



Model based prediction for steering response

Journal:	<i>Part D: Journal of Automobile Engineering</i>
Manuscript ID	JAUTO-16-0149.R3
Manuscript Type:	Original Article
Date Submitted by the Author:	28-Dec-2016
Complete List of Authors:	Lee, Eunjae; Korea Advanced Institute of Science and Technology, Future vehicle division Lee, Jonghyup; Korea Advanced Institute of Science and Technology, Mechanical Engineering Choi, Seibum; Korea Advanced Institute of Science and Technology, Mechanical Engr Dept Gil, Gibin; Hankook Tire R and D Center, Research division
Keywords:	Steering response, Steering agility, Handling objective test, Handling subjective test, Tire dynamics
Abstract:	To predict steering response, this paper proposes a string tire model based on relaxation length and a vehicle model which is designed by using effective cornering force. The cornering force, lateral static and distortion characteristics of a tire are significant factors that describe steering response. Nevertheless, it is difficult to define the contribution of each factor when a tire's cornering motion is evaluated through subjective assessment. A new concept of tire model, which adopts distortion stiffness(K_D), is defined. A modified vehicle dynamics model is adapted to represent tire force lag and converted cornering stiffness (C_a) based on effective cornering force. This integration scheme is proposed to provide a more accurate modelling for steering response performance. The integration model is confirmed by dynamic response and validated through comparison with a steering response assessment by a subjective test engineer. The test results demonstrate the ability of this approach to predict steering response.

1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
25
26
27
28
29
30
31
32
33
34
35
36
37
38
39
40
41
42
43
44
45
46
47
48
49
50
51
52
53
54
55
56
57
58
59
60

SCHOLARONE™
Manuscripts

For Peer Review

Model based prediction of steering response

Eun Jae Lee¹, Jong Hyup Lee², Seibum Ben Choi², and Gibin Gil³

¹Future Vehicle Division, KAIST, Daejeon, Republic of KOREA

²Department of Mechanical Engineering, KAIST, Daejeon, Republic of KOREA

³Research Division of Central Research Center, Hankook Tire co. Ltd., Daejeon, Republic of KOREA asphaltguy@kaist.ac.kr, sonic_92@kaist.ac.kr, seibumchoi@kaist.ac.kr, gibingil@hankooktire.com

Keywords

Tire model; Steering response; Steering agility; Handling objective test; Handling subjective test; Tire dynamics

Abstract

To predict steering response, this paper proposes a string tire model based on relaxation length and a vehicle model designed by using effective cornering force. The cornering force, lateral static, and distortion characteristics of a tire are significant factors that describe steering response. Nevertheless, it is difficult to define the contribution of each factor when a tire's cornering motion is evaluated through subjective assessment. A new concept of tire model, which adopts distortion stiffness (K_D), is defined. A modified vehicle dynamics model is adapted to represent tire force lag and converted cornering stiffness (C_α) based on effective cornering force. This integration scheme is proposed to provide a more accurate modelling for steering response performance. The integration model is confirmed by dynamic response and validated through comparison with a steering response assessment by a subjective test engineer. The test results demonstrate the ability of this approach to predict steering response.

Introduction

Contrary to common opinion, a strong coupling of a firm tire and chassis can generate fast steering response. To verify steering response, a huge variety of design parameters and testing methods have been developed. However, most of the existing solutions are too expensive to be practical due to cost of real test facilities and simulation software.

1
2
3
4
5
6 To predict steering response, this work integrated tire and vehicle modelling aims to
7
8 achieve the following objectives:

- 9
10
11 a. to predict steering response with a simplified model;
12
13 b. to utilize existing test machines and parameters globally;
14
15 c. to verify the validity in an easy way.
16

17
18 The three common parameters that have been widely used by numerous tire manufacturing
19
20 companies for indoor tests are cornering stiffness(C_α), lateral stiffness(K_L), and distortion
21
22 stiffness(K_D). These three test parameters comprise the fundamental approach to
23
24 determining the lateral motion of a tire. In addition, each parameter provides a specific
25
26 physical meaning of the tire's lateral motion. [1, 2] However, when these three tire test
27
28 procedures are conducted on a moving vehicle, observing and analysing the performance
29
30 results from each test procedure is very difficult because the exact initiation point of each
31
32 test is unknown.
33

34
35 Based on previous research, the typical dynamic model and string model with a relaxation
36
37 length is an appropriate tire model that can closely characterize a tire's lateral dynamic
38
39 motion. [1,2]
40

41
42 In this study, another equation is adopted that can be implemented in the tire model: an
43
44 equation for distortion stiffness(K_D), which can closely replicate the characteristics of
45
46 steering response. With the implementation of the proposed additional equation, the three
47
48 unknown property values can be mathematically determined, and this provides greater
49
50 accuracy than the previous model, which uses the controversial estimated value.
51
52

53
54 In the vehicle modelling part of this study, a single-track vehicle model was designed,
55
56 which mainly modified the tire force lag and cornering stiffness (C_α). From the tire
57
58
59
60

1
2
3
4
5
6 modelling, a dynamic tire model is adopted in the equation of motion to represent tire force
7
8 lag. To represent the steering compliance and roll behavior effect, the cornering stiffness
9
10 (C_{α}) is converted to the effective cornering force measured by a suspension parameter
11
12 measuring machine. Therefore, it is expected that this integration can be proposed to
13
14 predict the steering response.
15
16

17
18 The remainder of this paper is organized as follows. In the next section, the standard string
19
20 tire model is reviewed, and its three tire test properties are described in detail. The third
21
22 section describes the newly proposed tire model, which implements distortion stiffness
23
24 (K_D). The fourth section presents the single-track vehicle model with effective cornering
25
26 force. In the fifth section, the dynamic response based on the proposed tire and vehicle
27
28 models are evaluated and compared with actual tire performance subjective assessment
29
30 results to validate the feasibility of the proposed model. The last section concludes this
31
32 paper.
33
34
35

36 37 Review of previous research

38
39 A tire's lateral stiffness is closely related to its steering response. When the lateral stiffness
40
41 and cornering stiffness are high, the steering response becomes quicker. On the other hand,
42
43 when the performance results are analysed directly by subjective evaluation, each
44
45 parameter (lateral, distortion static stiffness, and cornering stiffness) cannot be correlated
46
47 clearly. To overcome this issue, appropriate tire modelling based on an evaluation method
48
49 is proposed in this paper. In previous tire models, the magic formula tire model and the
50
51 LuGre tire model were initially introduced, improved, and corrected [1, 3, 4]. These are
52
53 good representatives to obtain the cornering stiffness from experimental friction curves.
54
55 However, these models are not suitable to predict steering response with steady-state
56
57
58
59
60

values of lateral, distortion static stiffness, and cornering stiffness. In this work, only the cornering stiffnesses are obtained by the magic formula equation and an experimental indoor test. The details will be discussed in the next section.

The string tire model and relaxation length were selected to describe the performance appropriately. [1]

Guenther (1990) and Heydinger (1991) derived the typical dynamic model used for lateral tire force lag. This model expresses the relaxation length (L) as [5]

$$L = \frac{C_\alpha}{K_L} \quad (2.1)$$

where C_α and K_L represent the cornering stiffness at a steady state and lateral stiffness, respectively. Each variable can be further expressed as [5]

$$C_\alpha = \left. \frac{\partial F_y}{\partial \alpha} \right|_{\alpha=0} \quad (2.2)$$

$$K_L = \left. \frac{\partial F_y}{\partial y} \right|_{y=0} \quad (2.3)$$

where α and y are the slip angle and the lateral displacement of tire elements from the wheel plane centre, respectively. [5]

From equations (2.1) to (2.3), the relaxation time constant with respect to tire force lag (τ_{lag}) is related to the relaxation length as follows:

$$\tau_{lag} = \frac{C_{\alpha}}{K_L V_x} \quad (2.4)$$

The string tire model was first studied by Von Schelippe in 1941. [2] This specific tire model assumes that a tire is a set of endless strings, where each string provides a large number of tread elements. Figure 1 depicts the string tire model proposed by Von Schelippe and Pacejka. [1].

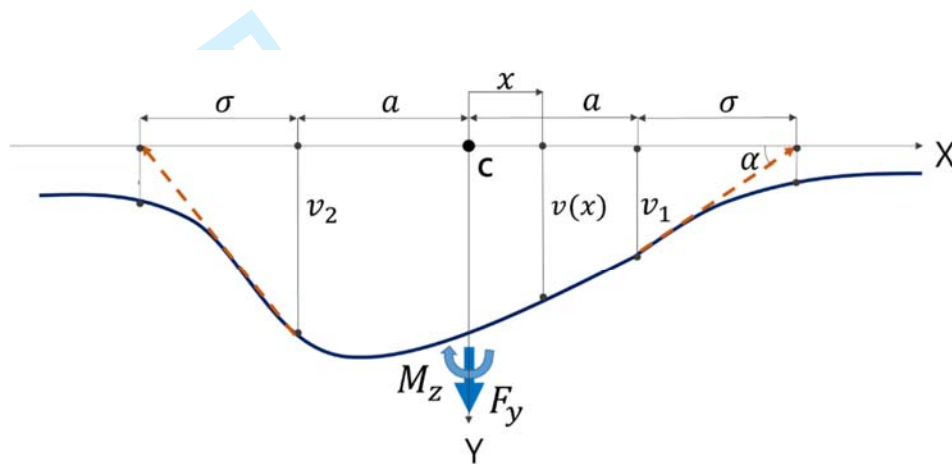


Figure 1 Top view of string tire model

where a , x , v , α , σ X and Y are the contact patch length, the longitudinal displacement and deflection of the string, the slip angle, the relaxation length, as well as the wheel plane axis and vertical axis of the wheel plane axis on the horizontal plane, respectively. [1, 2]

Proposed tire model

The three test parameters that have been commonly used by most tire companies for indoor testing are cornering stiffness (C_{α}), lateral stiffness (K_L), and distortion stiffness (K_D). These three test procedures comprise the fundamental approach to determine the lateral motion of a tire. In addition, each parameter provides a specific physical meaning of the tire's lateral motion from the previous study. These three factors measured through indoor

testing reveal the characteristics of a tire at a steady state, and they are represented by equations (3.1) to (3.3) from a previous study [1]:

$$K_L \triangleq \left. \frac{\partial F_y}{\partial y} \right|_{y=0} = 2C_c(\sigma + a) \quad (3.1)$$

$$C_\alpha \triangleq \left. \frac{\partial F_y}{\partial \alpha} \right|_{\alpha=0} = 2C_c(\sigma + a)^2 \quad (3.2)$$

$$K_D \triangleq \left. \frac{\partial M_z}{\partial \theta} \right|_{\theta=0} = 2C_c a \left\{ \sigma(\sigma + a) + \frac{1}{3}a^2 \right\} \quad (3.3)$$

Here, C_c was used to denote the carcass tension in the previous study. However, it is the total modulus of elasticity, between lateral force and deflection of the string, represented by not only carcass tension but also the sum of nylon, steel belt, and other reinforcement materials. Combining equations (3.1) and (3.2), the relaxation length can be derived as

$$\therefore \sigma = \frac{C_\alpha}{K_L} - a \quad (3.4)$$

The formula in equation (3.4) newly expresses the relaxation length which is comparable to that defined as 'L' in equation (2.1). The next subsection will discuss the accuracy of the newly expressed relaxation length σ .

Transient state values of string model characteristic

To analyse the response of tire performance, transient state values should be discussed as well as steady state values. In previous studies, three different string models have been developed to describe string shape within a contact patch [1, 2]: the single-point string model, the straight tangent model, and the exact string model, as shown Figures 2 and 3.

The exact string model proposed by Segel (1966) assumes that the velocity of each tread element with respect to the road is zero. The single-point string model assumes that the tire contact point is at one particular point. The straight tangent model is a simple linear approximation of the exact model. The contact patch line is defined as a linear extension of the deflection v_l at the patch leading edge. [1]

The time constant of the single-point string model can be defined as [1]

$$\tau_{single} = \frac{(\sigma + a)}{V_x} \quad (3.5)$$

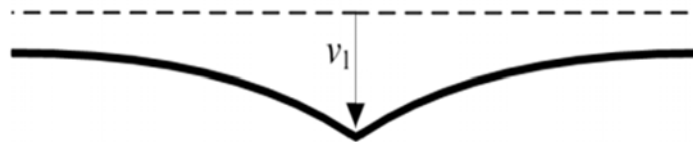


Figure 2 Single point string model

The transfer function of the straight tangent model is also a first-order function, and the relaxation time constant $\tau_{straight}$ is defined as [1]

$$\tau_{straight} = \frac{\sigma}{V_x} \quad (3.6)$$

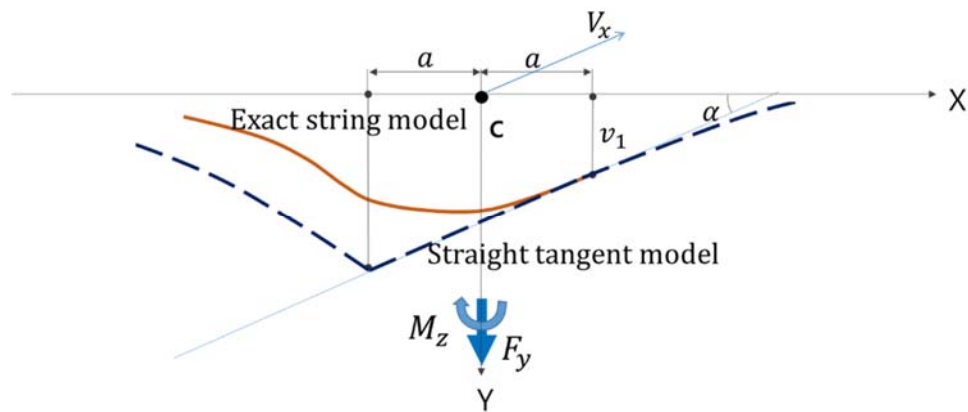


Figure 3 Exact and straight tangent model

The cornering stiffness of the exact model at a transient state cannot be described in a simple first-order form; therefore, the time constant is not defined explicitly. [1, 2, 5]

However, comparing the three transfer functions, the exact model describes a real tire best, and the others are approximations of the exact model.

The accuracies of the approximated models are investigated in terms of frequency response. Figure 4 can be expressed by the values of time constants in equation (3.5)-(3.6) and it shows that the straight tangent approximation is very close to the exact model.

Therefore, it is reasonable to substitute the exact model with the transfer function of straight tangent approximation, and the time constant of the exact model can be represented by the time constant of the straight tangent model $\tau_{straight}$. Therefore, it makes sense to calculate the relaxation length σ with proper expressions.

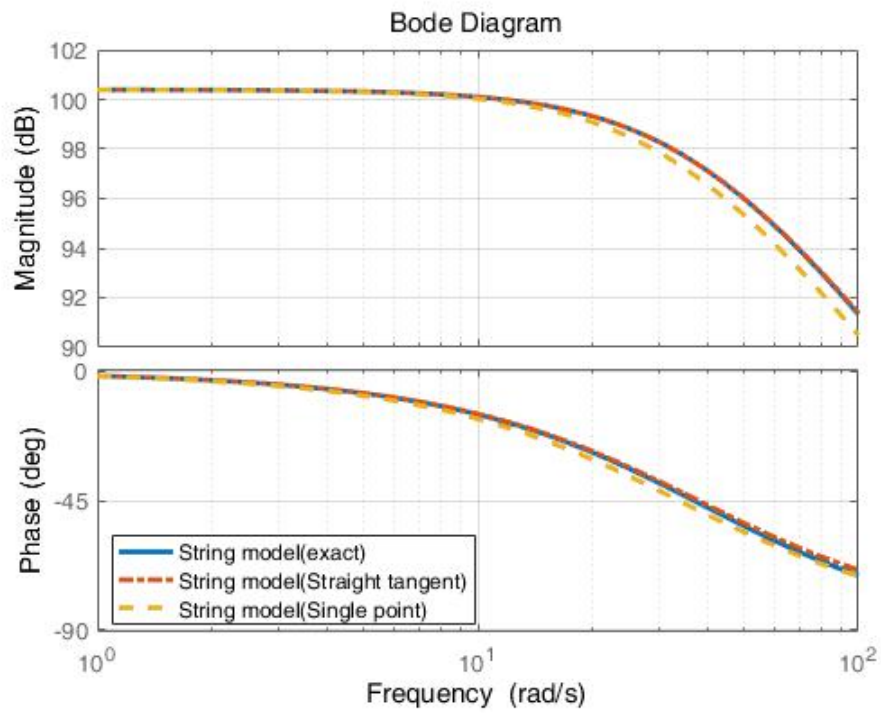


Figure 4 Bode plots of three models ($V_x = 100$ km/h)

Model identification

To calculate relaxation length σ using equation (3.4), the value of contact patch length 'a' is required. It is measured by indoor testing. Practically, determining the length of the contact patch is not a simple task. Even if each sample has the same pattern design, the sample-to-sample variations make the task quite delicate since the end of the contact patch has a highly non-linear shape.

Therefore, in this work, a new method of describing relaxation length σ using distortion stiffness K_D was investigated instead of the controversial contact patch length 'a'.

A new relaxation length (σ) is proposed, as expressed in equation (3.7), which can be derived by combining equations (3.1), (3.2), and (3.3):

$$\therefore \sigma = \left(\frac{C_\alpha^3}{K_L^3} - \frac{3C_\alpha K_D}{K_L^2} \right)^{\frac{1}{3}} \quad (3.7)$$

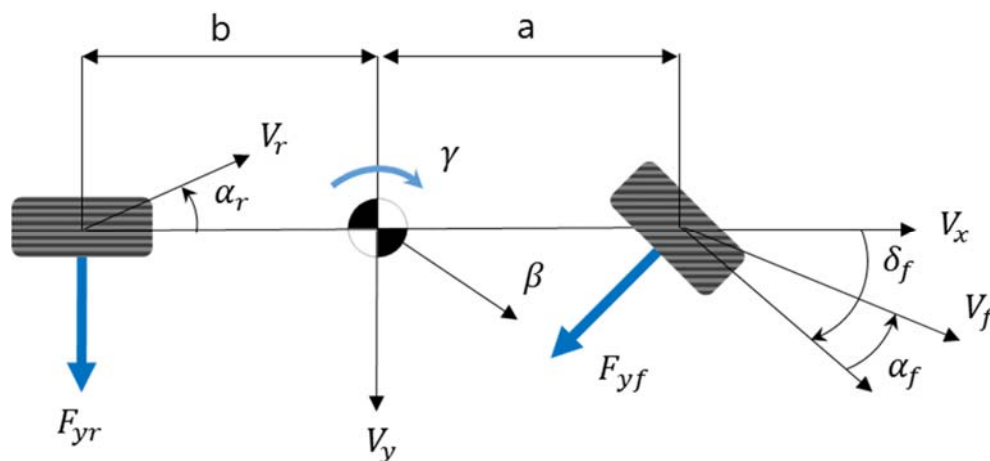
where C_α is the cornering stiffness, K_L is the lateral stiffness, and K_D is the distortion stiffness. This equation can closely replicate the characteristics of steering response by reflecting the effect of distortion static characteristics on the existing models. Furthermore, it can be formulated without the controversial factor 'a'.

Also, this work investigated the connection between the proposed tire model and the subjective evaluation method. A good starting point is acquiring the steering performance from a subjective test based on a test engineer's evaluation. The acquired steering feels from a subjective assessment based on the test engineer's evaluation is divided into two categories: steering response and torque. When the test engineer feels the steering performance of the tire, the most important factor is the visible reaction of the yaw motion and torque feedback felt by hands. After subjective testing, engineers express the distortion

1
2
3
4
5
6 motion of a tire as accurately as possible to the design engineers as well as the yaw
7 response of a vehicle, and this can be expressed in numerous ways. [1, 6-9] Therefore, it
8
9 makes a sense to include the distortion static stiffness K_D to the relaxation length σ .
10
11
12
13
14
15
16

17 Vehicle modelling

18
19 The single-track model depicted in Figure 5 is used to describe the vehicle lateral dynamics.
20
21 The three-degree-of-freedom (3DOF) linear model, which is a proper representation of
22 lateral dynamics in the steering response assessment region, is employed. Roll and pitch
23 motions are assumed to be not excited and are ignored.
24
25
26
27
28
29



46 Figure 5 Single-track model for vehicle lateral dynamics
47
48
49
50
51
52
53
54
55
56
57
58
59
60

Single track model equation of motion

The equations of motion for the model are given as

$$\dot{V}_y = \frac{2F_{yf}}{m} + \frac{2F_{yr}}{m} - V_r \quad (4.1)$$

$$\dot{r} = \frac{2aF_{yf}}{I_z} + \frac{2bF_{yr}}{I_z} \quad (4.2)$$

$$\dot{F}_{yf} = -\frac{F_{yf}}{\tau_f} + \frac{C_f}{\tau_f} \left(\delta_f - \frac{V_y}{V_x} - \frac{ar}{V_x} \right) \quad (4.3)$$

$$\dot{F}_{yr} = -\frac{F_{yr}}{\tau_r} + \frac{C_r}{\tau_r} \left(-\frac{V_y}{V_x} - \frac{br}{V_x} \right) \quad (4.4)$$

Here, the relaxation time constant (τ_f) from equation (3.6) and (3.7) is reflected in equations (4.3) and (4.4).

State space realization of the linear time invariant (LTI) system

The system modelled using a set of ordinary differential equations can be represented in the following state equations:

$$\begin{Bmatrix} \dot{V}_y \\ \dot{r} \\ \dot{F}_{yf} \\ \dot{F}_{yr} \end{Bmatrix} = \begin{bmatrix} 0 & -V_x & \frac{2}{m} & \frac{2}{m} \\ 0 & 0 & \frac{2a}{I_z} & -\frac{2b}{I_z} \\ -\frac{C_f}{\tau_f V_x} & -\frac{aC_f}{\tau_f V_x} & -\frac{1}{\tau_f} & 0 \\ -\frac{C_r}{\tau_r V_x} & -\frac{bC_r}{\tau_r V_x} & 0 & -\frac{1}{\tau_r} \end{bmatrix} \begin{Bmatrix} V_y \\ r \\ F_{yf} \\ F_{yr} \end{Bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{C_f}{\tau_f R} \\ 0 \end{bmatrix} \delta_f \quad (4.5)$$

$$Y(S) = \begin{Bmatrix} r \\ a_y \end{Bmatrix} = \begin{Bmatrix} r \\ \dot{V}_y + V_x r \end{Bmatrix} \quad (4.6)$$

Here, the transfer function $(Y(S) / \delta_f)$ can be obtained through the steering response using the front steering angle input δ_f for every vehicle dynamic response frequency, where the cornering stiffnesses C_f and C_r are changed to effective cornering stiffnesses, which are explained in detail in the next section.

Effective of suspension compliance effect on cornering force

Since the single-track model does not consider the vertical load transfer, it cannot express the understeering tendency of a vehicle. Therefore, the suspension compliance effect is also implemented as depicted in Figure 6. Suspension compliance expresses the input force variation in the suspension components and bushings resulting from changes in the road surface received through the tire.

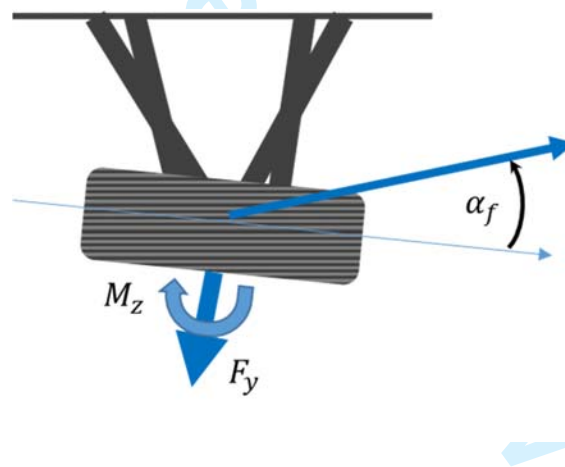


Figure 6 Steering due to wheel suspension compliance effect

Suspension compliance provides the benefit of extracting the cornering stiffness variable output. The effective cornering stiffness affected by the steering compliance can be calculated as follows [1]:

$$\alpha^e = \alpha + \frac{\partial \alpha}{\partial F_y} F_y + \frac{\partial \alpha}{\partial M_z} M_z \quad (4.7)$$

$$F_{yi} = C_i \alpha_i^e \quad i = (1, 2) \quad (4.8)$$

$$M_{zi} = (C_i \alpha_i^e) n_i \quad (4.9)$$

Then, equations (4.7) and (4.9) are substituted into equation (4.10) as follows [1]:

$$\alpha_i^e = \alpha_i + \frac{\partial \alpha}{\partial F_y} C_i \alpha_i^e + \frac{\partial \alpha}{\partial M_z} (C_i \alpha_i^e) n_i \quad (4.10)$$

Next, equation (4.10) is substituted into equation (4.11) as follows [1]:

$$F_{yi} = C_i \alpha_i^e = \frac{C_i}{1 - \frac{\partial \alpha}{\partial F_y} C_i - \frac{\partial \alpha}{\partial M_z} C_i n_i} \alpha_i = C_i^* \alpha_i \quad (4.11)$$

where C_i is the cornering stiffness of each axle, n_i is the pneumatic trail of the tire, and

C_i^* is the effective cornering stiffness. The cornering stiffness of each axle, and the

pneumatic trail of a tire were measured using an MTS Flat-Trac® test machine. Therefore,

the effective cornering stiffness can be calculated by equation (4.11).

Equation (4.11) shows that the cornering stiffness of the front axle tends to be decreased, and that of the rear axle tends to be relatively increased. The results presented in Table 1 show how much force can be changed through the suspension compliance effect. The compliance effects on the vehicle dynamics using a full vehicle model is presented in Figure 7. These obtained results are quite similar to actual evaluation results. To obtain the compliance coefficients $\partial \alpha / \partial F_y$ and $\partial \alpha / \partial M_z$, in equation (4.11), a compliance suspension parameter test machine was used for each test vehicle as shown Figure 7.

Table 1 Measured cornering stiffness and that converted to effective axle cornering stiffness

Axle	Measured by Flat-Trac®	Effective cornering force
Front	1860 N/deg	1075 N/deg
Rear	1700 N/deg	1477 N/deg



Figure 7 Suspension parameter test machine [11]

A simulation was conducted using a B segment sedan with 195/55R16 size tires. While the tire model is adopted in the vehicle model, the relaxation time constant and cornering stiffness values are implemented as described in equations (4.3) and (4.4). Figure 8 shows the dynamic response of yaw and lateral acceleration obtained using the measured cornering stiffness values and the steady-state gain at a low frequency region. Some of these values do not fully correspond with the results obtained from the field objective test. The magnitude of yaw rate and lateral acceleration without compliance effect have similar characteristics that appear in the sports car class as shown in Table 2.

Table 2 Practical magnitude of yaw rate and lateral acceleration of each vehicle class

	Normal range of SUV class	Normal range of Sports class
Magnitude of yaw rate @ 0.2 Hz (absolute value)	0.2~0.3	0.3~0.45

Therefore, it is crucial to use the effective cornering stiffness (C_{α}^*) for the proposed model.

It demonstrates that the newly configured value corresponds more accurately with the results from the field objective test.

The detailed description of the objective test method and the results are beyond the scope of this paper. To improve the accuracy of the proposed tire model, the effective axle cornering stiffness is used for the target value, and the adjustment factors are applied to the cornering stiffness transfer function.

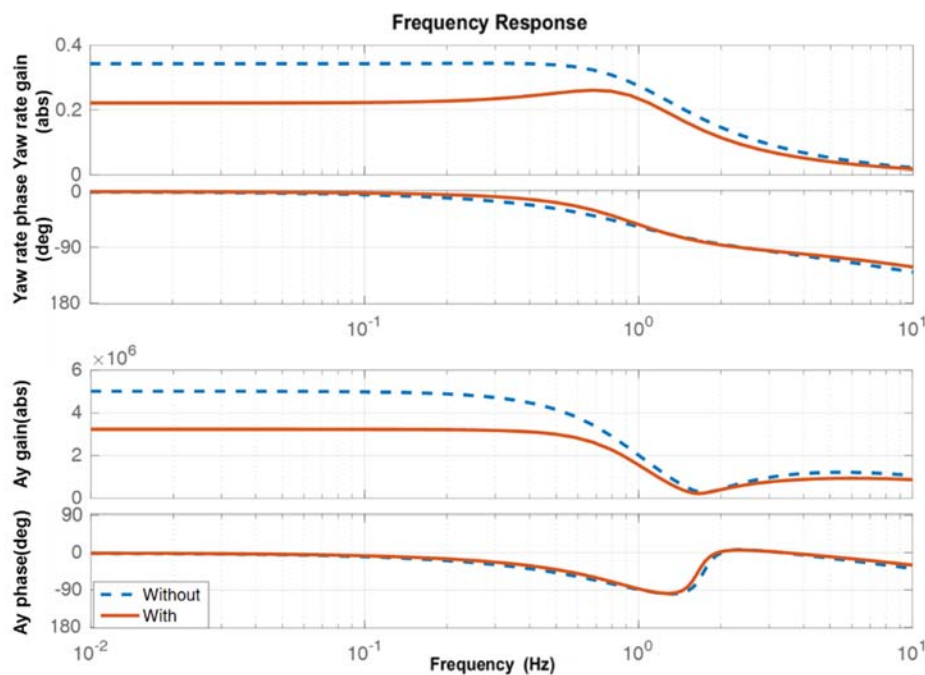


Figure 8 Dynamic response results of yaw and lateral acceleration by the effective cornering force

Model validation

The validity of the proposed tire and vehicle model, was verified by dynamic response tests and subjective assessment. There are two phases to identify the steering response. The initial phase is the front axle response time represented by the yaw rate. Afterwards, the second phase is the rear axle response expressed by the lateral acceleration. Therefore, the total response delay can be represented by the phase lag of lateral acceleration as shown Figure 8 and it has high correlation with subjective assessment according to field experiments. [12, 13]

The transfer function of the vehicle model based on the relaxation time constant and effective cornering force effect was implemented for each test tire. The relaxation time constant (τ_f, τ_r) in equation (3.6) is defined using equation (3.7). In the vehicle model, the dynamic response is represented by lateral acceleration, where input δ_f is applied to observe the dynamic response. [12, 13] Then it is evaluated at each frequency and the performance ranking is confirmed as shown in Figures 3 and 4.

Each case sample was tested on one day to reduce the effect of external noise, such as different test engineers as well as vehicle and environmental conditions.

A total of nine tire samples of two cases with the specifications in Table 3 were tested indoors under the same load condition (580 kgf) without being fitted to a vehicle. Those tires were made by the same manufacturer with a summer tire tread pattern design, and only minor design parameters were different. This was intended to evaluate performance differences in relation to very minor changes.

Tables 3 and 4 present the results of the indoor tire tests and subjective tests, respectively.

In Table 3, the proposed relaxation lengths were calculated based on three parameters:

lateral stiffness K_L , cornering stiffness C_α , and distortion stiffness K_D , which were obtained through the average of five iterative tests. The cornering stiffness C_α was measured using an MTS Flat-Trac® test machine. In Table 4, the subjective rating results are presented for the nine tire samples based on SAE evaluation method. [12]

Table 3 Tire indoor test results.

Sample	Case 1				Case 2				
	A	B	C	D	E	F	G	H	I
K_L ($\times 10^5 \text{N/m}$)	1.184	1.202	1.258	1.221	1.152	1.189	1.198	1.221	1.234
C_α ($\times 10^5 \text{N/rad}$)	1.250	1.256	1.242	1.235	1.226	1.219	1.224	1.220	1.260
K_D ($\times 10^3 \text{Nm/rad}$)	4.080	4.570	4.130	4.414	3.812	4.865	4.695	4.767	4.767
Relaxation length (m)	0.588	0.576	0.549	0.563	0.593	0.569	0.568	0.555	0.565

Table 4 Tire subjective test results.

Sample	A	B	C	D	E	F	G	H	I
Subjective Rating	6.500	6.625	6.750	6.625	6.500	6.750	7.000	7.250	7.500

In general, each subjective test engineer has a unique frequency of steering manoeuvres when assessing steering response performance. To analyse the correlation between the dynamic response and subjective test results, it is necessary to investigate the major frequency input used for the subject since results are greatly affected by the frequency contents. [6]

To minimize this issue, the specific steering manoeuvre frequencies performed by each test engineer need to be determined. Figure 9 shows the spectrum of steering input and vehicle yaw rate response for a steering response test.

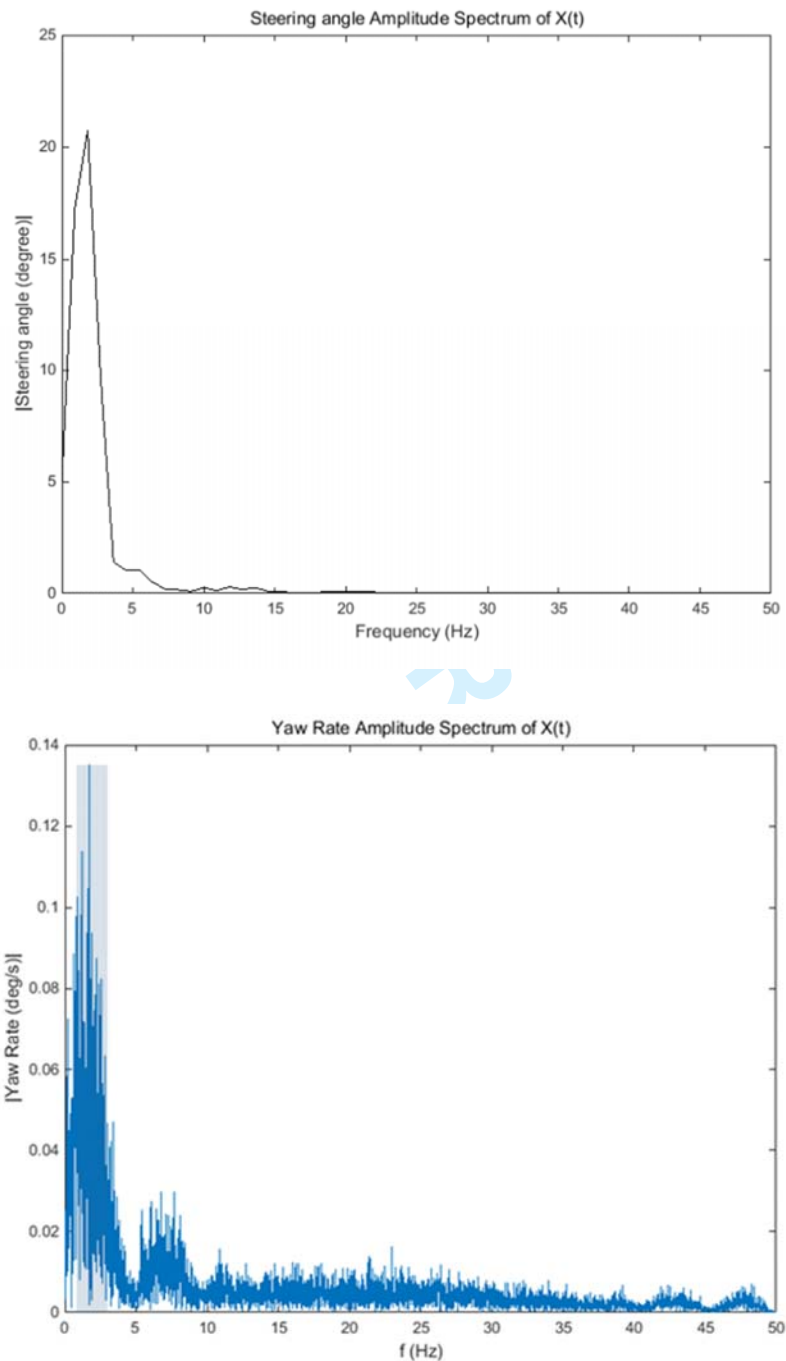


Figure 9 Steering manoeuvre and yaw rate spectrum in a subjective test

1
2
3
4
5
6 After each test engineer's major steering manoeuvre frequency is identified, more accurate
7
8 analysis of the dynamic response of yaw rate and lateral acceleration can be achieved. In
9
10 this work, the dynamic response was evaluated within the steering manoeuvre frequency
11
12 near 1 Hz reflecting each test engineer's driving style.
13
14

15 16 Model prediction results 17

18
19 In the vehicle model, the dynamic response is represented by lateral acceleration where
20
21 input δ_f is applied to observe the dynamic response. [5, 10, 14] The lateral acceleration
22
23 response of each sample A-I was evaluated in the frequency domain by using the
24
25 parameters in Table3 and equations (4.5) - (4.6). Also, the performance ranking is
26
27 confirmed as shown in Figures 10. The variation of the dynamic response test was
28
29 negligible. However, there are clear differences between subjective ratings in the scale of
30
31 0-1. [12]
32
33
34
35
36
37
38
39
40
41
42
43
44
45
46
47
48
49
50
51
52
53
54
55
56
57
58
59
60

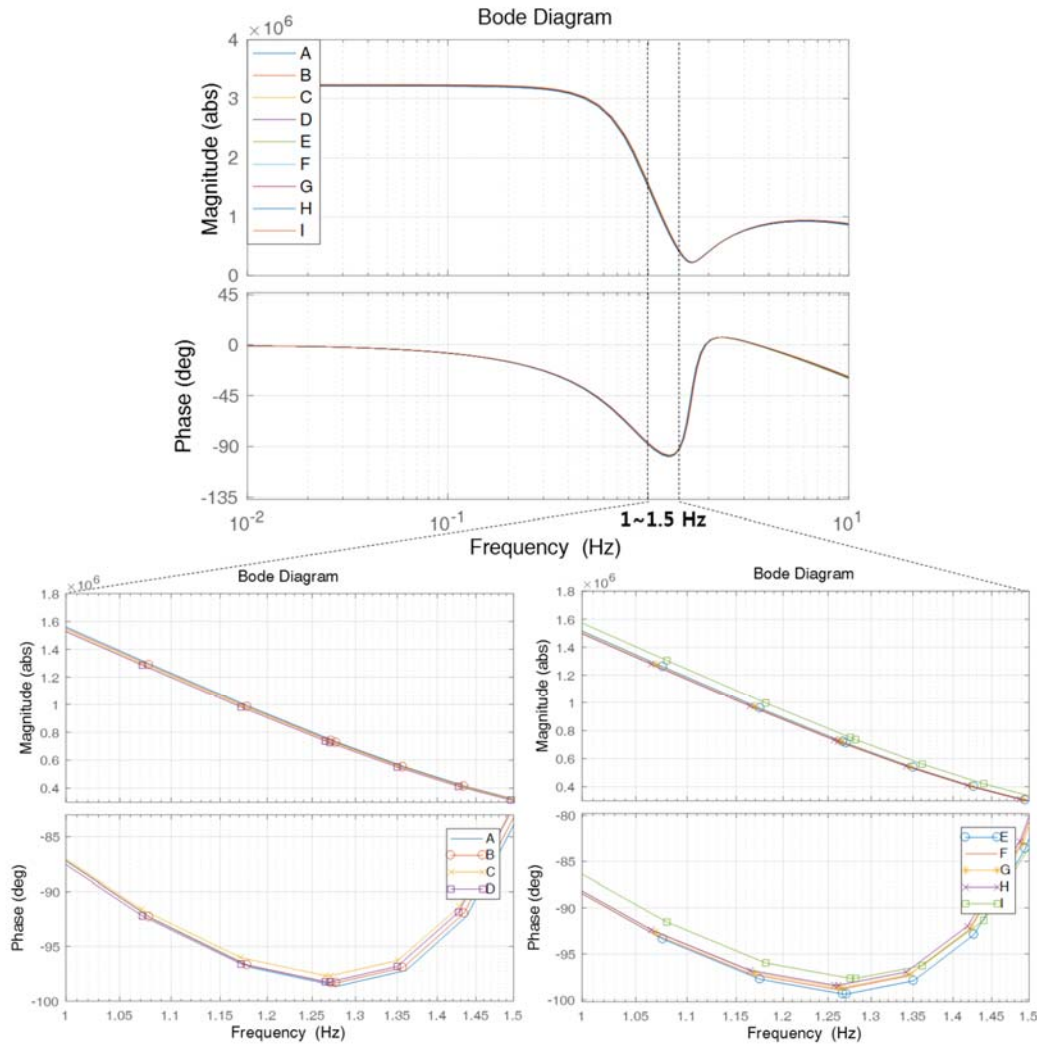


Figure 10 Dynamic response analysis of lateral acceleration at 1.2Hz

The integration model was confirmed by dynamic responses and validated through comparison with a steering response assessment by the subjective test engineer. Figures 11 and 12 show the correlation obtained using a linear least square regression method (Draper and Smith, 1981). The results demonstrate that the proposed tire dynamic model, described in equations (3.6) and (3.7) correlates with the subjective rating results much better than the single-point model, expressed in equation (2.1) does. Using the new model, r^2 values were improved from 0.84 and 0.86 to as much as 0.95. High r^2 value means that the model

reflects the actual movement.[15] Based on these results, it can be concluded that this integration model is suitable to predict tire steering responses.

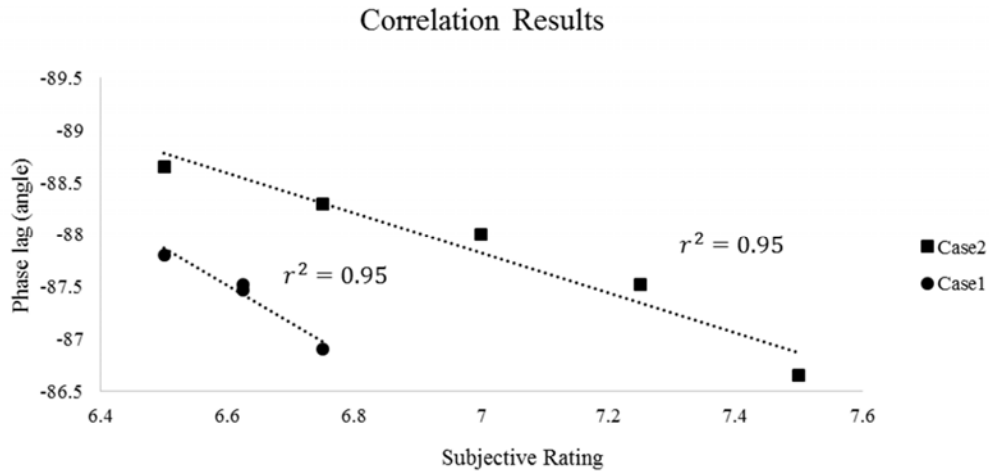


Figure 11 Correlation results of subjective rating and phase lag of lateral acceleration (with proposed straight tangent model)

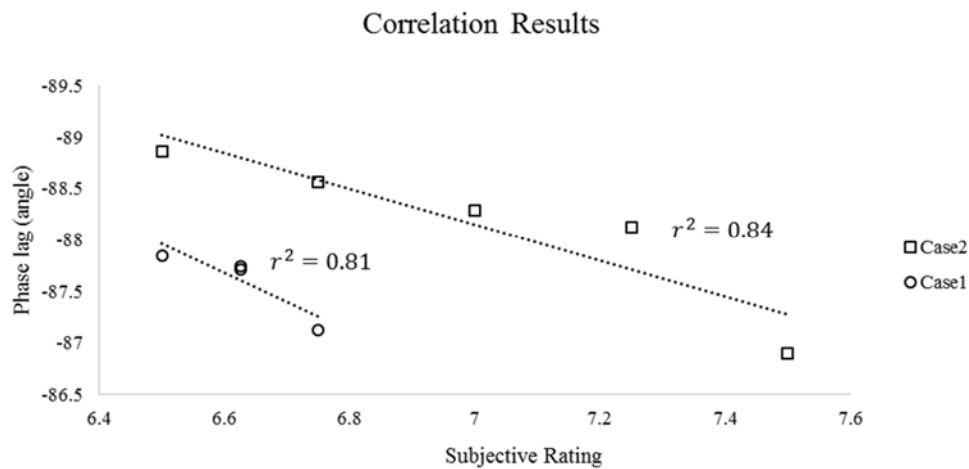


Figure 12 Correlation results of subjective rating and phase lag of lateral acceleration (with typical time constant)

Conclusions

The concept of using the relaxation length to predict the steering response performance has been applied to the string tire model. Three common parameters, stiffness (C_α), lateral stiffness (K_L), and the distortion stiffness (K_D), are considered at the same time while

1
2
3
4
5
6 relaxation length is defined to enhance the accuracy of the proposed tire steering response
7
8 model, which can be used even before a vehicle is built. The modified single-track vehicle
9
10 model is adapted to represent the relaxation time constant (τ_f, τ_r) and converted cornering
11
12 stiffness (C_α) based on effective cornering force. This integration is based on the fact that
13
14 tires influence the lateral dynamics. The modified vehicle dynamics model was applied to
15
16 nine tire samples with two different classes. This integration model yielded good estimation
17
18 results correlating with the subjective assessment very well.
19
20
21

- 22 ● The relaxation length equation (3.7) is proposed
- 23 ● The modified single-track vehicle model is adapted
- 24 ● This integration model was applied to the nine tire samples
- 25 ● Good estimation results were obtained correlating with the subjective assessment

26
27
28
29
30
31
32 The proposed model has the potential to significantly reduce the evaluation efforts
33
34 associated with field testing, and tire design engineers can conjugate the results of the tire
35
36 and vehicle characteristics used in indoor tests. Therefore, the model-based prediction of
37
38 steering response may play a significant role in replacing tire field tests on a vehicle with
39
40 just indoor tire tests.
41
42

43 Acknowledgements

44
45
46
47 The authors are grateful to Hankook Tire Co.Ltd
48
49

50 References

- 51
52
53 [1] H. Pacejka, *Tire and Vehicle Dynamics*: Elsevier, 2005.
54 [2] B. von SCHILIPPE, "Das Flattern eines bepneuten Rades, Bericht 140
55 der Lilienthal-Gesellschaft," *NACA TM 1365*, pp. 125-147, 1954.
56 [3] J. Deur, J. Asgari, and D. Hrovat, "A dynamic tire friction model for
57 combined longitudinal and lateral motion," in *2001 ASME International
58 Mechanical Engineering Congress and Exposition November 11-16, 2001*,
59
60

- 1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
25
26
27
28
29
30
31
32
33
34
35
36
37
38
39
40
41
42
43
44
- New York, NY, 2001.*
- [4] J. J. Zhu, A. Khajepour, and E. Esmailzadeh, "Total dynamic response of a PSS vehicle negotiating asymmetric road excitations," *Vehicle System Dynamics*, vol. 50, pp. 1835-1859, 2012.
- [5] R. Rajamani, *Vehicle dynamics and control*: Springer Science & Business Media, 2011.
- [6] G. L. Gil Gómez, M. Nybacka, E. Bakker, and L. Drugge, "Findings from subjective evaluations and driver ratings of vehicle dynamics: steering and handling," *Vehicle System Dynamics*, vol. 53, pp. 1416-1438, 2015.
- [7] H.-P. Krüger, A. Neukum, and J. Schuller, "A workload approach to the evaluation of vehicle handling characteristics," SAE Technical Paper 0148-7191, 2000.
- [8] C. B. Winkler, "Simplified analysis of the steady-state turning of complex vehicles," *Vehicle System Dynamics*, vol. 29, pp. 141-180, 1998.
- [9] K. Wolff, R. Kraaijeveld, and J. Hoppermans, "Objective evaluation of subjective driving impressions," *2008 JSAE Annual Cong. Yokohama, SAE Paper*, p. 0222, 2008.
- [10] K. A. Morris, *Introduction to feedback control*: Academic Press, Inc., 2000.
- [11] G. Gil, "Hybrid numerical integration scheme for highly oscillatory dynamical systems," 2013.
- [12] SAE, "Subjective rating scale for vehicle handling," in *SAE standard J 1441*, ed: SAE.
- [13] A. K. Zschocke and A. Albers, "Links between subjective and objective evaluations regarding the steering character of automobiles," *International Journal of Automotive Technology*, vol. 9, pp. 473-481, 2008.
- [14] G. J. Heydinger, W. R. Garrott, and J. P. Chrstos, "The importance of tire lag on simulated transient vehicle response," SAE Technical Paper 0148-7191, 1991.
- [15] S Data, F Frigerio, "Objective evaluation of handling quality" *Proc IMechE Part D: J Automob Eng*, vol. 216, 2002

Appendix

SA – Subjective Assessment

45
46
47
48
49
50
51
52
53
54
55
56
57
58
59
60