

Design and Evaluation of Output Profile Shaping of an Internal Combustion Engine for Noise & Vibration Improvement

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ABSTRACT

Engine torque profile shaping strategies have been proposed to reduce noise & vibration for passenger cars. However, it has not been sufficiently studied that feasible torque profile for vibration suppression is dependent on engine speed and target torque shape. On the other hand, combustion pressure profile shaping strategies have been proposed to reduce noise. However, there is almost no research of the quantitative evaluation of contribution of combustion pressure profile.

First, the torque profile shaping was studied. Pre-compensated torque and 2-step torque were selected as typical target torque profiles. An effectiveness of vibration suppression by two torque profiles was evaluated by both drivetrain vibration model and engine torque profile model which have been established well. As a result of studying the torque profile shaping, timing of torque rise by the 2-step torque generation is delayed or advanced. Torque profile by the pre-compensation is deformed in steep change part. Relative degree of deterioration in 2-step torque generation is smaller than that of the pre-compensator.

Second, the combustion pressure profile shaping was studied. A combustion noise generation model was constructed by the combustion pressure, a vibration transfer function in an engine structure, and a radiation factor at the engine surface. The surge in combustion pressure was modeled as triangular shape projection and analyzed in frequency domain to evaluate a contribution to combustion noise. As a result of studying the combustion pressure profile shaping, new analysis method by these models is proposed, so that radiation pressure for quantitative evaluation be obtained.

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1. INTRODUCTION

To suppress the amplitude of exciting force in a required frequency band is most effective for measures to solve noise and vibration problems. Because, preventing an exciting energy inflow to a system is based on a principle. Main exciting force of a vehicle is an internal combustion engine (ICE). Therefore, the exciting force suppression by output profile shaping of an ICE is obviously effective measures.

There are frequency bands to be required to suppress in exciting force. As shown in [figure 1](#), resonance of noise and vibration transfer paths depends on a frequency. Passenger's receptiveness to an unpleasant noise and vibration depends

also on the frequency. Therefore, exciting force suppression should be not over all amplitude suppression but effective frequency bands suppression with contribution analysis.

Moreover, as the factor to complicate this problem, it is mentioned that an ICE output is not only the exciting force but also given two functions. First, this is driving force and a response to operation (as expectation). Second, engine sound in acceleration is not noise but an acceleration feeling generator. It may be called "Sound" in the affirmative. However, any noise and vibration from a drivetrain which is not expected sound generation may be unpleasant.



Noise and vibration suppression may be classified into transfer path suppression and exciting force suppression. First, the active noise control is mentioned as an example of the transfer path suppression. This control aims at engine booming noise. The engine booming noise in a vehicle cabin is a resonance phenomenon at the second order frequency of engine rotation (in the case of 4 cylinders). The feature of this phenomenon is steady state sound. Adaptive feed-forward control, engine rotation speed used as a reference signal and measured acoustic pressure used as an error signal, is proposed and put to practical use to this phenomenon.[1] This control may be mentioned as the vibration suppression, for only the specific area, by control energy inflow after exciting energy inflow.

Second, output profile shaping of an ICE is mentioned as the exciting force suppression which prevents the exciting energy inflow. As the vibration suppression control, there are mean torque profile shaping and combustion pressure profile shaping.

The object of the mean torque profile shaping is suppression control of drivetrain torsional transient vibration. This vibration phenomenon is caused by a steep gradient, as step or impulse, of torque input from an ICE (or an electric motor), although it also caused by clutch engagement. This torque input is caused by tip-in or tip-out with an accelerator pedal. In contrast to engine booming noise, a steady state phenomenon of a specific frequency does not continue. As the measures of the vibration suppression, the mean torque profile shaping of an ICE has been proposed and put to practical use. [2, 3] This feasibility is dependent on both possibilities of arbitrary profile shaping in time series and restrictions by a frequency band of profile shaping. The mean torque may be understood intuitively as a direct current component, and be calculated quantitatively from a fuel injection quantity and an air flow rate. The quantitative calculation of pulsating torque of an alternate current component has not been accepted in general.

The object of the combustion pressure profile shaping is combustion noise suppression, for typical example: diesel knock sound. Although this combustion noise may be observed as a steady state phenomenon, it is a broadband noise which contains vibrations damped within a short time. In consideration of exciting source, the combustion noise may be expressed as a transient response phenomenon to the impulsive combustion pressure within a combustion cycle.

The combustion pressure profile shaping and the exhaust emission control have been performed by a multi-stage injection. By the measures, the combustion noise suppression is achieved without change of mean torque as driving force. A quantitative engine sound prediction technology is important for a reduction of development costs of a calibration process of the measures. As the sound prediction technology, it has been proposed that the combustion noise is dependent on maximum gradient of combustion pressure. However, there is almost no research of the quantitative contribution analysis (or engine sound quality prediction) with a model. Moreover, the sound

prediction technology has possibilities of application to a gear rattle problem. The gear rattle is a vibration phenomenon by impulsive inputs which impact the gear tooth flanks each other. [4] The phenomenon occurs when engine torque pulsating amplitude is larger than drag torque. The quantitative analysis to influences on the noise has not been also proposed.

Then, the engine torque profile model which is formulized and quantified by parameters of a frequency domain is proposed. About the mean torque profile shaping, vibration suppression effectiveness and feasibility are evaluated quantitatively by the model. Moreover, the prediction method of engine acoustic radiation from both the combustion pressure model and the transfer characteristic model is proposed newly for evaluation of a quantitative contribution of sound quality. A principle of the combustion pressure profile shaping is examined by this model.

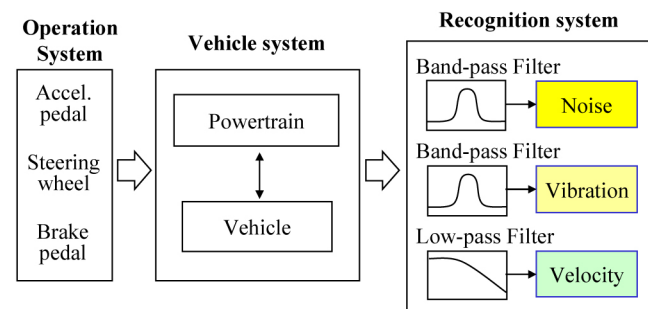


Figure 1. Noise & vibration excitation and recognition system

2. DRIVETRAIN VIBRATION SUPPRESSION CONTROL USING TORQUE PROFILE SHAPING

2.1. Engine Torque Profile Model

Here, an engine torque profile model is introduced for examining feasibility of the drivetrain torsional vibration suppression by the mean torque profile shaping. Priority is given to simple and linear engine plant modeling in a prospect of controller design.

It is proposed that a time series torque profile may be expressed by frequency domain parameters and a mean torque profile. [5] Although the engine torque profile is composed of both combustion pressure and reciprocating inertia forces, the method is simple and avoids these detail calculations. This concept is shown in figure 2. It may take into consideration the air intake response delay and the dead time of torque generating on demand. The harmonics basis function is expressed by the 8th order trigonometric functions so that the engine torque profile is approximated with sufficient accuracy. Then, it is described as the following.

$$T_i(t) = T_m(i) \left\{ 1 + \sum_{j=1}^8 \left[a_{j/2} \sin\left(\frac{j\omega_0 t}{2}\right) + b_{j/2} \cos\left(\frac{j\omega_0 t}{2}\right) \right] \right\} + m r \omega_0^2 b_{12} \sin(2\omega_0 t) \quad (1)$$

Here, t : time, T_i : engine torque, $T_m(i)$: mean torque at i combustion cycle, ω_0 : engine speed, a , b : torque coefficient, j : rotational order, m : reciprocating inertia mass, r : crank radius

The calculation result by the Eq. (1) is shown in figure 3. As calculation conditions, it is assumed that a 4-cylinder engine of passenger cars drives at 1500[rpm] steady state speed. It is shown that an accelerating transient torque profile may be expressed by a frequency band of less than the 0.5th order of rotation. Moreover, when the target torque rises at 0.1[sec] as a first-order delay characteristic, the dead time is expressed by the first peak of combustion torque pulsation at 0.16[sec]. About the total torque, it shows that the amplitude of the reciprocating inertia forces is too larger than that of the combustion pressure.

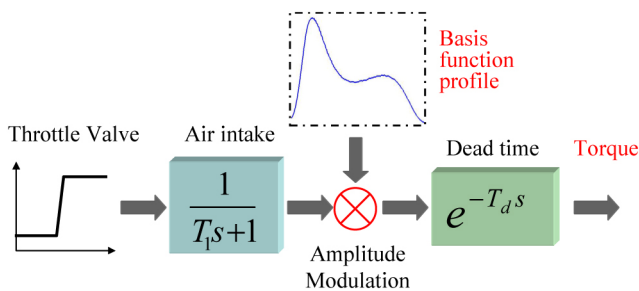


Figure 2. Engine pulsation torque generation

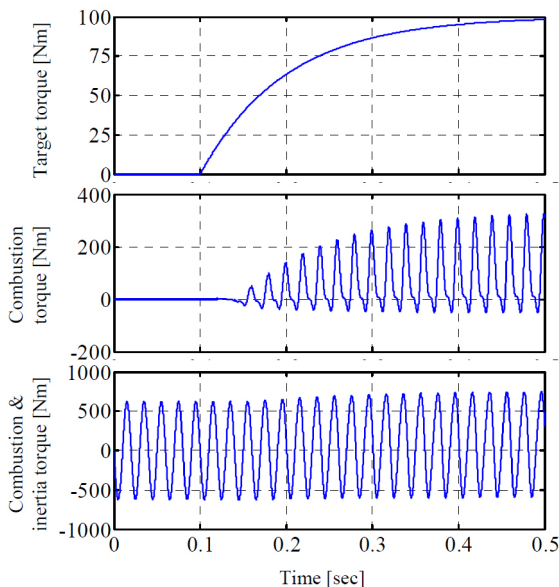


Figure 3. Target mean torque and torque profile with harmonics

2.2. Drivetrain Vibration Suppression Control

A pre-compensator which removes resonance frequency components of a drivetrain system from engine torque is an effective method to suppress a drivetrain torsional vibration. In particular, this technology has a wide scope because the drivetrain vibration by the tip-in or tip-out can be caused in any vehicles.

The concept of this method is shown in figure 4. A transfer function of a plant is defined as $G_p(s)$ which is approximated by a second-order delay system. When a target characteristic including a controller is defined as $G_t(s)$, a pre-compensator characteristic $W(s)$ can be described as the following.

$$W(s) = \frac{G_t(s)}{G_p(s)} = \frac{s^2 + 2\zeta_m \omega_{N1}s + \omega_{N1}^2}{s^2 + 2\zeta_d \omega_{N1}s + \omega_{N1}^2} \quad (2)$$

Here, ω_{N1} : resonance frequency of a plant, ζ_m : damping coefficient of a plant, ζ_d : designed damping coefficient of a system

As the meaning of Eq. (2), this method may be described as the change of the damping coefficient of a system. As the meaning of a frequency characteristic, the pre-compensator $W(s)$ may be called as a notch filter for the resonance frequency of the drivetrain system. Moreover, the compensated target torque of $W(s)$ output is modulated pulsating torque by the engine torque profile model. This pulsating torque is input into the spring-mass system of transmission (T/M). The drivetrain vibration model of T/M is established as a general one.

Moreover, in order to consider a feature of the method in time series, an example of a simulation result of a step response is shown in figure 5. The compensated target torque is formed as rapid falling after the rise at 0.1 [sec]. Here, control effectiveness decreases due to the following three reasons. [6] First, output torque can be changed only at interval of one combustion cycle. Second, in torque control by air flow, a torque change gradient is dependent on a time constant of an intake manifold. The target profile cannot be realized at low engine speed. [7] Third, vehicles model has an error between resonance frequency and notch frequency.

The evaluation on the pre-compensator has attracted attention only in the mean torque profile. This is because a high order pulsating component is damped by a torsion spring built in a clutch. Relationship between feasibility of the target profile and effectiveness of the actual profile has not been examined quantitatively.