Sensitivity Analysis of Transfer Mechanism Considering Dynamic Characteristic of a Bushing

Jin Hee Lee, Tae Won Park, Sung Pil Jung and Wook Hyeon Kim

Abstract — **The cause of brake judder is BTV (Brake Torque Variation) which is resulted from DTV (Disc Thickness Variation) and this phenomenon affects the ride comfort. In this paper, in order to analyze the transfer mechanism, a suspension system of an automotive is modeled using a multibody dynamic analysis program. Especially, nonlinear and frequency dependent characteristics of the bushing components are considered. BTV measured from brake dynamometer experiment is applied to the simulation model. Finally, sensitivity analysis of the suspension system which considers stiffness of a bushing as a design variable is carried out.**

Index Terms — **Brake Judder, Transfer Mechanism, Bushing, Sensitivity Analysis, Brake Torque Variation.**

I. INTRODUCTION

While driving a vehicle, frequent and sudden braking can cause vibration and noise problems like judder, shimmy and chattering. Especially, vibration, called brake judder, occurs rotational vibration (nibble), translational vibration (shake) of a steering wheel and pulsation of a brake pedal pass through the suspension and steering systems.

These vibrations that are caused by friction between the brake disc and pad can give an unpleasant feeling to passengers. In addition, if resonance is occurred, the suspension and steering systems can be severely damaged. Also this can cause of an accident. Therefore, brake judder should be evaluated and improved during the vehicle development.

The brake judder is generated from an irregular variation of brake force. This phenomenon is usually arisen by ununiformity of the disc friction surface and variation of the contact pressure between disc and brake pad. These conditions are happened by the DTV (Disc Thickness Variation), initial production defects and disc run-out. Eventually, it is the cause of the BTV (Brake Torque Variation) [1-2].

To reduce such brake judder, there are two main methods. The first way is to reduce brake torque variation due to exciting vibration forces for example improving the operation precision of the brake disc, the friction materials of the brake pad, etc. The other way is to differently set frequency range of natural frequency of the suspension system and frequency of the brake judder.

In this paper, numerical model is developed to find out vibration through the suspension system and rubber bushings. And the effect of vibration on the vehicle body is analyzed. Especially, sensitivity analysis due to change of the bushing stiffness is conducted and the influence is analyzed. If such evaluation and design improvement in virtual environment can apply at the initial design process, ride quality and developing advancement is possible [3].

II. CONSTRUCTION OF TRANSFER MECHANISM MODEL

A. Vehicle Model

To analyze dynamic behavior on judder, the multibody dynamic analysis program is used. The brake module and suspension system is assumed as a rigid body. The front suspension system of the vehicle consists of the Macpherson strut and each part is connected by joints, springs, dampers and bushings. The suspension and steering system model consist of 27 moving parts. The dimensions and material properties of each part of the system are obtained from the design process. Performance of the brake judder transfer mechanism is analyzed for the front suspension. An analysis model is shown in Fig. 1.

B. Bush Modeling

When the suspension system is modeled in analysis program, each part can be constrained as an ideal joint. However, the parts of actual vehicles are constrained by force element such as bushing. Table. I shows the type of bushing and mounting position used in front suspension.

The actual bushing components consist of rubber, urethane or fluid seal type. Such materials have viscoelastic behavior.

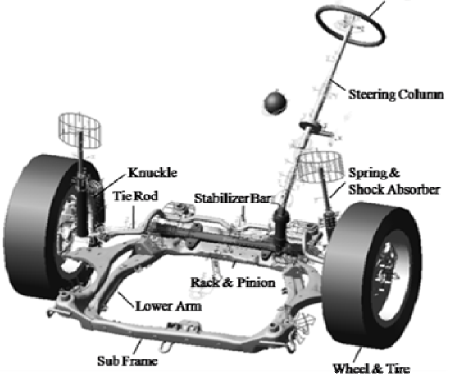

Fig. 1 Vehicle Multibody Dynamic Model

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Table. I Bushing Information

	Item	Location	No.
1	Sub Frame Mounting Bushing	FR	2
		RR	2
2	Lower Arm Mounting Bushing	FR	2
		RR	2
3	Stabilizer Bar Mounting Bushing	-	2
4	Spring Upper Strut Bushing	-	2
Total			12

Because bushing components support the vehicle body and constrain the movement of each part, it has a decisive effect on ride quality, driving stability, etc. Therefore, in order to more accurately evaluate chassis as a transfer mechanism, modeling of detailed bushing components is involved. Specially, bushings affect transmit of low frequency like judder. So this dynamic characteristic is needed to include.

In the bushing model, static nonlinear and frequency dependent characteristics are included using specific functions. The nonlinear characteristic of stiffness is created by applying experimental data using a spline curve - bushing force according to displacement. In other case as frequency dependent characteristic, the method that filters displacement variable x as a frequency function is applied using the static characteristic curve. In other words, to express characteristic of the frequency dependent characteristic of a rubber bushing, the transfer function formula based on the generalized Maxwell model is used. Equation (1) is the formula for calculating static bushing force.

$$F_{bush} = G(a_0 + x_f) \qquad (1)$$

Where, x_f is a filtered displacement value, $x_f = \frac{c+ds}{a+bs}x$. a, b, c and d are coefficients of the transfer function and a_0 is initial displacement. To use static displacement variable x as a frequency function, appropriate fitting process is needed using experimental data. Fig. 2 is the experimental data that is nonlinear bushing force according to the displacement. And Fig. 3 shows characteristic of the frequency dependent characteristic obtained tuning the transfer function coefficients.

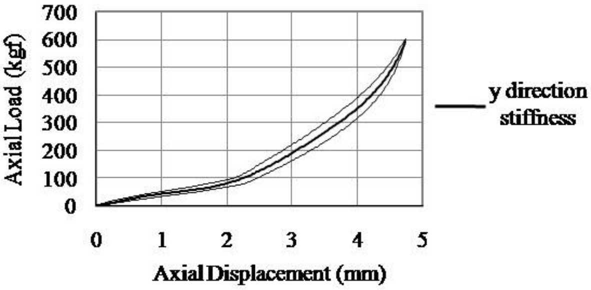

Fig. 2 Experimental Non-Linear Static Stiffness of Lower Arm Mounting FR bushing

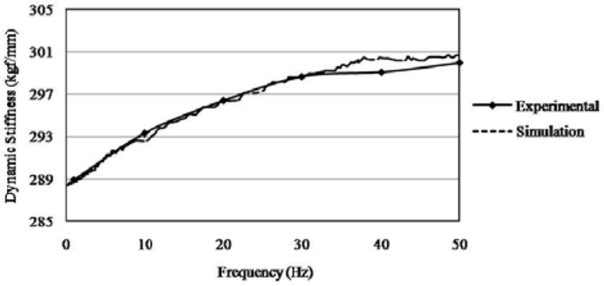

Fig. 3 Experimental Frequency Dependent Dynamic Stiffness of Lower Arm Mounting FR Bushing

Considering only 1st order is sufficient to express dynamic behavior because frequency range of the brake judder is less than 40Hz.

C. *BTV Measurement*

In developed virtual simulation environment, in order to represent the brake judder, BTV is inputted between the disc and pad. BTV is measured at the brake dynamometer test. Table. II shows test conditions. Fig. 4 and Fig. 5 show the test dynamometer and measured BTV, respectively.

Table. II Test Condition

Item	Unit	Value
Initial ang. velocity	Rad/s	67.4
Disc Thickness Variation	μm	8
Pressure	N/m^2	21×10^5
Dynamic Radius	m	0.330
Effective Radius	m	0.1336
Max Friction Coefficient	-	0.435
Contact Stiffness	N/m	1.5×10^7

Fig. 4 Brake Dynamometer

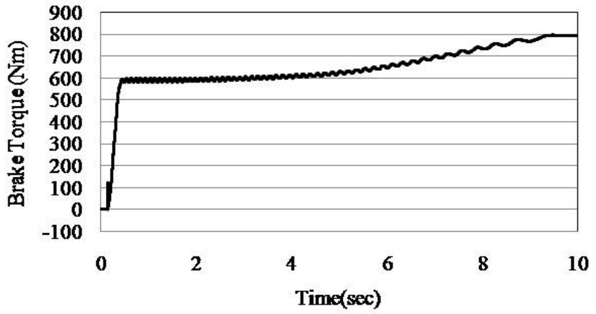

Fig. 5 Measured Brake Torque Variation

III. Sensitivity Analysis

A. Theoretical Background

Sensitivity is the derivative of the characteristic value in terms of design parameters. Sensitivity analysis allows main design parameters, which have a serious effect on the change of the characteristic value, to be selected. Since it is easier to create a Plackett-Burman design than the orthogonal array [4], it is widely used as a design table for sensitivity analysis. The Plackett-Burman design is a 2-level design of experiments and the interaction between design parameters can be neglected. n+1 experiments are carried out in terms of n design variables. The number of experiments is equal to a multiple of 4 and set according to the number of design parameters. In this study, the Plackett-Burman design of 8 runs is used. The relationship between design parameters and the characteristic value, which is called the response function, can be derived as (2).

$$y = \beta_0 + \sum_{i=1}^{k} \beta_i x_i + \varepsilon \qquad (2)$$

where x is the level of design parameters according to each experiment condition, β is the coefficients of design parameters, y is the characteristic value, ε is the error and k is the number of design parameters. Equation (2) is expressed in a matrix form as (3).

$$Y = XB + E \qquad (3)$$

According to the least square method, recursive coefficients (β) can be calculated by minimizing the square of error. The square of error can be written in the following matrix form.

$$E^T E = Y^T Y - 2B^T X^T Y + B^T X^T XB \qquad (4)$$

The error is eliminated by differentiating (4) with respect to B. This leads to

$$X^T Y = X^T XB \qquad (5)$$

Therefore, a set of recursive coefficients, B, can be obtained as

$$B = (X^T X)^{-1} X^T Y \qquad (6)$$

The magnitude of recursive coefficients indicates how much design parameters affect the change of the characteristic value [5].

B. Sensitivity Analysis of Bushing Component

The sensitivity analysis is implemented for 4 of 12 bushings which significantly affect vibrations. Selected components are the sub frame and lower arm mounting bushing. And total 5 design variables are chosen - stiffness of x, y and z direction of each bushing. Due to the suspension system is symmetry, stiffness of left and right variables have the same value. Fig. 6 shows the coordinate system of the lower arm mounting bushing and table. III represents design variables and level of bushing that sensitivity analysis is carried out. The stiffness of each design variable is applied at the experimental result of hardness 60 and 70. Each level of design variables is initial value of the nonlinear curve. Based on each design variable, total 8 times of simulation are performed using the multibody dynamic vehicle model. The objective of sensitivity analysis is to minimize the maximum rotational acceleration at the steering wheel. Therefore, maximum acceleration is obtained by the simulations. Experiment for sensitivity analysis is carried out using the Plackett-Burman design table. Fig. 7 shows the result of the simulations – experiment 3 and 4. Table. IV shows the level of design variables and the simulation results of each experiment.

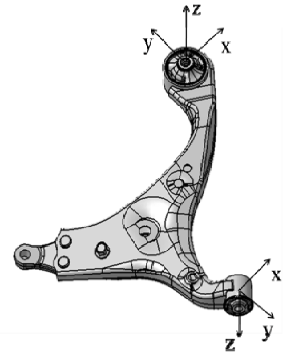

Fig. 6 Lower Arm(LH) and Coordinate of bush

Table. III Design Variable

Name	Design Variables		Stiffness(kgf/mm)	
			Min(-1)	Max(+1)
DV1	Lower arm mounting FR bushing	x	38.4	64.6
DV2		y	140.6	282.0
DV3		z	25.2	44.4
DV4	Lower arm mounting RR bushing	x/y	150.0	296.2
DV5		z	24.0	48.6

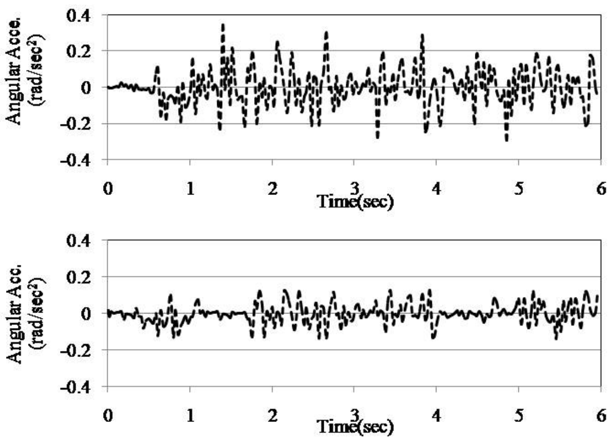

Fig. 7 Steering Wheel Angular Acceleration–exp.3 and 4

Table. IV Experiment Table and Result

Row	DV1	DV2	DV3	DV4	DV5	Result
1	+1	+1	+1	-1	+1	0.342
2	-1	+1	+1	+1	-1	0.213
3	-1	-1	+1	+1	+1	0.345
4	+1	-1	-1	+1	+1	0.130
5	-1	+1	-1	-1	+1	0.256
6	+1	-1	+1	-1	-1	0.286
7	+1	+1	-1	+1	-1	0.146
8	-1	-1	-1	-1	-1	0.288

Through the 2-level experiments, coefficients of response function in the form of polynomial function can be approximated. Equation (7) represents the response function. Sensitivity of bushing components is shown in Fig. 8. Change of the radial stiffness of the lower arm front and rear mounting bushing have a significant effect on the suspension system.

$$y = 0.2508 + 0.0247x_1 - 0.0565x_2 + 0.01558x_3 - 0.0255x_4 - 0.0108x_5 \tag{7}$$

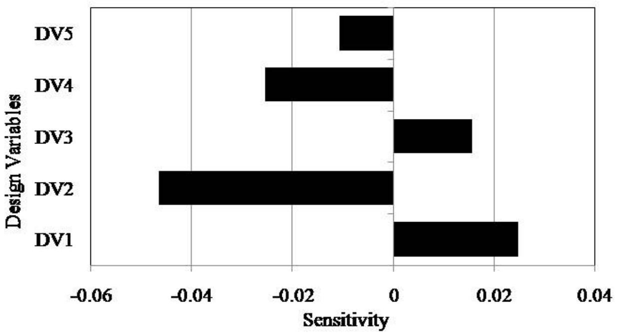

Fig. 8 Bush Sensitivity Analysis

IV. Conclusion

In this paper, using the multibody dynamic automotive model, the method which finds out influence of the brake judder is proposed in the virtual environment. The Mcpherson type suspension model is created and chassis performance is evaluated as a judder transfer path. Especially, a rubber bushing which plays an important role as vibration isolation is modeled in more detail. Nonlinear stiffness according to the displacement and frequency dependent characteristics are modeled applied actual experimental data. Applying a judder to brake system in the virtual model is carried out using BTV from the actual dynamometer. Through input BTV generated by judder phenomenon, the rotational acceleration is calculated. In order to find out impact judder on the transfer mechanism, sensitivity analysis due to change of the bushing stiffness is conducted for a lower arm mounting bushing. Sensitivity due to change of the each directional stiffness is analyzed using Plackett-Burman design table through 8 times simulation.

In order to verify results of this research, the vehicle test should be accompanied in the same environment. Furthermore, if sensitivity analysis which considers not only stiffness of bushings but also geometry of the suspension can be conducted, more useful analysis result can be obtained for the transfer mechanism. Finally, improvement of the vehicle ride quality is expected to contribute through optimization design for the suspension system and bushing components based on this result.

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