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Automotive Powerplant Isolation Strategies

R. Matthew Brach
Ford Motor Co.

ABSTRACT

Recently an increase in interest has occurred in automotive powerplant mounting. Evidence of this growth is the increase in the number of publications on the topic. The majority of this renewed interest has come from predicting and understanding the response of hydraulic engine mounts and the application of optimization techniques to the problem of powertrain vibration isolation, and occasionally to the combination of these two topics. However, it appears that these analytical techniques have been sufficiently developed and correlated to actual powertrain systems to have found widespread use by the automotive manufacturers. Subject to timing and packaging constraints, the more traditional mounting system design strategies are typically utilized. These strategies include natural frequency placement, torque axis mounting and elastic axis mounting. This paper presents a comprehensive review of these three strategies including a discussion of the assumptions associated with each method. In addition, the center of percussion mounting strategy, applicable to the isolation of transient inputs to the powertrain, is discussed in detail, including the technical basis for the theory.

INTRODUCTION

Since the inception of the automotive industry, isolation of the vibration of the engine from the rest of the vehicle has been desirable. Various isolation schemes have been devised to deal with this problem. Four strategies have been identified in the literature to address this problem. These strategies are: torque axis mounting, elastic axes mounting, natural frequency placement-optimization, and center of percussion mounting. A review of the literature indicates that the first two techniques are more widely used and investigated by vehicle manufacturers. All four techniques are described here in detail to establish a comprehensive background about engine mounting strategies that consider the motion of the engine as a rigid body only. Each description contains references that present the development and theory associated with each method. In each case, discussion regarding the assumptions made in the development of the method is presented. Where appropriate, the limitations of a method are also discussed. Other powerplant isolation strategies exist that consider the flexural motion of the automotive engine and drivetrain. These methods (Bolton-Knight, 1971) are outside the scope of this paper.

AUTOMOTIVE POWERPLANT ISOLATION STRATEGIES

NATURAL FREQUENCY PLACEMENT-OPTIMIZATION - Optimization algorithms have been applied to the problem of isolation of engine vibrations. The typical objective of this analysis is to move the rigid body natural frequencies of the powerplant away from the frequencies of the input sources. This alters the amount of modal coupling by changing the mode shapes and presumably reducing the displacements of the response, thereby reducing the transmitted forces. Numerous papers have been written about this topic, including Johnson and Subhedar, (1979); Bernard and Starkey, (1983); Geck and Patton, (1984); Spiekerman, Radcliffe, and Goodman, (1985); Staat, (1986); Saitoh and Igarashi, (1989), and Bretl (1993).

In these studies the powerplant is typically modeled as a rigid body with six degrees of freedom mounted on resilient supports. These supports are commonly attached to ground and the system is referred to as a "grounded" system. The design parameters usually include the stiffness values of the mount, and the mount locations and orientations. Viscous damping is occasionally included in the system model. However, it is usually treated as a fixed design parameter. These methods depend largely on the ability to predict or determine the natural frequencies of the powerplant on its mounts. This task can be difficult, and the results of this isolation method are only effective near the natural frequency. As such, this technique is used primarily to address vibrational problems that are due to the engine idle frequency, or one of its associated orders, being close to a system rigid body natural frequency. Modifications of this approach have been implemented that determine the stiffness values of the mount by directly minimizing the forces transmitted through the mounts to the vehicle structure at idle (Oh, Lim and Lee, 1991; Swanson, Wu, and Ashrafiuon, 1993).

More recently, optimization methods have been used to analyze a rigid body affixed to resilient mounts attached to a flexible support (Ashrafiuon, 1993; Lee, Yim, and Kim, 1995). In another recent publication, optimization techniques were applied to a system model that takes into account the general rotational motion of the engine as a rigid body (Snyman, et. al., 1995).

TORQUE AXIS MOUNTING - The torque axis, also referred to as the torque roll axis, is defined as the resulting fixed axis of rotation of an unconstrained three dimensional rigid body (i.e. either free or supported elastically on very soft springs) when a torque is applied along an axis not coincident with any of the principal axes (Timpner, 1965; Fullerton, 1984). For the case of an automotive engine, the axis about which torque is applied is the crankshaft axis. This axis is rarely coincident with a principal axis of the engine. It is hypothesized (Timpner, 1966) that the disturbances transferred to the vehicle can be reduced by positioning the engine mounts such that the engine oscillates predominantly about this torque axis. The two references above each present a method for determining the location of the torque axis of an engine.

In order to better understand the concept of torque axis, consider the general equations of motion governing the rotational behavior of a rigid body in three dimensions acted on by the moment vector **M**. These equations are given by (Greenwood, 1988):

$$
\begin{aligned}
M_x &= I_{xx}\dot{\omega}_x + I_{xy}(\dot{\omega}_y - \omega_x\omega_z) + I_{xz}(\dot{\omega}_z + \omega_x\omega_y) \\
&\quad + (I_{zz} - I_{yy})\omega_y\omega_z + I_{yz}(\omega_y^2 - \omega_z^2) \\[2ex]
M_y &= I_{xy}(\dot{\omega}_x + \omega_y\omega_z) + I_{yy}\dot{\omega}_y + I_{yz}(\dot{\omega}_z + \omega_x\omega_y) \\
&\quad + (I_{xx} - I_{zz})\omega_x\omega_z + I_{xz}(\omega_z^2 - \omega_x^2) \qquad (1) \\[2ex]
M_z &= I_{xz}(\dot{\omega}_x - \omega_y\omega_z) + I_{yz}(\dot{\omega}_y + \omega_x\omega_z) + I_{zz}\dot{\omega}_z \\
&\quad + (I_{yy} - I_{xx})\omega_x\omega_y + I_{xy}(\omega_x^2 - \omega_y^2)
\end{aligned}
$$

where I_{ij} are the elements of the rigid body inertia matrix, ω_x, ω_y, ω_z are the angular velocity components and $\dot{\omega}_x$, $\dot{\omega}_y$, $\dot{\omega}_z$ are the angular accelerations.

For the case of an automotive engine, it is common to assume the engine to be a rigid body on resilient supports that are attached to ground. It can be further assumed that the values of the moments, both the applied moments and the moments generated by the reaction forces at the mounts, and the inertia properties of the engine are known or can be calculated. Then, for a given set of initial conditions, the vector of angular velocities, $\boldsymbol{\omega}$, can theoretically be solved for as a function of time by numerical integration of equations (1). In general, the magnitude and direction of the axis of rotation of the rigid body will also be a function of time. Thus, the torque axis as defined previously, may not exist. Closer investigation of these equations for a free rigid body system will lead to a better understanding of the rotational response of the powertrain on its mounts and whether a single axis of rotation exists for a rigid body.

While the existence of an axis meeting the definition of the torque axis may be debated, the application of the torque axis theory generally involves the premise of placing the rigid body response that is predominantly about the axis parallel to the crank-shaft, commonly referred to as the roll-response, at as low a frequency as possible. This separates it from the other modes of vibration. This mounting philosophy does not require the existence of an axis, but merely uses the axis as a means to describe the response at that frequency.

It is further noted that equations (1) are derived under the assumption that the inertia properties of the powerplant are independent of time. This is not the case for an automotive engine. The motion of the pistons, cranks, connecting rods, and the crankshaft make the inertia properties periodic functions of time, independent of the coordinate system. This further influences the time dependent nature of $\boldsymbol{\omega}$ in an actual engine. However, this time dependence of the inertia properties is typically neglected in practice (Bachrach, 1995).

ELASTIC AXES - Elastic axes for an elastically supported rigid body system are those axes for which application of a force or torque, along or about a line produces only a corresponding translation or rotation on or about the same line. In the coordinate system defined by the elastic axes, the system response consists of decoupled translational and rotational modes. This decoupled state is frequently referred to as focalization.

The elastic axes of a rigid body on flexible supports can be determined using the flexibility matrix. Analytical modal decoupling of the flexibility matrix does not yield the elastic axes system because the eigenvectors do not typically span a physical space. Hence, the transformation to the elastic axes must be a physical coordinate transformation.

For a 2-dimensional system with three degrees of freedom, full decoupling can be accomplished since the six off-diagonal terms of the symmetric 3×3 flexibility matrix can be eliminated by two independent translations and one rotation. However, full decoupling of the symmetric 6×6 flexibility matrix for a 3-dimensional rigid body system with six degrees of freedom cannot be accomplished since 30 off-diagonal terms exist in the flexibility matrix (15 symmetric pairs) and only three independent coordinate translations and three independent coordinate rotations can be defined (Kim, 1991). This allows for only six symmetric off-diagonal pairs to be set to zero. Due to this limitation, effort has been spent in the area of partial decoupling of the modes of vibration (Derby, 1973; Ford, 1985, Xuefeng, 1991).

CENTER OF PERCUSSION MOUNTING - This isolation technique uses the mechanical phenomenon known as the center of percussion. One property of the center of percussion is that for a compound pendulum acted on by an impulse applied through the center of percussion and perpendicular to the line defined by the center of mass and the fixed point, no reactive impulse results at the fixed point. (A detailed presentation of the definition of the center of percussion and the properties associated with this point is given in Appendix A.)

The application to engine mounting follows directly. If the front and rear engine mounts are arranged such that their locations are reciprocal centers of percussion (see Appendix A), then an impulse to one mount from a road disturbance results in little or no reaction at the other mount (Wilson, 1959; Timpner, 1965; Bolton-Knight, 1971). This improves the overall isolation performance of the mounting system with respect to impulsive inputs since less vibrational input is imparted to the body of the vehicle. Although the application of this isolation scheme to actual mounting problems is typically not as simple as this description would indicate, placement of the engine mounts consistent with this theory will enhance their overall isolation effectiveness. This application of the center of percussion approach to powertrain mounting addresses the minimization of the reaction forces at the powertrain mounts to an externally applied impulsive load. This technique does not address the harmonic response of the system and is therefore supplemental to those methods that address the harmonic response of the powertrain.

Additionally, the system configuration used in the derivation of the theory is restricted to response in one plane. In the case of an automotive powertrain, the response of the system will be multi-planar even if the force applied by the front suspension is symmetric. This asymmetry of response is due to the lack of asymmetry of the mounting system and coupling. For this case, and the case of asymmetric loading of the front of the vehicle (only one wheel striking a pothole for example) the effectiveness of the technique as presented in this paper is not clear since the response will be multi-planar. No analysis has been done in this area in this paper or by others.

DISCUSSION

The complexity of the harmonic response of the automotive powerplant mounted on engine mounts in a vehicle cannot be understated. As mentioned in the previous sections, simplifications and assumptions are made in the development of the strategies currently used in industry to analyze this system. These simplifications and assumptions are made to facilitate the tractability of the analysis and permit expectations about the response of the system to be more easily formulated. These assumptions include: the time invariant nature of the inertia properties of the powerplant, the body-side of the engine mounts affixed to a rigid structure (a grounded system), and that the response of the system exhibits the same characteristics that its planar counterpart does, such as in the case of Elastic Axis theory and Center of Percussion theory.

In addition to these assumptions, other assumptions are made that have not yet been mentioned. It is typically assumed in many of the analyses that the powerplant system includes the engine, the transmission and the mounts. Occasionally the subframe (if applicable) and frame/body of the vehicle is included in the analysis. The exhaust sub-system, the drive-shaft, and hose connections are almost never included in the system model. The inclusion of each of these sub-systems in the system model increases the number of degrees of freedom thereby increasing the complexity of the problem. It has been shown that the these additional connections can affect the response of the system (Spiekerman, Radcliffe, and Goodman, 1985).

Furthermore, it is typically assumed that the force-displacement characteristics of the engine mounts are linear. Although this assumption may be valid for elastomeric mounts for small displacements, hydraulic engine mounts are inherently nonlinear and their response characteristics have been investigated and found to exhibit quite complicated nonlinear behavior (Kim, Singh, and Ravindra, 1992; Kim and Singh, 1993). In addition, it has been shown analytically that the response of a three degree-of-freedom system comprised of a rigid body on mounts with linear force displacement characteristics can be nonlinear (Brach and Haddow, 1996).

Lastly, it is commonly assumed that the effects of temperature on the properties of the engine mounts, and hence the performance of the powertrain isolation system, are negligible.

CONCLUSION

All of these assumptions are important in the modelling of automotive powertrain vibration isolation strategies. However, what has not been established, or at least not published, is the relative importance or ramifications of these assumptions. Of equal importance is assessing that when these assumptions are made, whether the models based on these assumptions correlate with actual systems.

Evaluation of these assumptions must take place so that the validity of the models can be established. This evaluation typically takes place on the simplest system that still exhibits the fundamental response characteristics of interest. In the case of a powertrain system, this model would likely consist of an engine on mounts attached to ground. This six degree-of-freedom system, the response of which will still likely be quite complicated, can be experimentally investigated effectively. Models of the engine-mount system have been established and correlation between the experimental response and the predicted analytical response can be done. With this simple system, the assumptions made in the development of the strategies discussed earlier that are currently used in the automotive industry can be evaluated.

With a fundamental understanding of the system response confirmed and a model that reliably predicts this response over a relevant range of parameters, parameter design studies can be performed to assess the performance of a mounting system against design requirements. Additional complexity can then be added to the system to broaden the range of parameters and circumstances over which the model is valid. This correlated model can then lead to more effective application to the optimization procedures already being applied to this system.

It should be noted that this correlated design process will not likely eliminate the fine tuning that inevitably occurs with the actual system. However, it can reduce the number of trials thereby reducing the time required to complete this fine tuning. This is important as it contributes to the overall reduction of the development time of the vehicle.

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APPENDIX

The Center of Percussion[1/]

Consider a rigid body in a horizontal plane, fixed at one point, and moving under applied loads as shown in Figure A.1(a). This system is referred to as a compound pendulum (Greenwood, 1988). The free body diagram for this rigid body is shown in Figure A.1(b) which includes the reaction force at O and applied loads. The external force system can be replaced by the resultants *ma* and $I\alpha$. The vector quantity ma can be broken down into its components $r\omega^2$ and $r\alpha$ which act through the center of mass in the normal and tangential directions, respectively, as shown in Figure A.1(c). Another resultant force diagram can be obtained by moving the force $mr\alpha$ to point Q along the line OG (actually beyond point G) such that the resulting moment created about O equals $I_G\alpha$. $I_G\alpha$ can then be removed from the resultant force diagram as shown in Figure A.1(d). This condition can be written as:

$$I_G\alpha + mr^2\alpha = mr\alpha\ell \qquad \text{(A.1)}$$

[1/]The topic of the center of percussion is covered in many books on dynamics and vibrations. This information is presented here for completeness and also since no references were found that contained this information relative to powertrain mounting. The material presented here most closely follows that presented in Meriam (1978) and Greenwood (1988).

Replacing I_G by $k_G^2 m$ where k_G is the radius of gyration of the rigid body about G:

$$k_G^2 m\alpha + mr^2\alpha = mr\alpha\ell \qquad \text{(A.2)}$$

which results in

$$k_G^2 + r^2 = r\ell \qquad \text{(A.3)}$$

Solving this for ℓ:

$$\ell = \frac{k_G^2 + r^2}{r} = \frac{k_O^2}{r} \qquad \text{(A.4)}$$

where k_O is the radius of gyration of the rigid body about O. The point Q defined by this procedure is called the center of percussion of a body of fixed point O. Note that $\sum M_Q = 0$.

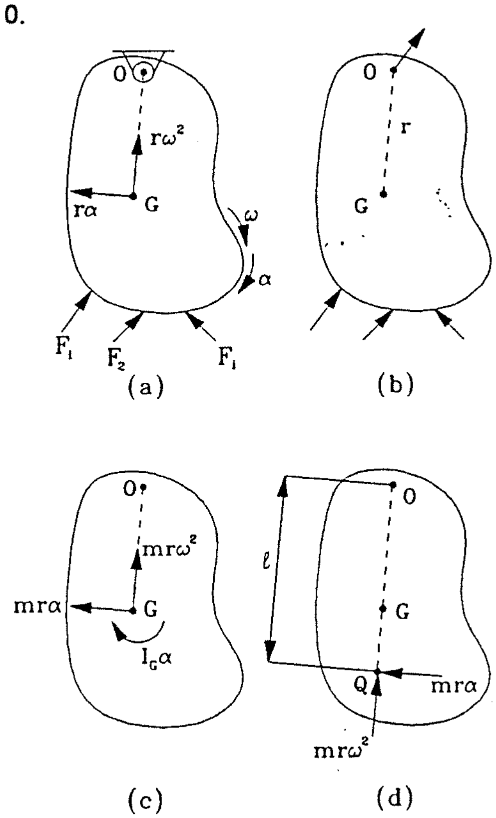

Figure A.1

The center of percussion has two interesting properties as shown by the following analysis. The first property involves the response of the rigid body to an impulsive load and the second shows a property about the natural frequency of a compound pendulum.

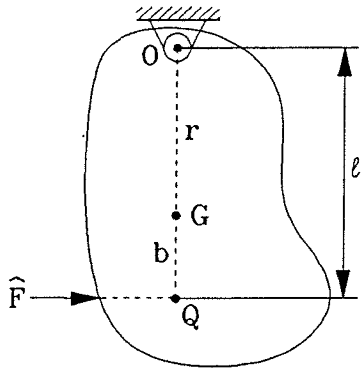

Figure A.2

Consider the same compound pendulum as shown in Figure A.1(a), initially at rest, impacted by an impulse $\hat{F} \perp$ OG, as shown in Figure A.2. Since by definition, the impulse is equal to the change in momentum:

$$\hat{F} = p_2 - p_1 = m(v_{G_2} - v_{G_1}) \qquad \text{(A.5)}$$

Since $v_{G_1} = 0$, let $v_{G_2} = v_G = \frac{\hat{F}}{m} = r\dot{\theta}$ where $\dot{\theta}$ is the angular velocity of the line OG. By definition, the angular impulse of $\hat{F}$ about G, $\hat{M}$, is:

$$\hat{M} = b\hat{F} \qquad \text{(A.6)}$$

Also,

$$\hat{M} = H_2 - H_1 = I_G\dot{\theta} \qquad \text{(A.7)}$$

Using the two previous results along with the fact that H_1 = 0 and $H_2 = H_G = I_G\dot{\theta}$, $\dot{\theta}$ can be solved for:

$$\dot{\theta} = \frac{b\hat{F}}{I_G} \qquad \text{(A.8)}$$

Using this relationship and that $\frac{\hat{F}}{m} = r\dot{\theta}$ it can be shown that $\frac{I_G}{m} = k_G^2 = rb$. With $b = \ell - r$, the same relationship for the location of Q is obtained as before:

$$\ell = \frac{r^2 + k_G^2}{r} \qquad \text{(A.9)}$$

This demonstrates that no impulsive reaction occurs at the point O for the applied impulse $\hat{F}$. Note that an impulsive reaction at O will occur if $\hat{F}$ is applied in any direction other than perpendicular to OC.

Now consider the natural frequency for small motion of the compound pendulum:

$$\omega_n = \sqrt{\frac{mgr}{I_O}} \qquad \text{(A.10)}$$

Consider the natural frequencies of the compound pendulum for motion about points O and Q. Using the previous result, the following relationships are obtained:

$$(\omega_n)_O = \sqrt{\frac{mgr}{I_O}} \qquad (\omega_n)_Q = \sqrt{\frac{mgb}{I_Q}} \qquad \text{(A.11)}$$

Using the fact that $I_O = k_G^2 + r^2$, $I_Q = k_G^2 + b^2$, and that $k_G^2 = rb$, it can be shown that $(\omega_n)_O = (\omega_n)_Q$. This relationship illustrates the second property of the center of percussion which is that point O is the center of percussion for the body when Q is the center of oscillation and vice versa. Points O and Q are then referred to as reciprocal centers of percussion.