

A Method for Evaluation of Low-frequency Noise and Vibration in Powertrain Mount Layouts using Degree of Coupling and Physical Matrices

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ABSTRACT

Ride comfort and idling performance are conflicting low-frequency noise and vibration issues in vehicle development. A convenient index for evaluation of these two phenomena at the design stage was formulated by defining the degree of coupling in a simple numerical expression representing the characteristic matrix based on a physical coordinate system. The degree of coupling was used for performance evaluations of idling and ride comfort in CAE impulse forced response simulation and in an actual vehicle, with good correlation. It was ascertained that this method could be used effectively as a design tool for performance evaluations of powertrain mount layouts in designing total vehicle performance.

1. Introduction

Design methods for the vibro-isolating support of automobile engines have focused on the coupling term of characteristics matrices obtained from equations of motion. Non-coupled support methods have been advocated for some time as a means of simultaneously damping the powertrain mount resonance area and preventing vibration in the vibro-isolating range.⁽¹⁾ Issues for powertrain mount design in automotive development include satisfying performance relative to various phenomena such as ride comfort relative to suspension input, vehicle longitudinal vibration relative to drive train torque input, and idling noise and vibration relative to engine torque fluctuation input. However, the powertrain mount resonance and the idling noise and vibration (vibro-isolating area) frequency bands are close. Additionally, the damping performance relative to suspension input and vibro-isolating performance relative to idling noise and vibration often have a mutually interacting relationship. When creating the actual design, it

is often difficult to place the powertrain mounts in the ideal locations to achieve non-coupled support inside the engine room of FF (front engine, front drive) vehicles, so a certain degree of coupled support is unavoidable.

With the construction of the CAE environment, which is representative of the improved computer performance and lower prices achieved over the past ten years, it is becoming possible to predict performance by response calculations using CAE software. This is generally done by processing large numbers of simulation proposals individually. In addition, design staff often merely evaluate the frequency response levels produced by calculation staff, and there are cases where discussion of physical issues is forgotten.

Therefore, this paper attempts to specify the degree of coupling using an index based on a simple physical coordinate system when setting the powertrain mount layout (geometry and spring value). In addition, ride comfort and idling, which are low frequency phenomena, were analyzed in regards to the following three items for a 2 liter displacement class FF transversally mounted engine passenger car, with good correlation.

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- (1) Degree of coupling obtained by arithmetically processing the characteristics matrices (mass, stiffness)
- (2) Frequency response analysis using structural analysis software
- (3) Actual vehicle tests

As a result, a degree of coupling guidelines was obtained for the FF transversally mounted engine powertrain mount layout.

2. Degree of Coupling in the Characteristics Matrices

In eigenvalue analysis, evaluation is performed according to a diagonal matrix on the inertial and elastic main axes. However, in degree of coupling analysis, evaluation is performed according to the relationship between non-diagonal terms, which are coupled, and diagonal terms, which are non-coupled. The mass matrix $[M_{gr}]$ (equation (1)) and stiffness matrix $[K_{gr}]$ (equation (2)) are analyzed from the discrete engine characteristics and the spring characteristics, respectively, with both matrices located on the grand coordinate axis. Here, 1 to 6 of subscripts i and j indicate in order the longitudinal, lateral, vertical, pitch, roll and yaw components. The mass and stiffness matrices generally contain not only diagonal terms, which are the inertial and elastic main axes on the grand coordinate axis, but also non-diagonal terms which are inertial and elastic products, indicating that the six characteristic modes are coupled.

$$[M_{gr}] = \begin{bmatrix} m & & & & & \\ & m & & & & \\ & & m & & & \\ & & & I_{xx} & -I_{xy} & -I_{xz} \\ & & & -I_{yx} & I_{yy} & -I_{yz} \\ & & & -I_{zx} & -I_{zy} & I_{zz} \end{bmatrix} \quad (1)$$

$$[K_{gr}] = \begin{bmatrix} k_{11} & & & k_{15} & k_{16} & \\ & k_{22} & & & k_{26} & \\ & & k_{33} & k_{34} & k_{35} & \\ k_{42} & k_{43} & k_{44} & k_{11} & k_{11} & \\ k_{51} & k_{53} & k_{11} & k_{55} & k_{11} & \\ k_{61} & k_{62} & k_{11} & k_{11} & k_{66} & \end{bmatrix} \quad (2)$$

To evaluate this mode coupling, $[D_0]$ is derived as the coupling matrix. The coupling matrix $[D_0]$ is calculated from the mass and stiffness matrices, which are characteristic matrices, and expressed as shown in equation (3). Here, ω is the angular speed [rad/s], and a value that matches the phenomena to be evaluated is set.

$$[D_0] = [K_{gr}] + \omega^2 [M_{gr}] \quad (3)$$

$$[D_0] = \begin{bmatrix} d_{11} & & & d_{15} & d_{16} & \\ & d_{22} & & d_{24} & d_{26} & \\ & & d_{33} & d_{34} & d_{35} & \\ & & & d_{44} & d_{45} & d_{46} \\ & & & & d_{55} & d_{56} \\ & & & & & d_{66} \end{bmatrix} \quad (4)$$

Degree of coupling is defined from the coupling matrix $[D_0]$ as the ratio of the diagonal and non-diagonal terms

that are the main components of the phenomena. For example, in the case of idling noise and vibration, the y-axis mode that is the main input direction and the coupled z-axis are calculated as shown in equation (5).

$$[D_{idle}] = d_{56} / d_{55} \quad (5)$$

Here, d_{55} is the y-axis diagonal term, and d_{56} is the z-axis non-diagonal term that is coupled to d_{55} .

The degree of coupling was set for both idling and ride comfort based on the features of each phenomenon.

3. Correlation between CAE Calculations and Actual Vehicle Tests

Three types of actual prototype powertrain mount layouts (Type A, Type B and Type C (conventional layout)) were manufactured and compared. Each layout was investigated using structural analysis software prior to confirmation in an actual vehicle. The results of investigating the degree of coupling are described below.

3.1 Idling Noise and Vibration

When this phenomenon occurs, the drive torque fluctuation is acting on the engine y-axis (hereafter “roll axis”), with the roll axis vibration mode as the main mode. The roll mode in the input direction is the main mode, and vibration is aggravated by coupling with the engine z-axis (hereafter “yaw axis”) mode. The degree of coupling in idling performance evaluation is given by equation (5), and is shown in Fig. 1 using Type C as the standard.

The Type A value is approximately 1.4 times, and the Type B value is approximately 4 times the value of the conventional layout Type C.

Next, the idling vibration mode of an actual vehicle was simulated using the CAE model shown in Fig. 2. This

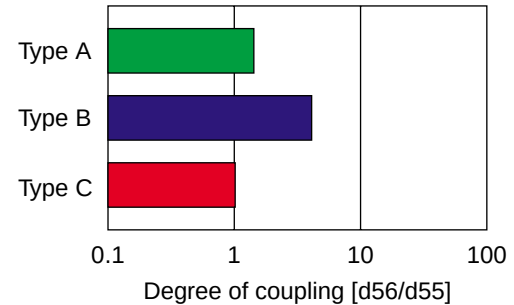


Fig. 1 Degree of coupling for idling noise and vibration

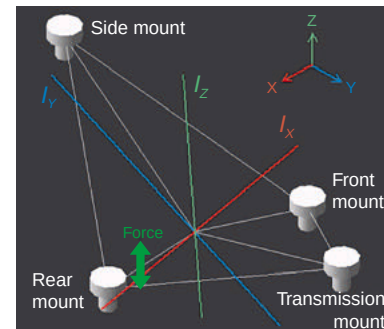


Fig. 2 CAE model for idling noise and vibration

model gives mass and inertia terms as the engine specifications, and the idling frequency characteristics as the mount spring characteristics. The response force and acceleration on the mounts are evaluated by applying the engine impulse force as shown in Fig. 3.

Fig. 3 shows that Type B has a high degree of coupling with the 20 Hz yaw rate, and this causes the vibration to increase in the idling noise and vibration range. Fig. 4 shows the test results using an actual vehicle, and the results are approximately the same as for CAE. Vibration clearly increases for Type B, and although vibration increases slightly for Type A, it is not enough to significantly affect idling noise and vibration in actual vehicle tests.

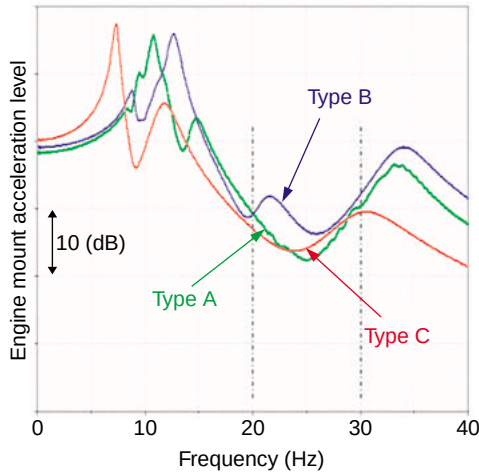


Fig. 3 Amount of engine mount vibration according to CAE

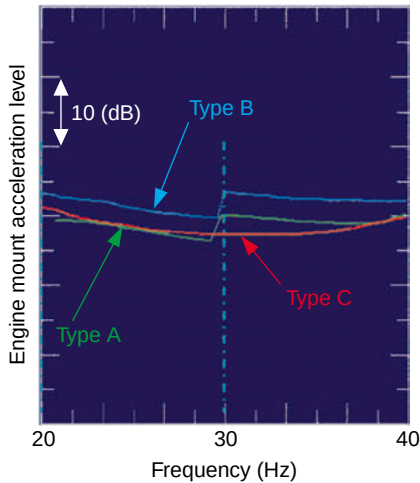


Fig. 4 Experimental level of engine mount vibration

According to the degree of coupling standard set by this study, coupling of approximately 1 to 1.4 does not adversely affect vibration at the powertrain mounting points while idling. However, Fig. 4 shows that the influence increases at 6, indicating that 1.4 is an index that should be observed.

3.2 Ride Comfort Relative to Longitudinal Input

Over the past ten years the application of sealed liquid mounts has become the general method of dealing with the phenomenon of ride comfort relative to powertrain mount vertical input. This paper tackles the longitudinal and lateral input response that is difficult to damp with the vertical

damping characteristics of sealed liquid mounts.

In the case of longitudinal and lateral input, it is already known that the vibration level can be reduced by coupling the rotational component to the main translation component^{(2)–(4)}.

The degree of coupling for longitudinal input ride comfort is as follows, and is shown in Fig. 5 using Type C as the standard.

$$|D_{logi}| = \frac{d_{33}}{d_{31}} \quad (6)$$

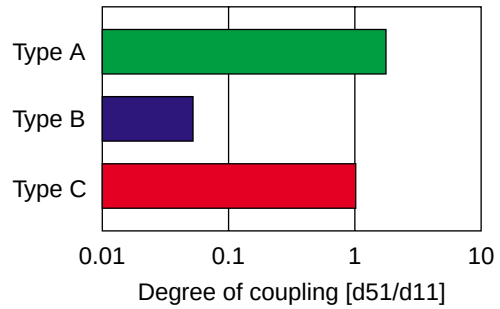


Fig. 5 Degree of coupling for ride comfort

The Type A value is approximately 1.1 times, and the Type B value is approximately 0.07 times the value of Type C.

Next, the ride comfort longitudinal vibration mode of an actual vehicle was simulated using a model such as that shown in Fig. 6 as the CAE model. Here, the engine specifications and mount characteristics were modeled in the same manner as the idling model, and the body, tires and a shaker were further modeled to evaluate ride comfort. The resulting body response force and acceleration were evaluated by exciting the shaker in the longitudinal or lateral direction. In addition, an actual vehicle was also evaluated using excitation equipment having this same structure. The simulation results are shown in Fig. 7.

Fig. 5 shows that Type B has a low degree of coupling between longitudinal characteristic vibration and the rotational component at 10 Hz, so the vibration level is higher than the other two types. In contrast to this, Types A and C have a high degree of coupling between the longitudinal and rotational components, and low vibration levels. In addition, Type A exhibits a high longitudinal

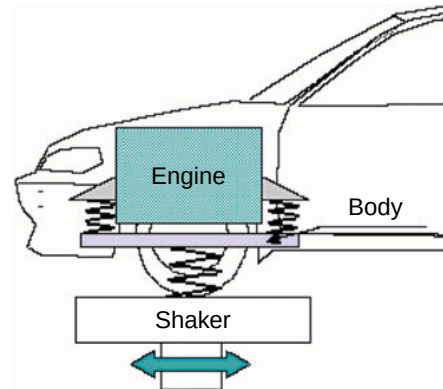


Fig. 6 CAE model for ride comfort

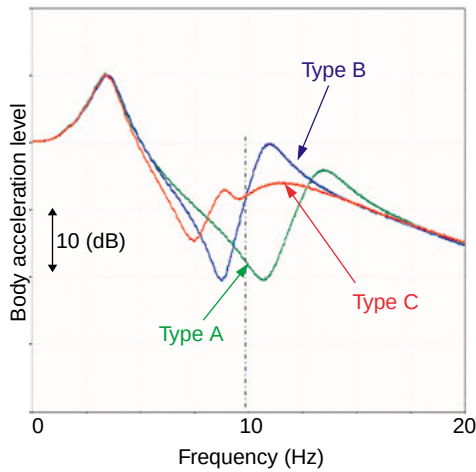


Fig. 7 Level of body vibration with CAE

characteristic value, with vibration around 10 Hz being lower than for Type C.

According to the degree of coupling standard set by this study, coupling of approximately 1 to 2 does not adversely affect body longitudinal vibration for ride comfort. However, Fig. 8 shows that the vibration level increases at 0.07, indicating that 1 is an index that should be observed.

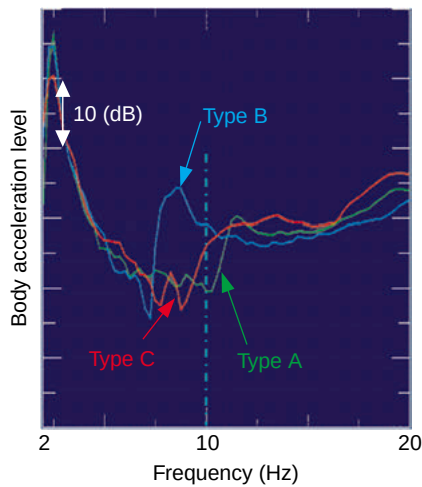


Fig. 8 Experimental level of body vibration

3.3 Ride Comfort Relative to Lateral Input

In the case of lateral input, the degree of coupling is expressed in the same manner as the longitudinal component as follows and is shown in Fig. 9 using Type C as the standard.

$$[D_{horizo}] = \frac{d_{42}}{d_{22}} \quad (7)$$

The Type A value is approximately 1.3 times, and the Type B value is approximately 0.3 times the value of Type C. As shown in Fig. 10, Type B has a low degree of coupling between lateral characteristic vibration and the rotational component at 10 Hz, with vibration levels higher than the other two types. In contrast, Types A and C have

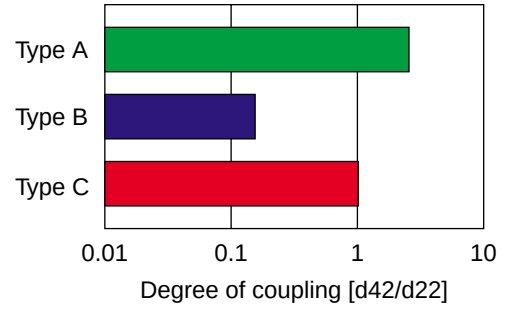


Fig. 9 Degree of coupling for ride comfort

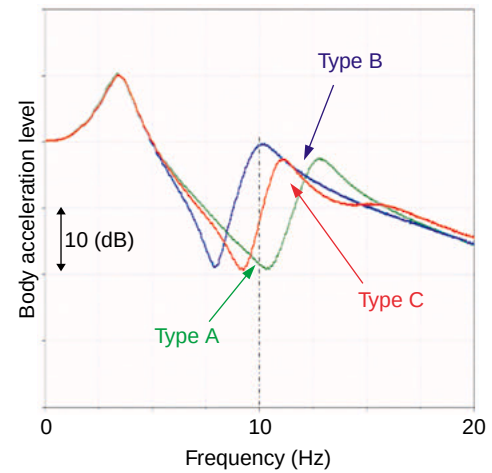


Fig. 10 Level of body vibration with CAE

a high degree of coupling between the lateral and rotational components, and low vibration levels. In addition, Type A exhibits a high lateral characteristic value, with vibration around 10 Hz being lower than for Type C.

According to the degree of coupling standard set by this study, coupling of approximately 1 to 3 does not adversely affect body lateral vibration for ride comfort. However, Fig. 11 shows that the vibration level increases at 0.2, indicating that 1 is an index that should be observed.

Viewed in terms of the overall completed vehicle, Type A can be considered the superior package.

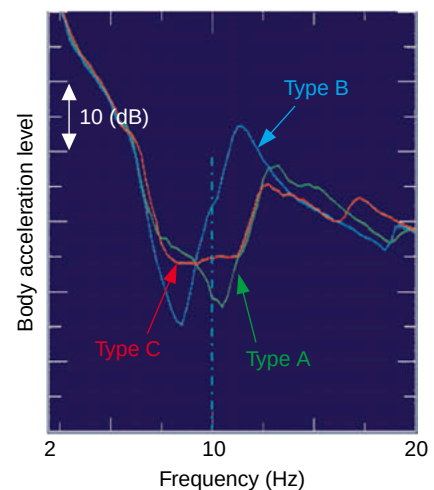


Fig. 11 Experimental level of body vibration

4. Conclusion

CAE investigations and actual vehicle tests of powertrain mount layouts verified that there is a high correlation between the degree of coupling advocated in this study and completed vehicle performance. The degree of coupling set in this study can be obtained by simple arithmetic calculations from characteristics matrices, making it easy for designers to understand ride comfort and idling noise and vibration performances relative to low-frequency phenomena in the layout stage. The non-diagonal terms in the matrix obtained by adding the mass and stiffness matrices are expressed by dividing the main mode by the related mode terms. Therefore, the meaning of the indices is also easy for designers to understand. Performance can be evaluated, even by designers without CAE calculation skills, and the indices can also be used for optimization in the stage before starting full-scale CAE investigations.

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