

Development of a Non-Linear Clutch Damper Experiment Exhibiting Transient Dynamics

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ABSTRACT

Many powertrain structural sub-systems are often tested under steady state conditions on a dynamometer or in a full vehicle. This process (while necessary) is costly and time intensive, especially when evaluating the effect of component properties on transient phenomena, such as driveline clunk. This paper proposes a laboratory experiment that provides the following: 1) a bench experiment that demonstrates transient behavior of a non-linear clutch damper under non-rotating conditions, 2) a process to efficiently evaluate multiple non-linear clutch dampers, and 3) generates benchmark time domain data for validation of non-linear driveline simulation codes. The design of this experiment is based on a previous experimental work on clunk. A commercially available non-linear clutch damper is selected and the experiment is sized accordingly. The stiffness and hysteresis properties of the clutch damper are assumed from the measured quasi-static torque curve provided by the manufacturer. Dissipation sources within the experiment are estimated as well. Instrumentation and time domain signal processing issues are discussed and typical measurements are compared to a simplified model for verification.

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INTRODUCTION

All powertrain or driveline systems have clearances due to design, manufacturing error, and wear. The presence of backlashes, gaps, or multi-staged elements generates transient phenomena depending on the mean and dynamic loads. The entire family of such events can be classified into three cases as shown in Figure 1, where θ is relative angular motion and T is transmitted torque through a component. The first is the no impact case (Figure 1(a)), which occurs when the range of motion does not cross a clearance transition ($\pm\Theta_b$). The second is the rattle case (Figure 1(b)), which occurs when pulsating motion about some mean displacement crosses one (single-sided impact) or two (double-sided impact) clearance transitions. The third is the clunk case (Figure 1(c)), which occurs when the mean operating point suddenly changes and the resulting motion crosses one or two clearance transitions. The severity of clunk depends on the mean operating points, transition rate, and dissipation mechanisms [1,2].

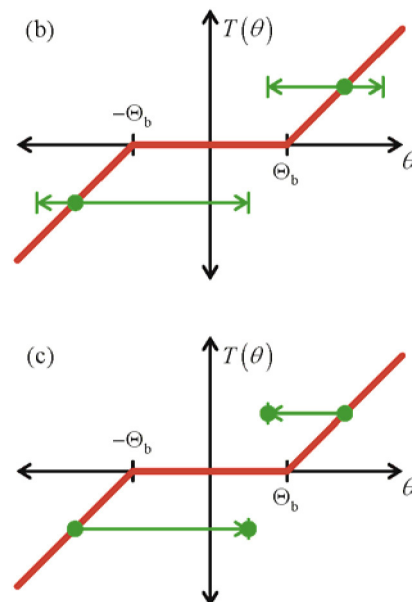
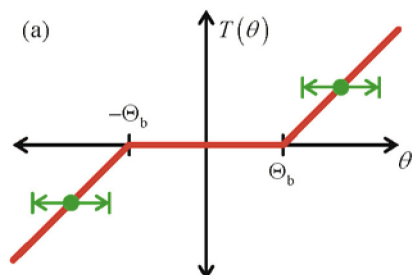


Figure 1. (cont.) Clearance-induced impact regimes: (a) no impact or constant contact, (b) rattle case with single and double-sided impacts, and (c) clunk case with single and double-sided impacts where θ is relative angular displacement, T is transmitted torque, and Θ_b is a clearance transition. Key: (—) - transmitted torque, (●) - mean displacement point, and (↔) - dynamic displacement range.

Figure 1.

Non-linear clutch dampers may be designed to mitigate clearance induced impacts in a powertrain system. This component serves to transmit torque and reduce vibration from the engine to the transmission [3]. A typical quasi-static performance curve of a three-stage, symmetric clutch damper is shown in Figure 2, where θ is the relative angular motion between the clutch damper and shaft, and T_D is transmitted torque. Positive angular displacement is referred to as “drive-side” and negative angular displacement as “coast-side.” Clutch dampers may include several non-linear features (intentionally or otherwise), such as clearances ($\pm\Theta_b$), multi-staged stiffness (K_I , K_{II} , K_{III}), pre-load (T_p), and frictional hysteresis (H_I , H_{II}) [3].

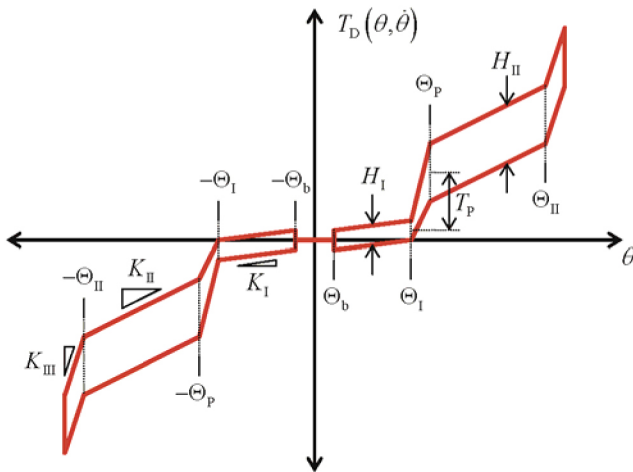


Figure 2. Typical quasi-static performance curve of a three-stage, symmetric clutch damper with relative angular displacement θ and transmitted torque T_D . Key: Θ - stage or clearance transition, K - torsional stiffness, and H - frictional hysteresis. Stages are denoted by subscripts I, II, and III (stopper), pre-load by subscript P, and clearances by subscript b.

Selection of non-linear clutch damper features could be more efficiently facilitated by experimentally validated non-linear driveline simulation codes. Powertrain structural sub-systems are often tested on dynamometers or in full vehicles under steady-state conditions. This process, while necessary, is costly in terms of time, expertise, and instrumentation, especially when evaluating the effects of component properties on transient phenomena. To address this, the current paper proposes a laboratory experiment that intends to achieve the following objectives:

1. Demonstrate the transient (clunk-like) behavior of a non-linear clutch damper under non-rotating conditions; and
2. Generate benchmark time domain data for validation of non-linear driveline simulation codes.

NEW NON-LINEAR CLUTCH DAMPER EXPERIMENT

The clunk phenomenon has been investigated by Gurm et al. [1] and Crowther et al. [2], who proposed non-linear vibratory (non-rotating) experiments that are excited by an external step-like torque. To ensure that these systems are definite, at least one component is grounded, such as a wheel rotor [1] or gear [2]. A torsion arm is attached to another component, which is free to vibrate, and a

step-like force is applied to the arm via a variable mass drop [1,2]. During the system response, impacts occur at clearance locations, such as gear meshes in transmissions or differentials [1,2].

A physical illustration of the new non-linear clutch damper experiment is proposed in Figure 3. The new experiment is a non-rotating system that is excited by an external step-like torque, similar to previous work [1,2]. For simplicity, a flywheel is chosen as the ground and a clutch assembly is bolted directly to it. The clutch assembly houses a greased sleeve bearing and two parallel non-linear clutch dampers. The clutch is engaged so that clutch dampers are compressed against the flywheel; a no-slip condition is assumed between the flywheel and clutch dampers. A shaft contacts the two clutch dampers through a spline, and is radially supported by the sleeve bearing and a ball bearing that is exterior to the clutch assembly. A torsion arm is attached to the shaft and a step-like force is applied to its free end. Due to the very high torque capacity of the clutch dampers, application of the force through a mass drop [1,2] would be impractical and unsafe. Therefore, two pneumatic cylinders with a quick-release mechanism are utilized. All components are supported by steel structures which are grounded to a bed plate.

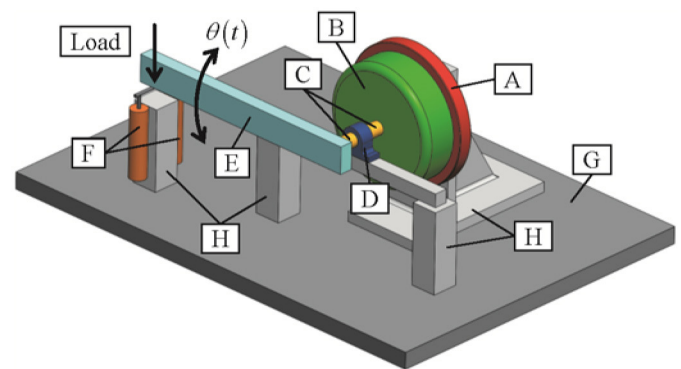


Figure 3. Physical illustration of the new non-linear clutch damper experiment. Key: A - flywheel (ground), B - clutch assembly, C - clutch shaft with spline, D - ball bearing, E - torsion arm, F - pneumatic cylinders with quick release mechanism, G - steel bed plate (ground), and H - structural support (ground).

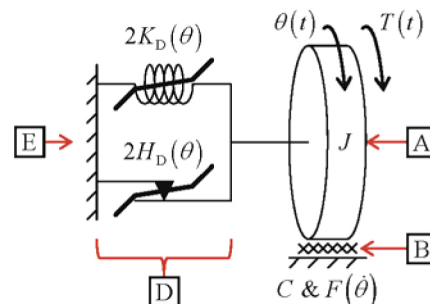


Figure 4. Non-linear model of the new clutch damper experiment. Key: A - lumped torsional inertia of torsion arm, shaft, and spline, B - lumped bearing interface, D - non-linear clutch dampers, and E - flywheel (ground).

For design purposes, the experiment may be simply described as a non-linear, single degree of freedom (SDOF) system shown in Figure 4. The equation of motion is the following

$$J\ddot{\theta}(t) + C\dot{\theta}(t) + F(\dot{\theta}) + T_D(\theta, \dot{\theta}) = T(t). \quad (1)$$

It is assumed that the torsion arm, clutch shaft, and spline are a single rigid body with torsional inertia J and angular motion $\theta(t)$. The interface between the shaft and the two bearings is modeled as a torsional viscous damping C and sliding friction $F(\dot{\theta})$. Multi-staged stiffness $K_D(\theta)$ and frictional hysteresis $H_D(\theta, \dot{\theta})$ are provided by the clutch dampers which are assumed to be identical, symmetric, synchronous, and have negligible torsional inertia. Stiffness and hysteresis properties are assumed from the quasi-static performance curves provided by the manufacturer. The non-linear function $T_D(\theta, \dot{\theta})$ is the torque transmitted through the clutch dampers in series with the spline clearance, as illustrated in Figure 2. The step-like external torque is denoted by $T(t)$.

SIZING AND EXCITATION CALCULATIONS

For sizing purposes, the operating range of the experiment is assumed to be within the drive-side stage II of the clutch dampers. The corresponding SDOF linear system has inertia J and stiffness K_{II} with a natural frequency of $\omega_n = (K_{II}/J)^{0.5}$ (rad/s). The target natural frequency is assumed to be approximately equal to the typical driveline surge mode (20π rad/s = 10 Hz) [2]. Since the clutch shaft and spline are production components with known inertia J_B , the inertia of the torsion arm (J_A) must satisfy the following

$$J_A \approx K_{II} (20\pi)^{-2} - J_B. \quad (2)$$

The external step-like torque $T(t)$ is approximated by a ramp function with initiation at $t = 0$ of duration of t_e , initial torque T_0 , and final torque T_f as illustrated in Figure 5(a). It is defined by the following where $\Xi(\theta) = 0.5(\text{sgn}(\theta) + 1)$

$$T(t) = \begin{cases} T_0(1 - \Xi(t)) + \dots \\ [T_0 + (T_f - T_0)t/t_e][\Xi(t) - \Xi(t - t_e)] + \dots \\ T_f\Xi(t - t_e) \end{cases} \quad (3)$$

To ensure a truly non-linear response, the initial and final operating points of the system are designed to lie on different stages of $T_D(\theta, \dot{\theta})$, as illustrated in Figure 5(b). The initial operating point (θ_0, T_0) lies on drive-side stage II and the final point (θ_f, T_f) can reside on drive-side stage I or pre-load. Achieving both (θ_0, T_0) and (θ_f, T_f) would require proper sizing of the pneumatic actuation system and torsion arm.

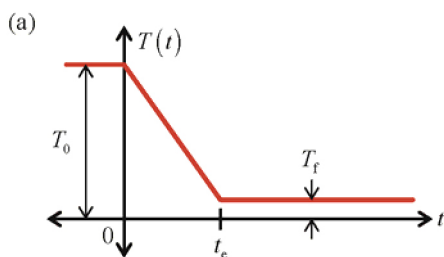


Figure 5.

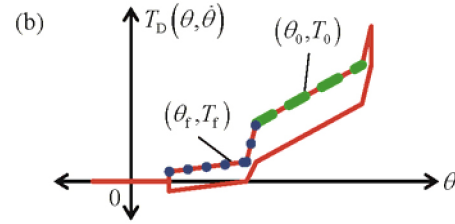


Figure 5. (cont.) Conceptual illustration of the external torque $T(t)$: (a) time-domain representation and (b) operating points on the clutch damper. Key: (θ_0, T_0) - initial operating point, and (θ_f, T_f) - final operating point.

INSTRUMENTATION AND TYPICAL MEASUREMENTS

A laser vibrometer [4] and data acquisition system [5] are chosen to directly measure the velocity of the torsion arm and shaft, as illustrated in Figure 6. The translational velocity of the laser point along the horizon is denoted by $\dot{u}(t)$, the radial distance from the shaft axis to the laser point by $r(t)$, and the angular position of the arm by $\theta(t)$ (w.r.t. the clutch damper datum) and $\phi(t)$ (w.r.t. the horizon). A reflective surface for the laser point is provided by a square finished block, which is rigidly attached to the torsion arm such that the measurement surface is in-plane with the shaft axis. Prior to excitation, ϕ_0 and r_0 are measured with a digital level and ruler, respectively; initial translational displacement u_0 is calculated from $u_0 = r_0 \sin(\phi_0)$. The system excitation is then applied and signal $\dot{u}(t)$ is measured with a sampling frequency of 1.28 kHz; triggering procedures are not discussed in this paper. The translational displacement $u(t)$ is then estimated by numerically integrating $\dot{u}(t)$ [6]. Angular displacement $\theta(t)$ is calculated using the following where θ_f is determined from the static balance of T_f

$$r(t) = \left[r_0^2 + (u_0 - u(t))^2 - 2r_0(u_0 - u(t))\sin(\phi_0) \right]^{0.5}, \quad (4)$$

$$\theta(t) = \theta_f + \phi_0 - \phi_f - \sin^{-1} \left(\frac{(u_0 - u(t))\cos(\phi_0)}{r(t)} \right). \quad (5)$$

Angular velocity $\dot{\theta}(t)$, acceleration $\ddot{\theta}(t)$, and jerk $\dddot{\theta}(t)$ are estimated through numerical differentiation [6], and all signals are synchronized so that the maximum value of $\ddot{\theta}(t)$ occurs at $t = 0$.

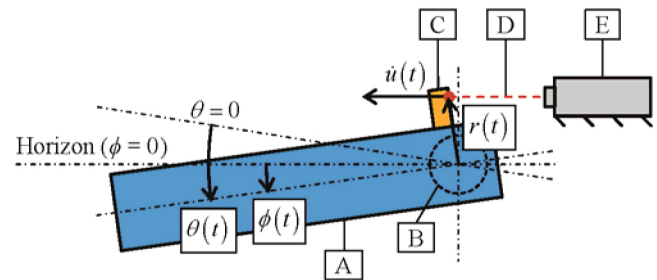


Figure 6. Schematic of the instrumentation system. Key: A - torsion arm, B - clutch shaft, C - block with finished surfaces, D - laser point, E - laser vibrometer.