

Optimization Design of a Six-Point Powetrain Mounting System with Flexible Support Rod

Zhengfei Tang and Yongfu Chen
Huazhong University of Science and Tech.

Jian Zeng and Yu Yang
C&C Trucks Co., Ltd.

Yunqing Zhang
Huazhong University of Science and Tech.

ABSTRACT

NVH quality is one of the most important criteria by which people judge the design of a vehicle. The Powertrain Mounting System (PMS), which can reduce the vibration from engine to vehicle cab as well as the inside noise, has attained significant attention. Much research has been done on the isolation method for three- and four-point mounting. But the six-point mounting system, which is usually equipped in commercial vehicle, is seldom studied and should be paid more attention.

In this paper, the support rod installed on the upside of the transmission case is considered as a flexible body. Thus a rigid-flexible coupling model of PMS is established and the necessity of the established model is analyzed by comparing the simulation results of the new model and those of the conventional model. Based on the traditional theory of energy decoupling and reasonable allocation of the natural frequencies, Adams and MATLAB are integrated into the optimization software iSIGHT to optimize the six-point PMS. The orthogonal experiment is used to determine the main factors that affect the optimization objective and the initial design parameters for further optimization. The Mixed Integer Sequential Quadratic Programming (MISQP) is used to obtain an optimal model of PMS. Finally the dynamic frequency response method is used to validate the optimal model.

CITATION: Tang, Z., Chen, Y., Zeng, J., Yang, Y. et al., "Optimization Design of a Six-Point Powetrain Mounting System with Flexible Support Rod," *SAE Int. J. Commer. Veh.* 7(1):2014, doi:10.4271/2014-01-1682.

INTRODUCTION

The powertrain is the main source of noise and vibration in a vehicle. Whether a powertrain mounting system is well designed directly affects the vibration characteristics and riding comfort. To make passengers feel more comfortable, engineers have introduced an isolation system between engine and frame. A typical isolation system consists of springs and dampers, and their properties are determined to minimize the transmissibility of the vibration.

To improve efficiency, the PMS is usually designed before the full vehicle design has been finished. Designers have developed many efficient method based on sub structuring of the PMS assuming the frame as a fixed rigid body [1].

In analyzing low-frequency vibrations of a powertrain under dynamic force and torque excitations, most studies have regarded the mounts in a PMS as rubber mounts [2, 3]. An analysis and design method was developed by Ishihama et al. [4] to control powertrain vibrations in pitch and bounce modes during low-frequency vibrations. They used a simple model with two degrees of freedom (DOF) to simulate the vibration characteristics of a PMS. However, the 2-DOF model seems to be too simple to simulate the performance of a PMS [5, 6].

Several powertrain mounting schemes have been developed to improve NVH properties concentrating on the positioning and design of resilient supports. However these methods are based on decoupling rigid body modes from a grounded powertrain model which ignores chassis and suspension system interactions. Courteille, E. [7] built a PMS model taking the chassis into consideration. Che et al. [8] considered the

compliance of the body and the suspension systems and performed the design optimization based on the whole vehicle system.

There are generally two approaches in the design optimization of the PMS. The first one is to decouple the roll mode from the others to reduce the transmissibility and isolate the rigid body modes from engine idle frequency [9]. Another is to reduce the acceleration levels directly at the response positions of interest [10].

Liu et al. [11] derived computational expressions of natural frequencies and kinetic energy distributions and investigated the optimization of the PMS. Shangguan et al. [9] developed a method for dynamic analysis and design calculation of a PMS with the aim of controlling powertrain motion and reducing low-frequency vibrations in rotation around the crankshaft and bounce along the vertical direction.

The literature in this area indicates that the six-point mount system is seldom studied and most authors have ignored the effects of the support rod in their optimization.

The main goal of this work is to introduce a method and procedure for optimizing the characteristics of PMS using Adams and iSIGHT. This article is organized as follows. In section 2, the rigid-flexible coupling PMS is established in Adams. Then the design variables are selected by orthogonal experiment analysis in section 3. The objective is the weighted sum of decoupling ratios. In section 4, the Mixed Integer Sequential Quadratic Programming (MISQP) [12] is used to obtain an optimal PMS. In section 5, the frequency response method is employed to validate the optimal design results. Some conclusions are shown in the last section.

THE SIMULATION MODEL BASED ON ADAMS FOR OPTIMIZATION

Powertrain Mounting System Model

The simplified model of a longitudinally placed PMS with four mounts is shown in figure 1. The powertrain is assumed as a 6-DOF rigid body which is composed of an inertial body (engine and transmission), a rigid base (chassis) and four undamped mounts. A mount is simplified as spring elements in three orthogonal directions determined by the direction of each mount's prime elastic axis. A right-hand Global Coordinate System (GCS), G_0 -XYZ, which has its origin at the C.G. of the powertrain, is built to describe the movements of the powertrain and the mounts' locations and orientations. The three orthogonal coordinate axes are set with Y- and Z- axes parallel to the crankshaft and axis of the piston cylinder respectively, X- axis normal to the YG_0Z plane, and the positive direction of Y- axis points to the front of a vehicle. A Local Coordinate System (LCS) Q_i - $u_i v_i w_i$, is built for each mount, where the origin is at the connecting point of the mount and the

powertrain, and the three coordinate axes are expressed by u_i , v_i and w_i ($i=1,2,\dots,n$, where n is the number of mounts), respectively.

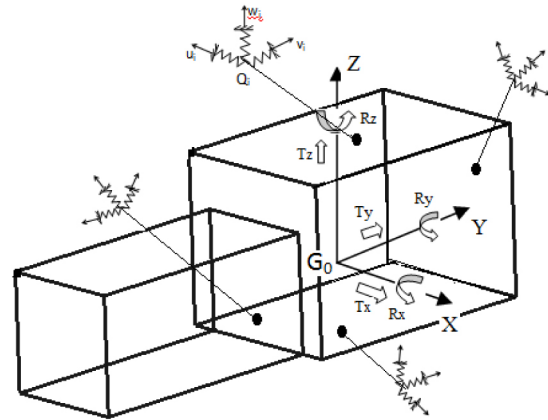


Figure 1. 6-DOF model of PMS

Kinetic Energy Distribution Theory

According to the general dynamic model, the vibration differential equation of powertrain mounting system could be represented as equation (1) when the external force and the damp of the mounts are assumed as zero.

$$[M]\{\ddot{X}\} + [K]\{X\} = \{0\} \quad (1)$$

$[M]$ is the (6×6) mass matrix containing inertia parameters of the powertrain, $[K]$ is the (6×6) stiffness matrix and is a function of the dynamic stiffness, locations and orientations of the mounts. $\{X\}$ is (6×1) displacement vector representing translation and angle displacements of the six DOFs of the powertrain with respect to its centre of gravity (C.G.) in GCS. The mass and stiffness matrix $[M]$ and $[K]$ can be calculated by design parameters including mass characteristics of the engine and the locations, orientations, stiffness of the mounts.

From equation (1) we can obtain the following equation:

$$[M]^{-1}[K]\{X_i\} = \omega_i^2 \{X_i\} \quad (2)$$

Where ω_i and $\{X_i\}$ ($i=1, 2, \dots, 6$) are the i -order natural circular frequency and mode shape of the PMS vibration model. So the natural frequency $f_i (= \omega_i / 2\pi, i=1, 2, \dots, 6)$ and mode shape $\{X_i\}$ can be obtained by calculating the eigenvalue and eigenvector of $[M]^{-1}[K]$.

When the powertrain vibrates at a natural frequency (f_i), the mode kinetic energy distribution $E(n,i)$ (usually defined as the decoupling ratio) in the n th ($n=1, 2, \dots, 6$) DOF of the powertrain and i -order mode is calculated by equation (5) [10], where T^i is

the kinetic energy of i -order vibration and T_n^i is the kinetic energy distributed in the n ($n=1,2 \dots 6$) DOF of i order vibration. T^i and T_n^i can be expressed as following equations:

$$T^i = \omega_i^2 \{X_i\}^T [M] \{X_i\} / 2$$

$$= \omega_i^2 \sum_{l=1}^6 \sum_{n=1}^6 (X_i)_l (X_i)_n m_{nl} / 2 \quad (3)$$

$$T_n^i = \omega_i^2 \sum_{l=1}^6 (X_i)_n (X_i)_l m_{nl} / 2 \quad (4)$$

Where $(X_i)_n$ is the n th element in the X_i , m_{nl} is the element of the mass matrix, $[M]$, in n - row and l -column.

$$E(n,i) = \frac{T_n^i}{T^i} = \frac{\sum_{l=1}^6 (X_i)_n (X_i)_l m_{nl}}{\sum_{l=1}^6 \sum_{n=1}^6 (X_i)_l (X_i)_n m_{nl}} \times 100\% \quad (5)$$

Rigid-Flexible Coupling PMS Model

The conventional three- or four-point PMS is modeled by connecting a rigid body to the rigid ground by rubber mounts. However, the six-point PMS with two added mounts, which is usually placed on the upside of the transmission, do not apply to the above traditional process of simplification. Because the transmission is connected to a support rod rather than directly to the rigid chassis frame, and the rod is so flexible that it can't be assumed as a rigid body.

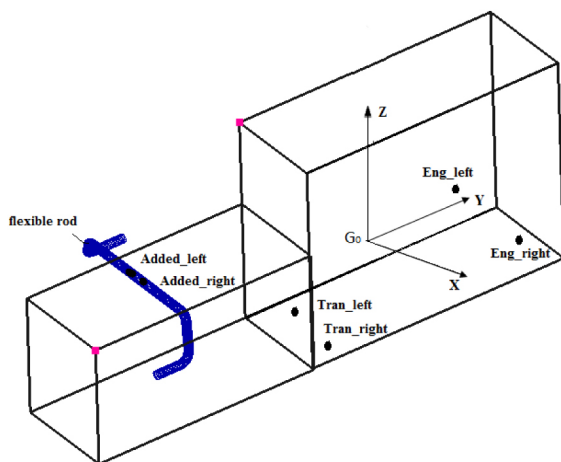


Figure 2. Rigid-flexible coupling model of PMS

To ensure the deviation of the PMS model is controlled within tolerable level, a rigid-flexible coupling PMS model, shown in Figure 2, is built in Adams. The rod model is generated in NASTRAN using meshing element and imported into the existing PMS model built in Adams via the mnf file.

The natural frequency and decoupling ratio have been calculated based both on the conventional model and the new flexible-rigid model with all the other parameters consistent. The powertrain weights 1526 Kg, and its inertia properties are given in Table 1. The location, orientation and dynamic stiffness of each mount are listed in Tables 2, 3 and 4 respectively.

Table 1. Inertia properties of a powertrain in GCS (Kg.m²)

m(kg)	lxx	lyy	lzz	lxy	lyz	lzx
1526	594.24	79.97	557.89	7.19	-69.46	-1.47

Table 2. Location of the mount in GCS(mm)

mount	Coordinates in GCS (mm)		
	X	Y	Z
Eng_left	-303	-548	-244
Eng_right	303	-548	-244
Tran_left	-357	474	-158
Tran_right	357	474	-158
Added_left	-63	1196	-63
Added_right	63	1196	-63

Table 3. Mount orientations (Degree)

Mount	LCS of a mount	GCS		
		X	Y	Z
Eng_left	u _i	0	90	90
	v _i	90	45	135
	w _i	90	45	45
Eng_right	u _i	0	90	90
	v _i	90	45	45
	w _i	90	135	45
Tran_left	u _i	0	90	90
	v _i	90	0	90
	w _i	90	90	0
Tran_right	u _i	0	90	90
	v _i	90	0	90
	w _i	90	90	0
Added_left	u _i	0	90	90
	v _i	90	0	90
	w _i	90	90	0
Added_right	u _i	0	90	90
	v _i	90	0	90
	w _i	90	90	0

Table 4. Mounts' Dynamic stiffness in LCS (N/mm)

Mount	Dynamic stiffness in LCS		
	u	v	w
Eng_left/right	320	398	1600
Tran_left/right	760	2160	2200
Added_left/right	540	1440	1440