[Power Electronics Technology]

Shaft Torque Excitation Control for Drivetrain Bench

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Abstract

We developed a technology for the excitation control of the input shaft torque to be carried out at a drivetrain bench at a specified amplitude.

In order to secure stable excitation at various frequencies, the following three control technologies have been combined:

(1) Resonance suppression control at a high-frequency resonance point where the amount of damping is small

(2) Steady-state torque control at a low-frequency resonance point where variation in resonance frequency is large and

(3) Automatic compensation control of excitation amplitude command

By using these control technologies, we realized control of the shaft torque to a stable amplitude without depending on the non-linear torsional characteristic of a product under test.

1 Preface

Against the background of rising social demands for better environmental performance, automotive industries are demanding more advanced performance and functions for dynamometer applied testing systems. In the field of dynamometer control, steady-state control of torque and revolution speed was a main goal in conventional practice. Recently, however, the control of torque and revolution speed has been a target to be attained through simulation of characteristics of real cars.

For driving motors of drivetrain benches to be used for the testing of transmissions and torque converters, simulation control of vibratory torques generated by engines is requested by the auto industry.

This paper introduces our newly developed shaft torque excitation control system for drivetrain benches.

2 System Configuration and Purpose and Challenges for Control

Fig. 1 shows a system configuration of the drivetrain bench. The input and output sides of the test piece are equipped with driving motors and power absorbing motors, respectively. Revolutions
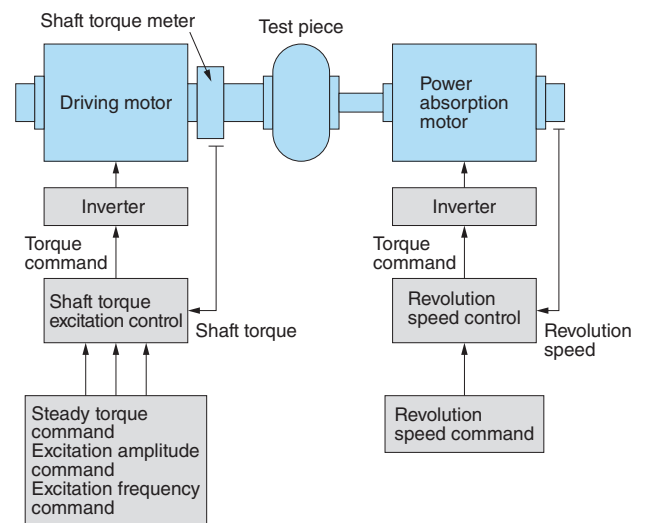

Fig. 1   System Configuration of the Drivetrain Bench

Mechanical configuration and control system configuration (names of detection signals, command signals, and control functions) are shown.

are controlled with power absorbing motors, while driving motors are used for the excitation control of input shaft torque of the test piece toward the level of engine torque.

The major purpose of the recently developed excitation control system is to control the average torque and the amplitude and frequency of the
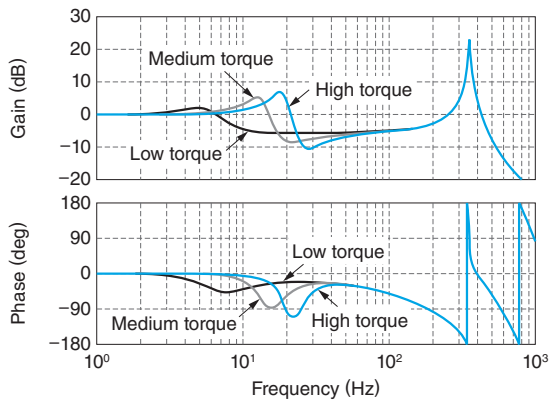
In a low-frequency band (several Hz to tens of Hz), the resonance frequency changes with the intensity of the torque. In a high-frequency band (hundreds of Hz), the resonance frequency does not change, but the phase lag becomes large (approx. 180 degrees).

vibratory torque of input shaft torques measured by the shaft torque meter installed between the driving motor and the test piece.

Fig. 2 shows frequency characteristics (from the inverter torque reference to the shaft torque.) of an ordinary drivetrain bench. The torque converter, which is a test piece of the drivetrain bench, is provided with a non-linear spring within its interior. A resonance frequency therefore varies depending on the steady intensity of the shaft torque.

In **Fig. 2**, the resonance frequency is changing in the frequency band from several Hz to tens of Hz. The resonance point appearing in the band of hundreds of Hz is attributable to the mechanical rigidity of driving motor, shaft torque meter, and their coupling.

The calculation below is shown in regard to the excitation frequency to be reproduced with a driving motor. For a 4-cycle engine, a large vibratory torque is generated, having a frequency of the number of cylinders $\times$ 0.5 $\times$ revolution. If the engine revolution is assumed to be 600min^{-1} to 6000min^{-1}, the excitation frequency will be 15 to 150Hz for a three-cylinder engine and 40 to 400Hz for an eight-cylinder engine. When simulation of vibratory torque is intended for three-cylinder to eight-cylinder engines, it is therefore, necessary to excite the shaft torque at an amplitude within the band of 15 to 400Hz.

3 Shaft Torque Excitation Control

3.1 Three-Inertia Model of the Drivetrain Bench

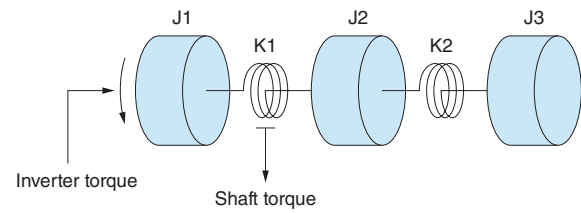**Fig. 3** shows a three-inertia model of the drive-

The diagram shows the corresponding parts for inverter torque and shaft torque detection when characteristics of the drivetrain bench are approximated with a three-inertia model.

train bench. For modeling, J1 mainly represents the moment of inertia of the driving motor, J2 is mainly the moment of inertia of the test piece, J3 is mainly the moment of inertia of the power absorbing motor, K1 is the torsional rigidity of the coupling shaft between the driving motor and the test piece, and K2 is the non-linear torsional rigidity of the test piece. Accordingly, J1, K1, J2, and J3 have almost no dependence on the intensity of shaft torque, but K2 has a characteristic that changes with the intensity of shaft torque.

The resonance point in the band of hundreds of Hz shown in **Fig. 2** is mainly determined by the characteristics of J1, K1, and J2. The resonance point in the band of several Hz to tens of Hz is mainly determined by the characteristics of J1 + J2, K2, and J3.

3.2 Resonance Suppression Control

According to the frequency characteristics shown in **Fig. 2**, a lag of about 180 degrees in phase can be recognized in the resonance frequency staying in the band of hundreds of Hz. This phase lag is attributable to the detection lag of the shaft torque meter and also to the sampling time of the controller. It is difficult for Proportional-Integral Derivative (PID) controller to accomplish resonance suppression at a resonance point where such a large phase lag is exhibited. For a solution, we established a resonance suppression controller by using μsynthesis approach where a controlled object is a two-inertia model consisting of J1, K1 and J2.

3.3 Steady Torque Control

Since the phase lag is not too big at a resonance frequency in the frequency band of several Hz to tens of Hz, we established a steady torque controller by using PID controller. Based on the assumption that a two-inertia model consisting of a
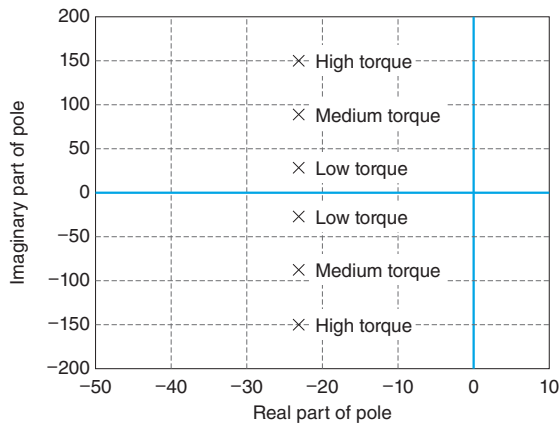
Fig. 4　Root Locus of Steady Torque Control

The diagram shows the behavior of changes in a predominant pole under steady torque control characteristic when the intensity of torque changes and these causes changes in low-level resonance frequency.

Fig. 5　Command Response of Steady Torque Control

The diagram shows the behavior of changes in command responses for steady torque control when the intensity of torque changes and this causes changes in low-level resonance frequency.

Fig. 6　Disturbance Response of Steady Torque Control

The diagram shows the behavior of changes in disturbance torque for steady torque control when the intensity of torque changes and this causes changes in low-level resonance frequency. At a high torque (high resonance frequency), the gain becomes high at tends of Hz, but this value remains to stay as high as a maximum gain at a low torque.

sum of J1 and J2, K2 where its resonance frequency (several Hz) becomes lowest as shown in **Fig. 2** and J3, is regarded as a nominal model of the PID controller, we determined a PID parameter by using the pole placement method so that the closed loop pole can be stabilized. In this case, I-PD control approach has been adopted for control system configuration.

　Fig. 4 shows the behavior of the root locus when the resonance frequency changes. Stability is maintained in the closed loop pole even though the resonance frequency should change from several Hz to tens of Hz. At the same time, however, as the shaft torque increases, the pole damping decreases. Consequently, we confirmed command response and disturbance response from the load side. **Fig. 5** shows the command response, and **Fig. 6** also shows the disturbance response. The command response hardly changes even though there are changes in shaft torque. In regard to the disturbance response, the gain seems to rise along with an increase in the shaft torque, but its rising mode is suspended almost at the same level of the maximum gain obtained with the nominal model used when determining the PID parameter. As such, we concur that there is no problem in terms of control stability.

3.4　Excitation Amplitude Control

　Since the nominal model used at the time of determining the PID parameter is set to have a mechanical characteristic at the lowest resonance frequency and a system configuration for I-PD control is adopted, the frequency band is maintained at
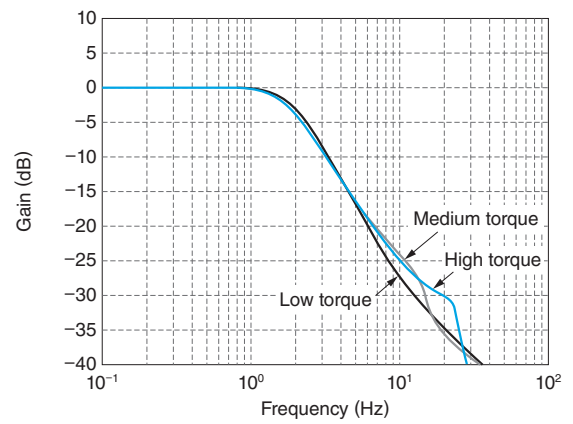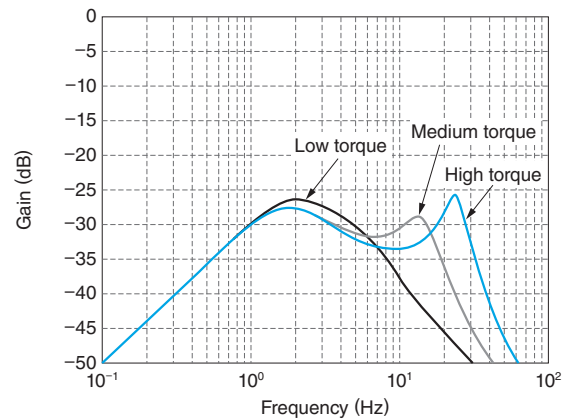
several Hz for command response in steady torque control. Accordingly, it is impossible to control the shaft torque excitation to a desired amplitude in a frequency band of 15Hz to 400Hz by simply applying an excitation torque command input to the shaft torque for steady torque control. Meanwhile, when trying to raise the frequency band of feedback control as high as 400Hz for an instantaneous value of shaft torque as in the case of steady torque control, it is generally difficult to achieve due to the presence of a resonance point that changes within a range of several Hz to tens of Hz. For this reason, we adopted a system configuration for excitation amplitude control instead.

Our system configuration is devised for automatic correction of excitation amplitude commands so that the shaft torque amplitude is detected from the excitation frequency command value and the detected shaft torque value, and then the detected amplitude is adjusted to the command value.

3.5 Result of Simulation

Fig. 7 shows an overall configuration of the shaft torque excitation control system. **Fig. 8** shows the result of shaft torque excitation control simulation where three-cylinder and eight-cylinder engines are used for simulation. Under the conditions that steady torque command is set at 100N·m and excitation amplitude command at ±500N·m, the revolution value is changed in a ramp state from 600min⁻¹ to 6000min⁻¹. "Without shaft torque excitation control" corresponds to a case when a steady torque
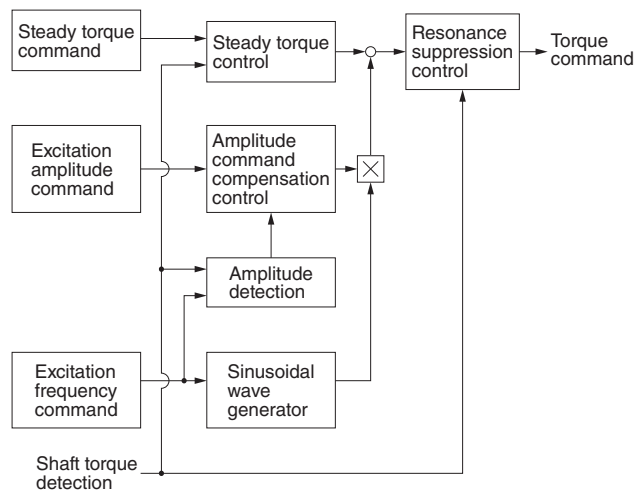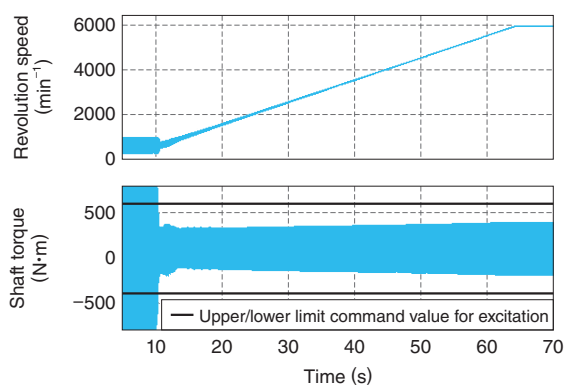
| Fig. 7 | Overall Configuration of the Shaft Torque Excitation Control System |

This diagram shows a combination of resonance suppression control at a high frequency range resonance point, steady torque control at a low frequency range, and compensation control for excitation amplitude commands.
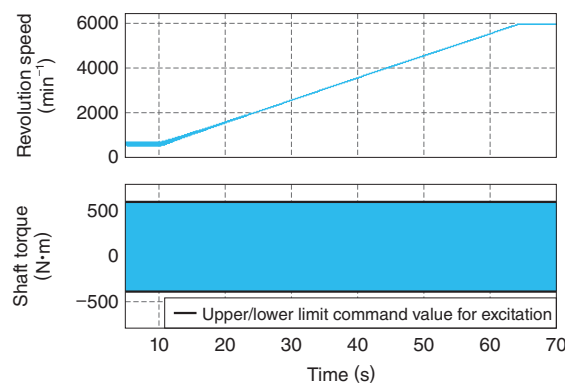
(a) Simulation of three-cylinder engine without shaft torque excitation control
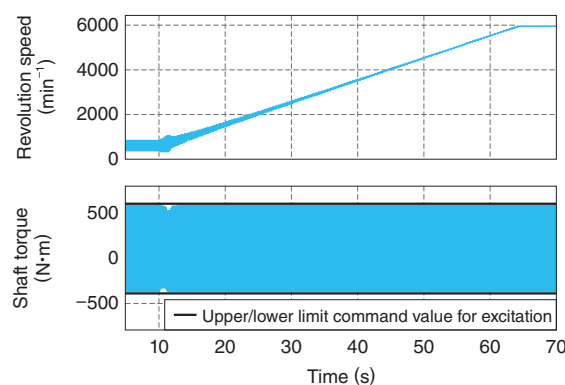
(b) Simulation of three-cylinder engine with shaft torque excitation control
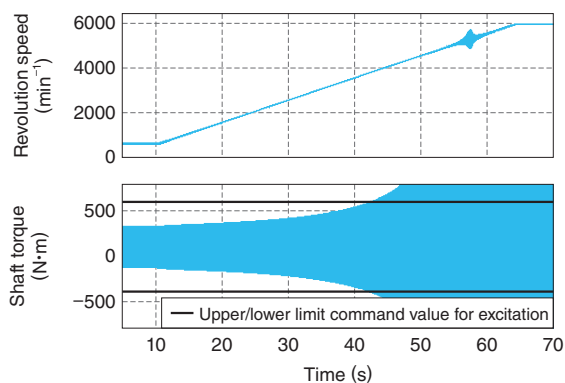
(c) Simulation of eight-cylinder engine without shaft torque excitation control

(d) Simulation of eight-cylinder engine with shaft torque excitation control

| Fig. 8 | Result of Shaft Torque Excitation Control Simulation |

A notation of "without shaft torque excitation control" corresponds to the waveforms observed when an excitation torque command is superposed without amplitude command compensation control. "With shaft torque excitation control" falls on the waveforms observed when amplitude command compensation control is carried out. When amplitude command compensation control is not performed in the case of simulation of three-cylinder engines, there is an increase in amplitude due to the effect of gain characteristics at the low-level resonance point (a). For the simulation of eight-cylinder engines, there is an increase in amplitude due to the effect of gain characteristics at the high-level resonance point (c). When amplitude command compensation control is carried out, shaft torque is excited at the commanded amplitude.

command input and an excitation command input are directly given to the inverter. In this case, the shaft torque amplitude changes in response to the gain characteristics as shown in **Fig. 2**. It is known that the system is controlled to the approximate desired amplitude in the case "with shaft torque excitation control."

4 Postscript

This paper has introduced our system to control the input shaft torque of a drivetrain bench to a desired steady torque and excitation amplitude. We will continue to make efforts to develop more advanced dynamometer controlling technologies that can reproduce real car characteristics as much as possible.

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