

Analysis of Speed-Dependent Vibration Amplification in a Nonlinear Driveline System Using Hilbert Transform

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ABSTRACT

The engine start-up process introduces speed-dependent transient vibration problems in ground vehicle drivelines as the torsional system passes through the critical speeds during the acceleration process. Accordingly, a numerical study is proposed to gain more insights about this transient vibration issue, and the focus is on nonlinear analysis. First, a new nonlinear model of a multi-staged clutch damper is developed and validated by a transient experiment. Second, a simplified nonlinear torsional oscillator model with the multi-staged clutch damper, representing the low frequency dynamics of a typical vehicle driveline, is developed. The flywheel velocity measured during the typical engine start-up process is utilized as an excitation. The envelope function of the speed-dependent response amplification is estimated via the Hilbert transform technique. Finally, the envelope function is effectively utilized to examine the effect of multi-staged clutch damper properties.

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INTRODUCTION

Transient vibrations in a ground vehicle driveline typically occur when the speed of the vehicle passes through one or more critical speeds of the driveline torsional system. In particular, the driveline in a heavy duty truck may encounter severe transient vibration and dynamic amplification during the engine start-up process [1]. Thus, new or improved models are necessary to gain more insights about this transient phenomenon in order to design effective vibration isolation devices.

The clutch damper assembly (with multi-staged properties) is a torque transmitting component, and it reduces the torsional vibration induced by dynamic torque from the engine. A nonlinear model of multi-staged clutch damper was discussed by Ohnuma et al. [2]. The driveline vibrations with the multi-staged clutch dampers have been investigated over the last three decades [3]. However, prior researchers [4, 5] have mainly focused on the periodic responses (including rattle type vibro-impacts) under pulsating torque excitation (with time-invariant frequencies). The transient response under an excitation with time-varying frequencies has not received much attention. Thus, the chief objectives of this paper include the development of a nonlinear model of a multi-staged clutch damper and the transient analysis of a simplified driveline system given motion input by the flywheel. The envelope function will be calculated to

examine the speed-dependent vibration amplification using a nonlinear simulation technique and the Hilbert transform method.

MULTI-STAGED CLUTCH DAMPER MODEL

Nonlinear Model

Based upon the transmitted torque T_T through the multi-staged clutch damper (Figure 1), a new nonlinear model of the multi-staged clutch damper is developed. The torsional characteristics and key parameters are extracted from the measured transmitted torque (T_T) through the relative angular displacement between the flywheel and clutch hub (θ_{cf}) from the conventional static test [2].

The torque characteristics of Figure 1 are assumed to be symmetric with respect to the origin, though some practical dampers may be designed otherwise. Symmetric multi-staged piecewise linear models are proposed in Figure 2 and Figure 3. In particular, Figure 2 shows the multi-staged stiffness model, and Figure 3 displays the multi-staged hysteresis model. In the multi-staged stiffness model, T_p represents the preload in a transition from stiffness K_1 to stiffness K_2 , and K_3 represents the stiffness of the stoppers at the two ends. In the multi-staged hysteresis model, the dashed line and solid

line describe the direction-dependent hysteresis values based upon the stages. In particular, the hysteresis value H is assumed to be zero in the pre-damper stage between $-\varphi_1$ and φ_1 , where the stiffness K_1 is relatively low. The mathematical expressions of these two multi-staged nonlinear models are as follows:

$$T_K(\theta_{cf}) = K_3\theta_{cf} + \dots + (K_3 - K_2) \frac{(\theta_{cf} - \varphi_2)\text{sgn}(\theta_{cf} - \varphi_2) - (\theta_{cf} + \varphi_2)\text{sgn}(\theta_{cf} + \varphi_2)}{2} + \dots + (K_2 - K_1) \frac{(\theta_{cf} - \varphi_1)\text{sgn}(\theta_{cf} - \varphi_1) - (\theta_{cf} + \varphi_1)\text{sgn}(\theta_{cf} + \varphi_1)}{2} + \dots + \frac{T_p}{2} (\text{sgn}(\theta_{cf} + \varphi_1) + \text{sgn}(\theta_{cf} - \varphi_1)) \quad (1)$$

$$T_H(\theta_{cf}) = \left(\frac{1}{2}H + \frac{1}{4}H \text{sgn}(\theta_{cf} - \varphi_1) - \frac{1}{4}H \text{sgn}(\theta_{cf} + \varphi_1) \right) \text{sgn}(\dot{\theta}_{cf}), \quad (2)$$

$$\text{sgn}(x) = \begin{cases} 1 & x > 0 \\ 0 & x = 0, \\ -1 & x < 0 \end{cases} \quad (3)$$

$$\text{sgn}(x) \approx \tanh(\sigma x). \quad (4)$$

In Eq. (1) and Eq. (2), the sign functions (Eq. (3)) are replaced by a hyperbolic function $\tanh(\sigma x)$ as given by Eq. (4). Note that a high value of the regularizing factor ($\sigma = 10^6 \sim 10^9$) is used to smoothen the discontinuities for the sake of numerical calculation.

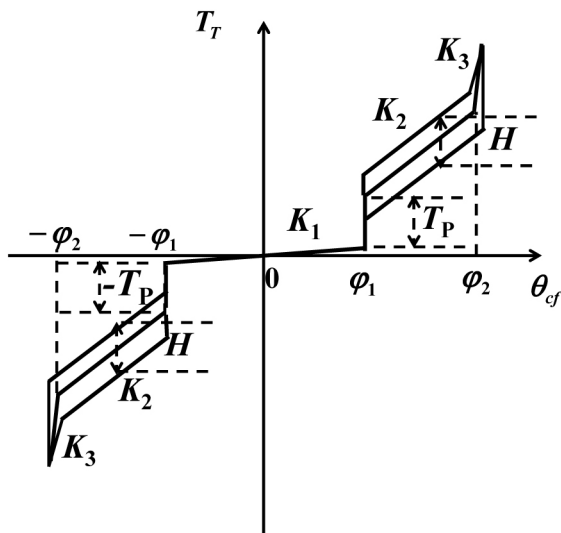


Figure 1. Static characteristics of multi-staged clutch damper.

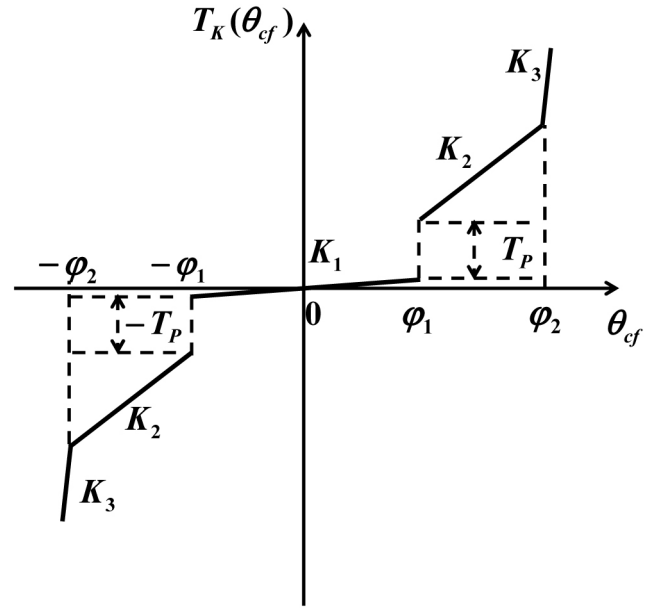


Figure 2. Nonlinear stiffness characteristics of a clutch damper in the proposed nonlinear model.

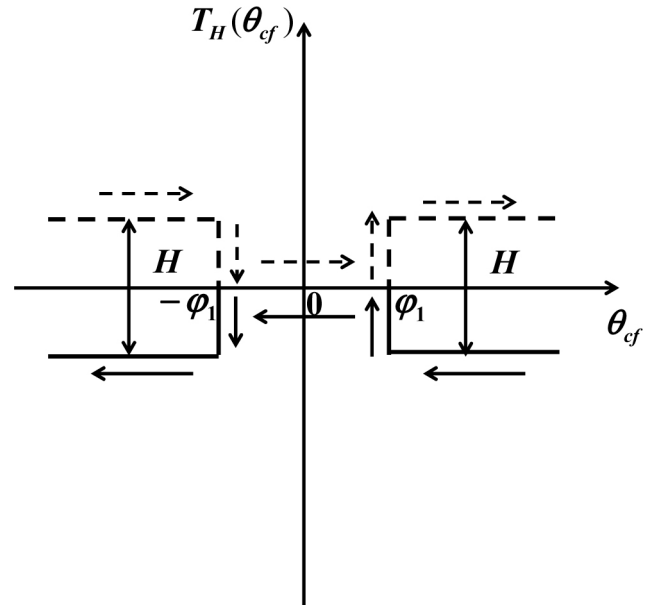


Figure 3. Nonlinear hysteresis characteristics of a clutch damper in the proposed nonlinear model.

Experimental Validation

The proposed nonlinear model is validated by a newly designed laboratory transient experiment. In this experiment, the multi-staged clutch damper is placed between a fixed flywheel and clutch hub. A torsion arm, on which the step-down torque excitation is applied, is attached to the clutch hub. The torsion arm is initially pulled down by a pneumatic actuator to displace the clutch damper (an initial relative angular displacement). Then, the constant torque applied on

the system from the actuator is removed quickly by releasing the torsion arm so as to generate the step-down torque input.

The acceleration of the torsion arm $\ddot{\theta}_2(t)$ is measured by a piezoelectric accelerometer. Then, $\ddot{\theta}_2(t)$ calculated from the proposed nonlinear model is compared with the measurement from the transient experiment.

Measured acceleration time history $\ddot{\theta}_2(t)$ ($\bar{t}$) is plotted in Figure 4 in a normalized form. The prediction matches well with the measurement for one case which validates the proposed nonlinear clutch damper model.

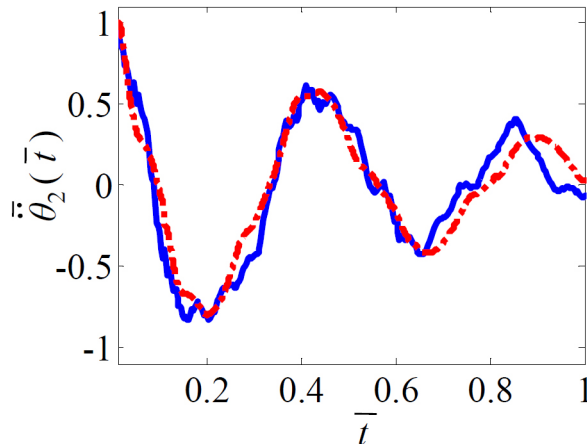


Figure 4. Validation of the proposed nonlinear clutch damper model. Key: —|—, nonlinear model; —, experiment.

TRANSIENT ENVELOPE ANALYSIS OF A NONLINEAR DRIVELINE SYSTEM

Nonlinear Model with Flywheel Speed Input

A nonlinear single-degree-of-freedom (SDOF) driveline model with focus on the multi-staged clutch damper is developed in Figure 5. The relative motion $\delta(t) = \theta(t) - \phi_F(t)$, which is related to the dynamic torque during the start-up process [1], is of interest in this study. Moreover, the angular motion of the flywheel $\dot{\phi}_F(t)$ would not be affected by the downstream drivelines due to the massive flywheel inertia. This indicates that the flywheel motion, instead of the dynamic engine torque, could be directly utilized as an excitation to the driveline system. Consequently, the measured flywheel velocity (Figure 6) is utilized. The measured flywheel speed is grouped into three regions:

Region 1 is located before $\bar{t}_1$, where the starter rotates the flywheel so as to provide the initial momentum to compress the gas in the cylinders; Region 2 is between $\bar{t}_1$ and $\bar{t}_2$, where

the engine starts to fire and the flywheel is rapidly accelerated; Region 3 is located beyond $\bar{t}_2$, where the vehicle computer reduces the speed of the flywheel and maintains the engine rotation with a constant speed (neutral status). The governing equation of motion of the nonlinear SDOF system is shown as follows:

$$I\ddot{\theta}(t) + T_K(\delta(t)) + T_H(\delta(t), \dot{\delta}(t)) + C\dot{\delta}(t) = 0, \quad (5)$$

$$\delta(t) = \theta(t) - \phi_F(t), \quad \dot{\delta}(t) = \dot{\theta}(t) - \dot{\phi}_F(t). \quad (6 \text{ a, b})$$

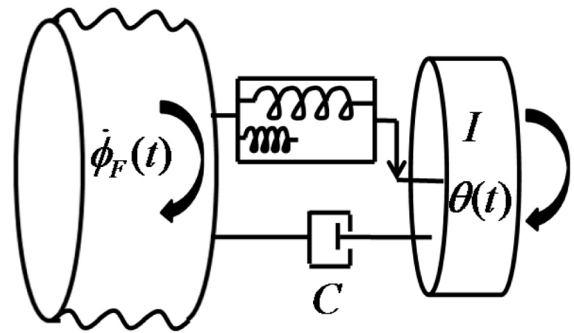


Figure 5. Schematic of a simplified nonlinear driveline model.

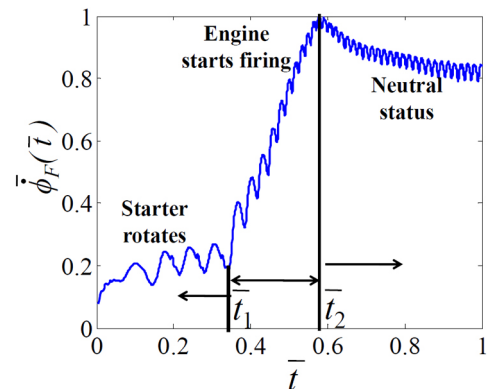


Figure 6. Measured flywheel speed (normalized).

Transient Envelope Analysis Using Hilbert Transform

The envelope function $E(t)$, which is the slowly varying curve describing the extremes of oscillatory response, will be utilized in this paper to evaluate the severity of the speed-dependent response and amplification of $\delta(t)$ during start-up process [6,7,8]. Hilbert transform, as introduced by Sen et al. [8], is an alternative method of finding the $E(t)$. The transient envelop function $E(t)$ is define as follows where $Y(t)$ is the analytic function of $\delta(t)$, and $h_\delta(t)$ is the Hilbert transform of $\delta(t)$: