

Ride Comfort Analysis Considering Suspension Friction with Series Rigidity

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ABSTRACT

A dynamics model considering series rigidity was constructed to examine suspension friction, which has a major effect on ride comfort on paved roads. The friction characteristics of the bushings, ball joints, and shock absorbers are expressed with series elastic elements such as arm rigidity and the spring constant of the oil seals. It was confirmed that the calculated values for the overall spring constant and damping coefficient of the suspension virtually matched values measured in a 4-post shaker test. In addition, the results of analysis using this dynamics model confirmed that the degree of friction affects both the damping coefficient and the spring constant of the suspension, especially when the series rigidity is high. Also highly rigid friction has an adverse effect on sprung motion in frequency ranges above 15 Hz. After suspension enhancements were adopted based on these findings, 4-post shaker tests confirmed that sprung motion above 2 Hz improved..

CITATION: Koumura, S. and Shionoya, T., "Ride Comfort Analysis Considering Suspension Friction with Series Rigidity," *SAE Int. J. Passeng. Cars - Mech. Syst.* 9(1):2016, doi:10.4271/2016-01-1679.

1. INTRODUCTION

It is generally known that suspension friction has a greater effect on ride comfort when the amplitude of road surface inputs is low. Recently, a friction model containing series elastic elements was constructed using suspension bench test data. This model was utilized to analyze suspension friction while also considering rigidity, as a method to enhance the accuracy of ride comfort prediction and the like [1]. However, virtually no research has been reported regarding the relationship between whole suspension friction characteristics and each component properties. This paper describes the construction of a dynamics model using the degree of friction and series rigidity obtained from each suspension component properties. It also details the analysis results of the overall spring constant and damping coefficient of the suspension, as well as the vehicle vibration characteristics.

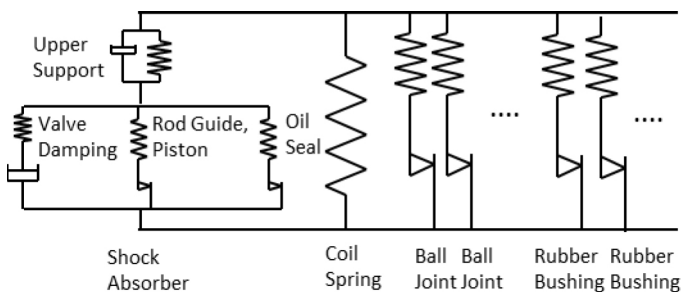


Figure 1. Dynamic model of Suspension vertical characteristics

2. FRICTION MODEL

Figure 1 shows the overall vertical characteristics of a suspension using the dynamics model. The following sections describe the calculation methods for the friction and hysteresis characteristics of the shock absorbers, rubber bushings and ball joints.

2-1. Shock Absorber/Coil Spring System

The force generated by the shock absorbers consists of the damping force created by the pressure difference of the valves (i.e., the valve damping force), the Coulomb friction created by the rod guide and piston, and the elastic friction caused by the oil seal. Each characteristic was identified from bench test results of stand-alone shock absorbers. In addition, the degree of Coulomb friction generated by the rod guide and piston also considers the effects of longitudinal and lateral force on the rods determined by the layout of the strut and the reaction force vectors of the coil spring

2-2. Rubber Bushings

The rotational characteristics (in the torsional and rotating directions) of stand-alone rubber bushings can be expressed by Equation (1) using steady-state terms in the Mats Berg model [2](Fig. 2).

$$T = T_s + T_H$$

$$\begin{cases} T_s = K_s \cdot \theta \\ T_H = \frac{K_H \cdot (\theta - \theta_0)}{(T_{H\max} \mp T_0) \pm K_H \cdot (\theta - \theta_0)} (T_{H\max} \mp T_0) \end{cases} \quad (1)$$

where

T : torque

T_S : torque (linear component)

T_H : torque (hysteresis component)

q : torsional and rotational angles

K_S : linear component spring constant

K_H : initial spring constant of hysteresis component

T_{Hmax} : degree of hysteresis

θ_0 : angle at reference condition*

T_0 : torque at reference condition*

*Latest angle or torque when a direction to transform changed.

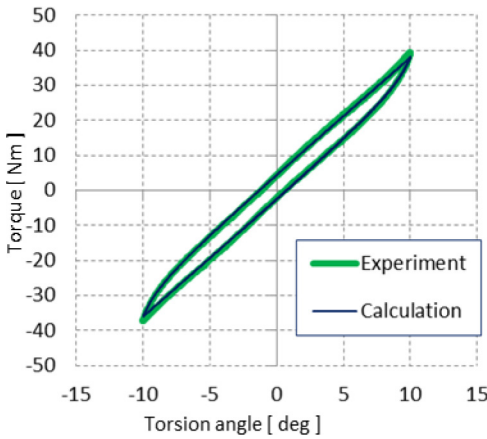


Figure 2. Bushing reaction torque

In addition the bushing characteristic parameters K_{SW} , K_{HW} , and F_{HmaxW} converted to the wheel center position can be calculated using Equations (2) to (4) using the bushing rotational angle per suspension stroke unit (Fig. 3). Figure 4 shows the rubber bushing characteristics converted to wheel center position.

$$K_{SW} = \sum_{j=1}^3 \left(\frac{\partial \theta_j}{\partial Z} \right)^2 K_{Sj} \quad (2)$$

$$K_{HW} = \sum_{j=1}^3 \left(\frac{\partial \theta_j}{\partial Z} \right)^2 K_{Hj} \quad (3)$$

$$F_{HmaxW} = \sum_{j=1}^3 \frac{\partial \theta_j}{\partial Z} T_{Hmax,j} \quad (4)$$

where

K_{SW} : bushing linear spring constant converted to wheel center position

K_{HW} : degree of the hysteresis converted to wheel center position

F_{HmaxW} : initial gradient of the hysteresis converted to wheel center position

$\partial \theta_j / \partial Z$: bushing rotational angle per suspension stroke unit (Fig. 3)

j : bushing rotation direction index

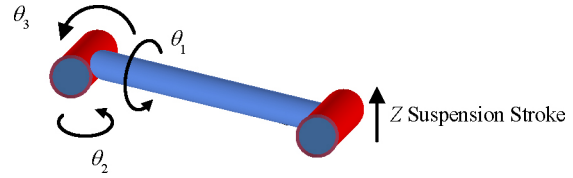


Figure 3. Bushing rotational angle per suspension stroke unit

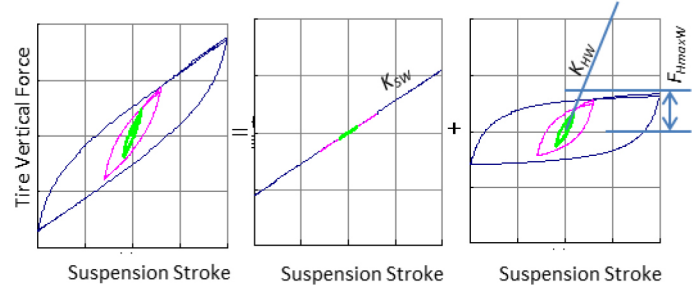
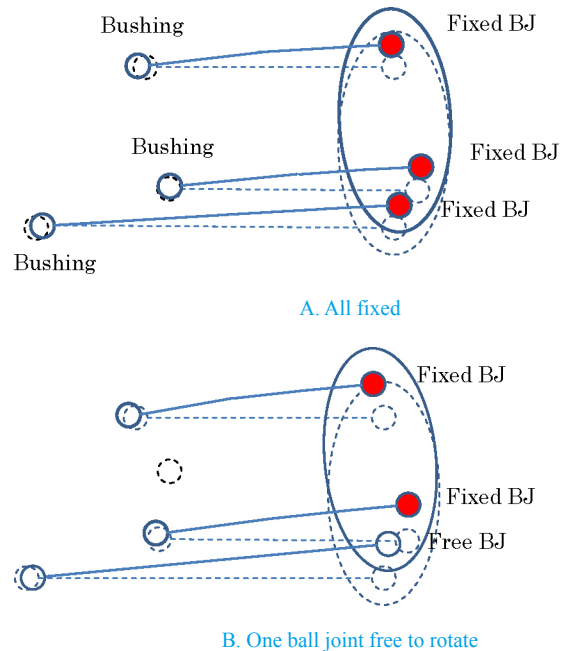


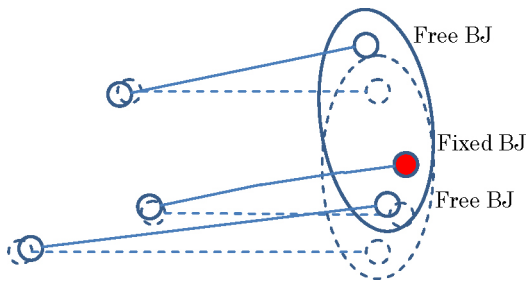
Figure 4. Tire vertical force due to rubber bushing

2-3. Ball Joints

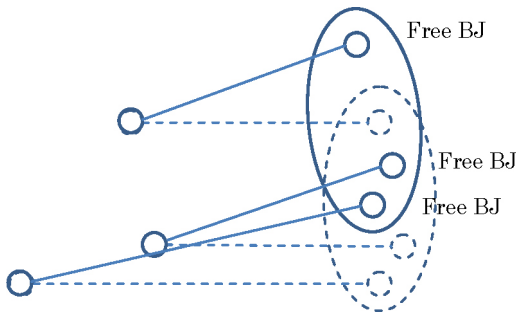
Two cases must be considered for ball joints: when the input torque is smaller than the friction torque and the ball joint is fixed, and when the input torque exceeds the friction torque and rotation is possible. Figure 5 shows the suspension motions when vertical displacement input is applied to the tire contact patch of a suspension with three ball joints. Figure 5(A) shows the case in which all three ball joint components of the suspension are fixed. Figure 5(B) shows the case when the input torque of one of the ball joints exceeds the friction torque and rotates. Figure 5(C) shows the case when the input torque of two of the ball joints exceeds the friction torque and the ball joints rotate. Figure 5(D) shows the case when all of the ball joints rotate.



B. One ball joint free to rotate



C. Two ball joints free to rotate



D. All ball joints free to rotate

Figure 5. Vertical behavior of suspension with ball joints

The line graph in Fig. 6 shows the relationship between the suspension stroke and the tire vertical force in states (A) to (D). The calculation methods for Z_A to Z_C and W_A to W_C in the figure are described below.

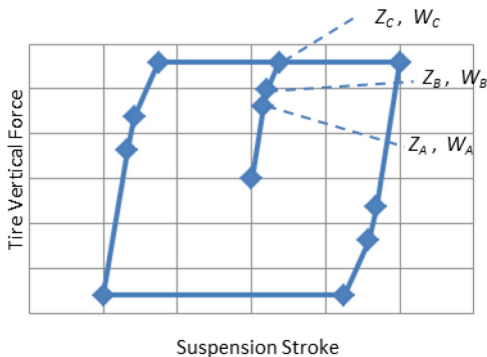


Figure 6. Tire vertical force due to ball joint friction

The characteristics of the rubber bushings and fixed state ball joints shown in Table 1 were used for calculation of the tire vertical force and the three-direction input torque of each ball joint per suspension stroke unit, by linkage mechanism analysis. The suspension stroke Z_A at which the input torque of the first ball joint reaches the friction torque, and the tire vertical force W_A were calculated by Equations (5) and (6).

Table 1. Joint rigidity used in linkage mechanism calculation

	Rigidity value
Rubber bushings	Sum of linear spring constant and initial spring constant of hysteresis component
Ball joints	Series sum of suspension arm rigidity and ball joint stud rigidity (Fig. 7)

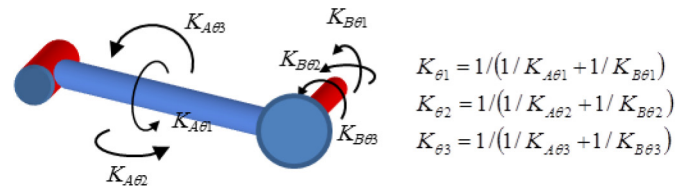


Figure 7. Series sum of bending and torsional rigidity of suspension arm and ball stud

$$Z_A = \min \left[\frac{T_{F0i}}{\sqrt{\sum_{j=1}^3 \left(\frac{\partial T_{ij}}{\partial Z} \right)_A^2}} \right] \quad (5)$$

$$W_A = \left(\frac{\partial W}{\partial Z} \right)_A \cdot Z_A \quad (6)$$

where

T_{F0i} : ball joint friction torque

$(\partial W / \partial Z)_A$: tire contact patch reaction force due to ball joint friction per suspension stroke unit in state A

$(\partial T_{ij} / \partial Z)_A$: ball joint input torque per suspension stroke unit in state A

i : ball joint index

j : ball joint rotation direction index

The residual friction torque T_{FAi} of the ball joints in Z_A and W_A is expressed by Equation (7)

$$T_{FAi} = T_{F0i} - \left[\sum_{j=1}^3 \left(\frac{\partial T_{ij}}{\partial Z} \right)_A^2 \right] \cdot Z_A \quad (7)$$

Next, the tire vertical force and the three-direction input torque of each ball joint per suspension stroke unit in Fig. 5(B) were calculated. The suspension stroke Z_B at which the input torque of the second ball joint reaches the friction torque, and the tire vertical force W_B were calculated by Equations (8) and (9)

$$Z_B = Z_A + \min \left[\frac{T_{FAi}}{\sum_{j=1}^3 \left(\frac{\partial T_{ij}}{\partial Z} \right)_B^2} \right] \quad (8)$$

$$W_B = W_A + \left(\frac{\partial W}{\partial Z} \right)_B \cdot (Z_B - Z_A) \quad (9)$$

T_{FBP} , Z_C , and W_C were calculated using Equations (10) to (12) in the same way.