

Clarification of Transient Characteristics by Coupled Analysis of Powertrains and Vehicles

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ABSTRACT

With the goal of improving drivability, this research aimed to clarify the mechanism of vehicle longitudinal acceleration, focusing on tip-in acceleration. Conventional typical analysis methods include experimental modal and model-based analysis. However, since the former requires the measurement of impulses and other input forces while the vehicle is stopped, measurement under actual driving conditions is difficult. The latter requires characteristic values such as the stiffness and damping coefficients to be identified in advance, which cannot be achieved either easily or precisely. Therefore, this paper proposes a new experiment-based analysis method. This method enables the acquisition of engine torque and transmission torque/force by measuring only the acceleration values of some components under driving conditions. The key to realizing this method is the measurement of motions with the necessary and sufficient degrees of freedom (e.g., pitch motion) since these greatly affect vehicle behavior. Another aspect of this method is the adoption of appropriate equations to express the relationship between action and reaction forces and the measured acceleration (inertial force). Once the torque and force are obtained, a physical model can be devised even if the characteristic parameters of the vehicle are unknown. For the vehicle used in this study, tip-in acceleration behavior can be expressed by a model with only seven degrees of freedom, which expresses the important motions of the powertrain, driveline, and chassis. This model allows a simple understanding of the acceleration behavior and can be used for concept design.

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1. INTRODUCTION

Drivability is one of the basic vehicle performance indices. A typical method for estimating this performance characteristic is vehicle-based tests of tip-in acceleration. Transient vibration often occurs in the floor of the vehicle body during such tests due to the transmission of transient torque from the engine to the body and the backlash between the gears of the transmission and driveline, which causes rapid changes in the torque. Therefore, vibration is a key issue in vehicle development. This paper proposes a useful analysis method that aims to resolve this issue.

Various studies of control methods and phenomenon analysis have been carried out with the aim of improving vibration [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15]. Some have addressed control or analysis methods for the engine and the driveline [1, 2, 3, 4] because these components are important for explaining vibration phenomena. However, in an actual vehicle, the driveline is connected to the chassis (and body) system, and floor longitudinal acceleration is attained through the chassis system.

Therefore, the analysis must address the entire vehicle, including components from the engine to the chassis [9, 10]. Therefore, it is necessary to consider the interaction between the vehicle components to clarify the vibration phenomena. This process is challenging.

One possible analysis method is based on CAE models [7, 8, 9, 10]. This is useful for understanding vibration phenomena because CAE models are capable of identifying various states that cannot usually be measured in a real vehicle. In contrast, issues with this method include the fact that various characteristic values must be measured in advance, such as the stiffness and damping coefficients of the engine mounts, suspension bushings, and the like. For example, to measure the characteristics of a simple bushing, it might be necessary to lower the engine block from the vehicle body, which is very time-consuming. Another issue is the difficulty of understanding the causal relationships between various motions because a CAE model for an entire vehicle (i.e., a multi-body system) exhibits a high-order degree of freedom.

Other analysis methods are based on experiments. Experimental modal analysis (EMA) is a typical example [16, 17]. However, it can be difficult to apply this method except when the vehicle is stopped because impulse forces have to be applied with a hammer or the like. It is also difficult to apply exciting forces to a rotating shaft of an engine, which is the input point in a real vehicle.

Recently, transfer path analysis (TPA) has been adopted as a method that enables analysis under actual operating conditions [14, 15]. Although this method enables identification of the transfer forces acting on the engine mounts, mount stiffness measurement remains an issue. Furthermore, physical information cannot be obtained because only the system transfer function is fitted to the experimental data. In contrast, the operational TPA method does not require the measurement of mount stiffness [18, 19]. However the contribution ratio of the acceleration to each mount can be obtained only by measuring acceleration data. Since this ratio differs in accordance with the transfer function (input: force, output: velocity or acceleration), the wrong contribution ratio may be obtained. It is also not clear whether the exciting force required by the analysis can be obtained because the test is performed under actual operating conditions.

To resolve these issues, this paper proposes a new analysis method based on the results of experiments. This method is called hierarchical operational transfer path analysis and separation (HOTPAS). The ultimate goal of this method is to develop a principle model. The next section explains the concept of the analysis method and Section 3 describes the specific analysis procedure. This method is used to calculate the torque and force required to create the principle model. Section 4 uses this principle model to validate the proposed analysis method. The effectiveness of the method is demonstrated and compared with conventional methods.

2. CONCEPT

Figure 1 shows the issue relating to tip-in acceleration. This figure shows acceleration time series data for the floor of a vehicle body in 2nd gear at 40 km/h. The undamped oscillation that continues as the vehicle acceleration increases is one cause of a degradation in drivability. To resolve this issue, it is necessary to clarify the oscillation mechanism. However, this mechanism is not simple to identify because the engine torque is transmitted to the vehicle body through the transmission, tires, suspension, and engine mounts, as shown in Fig. 2. Therefore, the floor longitudinal acceleration is a result of the interactions between different components.

The proposed analysis method has two advantages. The first is that the engine torque and the torque and force transmitted between components can be obtained by measuring only the acceleration of the components under driving conditions. Sensor installation issues are often encountered when measuring the torque and force under driving conditions. This is because, in a powertrain, the sensors have to be installed on a rotating shaft and, in an engine mount system, the sensors have to be attached to the transfer path of the force. Equation (1) expresses the basic concept.

$$f = m\ddot{x} \quad (1)$$

where, f is the force, m is the mass, and $\ddot{x}$ is the acceleration. It is clear that Equation (1) can be used to obtain the force if the mass and the acceleration are measured, even though the force itself cannot be measured. The acceleration can be measured easily relative to the force. However, Equation (1) cannot be directly applied to an actual vehicle. An example is shown in Fig. 3(a). Each component (mass) of the vehicle is usually connected to two points, one upstream and one downstream, such as the gear train, such that the two forces f_1 and f_2 cannot be calculated with the measured acceleration data for one point.

Here, the conceptual system shown in Fig. 3(b) is considered. In this system, the transferred forces can be calculated by inverse transformation because the number of the force to be found is equal to that of the acceleration. In an actual vehicle, this idea cannot be applied because the number of the force is usually greater than that of the acceleration. However an idea like that shown in Fig. 3(b) may be applied in some cases provided that the relationships between the force (torque) and the corresponding acceleration (angular acceleration) are established adequately. Another key point is that the type of the transferred force (such as stiffness, damping, clutch, or backlash) does not matter because the transfer force only implies the relationship between the action and reaction forces. This is an advantage in performing the analysis.

The second advantage is that, whenever possible, the analysis creates a model with a low degree of freedom while maintaining the physical structure of the model. As a result, the analysis results can be understood easily. All the components of the chassis system, such as the engine block, which influence the acceleration phenomenon, have six degrees of freedom (rigid-body motions are considered). However, not all of these degrees of freedom exhibit large movements under driving conditions. Therefore, only those degrees of freedom with a large influence are selected.

Figure 4 shows the flow of the proposed analysis method from the experimental measurements to the making of the principle model. Unlike conventional CAE methods, this modeling process can be applied even to a vehicle for which the characteristic parameters cannot be obtained because it does not require these parameters in the first instance. The next section describes the analysis processes in detail, according to the flow shown in Fig. 4.

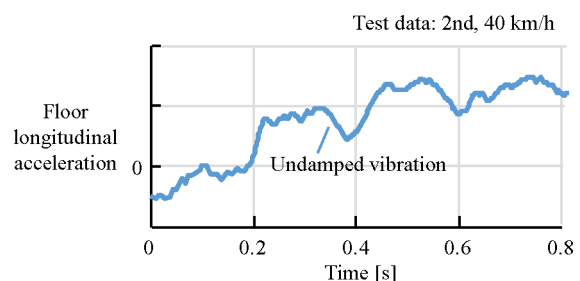


Figure 1. Floor longitudinal acceleration under tip-in acceleration

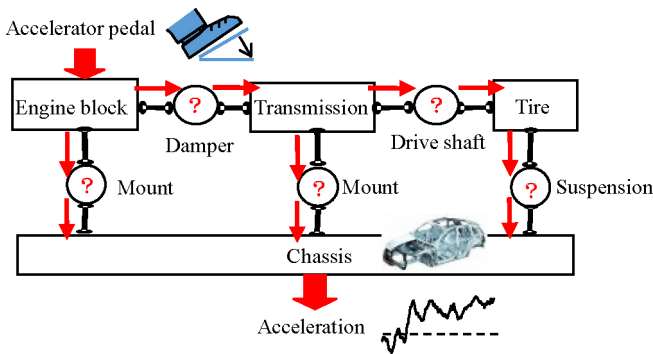
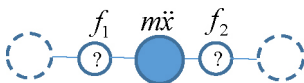
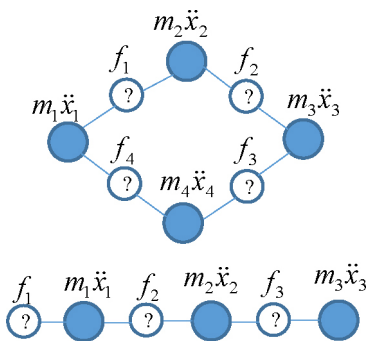


Figure 2. Issues addressed by proposed analysis method.



(a). One component considered



(b). All components considered

Figure 3. Concept of force calculation.

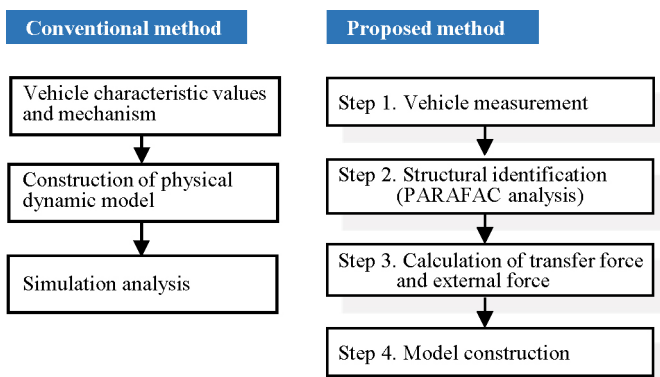


Figure 4. Flow of proposed analysis method.

3. EXPERIMENT-BASED ANALYSIS METHOD

Figure 5 shows a diagram of the overall system as it relates to the acceleration phenomenon. Paths from the engine torque to the vehicle include through the transmission and the engine mounts. Therefore, this analysis targeted the engine, the transmission, the driveline, and the chassis system, which contains the engine block and unsprung systems.

The frequency band for the analysis is from DC to around 20 Hz, in which the motions of the components shown in Fig. 5 can be expressed. In this frequency band, the motions of the chassis components (the engine block and the unsprung mass) can be considered as rigid-body motions. The analysis assumes that the following values are known.

- The mass or the inertia of each component
- The position of the center of gravity of the rigid body
- The running resistance of the vehicle

3.1. Vehicle Measurement Method

The following section describes step 1 (vehicle measurement) from the analysis flow in Fig. 4. As mentioned in the previous section, the motion of the chassis components can be regarded as rigid-body motions under tip-in acceleration. Therefore, these motions can be expressed as translation and rotational motions with six degrees of freedom around the center of gravity. For this reason, the chassis measurement is performed so that the motion of the center of gravity can be identified.

Three-axis acceleration sensors were used for the measurement. These were attached to three different points on components located far from the center of the gravity, such as the engine block, as shown in Fig. 6. The motions with six degrees of freedom around the center of the gravity can be calculated based on the acceleration data obtained from these three triaxial sensors (appendix). This figure also shows the acceleration sensors attached to the lower arm, sub-frame, and body floor. At this point, since it is not clear to which components the sensors should be attached, the sensors were attached to all possible components that might influence acceleration.

Only one sensor was attached to the vehicle floor. This is because the motion of the body (pitch, heave, and the like) is not a target of the analysis. Therefore, the attachment of only one sensor close to the center of gravity is not a problem because the acceleration is analyzed only in the longitudinal direction.

Figure 7 shows the acceleration sensors attached to the actual vehicle. Small sensors that can be attached in the limited space are used as much as possible.

At the same time, the rotational speed of each component listed in Table 1 is also measured to identify the rotational motion of the powertrain. For the analysis, angular velocity is converted to angular acceleration using the time difference of the rotational speed data. Table 1 also lists the measurement conditions.

In addition, although this method does not require the driveshaft (DS) torque to be identified, it was measured in the steps described in the following sections. This is because it can be measured easily and useful information for confirmation and verification can be obtained from the measured torque value.

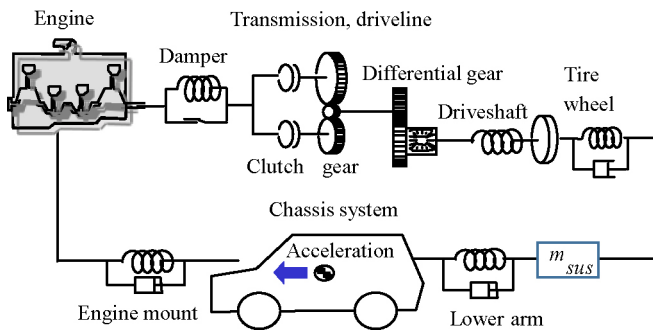


Figure 5. Target of analysis.

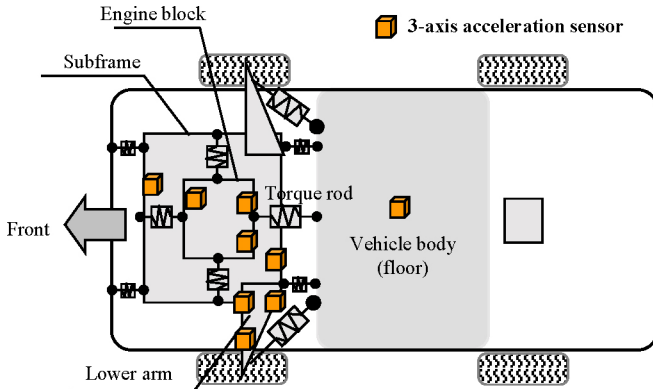
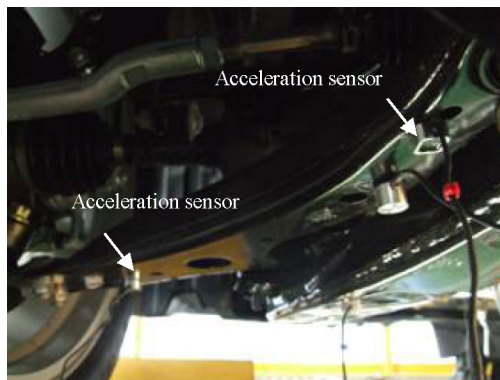
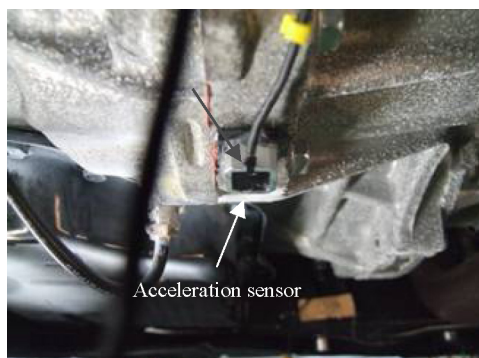


Figure 6. Measurement points with 3-axis acceleration sensors in chassis system.



(a). Lower arm



(b). Bottom of engine block

Figure 7. Sensors attached to chassis system.

Table 1. Measured components and conditions.

Acceleration in chassis system	Engine block, lower arm, subframe
Rotational speed	Engine, transmission, wheel
Measurement conditions	2nd, 40 km/h, tip in acceleration
Target vehicle	Naturally aspirated 2.5 L engine, 6-speed AT

3.2. Structural Identification (PARAFAC Analysis)

This section explains step 2 (structural identification) of the analysis flow. In this step, the important degrees of freedom are selected for tip-in acceleration to simplify the analysis and the modeling of the chassis system (engine block and unsprung mass). A method called parallel factor analysis (PARAFAC) [20, 21, 22] is applied. This is based on wavelet analysis of the acceleration data in the preceding section because the target phenomenon is transient.

Figure 8 shows the data structure used for the wavelet analysis in this method. This structure is a three-way data array consisting of frequency, time, and motion data. The motion refers to the six degrees of freedom of each element. However it is not simple to analyze this array directly since it contains an enormous amount of data. Therefore the array must be converted into as simple a data structure as possible. The adopted PARAFAC analysis method is a kind of tensor decomposition expressed by Equation (2) and Fig. 9.

$$x_{ijk} = \sum_{f=1}^F a_{if} b_{jf} c_{kf} \quad \text{for } i = 1, \dots, I, j = 1, \dots, J, k = 1, \dots, K \quad (2)$$

where x_{ijk} is the element of the tensor decomposition (which corresponds to the (i, j, k) element of the array data in Fig. 8, a is the element of frequency, b is the element of time, c is the element of motion, and f is the factor corresponding to each mode of the conventional mode analysis. Using tensor decomposition, the equation and the figure can be expressed simply. The coefficient of each a, b, c element is calculated so that the error between the original data shown in Figure 8 and the decomposed data is as small as possible.

As an example of the result, the elements of frequency $A = [a_1, \dots, a_I]$, which are calculated for every factor from 1 to 5, are shown in Fig. 10. Distinctive waveforms can be acquired, which have peaks at frequencies of 4, 6, 11, 12, and 16 Hz. The elements of factors 2, 4 are not considered in this analysis because these peaks correspond to stationary vibration occurring before the start of acceleration. Figure 11 shows the elements of motion $C = [c_1, \dots, c_K]$ of factor 1 (peak frequency of 4 Hz) as a bar graph. Factor 1 is the mode in which the drive shaft is mainly twisted. This graph shows the rate of contribution of each degree of freedom. Therefore, the important degrees can be selected with this graph.

Since the values of the motions (longitudinal motion of the unsprung mass, and the longitudinal and pitch motion of the engine block) indicated by the red circles in this figure are large, they are used for the analysis and modeling. The results for factors 3 and 5 exhibit