

Development of Refined Clutch-Damper Subsystem Dynamic Models Suitable for Time Domain Studies

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ABSTRACT

This study examines clutch-damper subsystem dynamics under transient excitation and validates predictions using a new laboratory experiment (which is the subject of a companion paper). The proposed models include multi-staged stiffness and hysteresis elements as well as spline nonlinearities. Several example cases such as two high (or low) hysteresis clutches in series with a pre-damper are considered. First, detailed multi-degree of freedom nonlinear models are constructed, and their time domain predictions are validated by analogous measurements. Second, key damping sources that affect transient events are identified and appropriate models or parameters are selected or justified. Finally, torque impulses are evaluated using metrics, and their effects on driveline dynamics are quantified. Dynamic interactions between clutch-damper and spline backlash nonlinearities are briefly discussed.

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INTRODUCTION

Vehicle powertrains are usually subjected to transient loads that result from abrupt changes in throttle position, clutch engagement/disengagement, and braking. Such transient events lead to problematic noise and vibration phenomena and excessive impulsive torsional vibrations that are eventually transmitted to the cabin. Few recent papers have employed a lumped parameter approach to address impulsive responses in vehicle drivelines [1, 2, 3, 4, 5, 6]. For example, Crowther et al. [2] have studied the driveline clunk in a rear wheel drive driveline with a 4 degree of freedom (4DOF) nonlinear torsional model. It has been validated experimentally in the time domain using a non-rotating driveline experiment. Their model, with minimal components, included both backlash and friction nonlinearities within the differential. Li and Singh [7] have simplified the driveline system to study transient response associated with the engine start-up process with focus on vibration amplification. They assumed that flywheel motion is not affected by the downstream driveline components due to the massive flywheel inertia, and the downstream system behaves as an integral inertial component during low frequency dynamics of a typical vehicle driveline. Accordingly, they developed a nonlinear single degree of freedom (SDOF) system with a multi-staged clutch damper including hysteresis and multi-staged stiffness elements. This article aims to extend the prior study [7] by proposing alternate models with focus on the identification of damping elements during transient events. In particular, both computational and experimental methods are adopted to examine the transient dynamics of a clutch-damper subsystem. The clutch-damper experiment (published in a companion paper [8]) simulates an

impulsive excitation transmitted through the clutch-damper subsystem to the driveline. In the current paper, the applicability of nonlinear hysteresis models for clutch-damper subsystems under transient conditions is explored. Then, key damping sources that affect transient events are analyzed and finally a refined model is proposed. Specific objectives of this paper are: (1) Develop a refined nonlinear model and investigate dynamic characteristics that are not fully addressed in the companion paper [8]; and (2) Compare predictions with measurements and define damping mechanisms during the transient events.

NEW CLUTCH-DAMPER EXPERIMENT

Figure 1 shows the physical schematic of a clutch-damper experiment used to examine only transient oscillations for a nonrotating system using a pneumatic actuator for loading (refer to [8] for more details). The clutch-damper assembled on a shaft is supported by two bearings. The front end of the shaft is connected to a torsion bar. The clutch at the flywheel side is grounded to the fixtures that are bolted to a heavy test bed. On the far end of the cantilevered torsion bar, a manual release mechanism is bolted to the bar to which the pneumatic actuator tip is attached. It is released to provide excitation during the experiment. This setup allows various levels of step-like torque excitations (as shown in Fig. 2) to the clutch-damper subsystem as listed in Table 1. Time domain torque responses and vibrational accelerations at various locations are measured for subsequent damping mechanisms identification and validation purposes.

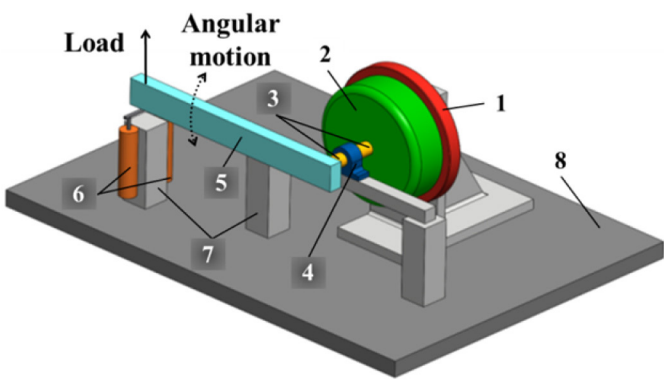


Figure 1. Physical schematic of the new non-linear clutch-damper experiment. Key: 1- flywheel (ground), 2- clutch assembly, 3- clutch shaft with spline, 4-ball bearing, 5- torsion arm, 6- pneumatic cylinders with quick release mechanism, 7- steel bed plate (ground), 8- structural support (ground). See [8] for more details.

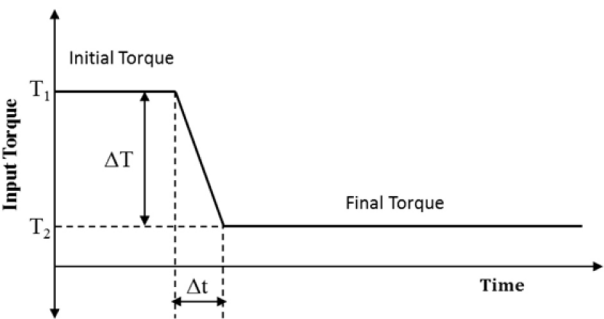


Figure 2. Transient excitation torque simulated by step-like event. Values for 4 experimental runs are listed in Table 1.

Table 1. Transient excitation torque characteristics with reference to Fig. 2.

Experimental Run	T_1 [Nm]	Δt [ms]	ΔT [Nm]
1	645	1	595
2	1010	1	960
3	1350	1	1300
4	1730	1	1680

MINIMAL ORDER NONLINEAR SYSTEM MODELS

For the sake of simplicity, the powertrain system could be divided into connected subsystems to provide some physical insights regarding the nature of nonlinear problems, as shown schematically in Figure 3. Several approaches have been utilized to examine the impulsive loading responses in drivelines, such as finite element analysis [6], though this method is time intensive. In addition, it is difficult to include accurate damping models since dissipation mechanisms during transient events (unlike steady state conditions) are still poorly understood. The lumped parameter method, with a minimal order, is a powerful technique to model the dynamic response of torsional systems subjected to impulsive loads [2, 3, 4, 5, 6, 7]. Moreover, the inclusion of clearance and friction nonlinearities in this approach is more tractable. Two dynamic models for the

clutch-damper subsystem are proposed in this paper: (1) A nonlinear SDOF system with a rigid spline contact model as in [7]; and (2) A 3DOF nonlinear model including a compliant spline contact model.

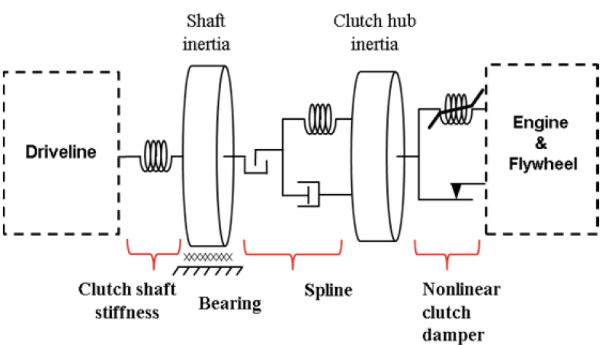


Figure 3. Vehicle subsystem of interest described by a lumped torsional model.

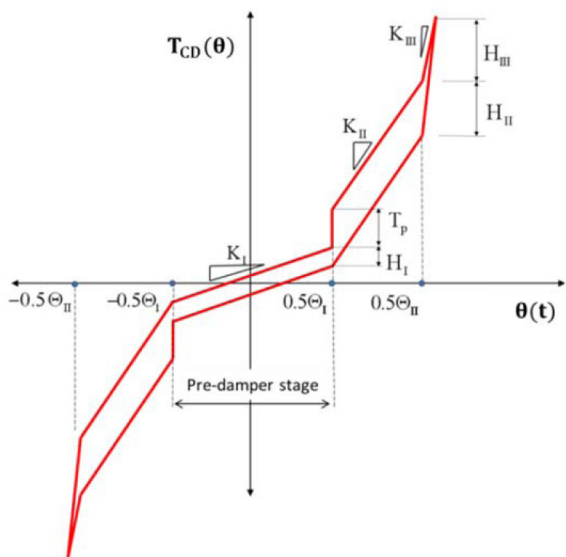


Figure 4. Quasi-static torque (T_{CD}) curve of a typical symmetric clutch-damper. Key: θ -relative angular displacement, (I, II, and III) are stages indices, K- stiffness constant, H- hysteresis, T_p - preload; and Θ - stage length. See Appendix A for a list of symbols.

Table 2. An overview of clutch-damper cases and properties with reference to Figure 4.

Clutch-Damper Case	Pre-damper stage	Operating Stiffness K_{II}	Hysteresis H_{II}
A1	Yes	High	Low
A2	No	Low	Low
A3	No	High	Low
A4	No	High	High

MATHEMATICAL FORMULATION

Four clutch-dampers with multiple hysteresis and stiffness stages, as illustrated in Table 2, are considered. For each case, a torque-angular deflection curve is assumed to be available similar to the quasi-static curve shown in Figure 4 for a typical clutch-damper assembly with a pre-damper. The hysteresis in the system can be observed by observing the resulting area between loading and unloading traces.

The torsional stiffness for each stage can be increased abruptly by applying a preload on the helical springs for a particular stage. On the other hand, adding a pre-damper component to the clutch leads to considerable softening of the first stage stiffness while enlarging the transition to the next stage. A 3DOF nonlinear model including a compliant spline contact model (as displayed in Figure 5a) is compared with a simplified SDOF nonlinear model with a rigid spline mesh model (as shown in Figure 5b). The equations of motion for the 3DOF model in matrix form is

$$\underline{\mathbf{J}}\ddot{\underline{\theta}}(t) + \underline{\mathbf{C}}\dot{\underline{\theta}}(t) + \underline{\mathbf{K}}\underline{\theta}(t) = \underline{\mathbf{T}}(t). \quad (1)$$

where $\underline{\mathbf{J}}$, $\underline{\mathbf{C}}$, and $\underline{\mathbf{K}}$ are inertia, damping, and stiffness matrices, respectively. Note that the stiffness matrix includes two nonlinear stiffness elements. First, the multi-staged clutch model is represented by the following:

$$T_{cd}(\theta_3(t)) = T_D(\theta_3(t)) + T_p(\theta_3(t)) + H(\theta_3(t), \dot{\theta}_3(t)). \quad (2)$$

Here, T_D is the torque transmitted through elastic elements (springs), T_p is the internal clutch preload, and H is the clutch-damper hysteresis torque (dry-friction torque losses inside the clutch-damper subsystem); these are estimated from the quasi-static torque curve for each clutch.

$$\begin{aligned} T_D(\theta_3(t)) &= K_I \theta_3(t) \\ &+ 0.5(K_{II} - K_I) \\ &\left[(\theta_3(t) - \Theta_I) [\tanh \sigma(\theta_3(t) - \Theta_I) + 1] - (\theta_3(t) + \Theta_I) [\tanh \sigma(\theta_3(t) - \Theta_I) - 1] \right] \\ &+ 0.5(K_{III} - K_{II}) \\ &\left[(\theta_3(t) - \Theta_{II}) [\tanh \sigma(\theta_3(t) - \Theta_{II}) + 1] - (\theta_3(t) + \Theta_{II}) [\tanh \sigma(\theta_3(t) - \Theta_{II}) - 1] \right]. \end{aligned} \quad (3)$$

Here, σ is a regularizing factor. Define T_p and H as follows:

$$T_p(\theta_3(t)) = 0.5 \left[T_p \left(\tanh \left[\sigma(\theta_3(t) + \Theta_I) \right] + \tanh \left[\sigma(\theta_3(t) - \Theta_I) \right] \right) \right]. \quad (4)$$

$$\begin{aligned} H(\theta_3(t), \dot{\theta}_3(t)) &= [0.5H_{II} + 0.25H_{II} \tanh \sigma(\theta_3(t) - \Theta_I) \\ &- 0.25H_{II} \tanh \sigma(\theta_3(t) + \Theta_I)] \tanh(\sigma \dot{\theta}_3(t)). \end{aligned} \quad (5)$$

Note, hysteresis is only effective in the second stage (operating stage) for all clutches considered in this paper. Further, the compliant contact model for the clutch spline with an angular clearance b is given by the following:

$$K_{23}(\delta(t)) = K_{23}\delta(t), \text{ and } C_{23}(\delta(t)) = C_{23}\delta(t). \quad (6)$$

$$\text{where, } \delta(t) = \begin{cases} 1 & |\theta_2 - \theta_3| \geq 0.5b \\ 0 & \text{Otherwise.} \end{cases}$$

The spline mesh offset torque [3] is:

$$T'_{23} = 0.5 \text{sgn}(\theta_{23}(t)) k_{23} b. \quad (7)$$

Now the torque vector could be constructed from excitation torque, clutch hysteresis torque, and a spline mesh torque offset as follows:

$$\underline{\mathbf{T}}(t) = [T(t) \quad T'_{23} \quad T'_{23} - H(\theta_3(t), \dot{\theta}_3(t)) - T_p(\theta_3(t))]^T. \quad (8)$$

where, $T(t)$ is the step-like external torque as shown in Figure 2. Applying a load starting from zero, the upstream elements (J_1 , J_2) are rotated freely through the backlash region with an angular displacement equivalent to $\pm b$. Therefore, the initial condition displacements need to be adjusted accordingly.

$$\underline{\theta}(0) = \underline{\mathbf{K}}^{-1} \underline{\mathbf{T}}(0) + \text{sgn}(T(0)) b [1 \quad 1 \quad 0]^T \text{ and } \dot{\underline{\theta}}(0) = \underline{\mathbf{0}}. \quad (9)$$

Additionally, non-dimensional variables are introduced below based on the system characteristics such as the second stage stiffness K_{II} of the clutch and the first natural frequency ω_1 .

$$\begin{aligned} \bar{t} &= 2\pi \frac{t}{\omega_1}, \quad \bar{\lambda} = 2\pi \frac{\lambda}{\omega_1}, \quad \bar{T}(\bar{t}) = \frac{T(t)}{bK_{II}}, \quad \bar{\theta}(\bar{t}) = \frac{\theta(t)}{b}, \quad \bar{\dot{\theta}}(\bar{t}) = \frac{\dot{\theta}(t)}{b\omega_1}, \\ \text{and } \bar{\ddot{\theta}}(\bar{t}) &= \frac{\ddot{\theta}(t)}{b\omega_1^2}. \end{aligned} \quad (10 \text{ a-f})$$

Here, λ is the time period between collision impulses; it will be used as a metric in a latter section. Finally, Equations (1), (2), (3), (4), (5), (6), (7), (8), (9) are implemented and solved by a commercial code [9] as demonstrated in Figures 6 and 7 for the 3DOF system model of Figure 5 (a).

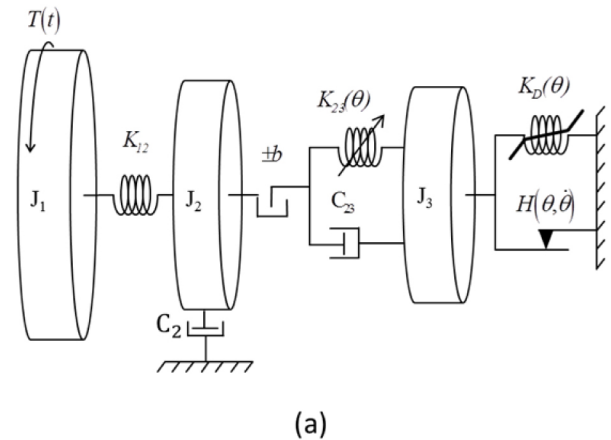


Figure 5. Minimal order nonlinear torsional system models of the experiment of Figure 1. (a) 3DOF nonlinear system; (b) SDOF nonlinear system. Key: J_1 - lumped inertia of torsion arm, J_2 - lumped inertia of clutch shaft with spline, J_3 - lumped inertia of clutch hub, K_{12} - torsion arm stiffness, K_{12} - spline contact stiffness, K_D - non-linear clutch stiffness, H - non-linear clutch hysteresis, and C_3 - bearing losses