

Start-Up Transient Vibration Analysis of a Vehicle Powertrain System Equipped with a Nonlinear Clutch Damper

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ABSTRACT

The transient vibration phenomenon in a vehicle powertrain system during the start-up (or shut-down) process is studied with focus on the development and experimental validation of the nonlinear powertrain models. First, a new nonlinear four-degree-of-freedom torsional powertrain model for this transient event, under instantaneous flywheel motion input, is developed and then validated with a vehicle start-up experiment. Second, the interactions between the clutch damper and the transmission transients are established via transient metrics. Third, a single-degree-of-freedom nonlinear model, focusing on the multi-staged clutch damper, is developed and its utility is then verified.

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INTRODUCTION

Transient events during the engine start-up (or shut-down) process are receiving attention recently for both conventional internal combustion (IC) engines [1-2] and hybrid electrical vehicles [3, 4, 5, 6, 7]. Such vibration issues are now more noticeable due to two reasons. First, downsizing of the IC engine including the fewer cylinders, cylinder deactivation, and light-weighting has resulted in higher pulsating torques. Second, new start-up/shut-down methods induce more frequent resonant transient vibration amplifications [3, 4, 5, 6, 7]. The engine instantaneous firing frequency passes through the natural frequencies of the powertrain torsional modes, resulting in a vibration amplification and thus increasing human discomfort. Due to multiple discontinuous nonlinearities in the clutch damper and the transmission [8-9], the transient vibration in the nonlinear powertrain subsystem is yet to be fully analyzed. Thus this article intends to develop powertrain system nonlinear models and examine the non-stationary process using numerical, experimental, and analytical approximation methods.

PROBLEM FORMULATION

The effects of discontinuous nonlinearities, such as piecewise linear stiffness, hysteresis, and preload of a multi-staged clutch damper on the stationary periodic vibro-impacts events have been studied in frequency and time domains [10, 11, 12, 13, 14, 15, 16, 17, 18]. For a non-stationary process, the transient envelope of oscillations is normally utilized to evaluate the peak amplification level. Li and Singh [2] and Sen et al. [19-20] utilized the Hilbert transform [21] to establish the transient envelope of nonlinear single-degree-of-freedom system. Earlier, indirect analytical methods were developed by Markert and Seidler [22] and Hok [23] to estimate transient envelope of linear SDOF system. Also, closed form solutions of the transient envelope of a linear oscillator have been developed in the

instantaneous frequency domain by Li and Singh [24, 25, 26]. However, the transient events occurring in the nonlinear multi-degree-of-freedom (MDOF) torsional system have not been quantitatively interpreted. Therefore, a MDOF powertrain system of a conventional vehicle with a clutch damper and manual transmission is considered in this article. As shown in Figure 1(a), a 4DOF nonlinear torsional model excited by instantaneous flywheel torsional displacement $\bar{\theta}_0(\bar{t})$ and torsional velocity $\bar{\dot{\theta}}_0(\bar{t})$ will be developed where a bar over a symbol indicates a dimensional parameter. This MDOF model and its reduced order version will be experimentally validated and their utility will be discussed.

NONLINEAR MDOF MODEL OF POWERTRAIN SYSTEM

The 4DOF nonlinear model is developed based on the assumption that the flywheel motion (displacement and velocity) can be employed as the system input since the flywheel motion should not be affected by the motion of other downstream elements due to its massive inertia. The downstream driveline components including the propeller shaft, axles, and the differential are decoupled from the transmission during the engine start-up. Here, $\bar{I}_i$, $\bar{C}_{ij}$, and $\bar{K}_{ij}$ indicate the torsional inertia, viscous damping, and stiffness of the lumped element, respectively. Two gear backlashes with the backlash size $\bar{b}$ (mm) between the headset gear pair and the unloaded gear pair are present. An asymmetric multi-staged clutch damper is placed between the flywheel and clutch hub; its characteristics curve is given in Figure 1(b), where $\bar{T}_{01}$ is the torque transmitted through the clutch damper, $\bar{K}$ and $\bar{H}$ represent the stiffness and hysteresis, and *I*, *II*, and *III* indicate the pre-damper, main spring, and stopper stages; and subscripts *c* and *d* imply coast and driving sides, respectively. The detailed nonlinear function $\bar{T}_{01}(\bar{\theta}_{01}, \bar{\dot{\theta}}_{01})$, where the hysteresis is

determined by $\bar{\theta}_{01}$ and $\bar{\dot{\theta}}_{01}$, can be found in [26]. The system governing equations under the flywheel motion input $\bar{\theta}_0(\bar{t})$, $\bar{\dot{\theta}}_0(\bar{t})$ are given as follows:

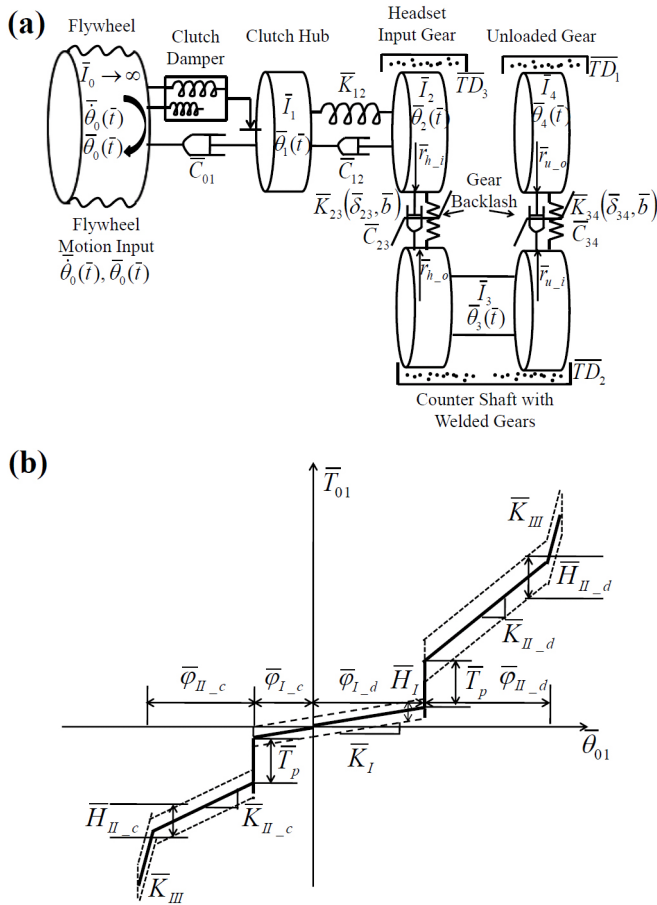


Figure 1. Powertrain torsional system model. (a) 4DOF nonlinear powertrain model given motion input from the flywheel; (b) Asymmetric nonlinear characteristics of a multi-staged clutch damper.

$$\begin{aligned} \bar{I}_1 \ddot{\bar{\theta}}_1(\bar{t}) + \bar{C}_{01}(\bar{\dot{\theta}}_1(\bar{t}) - \bar{\dot{\theta}}_0(\bar{t})) - \bar{T}_{01}(\bar{\theta}_{01}(\bar{t}), \bar{\dot{\theta}}_{01}(\bar{t})) + \\ \bar{C}_{12}(\bar{\dot{\theta}}_1(\bar{t}) - \bar{\dot{\theta}}_2(\bar{t})) + \bar{K}_{12}(\bar{\theta}_1(\bar{t}) - \bar{\theta}_2(\bar{t})) = 0 \end{aligned} \quad (1)$$

$$\begin{aligned} \bar{I}_2 \ddot{\bar{\theta}}_2(\bar{t}) + \bar{C}_{12}(\bar{\dot{\theta}}_2(\bar{t}) - \bar{\dot{\theta}}_1(\bar{t})) + \bar{K}_{12}(\bar{\theta}_2(\bar{t}) - \bar{\theta}_1(\bar{t})) + \\ \bar{F}_{head}(\bar{\delta}_{23}(\bar{t}), \bar{b}) \bar{F}_{h-i} + \bar{C}_{23} \bar{\dot{\delta}}_{23}(\bar{t}) \bar{F}_{h-i} = \bar{T}_{D3} \end{aligned} \quad (2)$$

$$\begin{aligned} \bar{I}_3 \ddot{\bar{\theta}}_3(\bar{t}) + \bar{F}_{head}(\bar{\delta}_{23}(\bar{t}), \bar{b}) \bar{F}_{h-o} + \bar{C}_{23} \bar{\dot{\delta}}_{23}(\bar{t}) \bar{F}_{h-o} + \\ \bar{F}_{unload}(\bar{\delta}_{34}(\bar{t}), \bar{b}) \bar{F}_{u-i} + \bar{C}_{34} \bar{\dot{\delta}}_{34}(\bar{t}) \bar{F}_{u-i} = \bar{T}_{D2} \end{aligned} \quad (3)$$

$$\bar{I}_4 \ddot{\bar{\theta}}_4(\bar{t}) + \bar{F}_{unload}(\bar{\delta}_{34}(\bar{t}), \bar{b}) \bar{F}_{u-o} + \bar{C}_{34} \bar{\dot{\delta}}_{34}(\bar{t}) \bar{F}_{u-o} = \bar{T}_{D1} \quad (4)$$

Here, $\bar{T}_{Di}$ ($i = 1, 2$, and 3) represents the drag torques on the gears and shafts in the transmission that are caused by the lubricant and bearings and assumed to be constants; $\bar{F}_{head}(\bar{\delta}_{23}(\bar{t}), \bar{b})$ and $\bar{F}_{unload}(\bar{\delta}_{34}(\bar{t}), \bar{b})$ are the nonlinear gear mesh forces from the headset gear pair and unloaded gear pair with the backlashes, respectively. Detailed nonlinear function expressions can be found in [26].

EXPERIMENTAL VALIDATION

In order to validate this nonlinear model, a vehicle start-up experiment is conducted on a medium-duty truck with a 4-cylinder gasoline engine and a 6-speed manual transmission. The schematic of the test rig is shown in Figure 2; other downstream driveline components including the propeller shaft, axles, and the differential are not shown since they are decoupled during the engine start-up with manual transmission. Two magnetic sensors are utilized to measure the absolute torsional velocities of the flywheel ($\bar{\dot{\theta}}_0(\bar{t})$) and clutch hub ($\bar{\dot{\theta}}_1(\bar{t})$), and the key system parameters are estimated from the physical dimensions or vehicle measurements. For the sake of comparison, modal properties of the linearized model are calculated as reported in Table 1. In the linearization, net backlash size $\bar{b}$ is assumed to be zero and only one stage of the clutch damper is activated. Then, the measured instantaneous flywheel motion ($\bar{\theta}_0(\bar{t})$) is applied to the nonlinear 4DOF model as an input. Finally, the relative velocity through the clutch damper ($\bar{\dot{\theta}}_{01}(\bar{t}) = \bar{\dot{\theta}}_0(\bar{t}) - \bar{\dot{\theta}}_1(\bar{t})$) is predicted via the nonlinear model and compared with test measurements in both time and time-frequency domains. The comparison shown in Figure 3(a), (b) and Table 2 clearly indicate that the 4DOF nonlinear model yields accurate predictions, especially for $\bar{\theta}_{01_max}$, $\bar{\theta}_{01_min}$, $\bar{\theta}_{01_pp}$, and $\bar{t}_{max}$. Also, the time-frequency domain results, calculated by short-time Fourier transform algorithm, in Figure 3(c), (d) suggest that the 4DOF nonlinear model is able to accurately capture first two dominant orders of the instantaneous frequency and the instant (around 0.4 s) when the maximum amplification ($\bar{\theta}_{01_max}$ or $\bar{\theta}_{01_min}$) occurs (dashed circle). In addition, the nonlinear model shows that the clutch mode with stage I activated (12 Hz) results in the amplification, which is consistent with the test measurement. The minor discrepancies in the comparison may be caused by other damping mechanisms; a discussion of these is beyond the scope of this article.

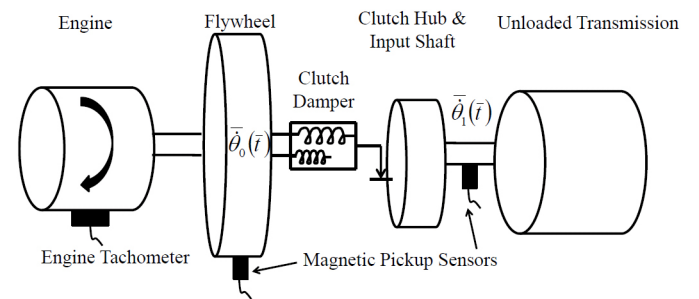


Figure 2. Schematic of start-up test for a medium-duty truck with a 4-cylinder gasoline engine and 6-speed manual transmission.

CORRELATION BETWEEN CLUTCH VIBRATION AMPLIFICATION AND GEAR RATTLE

By utilizing the validated 4DOF nonlinear model with the same parameters and measured flywheel input $\bar{\theta}_0(\bar{t})$ in Figure 4(a)), the correlation between the transient vibration amplification at the multi-staged clutch damper and the gear rattle in the transmission are found. As shown in Figure 4(b), (d), while flywheel speed $\bar{\theta}_0(\bar{t})$ rapidly increases up to 1600 rpm within the first 1.0 s, $\bar{\theta}_{01}(\bar{t})$ exhibits significant vibration between 0.3 and 0.5 s, which induces vibro-impacts at the transitions between clutch stages I and II on both driving and coast sides. Over the same duration, $\bar{\delta}_{23}(\bar{t})$ and $\bar{\delta}_{34}(\bar{t})$ exhibit severe transient gear rattle events (single- and double-sided impacts).

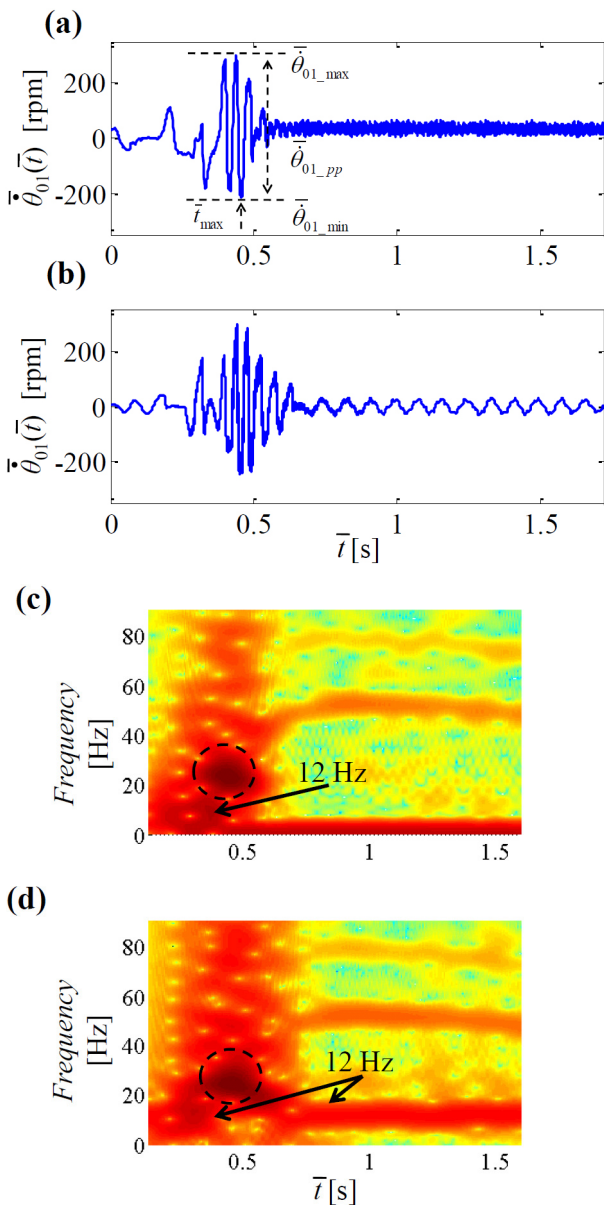


Figure 3. Experimental validation of the 4DOF nonlinear model given instantaneous flywheel motion input. (a) $\dot{\theta}_0(\bar{t})$ from the test; (b) $\dot{\theta}_0(\bar{t})$ from the 4DOF model; (c) Spectrogram of $\dot{\theta}_0(\bar{t})$ from the test; (d) Spectrogram of $\dot{\theta}_0(\bar{t})$ from 4DOF model.

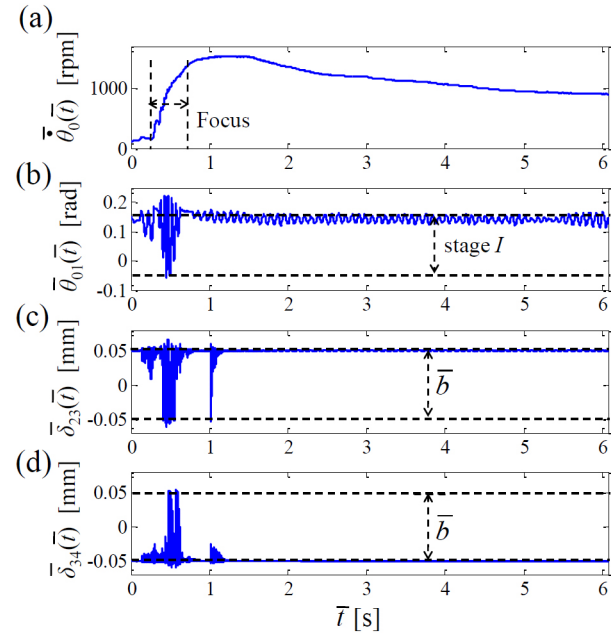


Figure 4. Time domain responses of the nonlinear 4DOF system given the measured instantaneous input $\bar{\theta}_0(\bar{t})$.

Table 1. Modal analysis of the linearized 4DOF powertrain system model.

	Mode (Stage <i>I</i> activated)				Mode (Stage <i>II_d</i> activated)			
	1	2	3	4	1	2	3	4
Natural Frequency (Hz)	12	446	2172	2515	52	460	2172	2515
	Modal Vector				Modal Vector			
Flywheel	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Clutch Hub	1.00	1.00	0.01	-0.02	0.93	1.00	0.01	-0.02
Headset gear and input shaft	1.00	-0.19	-0.19	0.80	1.00	-0.18	-0.20	0.86
Counter shaft and “welded” gears	1.00	-0.22	0.01	-0.33	0.93	-0.20	0.01	-0.37
Output gear	1.00	-0.23	1.00	1.00	0.93	-0.20	1.00	1.00

Table 2. Validation of the 4DOF nonlinear powertrain system model given measured flywheel motion $\bar{\theta}_0(\bar{t})$.

Symbol (Units)	Experiment	4DOF Nonlinear Model	Prediction Error
$\dot{\theta}_{01_max}$ (rpm)	299	298	0.3%
$\dot{\theta}_{01_min}$ (rpm)	-215	-232	8.2%
$\dot{\theta}_{01_pp}$ (rpm)	513	531	3.3%
t_{max} (s)	0.44	0.44	0%

Evaluation metrics are proposed next to further describe the correlation observed in Figure 4. First, $\bar{\theta}_{01_pp} = \max(\bar{\theta}_{01}(\bar{t})) - \min(\bar{\theta}_{01}(\bar{t}))$ is utilized, and then two new metrics $\bar{\delta}_{23_rattle}$ (%) and $\bar{\delta}_{34_rattle}$ (%), as originally developed in [26], are used to evaluate the rattle severities of headset and unloaded gear pairs respectively. The stiffness of the pre-damper ($\bar{K}_I$) is redefined as $\beta_I^K \bar{K}_r$, where $\bar{K}_r$ is the reference stiffness (base design), and β_I^K denotes the variation. Similar trends between the metrics $\bar{\theta}_{01_pp}$ and $\bar{\delta}_{23_rattle}$ in Figure 5, as β_I^K varies from 0 to 2, suggest that these two metrics quantitatively correlate the transients between two powertrain components. Given the fact that it