to the change of sensitivity parameter during the lane change by feeling the sensitivity of the vehicle to the disturbance. The higher the sensitivity of the vehicle is, the lower $\tau_L$ the driver adopts to control the vehicle and the lower the handling quality evaluation becomes.

**FIGURE 12.19**

Four drivers’ parameter $\tau_L$ and vehicle parameters with respect to vehicle weight.
12.5.4 EFFECT OF STABILIZING VEHICLE MOTION [1,2]

As shown in Subsection 3.3.2.3, a body side-slip angle, $\beta$, of a normal understeer vehicle produces a positive yaw moment, $-2(l_f K_f - l_r K_r)\beta$, so far as the tire characteristics of the vehicle remain within a linear relation to side-slip angle. However, a saturation property of the tire lateral force to side-slip angle reduces the positive yaw moment with increase of the side-slip angle, which makes the vehicle less stable. In order to compensate for that, it is possible to exert the yaw moment produced by longitudinal forces of the tires on the vehicle body for stabilizing the vehicle motion. This is a kind of DYC that controls the tire longitudinal forces to give the vehicle the same yaw moment as linear tires can generate as much as possible, even when the vehicle motion gets into tire nonlinear ranges in order to augment the stability of the vehicle.

Figure 12.20 shows the measured time histories of the driver-vehicle system behavior during lane changes on a proving ground for different vehicles with and without DYC. Though the driver feels it is easy to control the vehicle with DYC and gives a better subjective handling quality evaluation, almost no specific difference that supports the driver’s opinion is found between the measured time histories of the vehicles with and without DYC.

On the other hand, there are significant differences in the driver parameters identified by the same method as described in Section 12.2 using the time histories of driver-vehicle system during the lane changes. Figure 12.21 shows the identified driver parameters of the driver. It is found in the figures that the vehicle with DYC allows a larger time delay of the drivers, $\tau_L$, and smaller

![Graphs showing vehicle responses during lane change.](image-url)
preview time, $\tau_h$, compared with the identified parameters of the vehicle without control. Especially, a larger $\tau_L$ reasonably corresponds to the driver’s evaluation on the vehicle handling quality [2].

**REFERENCES**


