

Analysis of Motor Vibration Isolation System with Focus on Mount Resonances for Application to Electric Vehicles

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ABSTRACT

The vibration isolation effectiveness of powertrain mount configurations is examined for electric vehicle application by considering the effect introduced by internal mount resonances. Unlike internal combustion engines where mounts are typically designed only for static support and low frequency dynamics, electric motors have higher excitation frequencies in a range where mount resonances often occur. The problem is first analytically formulated by considering a simple 3-dimensional powertrain system, and the vibration isolation effectiveness significantly deteriorates at the mount resonance(s). It is shown that by modifying the mount shape, the mount resonance(s) can be shifted while maintaining the same static rate, tuning the frequency away from any engine excitation or natural frequencies. Further, internal mount resonances are utilized to improve vibration isolation over a narrow frequency range, using non-identical mounts to split mount resonance peaks. Then a computational model for a realistic drive unit (containing electric motor, power inverter, and gearbox) is considered. The mount resonance phenomenon is replicated in the computational model, and the effect on non-identical mounts is again examined. Finally, 3 and 4-point mounting schemes are compared, and their parameters (if selected properly) exhibit better vibration isolation over a tunable frequency range of interest.

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INTRODUCTION

Conventional automotive powertrains generally have low frequency vibration excitations, and the primary function of engine mounts is to reduce the transmission of dynamic forces at lower frequencies, as well as provide static support for the powertrain weight. In the low frequency regime, these mounts are usually characterized by simple Kelvin-Voigt type visco-elastic models with inherent stiffness and damping properties, illustrated in Fig. 1(a). Here, k is stiffness, c is viscous damping, $X(\omega)$ is the mount amplitude at frequency ω , and $F_T(\omega)$ is the transmitted force at ω . Some recent electric vehicle powertrains produce significant mid and high frequency excitations (say from 100 to 1000 Hz) that amplify the source regime, creating significant structure-borne disturbance that propagates through the vehicle structural paths to the driver for undesirable noise and vibration performance. Additionally, there is no masking effect from an internal combustion engine (as in hybrid vehicles), and the mid frequency noise and vibration becomes objectionable. New mount design considerations are therefore needed to achieve acceptable vibration isolation.

In the mid frequency regime, a purely visco-elastic mount model (Fig. 1(a)) is no longer valid, and dynamic mass participation effects must be considered. This concept is illustrated by Du et al. [1] by replacing the elastic model with a continuous mount model consisting of distributed mass, stiffness, and damping properties. Low frequency

isolation characteristics remain similar for the force transmissibility, though multiple internal mount resonances now exist in the mid-to-high frequency range. The first continuous system resonance in particular is in the excitation frequency range of a typical electric engine [2, 3, 4, 5], and the stator bending mode or motor drive system resonance can also fall in the same frequency range [4-5]. This article proposes to capture mount mass participation through an equivalent single degree of freedom model, as shown in Fig. 1(b), and explore the effects on vibration isolation. This model will be utilized to resolve the following issue: whether the internal mount resonance strictly has a detrimental effect on vibration isolation or can be exploited for improved isolation.

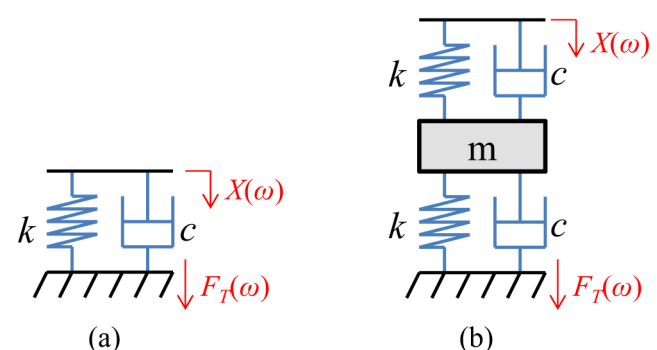


Figure 1. Discrete powertrain mount models: (a) massless visco-elastic elements and (b) visco-elastic elements with effective mass.

PROBLEM FORMULATION

The scope of this paper is limited to a generic electric powertrain supported by multiple elastic mounts. It is shown in Fig. 2 as a conceptual schematic of a realistic drive unit (containing electric motor, power inverter, and gearbox). This unit is a representative subsystem of the vehicle powertrain and driveline for motion analysis (as opposed to torsional system analysis). It is analyzed using both a simplified analytical model with two mounts and a representative computational model with three or more mounts, shown in Fig. 2(a) and Fig. 2(b), respectively. The analytical model serves as a proof-of-concept on the effect of internal mount resonances on vibration isolation effectiveness and provides guidance for computational analysis. The computational model (created using a commercial code [6]) enables analysis of this effect on the driveline subsystem with a more realistic layout and response. While both radiated sound and transmitted forces to the base must be minimized, the scope of this article is strictly limited to reducing the transmitted forces for effective vibration isolation over the mid frequency range (say from 100 to 1000 Hz.) Specific objectives include the following: 1. Develop an analytical and computational model of the generic drive unit to replicate internal mount resonances and assess vibration isolation effectiveness, 2. Examine the models with both identical and non-identical mounts, and 3. Compare 3-point and 4-point mounting schemes for the computational model.

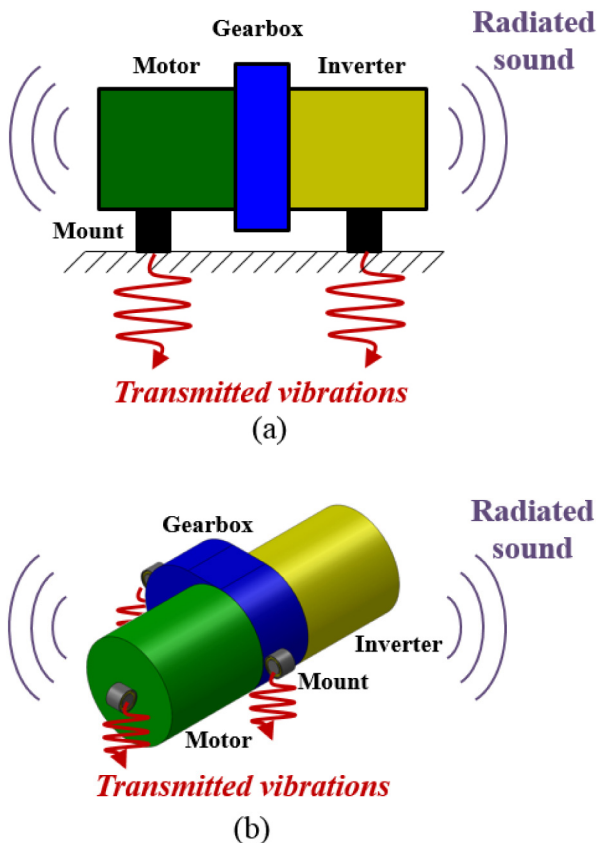


Figure 2. Typical electric vehicle powertrain and driveline subsystem: (a) simplified schematic and (b) representative computational model

For all models presented in this paper, the drive unit is assumed to be discrete, deterministic, frequency-invariant, and linear about an operating point with no kinematic nonlinear effects. Additionally,

lumped masses are assumed to be rigid with no flexural mode participation, and a known external harmonic disturbance force excites the motor only. The powertrain sub-frame is assumed to be a rigid base to simplify the system, though it is suggested that a flexible base does not significantly alter the vibration isolation capabilities of the mounts [1]. Lastly, it is assumed that motion control of the drive unit is not of concern in the mid-frequency range (where response amplitudes are small). Thus, no hydraulic mounts are considered in this article. Model parameters are selected to be representative of realistic electric engines to reasonably simulate the mount resonances in the appropriate mid frequency range.

ANALYTICAL MODEL

The drive unit is first analyzed as a simplified model (denoted as A), shown in Fig. 3. The motor, gearbox, and inverter components are lumped into a single mass with two supporting mounts (each possessing mass); and motion is only considered in one translational direction. Excitation force amplitude F and complex valued response motion amplitudes $\tilde{X}$ are assumed at any frequency, and the system is analyzed in the frequency domain. Powertrain mounts generally consist of rubber (elastomeric) components to provide damping, and structural damping is a more valid model for rubber, especially at higher frequencies [7, 8]. Thus, a complex stiffness $\tilde{k} = k(1 + i\eta)$ describes the mount compliance at a given frequency where k is the static stiffness, η is the loss factor, and $i = \sqrt{-1}$.

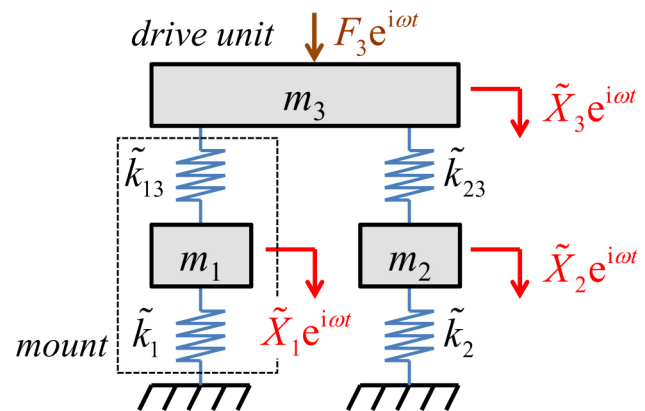


Figure 3. Conceptual analytical model of electric vehicle powertrain.

The governing equations at frequency ω are written in compact matrix form as

$$\tilde{\mathbf{X}}(\omega) = [-\omega^2 \mathbf{M} + i\tilde{\mathbf{K}}]^{-1} \mathbf{F} \quad (1)$$

where $\tilde{\mathbf{X}}(\omega) = \{\tilde{X}_1(\omega) \quad \tilde{X}_2(\omega) \quad \tilde{X}_3(\omega)\}^T$, superscript T indicates a transpose, $\mathbf{F} = \{0 \ 0 \ F_3\}^T$,

$$\mathbf{M} = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix}$$

is the inertia matrix, and

$$\tilde{\mathbf{K}} = \begin{bmatrix} \tilde{k}_1 + \tilde{k}_{13} & 0 & -\tilde{k}_{13} \\ 0 & \tilde{k}_2 + \tilde{k}_{23} & -\tilde{k}_{23} \\ -\tilde{k}_{13} & -\tilde{k}_{23} & \tilde{k}_{13} + \tilde{k}_{23} \end{bmatrix} \quad (3)$$

is the complex valued stiffness matrix. For a given excitation frequency,

$$\tilde{F}_T(\omega) = \tilde{X}_1(\omega)\tilde{k}_1 + \tilde{X}_2(\omega)\tilde{k}_2 \quad (4)$$

is the force transmitted to the rigid base. The magnitude of the transmitted force for vibration isolation characteristics is the metric to determine effectiveness of the mounting scheme and the effect of the internal mount resonance. Both identical and non-identical mount sets are examined, denoted as Model A0 and Model A1, respectively. All model and mount set designations examined in this paper are defined in Table 1.

Table 1. Model and mount set designations.

Model	Model Type	# Mounts	Configurations
A	Analytical	2	0, 1
B	Computational	3	0, 1, 2
C	Computational	4	0, 1, 2

KEY: Configuration 0 – Identical mounts

Configuration 1 – Non-identical mount set #1

Configuration 2 – Non-identical mount set #2

TUNING OF MOUNT RESONANCES

Parameters for Model A0 are shown in Table 2, selected as representative order of magnitude values to replicate a realistic system response. Mount rates are high, typical for electric powertrains to accommodate increased output torque over combustion engines [3]; and the drive unit mass includes electric motor, inverter, and gearbox powertrain/driveline components. The loss factor is a low value representative of natural rubber in the mid-frequency range [8] and is chosen in part to highlight the mount resonance effect. These values are used for an illustrative example case, not a specific design process.

Table 2. Model A0 parameters.

Parameter	Value	Units
$m_1 = m_2$	1.0	kg
m_3	240	kg
$\tilde{k}_1 = \tilde{k}_{13}$	$8.4(1 + i0.05)$	kN/mm
$\tilde{k}_2 = \tilde{k}_{23}$	$8.4(1 + i0.05)$	kN/mm

The frequency response of the Model A0 transmitted force is first examined from 1 to 1000 Hz in 0.1 Hz excitation frequency increments, shown in Fig. 4(a). Resonance frequencies Ω are evident. Namely, the drive unit resonance $\Omega_1/2\pi = 29.7$ Hz and internal mount resonances $\Omega_2/2\pi \approx \Omega_3/2\pi \approx 653$ Hz. Note that the large amplitude at Ω_1 is caused by the low η value. Increased rubber damping or a hydraulic mount could be used in practice to lower this amplitude to a reasonable level. Since both mounts are identical, their resonance frequencies coincide, and vibration isolation capabilities are degraded near $\Omega_2 \approx \Omega_3$. Additionally, a single degree of freedom estimation can be made for $\Omega_2 \approx \Omega_3$ as

$$\frac{\Omega_2}{2\pi} \approx \frac{\Omega_3}{2\pi} \approx \frac{1}{2\pi} \sqrt{\frac{2k_1}{m_1}} \quad (5)$$

which is approximately 653 Hz. Both Ω_2 and Ω_3 can be tuned independently of the powertrain properties. In this manner, the mount resonance(s) can be shifted to avoid a prominent powertrain excitation frequency or drive unit structural resonance, such as the stator bending mode. The two mounts can also be tuned independently such that $\Omega_2 \neq \Omega_3$.

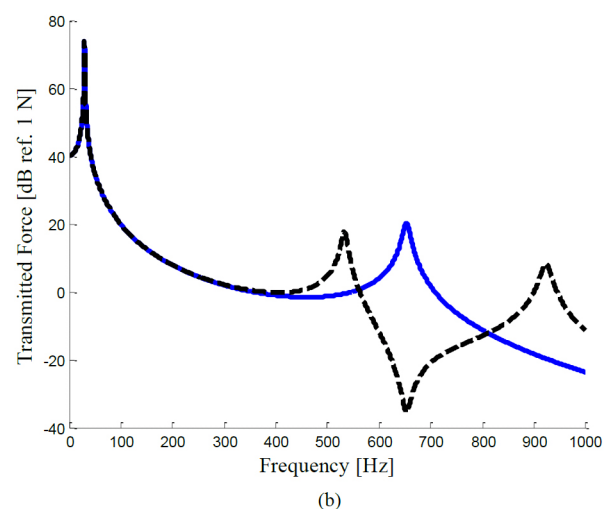
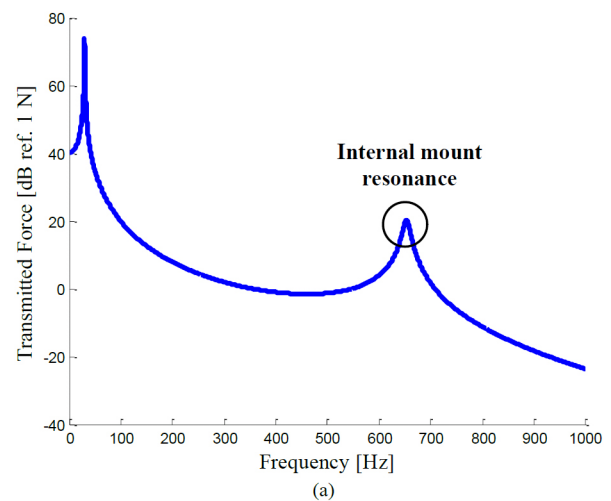


Figure 4. Transmitted force of the conceptual model for (a) identical mounts (Model A0) and (b) non-identical mounts (Model A1). Key: —, A0 and ---, A1.