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A method for estimating mount isolations of powertrain mounting systems

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ABSTRACT

A method for calculating isolation ratios of mounts at a powertrain mounting systems (PMS) is proposed assuming a powertrain as a rigid body and using the identified powertrain excitation forces and the measured IPI (input point inertance) of mounting points at the body side. With measured accelerations of mounts at powertrain and body sides of one Vehicle (Vehicle A), the excitation forces of a powertrain are identified using conversational method firstly. Another Vehicle (Vehicle B) has the same powertrain as that of Vehicle A, but with different body and mount configuration. The accelerations of mounts at powertrain side of a PMS on Vehicle B are calculated using the powertrain excitation forces identified from Vehicle A. The identified forces of the powertrain are validated by comparing the calculated and the measured accelerations of mounts at the powertrain side of the powertrain on Vehicle B. A method for calculating acceleration of mounting point at body side for Vehicle B is presented using the identified powertrain excitation forces and the measured IPI at a connecting point between car body and mount. Using the calculated accelerations of mounts at powertrain side and body side at different directions, the isolation ratios of a mount are then estimated. The isolation ratios are validated using the experiment, which verified the proposed methods for estimating isolation ratios of mounts. The developed method is beneficial for optimizing mount stiffness to meet mount isolation requirements before prototype.

1. Introduction

1.1. Motivations

Nosie, Vibration and Harshness (NVH) are one of top five attributes for a passenger car, and the NVH from powertrain is one of main source of NVH in automobile, especially if a car is at idle or cruising with high speed [1–4]. One method to reduce or isolate vibration from powertrain is to optimize the mounts at a powertrain mounting system (PMS) [5–7]. A PMS consists of a powertrain (engine and transmission) and three or four mounts. One of the important evaluation indexes of a PMS is the isolation ratio defined as a ratio between mount accelerations at powertrain side and body side at three perpendicular direction and is defined as:

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$$T = 20\log(a^e/a^c) \tag{1}$$

where $a^e$and a^c are acceleration amplitudes of a mount in one direction at powertrain side and body side, respectively. The isolation ratios of mounts is called "mount isolations" in this paper.

To the best of author knowledge's, there are few methods or researches involving the estimation of mount isolations ratios before prototype of mounts. Mount isolations are usually obtained by measurement after prototype of mounts and car body if a powertrain is specified. To meet the required mount isolations, the mount stiffness are usually adjusted by trial and error, which increases product development period and costs [8]. Difficulties for estimating mount isolations are to identify powertrain excitation forces, to obtain the IPI (input point inertance) of car body at the connecting point between mount and body and calculate the dynamic stiffness of mounts.

Now, in order to reduce development cost, one powertrain is usually equipped for different cars with different bodies, mount stiffness and mount locations. In design of a PMS, the mount isolations at different directions are required and usually greater than 20 dB [9]. If a car with one powertrain is prototyped and mount isolations are measured, is it possible to estimate mount isolations when the powertrain is installed in another car and IPI of the car and mount stiffness are known?

So the motivations of this study are to develop methods or procedures to calculate mount isolations before mount prototype to reduce design period and cost for the PMS.

1.2. Literature review and objectives

The mount isolations in each direction currently are rarely analytically calculated in the initial design stage, and it is usually obtained by measurement after the powertrain has been mounted on the car body. In Ref. [8], an experimental method for optimizing the mount stiffness and locations to minimize the accelerations of mounts at body side is proposed. In Ref. [10], to reduce the vibration for a city bus, the PMS is modeled as 6 DOFs (degrees of freedom) and location of mounts are optimized by using the decoupling techniques and minimizing the mount transmitted forces. Ref. [8] does the same work as Ref. [10], but the method in Ref. [10] is more close to the engineering applications since optimal the locations of mounts meet the mount isolations greater than 20 dB and the NVH requirements of a car [9].

The excitation forces of a powertrain are important input parameters for estimating mount isolations. The excitation forces of a powertrain are defined as the forces and moments applied at one coordinate system connected with the powertrain and its origin is located at the center of gravity (CG) of the powertrain, with the Z-axis along the piston movement and the Y-axis along the crankshaft if the powertrain is transverse positioned. Identification of excitation forces is a typical inverse problem in vibration analysis and has been paid much attention [11–14]. Methods to obtain excitation forces of powertrain presented in Refs. [15–20] are classed as direct method and indirect method.

If direct method is used, the excitation forces of the engine of a powertrain are obtained from the engine dynamic model using input parameters, such as gas pressure at each cylinder, mass and moment of inertias for different parts. In Ref. [15], to optimize the transmitted force from the engine to the car body, the excitation forces of three-cylinder engine are calculated by the values of these parameters. The main drawback of direct method is that it requires the accurate values [16] and excitation forces from transmission are not included.

Indirect method is based on the frequency response of the powertrain [17] or the dynamic equations of force re-construction [18–20]. For the indirect method based on the frequency response transfer function (FRF), the response of powertrain is measured at different points to build transfer function. In Ref. [17], a method of identifying excitation forces for a single cylinder outboard engine is presented. A set of equivalent forces vectors are formulated by combining the dynamic motion of piston-crank mechanism and the FRF matrix inversion. The equivalent force is transformed to the point at the intersection between the center line of the crankshaft and connection rod. Because the identified forces and moments are applied at the CG of a powertrain, the method based FRF analysis is difficult to identify the excitation forces and moments. In this method, because the inverse of FRF must be estimated, this will amplify the errors for the identified forces since measurement errors of FRF [13,14].

Indirect method is based on the dynamic equations of force re-construction, and the dynamic equations of a powertrain with 6 DOFs are established firstly assuming the car body as a rigid or flexible body. Then, using the measured velocity or acceleration (amplitude and phase) at all mounting points, excitation forces at the CG of a powertrain are identified [18–20]. In Ref. [18], based on an assumption that the phase of excitation forces at the CG are zeroes, a method of identifying excitation forces for an engine is proposed. In Ref. [19], using the identified excitation forces and admittance matrix, a method of optimizing the stiffness of mounts is presented by minimizing transfer power from the engine to the car body. In Ref. [20], considering the analysis error of amplitude and phase of accelerations or velocities due to the discrete spectrum analysis method, the interpolation method of discrete spectrum is applied to the engine excitation force identification.

In calculating the accelerations mounts at the body side, the car body is assumed as a flexible body. A coupled dynamic equations between the PMS and car body is constructed by modelling car body as flexible body and using IPI at a connecting point between car body and mount.

The IPI is an index for describing flexibility of a point, and is defined as acceleration at one point over the corresponding force at one point. One of method to obtain the IPI is to calculate the dynamic stiffness at a connecting point between car body and mount using FEA [21,22]. In Refs. [21,22], the coupled equations between powertrain and car body are constructed using