



Optimization of the lower arm of a vehicle suspension system for road noise reduction by sensitivity analysis



Yong-Dae Kim^a, Jae-Eun Jeong^a, Jin-Su Park^a, In-Hyung Yang^a, Tae-Sang Park^a,
Pauziah Binti Muhamad^b, Dong-Hoon Choi^c, Jae-Eung Oh^{d,*}

^a Graduate School of Mechanical Engineering, Hanyang University, 216 Engineering Center, Hanyang University, Haengdang-dong, Seongdong-gu, Seoul 133-791, Republic of Korea

^b Mechanical Precision Engineering Department, Universiti Teknologi Malaysia Kuala Lumpur, Jalan Semarak, 54100 Kuala Lumpur, Malaysia

^c School of Mechanical Engineering, Hanyang University, 209 Engineering Center, Hanyang University, Haengdang-dong, Seongdong-gu, Seoul 133-791, Republic of Korea

^d School of Mechanical Engineering, Hanyang University, 211-1 Engineering Center, Hanyang University, Haengdang-dong, Seongdong-gu, Seoul 133-791, Republic of Korea

ARTICLE INFO

Article history:

Received 16 February 2013

Received in revised form 19 June 2013

Accepted 24 June 2013

Available online 10 July 2013

Keywords:

Vehicle suspension system

Road noise

Sensitivity analysis

Optimization

Operation transfer path analysis

Response surface method

ABSTRACT

This study analyzed characteristics of road noise using vehicle tests and identified the 200–230 Hz range as the most important frequency for road noise reduction. Moreover, vibration sources in the vehicle suspension system were identified through transfer path analysis and coherence analysis. Using a finite element model of a vehicle suspension system, sensitivity analysis was performed to determine sensitive design factors. In order to achieve noise reduction using sensitivity analysis, the lower arm of the vehicle suspension system was found to be the most important design variable. For design optimization, we employed a robust and efficient sequential approximate optimization method, named PQRS (Progressive Quadratic Response Surface Method) suitable for solving practical design optimization problems. The estimates based on a model proposed from optimization were in accord with the results of the experiment and road noise reduction was achieved by applying the optimally designed lower arm of the vehicle suspension system to a real vehicle.

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1. Introduction

As advances in automotive design and manufacturing technology have emerged, the demand for noise reduction and ride quality improvement has increased. In particular, under normal driving conditions, drivers are most exposed to road noise, which can be unpleasant. Furthermore, it can reduce ride quality and brand awareness. Noises within the passenger compartment are categorized as structural and air-borne noises, depending on the transfer path characteristics. Road noise is the most common structural noise and is caused by road excitation to the tires in the medium- to low-frequency range (0–500 Hz). This energy is transferred to the vehicle suspension system [1]. Thus, each component of the vehicle suspension system should work to isolate against road vibrations, this requires various studies to improve vehicle suspension systems and reduce road noise [2–4].

Road noise transferred to the passenger compartment has a complex vibration mechanism and multiple vibration sources, making it difficult to develop precise countermeasures. In particular, when road vibration is transmitted to the vehicle suspension system and car frame, the vibration generated from each element in the transfer path, as well as the specific transfer path characteristics, is modeled in multiple input and single output systems to perform effective noise reduction strategies using

* Corresponding author. Tel.: +82 2 2294 8294; fax: +82 2 2299 3153.

E-mail addresses: ydkim200@naver.com (Y.-D. Kim), jjaeui@naver.com (J.-E. Jeong), jinsu96@naver.com (J.-S. Park), kornblind@naver.com (I.-H. Yang), tag1987@hanmail.net (T.-S. Park), pauziah@ic.utm.my (P.B. Muhamad), dhchoi@hanyang.ac.kr (D.-H. Choi), jeoh@hanyang.ac.kr (J.-E. Oh).

contribution estimations. Moreover, it is critical for the designer to have detailed information on vibration sources to reduce noise in the passenger compartment. This information will help save a lot of time and effort compared to the traditional method of relying on experience and intuition. In this respect, this study uses sensitivity analysis to develop a new vibration isolation design in order to identify the relationship between design variables and objective functions [5,6]. In terms of design sensitivity, Haftka and Adelman reported results on design sensitivity in discrete systems up to the late 1980s [7], whereas in 1968, Fox and Kapoor reported changes in the eigenvalue and eigenvector of the symmetric matrix [8]. C.C. Hsieh and J. S. Arora set up a first-order differential equation for mechanical structures to present a sensitivity analysis method using direct differentiation and an adjoint variable method [9]. However, these methods require a precise understanding of the design object and complicated kinematic equations. Thus, FE analysis is needed for simpler applications.

This study performed a vehicle driving test to analyze road noise characteristics and identified the frequency appropriate for noise reduction. Moreover, this study also developed a model of road noise delivered to the passenger compartment using multiple input and single output systems to quantify coherent output estimates through transfer path analysis (operation transfer path analysis: OTPA). For components that serve as the vibration source, we performed finite element analysis using Abaqus and sensitivity analysis using direct differentiation [10,11] to identify important design variables. To obtain the optimum values of the important design variables identified, we employed a robust and efficient sequential approximate optimization method, named PQRS (Progressive Quadratic Response Surface Method), suitable for solving practical design optimization problems [12]. Then, we applied an optimally designed lower arm of the vehicle suspension system to a real car to assess noise reduction and compared the estimates to the actual test results.

2. Sensitivity analysis and design optimization method

2.1. Sensitivity analysis using direct differentiation

Design sensitivity analysis refers to the changes in response function after changes in design variables. Direct differentiation for sensitivity analysis is direct differentiated eigenvalue of model with design variables. The following is the normalization equation and the eigenvalue of the discrete model.

$$[K]\{u\} = \lambda[M]\{u\} \quad ([K] : \text{stiffness}, \{u\} : \text{eigen vector}, [M] : \text{mass}) \tag{1}$$

$$\{u\}^T [M] \{u\} = 1 \tag{2}$$

When Eq. (1) is directly differentiated with design variable b and transposed to the left, the following equation results

$$\frac{\partial [K]}{\partial b} \{u\} + [K] \frac{\partial \{u\}}{\partial b} - \frac{\partial \lambda}{\partial b} [M] \{u\} - \lambda \frac{\partial [M]}{\partial b} \{u\} - \lambda [M] \frac{\partial \{u\}}{\partial b} = 0. \tag{3}$$

Inputting design variable u^T to Eq. (3) results in Eq. (4).

$$\{u\}^T \frac{\partial [K]}{\partial b} \{u\} + \{u\}^T [K] \frac{\partial \{u\}}{\partial b} - \{u\}^T \frac{\partial \lambda}{\partial b} [M] \{u\} - \{u\}^T \lambda \frac{\partial [M]}{\partial b} \{u\} - \{u\}^T \lambda [M] \frac{\partial \{u\}}{\partial b} = 0. \tag{4}$$

Finally, Eq. (4) becomes Eq. (5).

$$\{u\}^T \left(\frac{\partial [K]}{\partial b} - \lambda \frac{\partial [M]}{\partial b} \right) \{u\} + \{u\}^T ([K] - \lambda [M]) \frac{\partial \{u\}}{\partial b} - \{u\}^T \frac{\partial \lambda}{\partial b} [M] \{u\} = 0 \tag{5}$$

Eigenvalue design sensitivity calculated using Eqs. (1) and (5) is as follows:

$$\frac{\partial \lambda}{\partial b} = \{u\}^T \left(\frac{\partial [K]}{\partial b} - \lambda \frac{\partial [M]}{\partial b} \right) \{u\}. \tag{6}$$

The eigenvalue design sensitivity can be calculated using the equation found by differentiating the normalization condition equation and Eq. (6) and then forming the matrix formula. The differentiation equation of the normalization condition equation is as Eq. (7) and the matrix formula is as Eq. (8).

$$2\{u\}^T [M] \frac{\partial \{u\}}{\partial b} + \{u\}^T \frac{\partial [M]}{\partial b} \{u\} = 0 \tag{7}$$

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