

Research on the Optimal Design of Engine Mounts by Minimizing Transfer Power based on Identifying Excitation and Admittance Matrix

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ABSTRACT

Engine mounts play important roles in interior noise of automobiles. Decoupling optimal design of mounts has been researched for long, but reducing vibration power into body transmitted from engine can be a more intuitive way to improve NVH performance. Some approaches for minimizing transfer power through engine mounts based on finite element model were reported, whose disadvantages are lack of data and inaccuracy at high frequency in some cases. To get an analytic formula of transmitted power, a model considering coupled vibration between the body and the engine is presented here. An admittance function matrix is used to describe the dynamic relationship between the mounting points on the body side. Based on this admittance matrix measured on the full vehicle, and excitation forces identified with acceleration data measured on all mounts, the vibration equation of the coupled model can be established by using Lagrange's methodology. Vibrational response of mounting points, as well as the vibration power transmitted to the body in each direction of mounts, can be obtained by solving the equations in complex-frequency within band of 20Hz~200Hz. The overall transfer power can be expressed by summation of each direction of all mounts. The particle swarm optimization algorithm is used to minimize the transmitted power and find out an optimum solution for stiffness of the rubber mounts. It is confirmed by examples that power transmitted to the body was reduced observably after optimized with this method, which is better than optimization with TRA decoupling method.

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INTRODUCTION

Engine is the main exciting source of vibration in automobiles, and as the connection of the engine and the body, engine mounts have drawn much attention in vibration isolation. Some approaches to the optimum design of engine mount have been employed to improve the NVH behavior of vehicles. Jeong and Singh used torque roll axis (TRA) decoupling method to decouple the TRA mode from the remaining five rigid body modes [1], Park and Singh improved this method by considering the non-proportionality of damping [2]. Wang designed a mixed mode magnetorheological fluid mount based on displacement transmissibility and force transmissibility [3], Madjlesi used Advanced Noise Path Analysis tool based on substructuring method to optimize the mounting system [4]. Liu and Lim proposed an inverse substructuring approach to determine the vibration transmissibility through chassis mounts [5]. Some optimizations based on finite model have also been proposed [6, 7, 8, 9].

An analytic formula of the transmitted power is derived in this presentation, which is used as the objective function. The particle swarm optimization (PSO) algorithm is used to minimize the transmitted power and find out an optimum solution for stiffness of mounts.

COUPLED VIBRATION EQUATION

As shown in Figure 1, an engine-mount-body system with 3 mounting points is simplified. In this model, the engine is modeled as a rigid body, and the rubber mounts are modeled as springs and dampers. The mounting points are modeled as massless nodes, and the bodywork is modeled flexibly. To describe the vibration of this coupled system, a 15 DOFs global coordinate $q = \{q_1, q_2\}^T$ is employed, where $q_1 = \{x_0, y_0, z_0, \alpha_0, \beta_0, \gamma_0\}^T$ describes the motion of c.g. of the engine, $q_2 = \{x_1, y_1, z_1, x_2, y_2, z_2, x_3, y_3, z_3\}^T$ describes the motion of the mounting points on the body side, and the vertical axes at the mounting points are all parallel to that at the c.g. respectively. Based on the linear assumption of displacement, the engine-mount-body system equation can be written as

$$[M]\{\ddot{q}\} + [K]\{q\} = \{f\}e^{j\omega t} \quad (1)$$

Where $[M]$, $[K]$ and $\{f\}$ are derived by Lagrange's methodology based on identifying excitation force and admittance matrix. Energy transmitted to the body is considered as potential energy of this system. $[M]$ and $[K]$ are extracted from kinetic energy and potential energy of the system respectively.

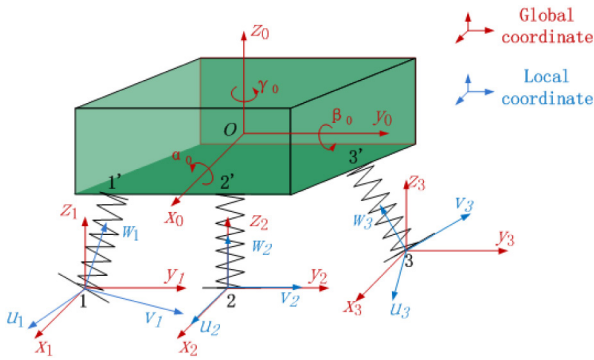


Figure 1. Simplification of an engine-mount-body system

Mass Matrix [M]

As the mounting points are modeled as massless nodes, kinetic energy of the system comprises only that of the engine:

$$T = \frac{1}{2} m_0 (\dot{x}_0^2 + \dot{y}_0^2 + \dot{z}_0^2) + \frac{1}{2} J_{xx} \dot{\alpha}_0^2 + \frac{1}{2} J_{yy} \dot{\beta}_0^2 + \frac{1}{2} J_{zz} \dot{\gamma}_0^2 - J_{xy} \dot{\alpha}_0 \dot{\beta}_0 - J_{xz} \dot{\alpha}_0 \dot{\gamma}_0 - J_{yz} \dot{\beta}_0 \dot{\gamma}_0 \quad (2a)$$

It can be expressed in matrix form

$$T = \frac{1}{2} \dot{q}^T \begin{bmatrix} M_0 & \\ & 0_{9 \times 9} \end{bmatrix} \dot{q} \quad (2b)$$

Where

$$M_0 = \begin{bmatrix} m_0 & & & & & \\ & m_0 & & & & \\ & & m_0 & & & \\ & & & J_{xx} & -J_{xy} & -J_{xz} \\ & & & -J_{xy} & J_{yy} & -J_{yz} \\ & & & -J_{xz} & -J_{yz} & J_{zz} \end{bmatrix} \quad (2c)$$

Compare it with equation (3)

$$T = \frac{1}{2} \dot{q}^T M \dot{q} \quad (3)$$

Mass matrix [M] is obtained as following:

$$M = \begin{bmatrix} M_0 & \\ & 0_{9 \times 9} \end{bmatrix} \quad (4)$$

Stiffness Matrix [k]

Potential energy of the system comprises two parts: energy transmitted to the body and energy of the mounts.

For the first part, under the assumption of “small motion” and “linear system”, force transmitted to the body is approximately linear to the displacement of each mounting point on the body in each direction. The overall energy transmitted to the body can be expressed as following:

$$P_4 = \frac{1}{2} \{q_2\}^T \{F_2\} \quad (5a)$$

Where

$$F_2 = \{F_{x1}, F_{y1}, F_{z1}, F_{x2}, F_{y2}, F_{z2}, F_{x3}, F_{y3}, F_{z3}\}^T \quad (5b)$$

It represents forces at mounting points in all directions. Additionally, there is an inherent relationship between F_2 and q_2 , which can be written as following:

$$\{q_2\} = [Y] \{F_2\} \quad (6)$$

Where [Y] is a 9×9 admittance matrix between these three mounting points on the body side. It is also a function of frequency ω , and can be tested by impulse technique after removing the engine.

From simultaneous equations (5) and (6), P_4 can be expressed in global coordinate:

$$P_4 = \frac{1}{2} \{q_2\}^T [Y]^{-1} \{q_2\} = \frac{1}{2} \{q\}^T \begin{bmatrix} 0_{6 \times 6} & \\ & [Y]^{-1}_{9 \times 9} \end{bmatrix} \{q\} \quad (7)$$

The potential energy of mounts can be determined by the relative displacement between the mounting points on the body and engine side. Coordinates of the points on the engine side can be expressed by that of the c.g., for i 'th mounting point:

$$\{x_i', y_i', z_i'\}^T = [E_i] \{q_i\} \quad (8)$$

Where $[E_i]$ is the transfer matrix from the c.g. to the i 'th mounting point on the engine side. The relative displacement:

$$\{\Delta x_i, \Delta y_i, \Delta z_i\}^T = [E_i] \{q_i\} - \{x_i, y_i, z_i\}^T \quad (9)$$

Transform it into its local coordinate:

$$\begin{aligned} \{\Delta u_i, \Delta v_i, \Delta w_i\}^T &= [T_i] \{\Delta x_i, \Delta y_i, \Delta z_i\}^T \\ &= [T_i] [E_i] \{q_i\} - [T_i] \{x_i, y_i, z_i\}^T \end{aligned} \quad (10)$$

Where $[T_i]$ is the Euler rotation matrix.

The potential energy of the i th mount: