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Analysis of the effects of main design parameters variation on the vibration characteristics of a vehicle sub-frame †

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Abstract

In the design process of an automobile part, several analysis methods are usually used to evaluate the performance of the part. However, most automobile design engineers do not directly use CAE (computer aided engineering) tools since specific skills are required to obtain practical results. Moreover, CAE requires a huge amount of computation time and cost. To resolve these problems, a new design approach, termed first order analysis (FOA), has been proposed. In this paper, the FOA technique is employed to design a vehicle sub-frame. An equivalent model of the vehicle sub-frame which only consists of beam elements is proposed and the modal properties obtained with the model are compared to those obtained with a full scale finite element model. The effects of some parameter variations on the modal characteristics of the vehicle sub-frame are investigated by employing the FOA equivalent model.

Keywords: FOA equivalent model; Variation analysis; Vehicle sub-frame; Vibration characteristics

1. Introduction

Analysis based on the finite element method technique(FEM) plays a major role in virtual engineering for automotive design without using prototypes. Due to the fast spread of CAE (computer aided engineering) and the improvement of its function over the past few years, it is possible to quantitatively estimate the performance of an automobile before actually building a prototype. However, the FE model is based on a CAD model so engineers cannot develop a revolutionary shape or make drastic structural modifications using this detailed mesh model. Furthermore, CAE requires a huge amount of time and special knowledge to construct the analysis model.

To overcome these drawbacks, a new design approach, FOA (first order analysis), has been proposed by Nishigaki [4] and Yasuaki [6]. The FOA technique is a very quick and easy way of deciding and analyzing structures in a conceptual design phase. Furthermore, it is a CAE tool for design engineers that can be used with no special knowledge or skills in modeling or analysis.

In this paper, the FOA technique is employed to design a vehicle sub-frame. An equivalent model of the vehicle sub-frame that only consists of beam elements is proposed and the accuracy of FOA model is verified. And then the effects of some parameter variation on the modal characteristics of the vehicle sub-frame are investigated.

2. FOA equivalent model

Most medium class passenger cars usually have adopted #-type front sub-frame. The principal purpose of using a front sub-frame is to isolate vibration and harshness from the rest of the body.

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Fig. 1. Finite element model of sub-frame.

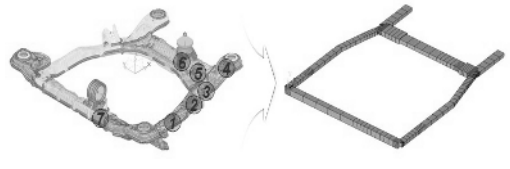

Fig. 2. FOA equivalent model of sub-frame.

Fig. 1 shows the finite element model of #-type front sub-frame. The #-type front sub-frame consists of total eight subparts, that is, upper and lower cross member, left and right side members, A and G-point brackets.

An equivalent model of the vehicle sub-frame that only consists of a beam element is constructed based on the FOA technique, as shown in Fig. 2. The #-type front sub-frame is a symmetric structure except for front and rear engine mounts. So, by dividing the finite element model into seven pairs of symmetric sides, a total of fourteen areas, the section properties, such as cross-sectional area, area moment of inertia and shear coefficient are extracted for constructing an FOA equivalent model.

3. Verification of FOA model

To verify the accuracy of the FOA model that only consists of beam elements, the modal properties obtained with the model are compared to those obtained with a full scale finite element model.

3.1 Mass and stiffness matrix

The geometry and the coordinate systems employed for a beam element used for modeling are shown in Fig. 3. The element stiffness matrix of a beam element having arbitrary orientation in space is given as follows:

$$\hat{\mathbf{k}}^e = \begin{bmatrix} \hat{K}_{11} & \hat{K}_{12} \\ \hat{K}_{21} & \hat{K}_{22} \end{bmatrix} \tag{1}$$

where the 6×6 sub-matrix is comprised of bending,

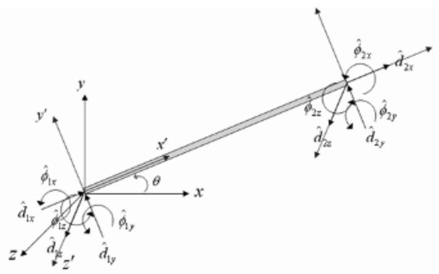

Fig. 3. Three dimensional beam element model.

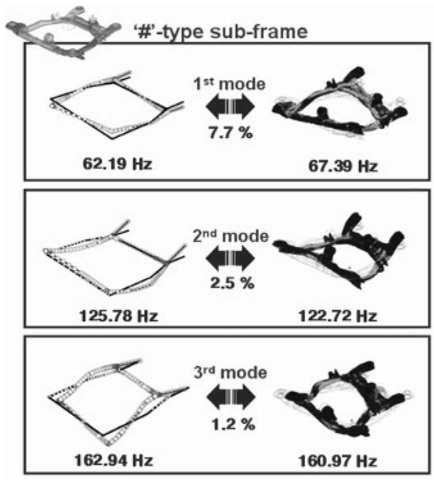

Fig. 4. Comparison of natural frequencies and mode shapes between FOA equivalent model and finite element model.

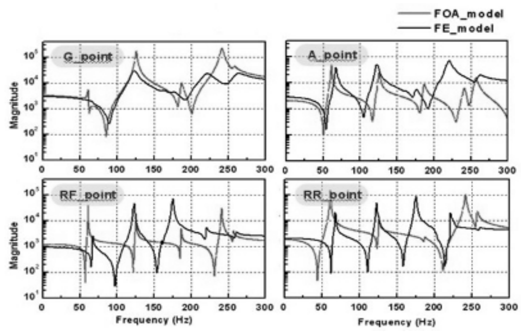

Fig. 5. Comparison of FRF results obtained by FOA equivalent model and finite element model.

axial and torsional stiffness matrices by direct superposition.

The element mass matrix for the beam element is given by

$$\hat{\mathbf{m}}^e = \iiint_V \rho \mathbf{N}^T \mathbf{N} \, dV \tag{2}$$

In this study, a consistent mass matrix, which is derived from the same shape functions that are used to obtain the stiffness matrix, is used for modeling.

3.2 Verification of effectiveness of FOA model

To check the accuracy of the FOA equivalent model, the natural frequency, mode shape and FRFs are compared with each other.

First, we calculate the natural frequencies. As shown in Fig. 4, the values of natural frequencies are in good agreement only within 10 percent error. And the mode shapes obtained with the FOA modeling method are in good agreement with those obtained with the finite element.

Fig. 5 shows the frequency response function according to the variation of input/output point in the 300Hz range. The frequencies where a peak is generated agree with the modal analysis results within 10 percent and the magnitude of the overall response also has the same tendency.

4. Analysis of variation

From the FOA equivalent model, vibration and variation analysis can be done. The effects of some parameters' variation on the modal characteristics of the vehicle sub-frame are investigated by employing the FOA equivalent model.

4.1 Variation analysis

The deviation of the performance index can be obtained if the sensitivity of the performance index is multiplied by the deviation of a design variable b as shown below :

$$\delta g = \left(\frac{\partial g}{\partial b}\right)\delta b \tag{3}$$

Since the normal distribution has infinite range, the 3-sigma span of a dimension is taken as the variation with 99.73% confidence. Then, the deviation of the performance index can be expressed as follows [2].

$$\delta g = \frac{1}{3}\left(\frac{\partial g}{\partial b}\right)T \tag{4}$$

where T denotes the variation of the design variable b.

4.2 Sensitivity equations

The sensitivity of the natural frequency with respect to the design variable can be obtained in an analytical way. The eigenvalue problem of an undamped system can be stated as follows:

$$\hat{\mathbf{K}}\varphi_j = \lambda_j \hat{\mathbf{M}}\varphi_j \tag{5}$$

Differentiating Eq. (5) with respect to a design variable b and pre-multiplying Eq. (5) by φ_j^T, and employing the normalized condition, the sensitivity of the eigenvalue for the design variable can be obtained as follows [5]:

$$\frac{\partial \lambda_j}{\partial b} = \varphi_j^T \left(\frac{\partial \hat{\mathbf{K}}}{\partial b} - \lambda_j \frac{\partial \hat{\mathbf{M}}}{\partial b}\right)\varphi_j \tag{6}$$

Meanwhile, the frequency response function can be expressed as a receptance matrix shown below [3].

$$\alpha(\omega) = \sum_{i=1}^{n}\left[\frac{\mathbf{u}_i \mathbf{u}_i^T}{(\omega_i^2 - \omega^2) + (2\xi_i \omega_i \omega) j}\right] \tag{7}$$

Since $\alpha(\omega)$ is a matrix and each element is a transfer function, a simple finite difference method is employed to obtain the sensitivity equations of frequency response function.

4.3 Results of variation analysis

Fig. 6 shows the standard deviations of the first natural frequency versus the variation for various design variables. The effects of design variable variation on the first natural frequency are quite different from each other. The variation of G lower arm mount location has the biggest effect on the first natural frequency deviation. Next, the standard deviations of the first natural frequency with the layout of the sub-frame variation, such as length of the side frame and the cross frame are obtained. The effects of length of the side and cross frame variation are much bigger than those of the lower arm and engine mount locations.

Fig. 7 shows the variation of standard deviation of the frequency response function when lower arm / engine mount location and the length of the side and cross frame have 1~10% variations. The frequency response function is compared at the 400Hz; other frequencies also yielded similar results. The variation of rear engine mount location has the biggest effect on the frequency response deviation, and the effects of the layout of sub-frame varia-

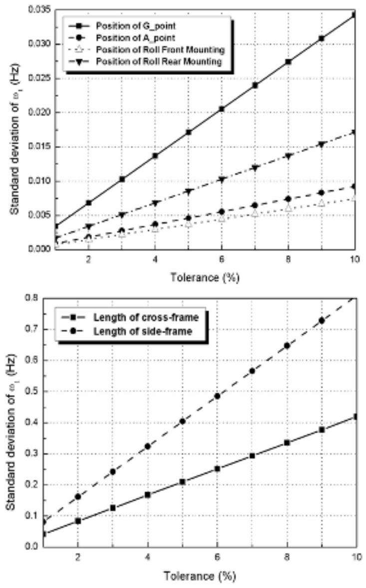

Fig. 6. Standard deviations of the first natural frequencies for various variations.

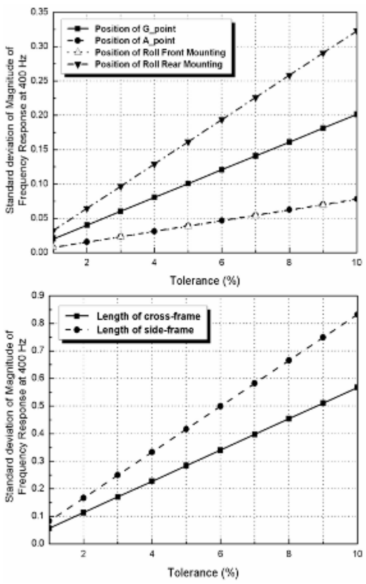

Fig. 7. Standard deviations of the frequency response functions for various variations.

tion have a relatively big impact, in line with the variation analysis results of first natural frequency.

5. Conclusions

An equivalent model of the vehicle front sub-frame that only consists of beam elements has been proposed and the effect of some parameters variation on the modal characteristics of the vehicle sub-frame are investigated. The results of variation analysis show that the variation of the overall layout of the sub-frame has a bigger effect than the mount location. The variation of G lower arm mount location has a great effect on the first natural frequency deviation, while the variation of rear engine mount location has the biggest effect on the frequency response function deviation.

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