

Research on Mechanism of Change in Suspension Transfer Force in Relation to Low-Frequency Road Noise

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ABSTRACT

Cabin quietness is one of the important factors for product marketability. In particular, the importance of reducing road noise is increasing in recent years. Methods that reduce acoustic sensitivity as well as those that reduce the force transferred from the suspension to the body (the suspension transfer force) are used as means of reducing road noise. Reduction of the compliance of the body suspension mounting points has been widely used as a method of reducing acoustic sensitivity. However, there were cases where even though this method reduced acoustic sensitivity, road noise did not decrease. This mechanism remained unclear. This study focused on the suspension transfer force and analyzed this mechanism of change using the transfer function synthesis method. The results showed that the balance between the body's suspension mounting points, suspension bush, and suspension arm-tip compliance is an important factor influencing the change in suspension transfer force. For example, the suspension transfer force easily increases in the frequency range of 100-300 Hz in the axial direction of the rear lower arm of a multi-link suspension. In this case, the compliance of the body's suspension mounting points and that of the suspension bushes are close to each other and the compliance of the suspension arm-tips is less. To reduce road noise in such cases, effective methods of reducing acoustic sensitivity are countermeasures that do not reduce the compliance of the body's suspension mounting points in order to avoid increasing the suspension transfer force.

CITATION: Ichikawa, K., "Research on Mechanism of Change in Suspension Transfer Force in Relation to Low-Frequency Road Noise," *SAE Int. J. Passeng. Cars - Mech. Syst.* 8(1):2015, doi:10.4271/2015-01-0667.

INTRODUCTION

Reducing noise in the vehicle cabin is one of the most important aspects of vehicle product performance. Particularly in recent years as noise associated with engine input has been reduced, reducing road noise has become more important. Techniques commonly used to reduce road noise include reducing both body acoustic sensitivity and the force transferred to the body from the suspension. Several methods have been proposed for reducing body acoustic sensitivity, including reducing excitation point compliance, reducing excitation point conductance(1), and attaining a balance of low sensitivity and low weight by using optimization technology(2). Methods proposed for reducing the suspension transfer force include controlling the suspension mode shape(3) and controlling the tires' eigenvalue(4). In general, when developing vehicles, targeted values are individually determined for the vehicle body as the response side and suspensions as the input side. However, when making changes to the body in order to reduce body acoustic sensitivity, there are cases where road noise does not decrease and also where road noise actually increases. In these cases, the suspension transfer force is expected to increase. This study focuses on changes in suspension transfer force, because it is unclear in what situations the suspension transfer force does change. Using a substructure synthesis method known as the transfer function synthesis method(5), a theoretical formula for change in suspension transfer force was derived. The results show that the balance between the compliance of the body suspension mounting points, the suspension bushes, and the suspension arm-tips is an important factor influencing the change in the suspension transfer force. For example, the

suspension transfer force easily increases in the frequency range of 100-300 Hz in the axial direction of the rear lower arm of a multi-link suspension. Here, the compliance of the body suspension mounting points and that of the suspension bushes are close to each other and the compliance of the suspension arm-tips is less. To reduce road noise in such cases, effective methods of reducing acoustic sensitivity are countermeasures that do not reduce the compliance of the body suspension mounting points in order to avoid increasing the suspension transfer force. To verify this mechanism, finite element method was used to calculate the suspension transfer force, which is rarely accurately measured using vehicle experimentation. In addition, a method was also proposed for determining whether the compliance of the body suspension mounting points can be reduced to avoid increasing the suspension transfer force. This method clarified the direction of body countermeasures for road noise, thereby increasing Honda development efficiency by reducing reworking when investigating the reduction of suspension mounting point compliance.

ANALYSIS OF MECHANISM OF CHANGE IN SUSPENSION TRANSFER FORCE

Derivation of Theoretical Formula of the Suspension Transfer Force

The transfer function synthesis method, which is a substructure synthesis method, had been used to determine the mechanism of the suspension transfer force changes. As Fig. 1 shows, the vehicle was

divided into three components: the body, the suspensions and the suspension bushes. After decomposing, the boundary condition of each node of these components is unconstrained, and the suspension transfer force can be dealt with as external force.

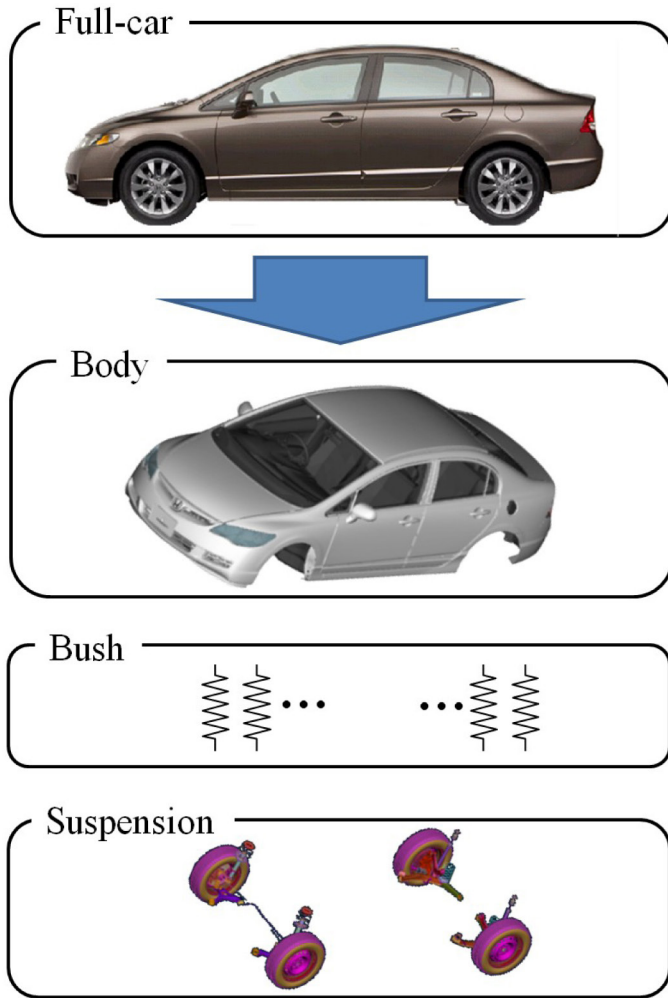


Figure 1. Decomposition to three substructures

The body's suspension mounting point displacement $\{X_b\}$ in the separated body is expressed as the following equation using the suspension transfer force $\{F\}$ and the body's suspension mounting point compliance matrix $[G_b]$. All these matrices are complex with each element containing amplitude and phase information.

$$\begin{Bmatrix} X_{b1} \\ \vdots \\ X_{bn} \end{Bmatrix} = \begin{bmatrix} G_{b11} & \cdots & G_{b1n} \\ \vdots & \ddots & \vdots \\ G_{bn1} & \cdots & G_{bnn} \end{bmatrix} \begin{Bmatrix} F_1 \\ \vdots \\ F_n \end{Bmatrix} \quad (1)$$

A schematic diagram of the body's suspension mounting point compliance matrix $[G_b]$ is shown in Fig. 2. The subscripts i and j represent joints, while G_{bij} represents compliance characteristics between the mounting points.

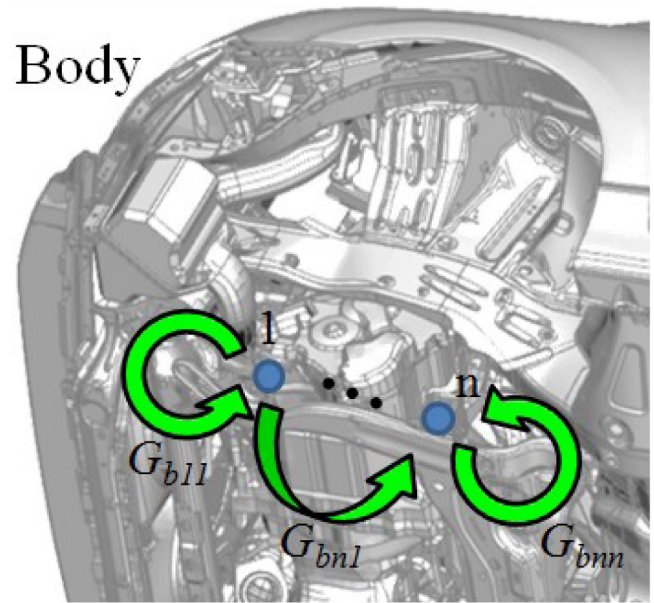


Figure 2. Schematic diagram of the body compliance matrix

Similarly in the suspension, arm-tip displacement $\{X_s\}$ is expressed as the following equation using the counterforce to the suspension transfer force in the suspension arm-tips $\{-F\}$, road input D , transfer characteristics from the tire ground-contact point to the suspension arm-tips in the suspension $\{G_{st}\}$, and the compliance matrix between the suspension arm-tips in the suspension $[G_s]$.

$$\begin{Bmatrix} X_{s1} \\ \vdots \\ X_{sn} \end{Bmatrix} = - \begin{bmatrix} G_{s11} & \cdots & G_{s1n} \\ \vdots & \ddots & \vdots \\ G_{sn1} & \cdots & G_{snn} \end{bmatrix} \begin{Bmatrix} F_1 \\ \vdots \\ F_n \end{Bmatrix} + \begin{Bmatrix} G_{st1} \\ \vdots \\ G_{stn} \end{Bmatrix} D \quad (2)$$

A schematic diagram of the compliance matrix between the suspension arm-tips $[G_s]$, transfer characteristics from the tire ground-contact point to the suspension arm-tips $\{G_{st}\}$, and road input D is shown in Fig. 3.

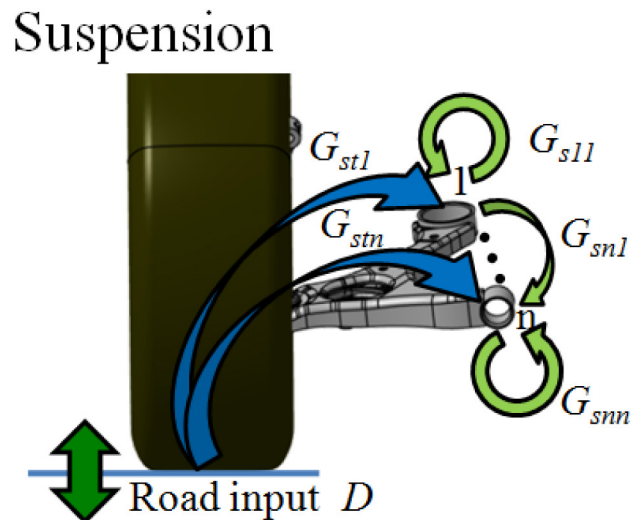


Figure 3. Schematic diagram of suspension compliance matrix and transfer function from tire ground-contact point to connecting point

Next, if the dynamic stiffness of the bush, which joins the suspension at some joint j , is denoted as K_j , then based on Hooke's law, the suspension transfer force $\{F\}$ can be expressed as the following equation, using the relative displacement of the body's suspension mounting point displacement $\{X_b\}$ and suspension arm-tip displacement $\{X_s\}$.

$$\begin{Bmatrix} F_1 \\ \vdots \\ F_n \end{Bmatrix} = \begin{bmatrix} K_1 & & 0 \\ & \ddots & \\ 0 & & K_n \end{bmatrix} \left(\begin{Bmatrix} X_{s1} \\ \vdots \\ X_{sn} \end{Bmatrix} - \begin{Bmatrix} X_{b1} \\ \vdots \\ X_{bn} \end{Bmatrix} \right) \quad (3)$$

Using Eqs. (1), (2), (3) above, the following equation is derived by compiling the suspension transfer force in the body $\{F\}$.

$$\begin{Bmatrix} F_1 \\ \vdots \\ F_n \end{Bmatrix} = - \left\{ \begin{bmatrix} G_{b11} & \cdots & G_{b1n} \\ \vdots & \ddots & \vdots \\ G_{bn1} & \cdots & G_{bnn} \end{bmatrix} + \begin{bmatrix} K_1 & & 0 \\ & \ddots & \\ 0 & & K_n \end{bmatrix}^{-1} + \begin{bmatrix} G_{s11} & \cdots & G_{s1n} \\ \vdots & \ddots & \vdots \\ G_{snn} & \cdots & G_{snn} \end{bmatrix} \right\}^{-1} \begin{Bmatrix} G_{st1} \\ \vdots \\ G_{stm} \end{Bmatrix} D \quad (4)$$

Going further, the following equation is derived by compiling the middle of the inverse matrix of the above equation.

$$\begin{Bmatrix} F_1 \\ \vdots \\ F_n \end{Bmatrix} = - \left\{ \begin{bmatrix} G_{b11} + K_1^{-1} + G_{s11} & \cdots & G_{b1n} + G_{s1n} \\ \vdots & \ddots & \vdots \\ G_{bn1} + G_{snn} & \cdots & G_{bnn} + K_n^{-1} + G_{snn} \end{bmatrix} \right\}^{-1} \begin{Bmatrix} G_{st1} \\ \vdots \\ G_{stm} \end{Bmatrix} D \quad (5)$$

If a matrix that adds together the body's suspension mounting point compliance matrix $[G_b]$, the inverse of the bush's dynamic stiffness $[K^{-1}]$, and the suspension arm-tips compliance matrix $[G_s]$ is defined as the total mounting point compliance matrix $[G_{total}]$, then the following equation is applied.

$$[G_{total}] = \begin{bmatrix} G_{b11} + K_1^{-1} + G_{s11} & \cdots & G_{b1n} + G_{s1n} \\ \vdots & \ddots & \vdots \\ G_{bn1} + G_{snn} & \cdots & G_{bnn} + K_n^{-1} + G_{snn} \end{bmatrix} \quad (6)$$

Then, Eq. (4) becomes Eq. (7).

$$\begin{Bmatrix} F_1 \\ \vdots \\ F_n \end{Bmatrix} = -[G_{total}]^{-1} \begin{Bmatrix} G_{st1} \\ \vdots \\ G_{stm} \end{Bmatrix} D = -[G_{total}]^{-1} \{G_{st}\} D \quad (7)$$

Based on Eq. (7), the suspension transfer force $\{F\}$ is determined by the total mounting point compliance matrix $[G_{total}]$, transfer characteristics from the tire ground-contact points to the suspension arm-tips in the suspension $\{G_{st}\}$, and road input D . Moreover, the above shows that even if road input D and transfer characteristics from the tire to the suspension arm-tips $\{G_{st}\}$ do not change, the suspension transfer force will change if the total mounting point

compliance matrix $[G_{total}]$ changes. In other words, the suspension transfer force changes because it is affected by changes in the body's suspension mounting point compliance, the bush's dynamic stiffness, and the suspension's arm-tip compliance.

Concept of Road Noise Reduction

Denoting the acoustic sensitivity of the body from the suspension mounting points to the driver ear-point as $[H_b]$, sound pressure at the driver ear-point $\{P\}$ can be expressed as the following equation.

$$\begin{aligned} \{P\} &= [H_b] \{F\} \\ &= -[H_b][G_{total}]^{-1} \{G_{st}\} D \end{aligned} \quad (8)$$

To simplify the discussion, in this chapter, a case that was jointed in only one bushing with one degree of freedom was investigated. A schematic diagram is given in Fig. 4.

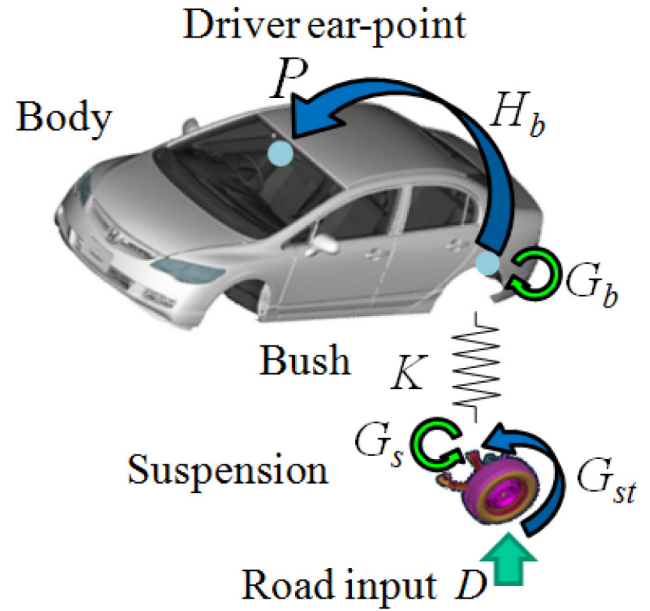


Figure 4. Schematic diagram of the car jointed in one bushing

The suspension transfer force F in this case can be expressed as follows:

$$\begin{aligned} F &= -(G_b + K^{-1} + G_s)^{-1} G_{st} D \\ &= \frac{-G_{st} D}{G_b + K^{-1} + G_s} \end{aligned} \quad (9)$$

The sound pressure P at driver ear-point in this case is as follows:

$$P = H_b F = \frac{-H_b \cdot G_{st} D}{G_b + K^{-1} + G_s} \quad (10)$$