

Effect of Limited Slip Clutch Friction on the Driveline Dynamics of a Rear Wheel Drive Vehicle Coasting in a Turn

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ABSTRACT

A model and simulation results are presented for the torsional dynamics of a rear wheel driveline while the vehicle is coasting in a turn. The model includes the effects of road load and powertrain drag, limited slip differential clutch friction, the inertias of the vehicle, wheels, axles, differential carrier, and driveshaft, the final drive ratio, torsional stiffnesses of the axles and driveshaft, vehicle track width, and radius of the turn. The dynamics of coasting in a turn differ from powered driving due to changes in the inertia loading the driveshaft, the damping effect of the disengaged transmission, and nonlinearities in the clutch friction.

Specific focus is given to vibration in the axles and driveshaft due to variations in the torque-speed slope of the clutches, which is determined by the slope of the friction coefficient ' μ ' versus sliding speed ' v ' in the limited slip clutches. The clutch μ - v slope is strongly influenced by the lubricant in the differential and understanding its effect on driveline vibration will help in the development of future lubricants.

Three cases are presented: (1) two clutches with the same static breakaway torque and positive torque-speed slopes, (2) one clutch with a lower breakaway torque and zero torque-speed slope, and (3) one clutch with a lower breakaway torque and negative torque-speed slope.

INTRODUCTION

Several operating conditions – acceleration, steady driving, coasting and braking – arise in normal driving when the drive tires have good traction. These operating conditions may occur during cornering as well as straight-ahead driving. Different operating conditions often generate different frequencies of noise and vibration since the effective dynamic properties can change. For example, in a vehicle with slip-limiting clutches in the differential the clutches may be locked during straight ahead driving, so that the inertias inside the differential housing act as a single inertia [cf., 12], but while turning or cornering the clutches slip and the inertias of the side gears

and carrier are decoupled. This changes the effective system inertias and the damping on the side gears, axles and differential assembly.

This study focuses on rear wheel drive vehicles, with limited slip differential (LSD) clutches, that are coasting down while turning, which often occurs when exiting a highway. The specific objectives are (1) to determine the effect of the friction torque-speed slope of the limited slip clutches on vibration and (2) to identify the dominant vibration characteristics such as what components are vibrating, at what frequencies, and at what times during the coast down.

The torque-speed slope of the clutches is largely controlled by the lubricant and friction material μ - v slope, $d\mu/dv$, which represents the variation in the coefficient of friction, μ , with respect to variation in the clutch sliding speed, v . During coast down there is essentially no torque input from the transmission; motion occurs due to the translational kinetic energy of the vehicle and the rotational kinetic energy of driveline components. As the vehicle starts to turn the clutches slip. The slip can cause vibration by releasing strain energy in the axles and carrier. The transmission components connected to the driveshaft apply a damping load, which, combined with wheel rolling resistance and damping losses in other driveline and clutch elements, dissipates the kinetic energy of the vehicle until it comes to rest. As the vehicle comes to rest the clutches can start to stick-slip as they approach lock-up, which introduces impulses that also cause vibration.

REAR WHEEL DRIVE SYSTEM MODEL

The main elements of the rear wheel drive driveline model are shown in Figure 1. The inertia of the transmission and a portion of the driveshaft is I_{Tr} . The damping in the transmission is R_{Tr} . The driveshaft stiffness is k_D . The carrier inertia is I_C . The side-gear ends of the axles have inertias I_A and I_B , which include the effective axle inertias. The axle stiffnesses are k_A and k_B . When both tires have good traction, so that neither tire is slipping, J_A and J_B are the equivalent proportions of vehicle

inertias at the wheel ends of the axles. The rolling resistances of the tires are R_A and R_B , and R_G represents transmission error in the gears.

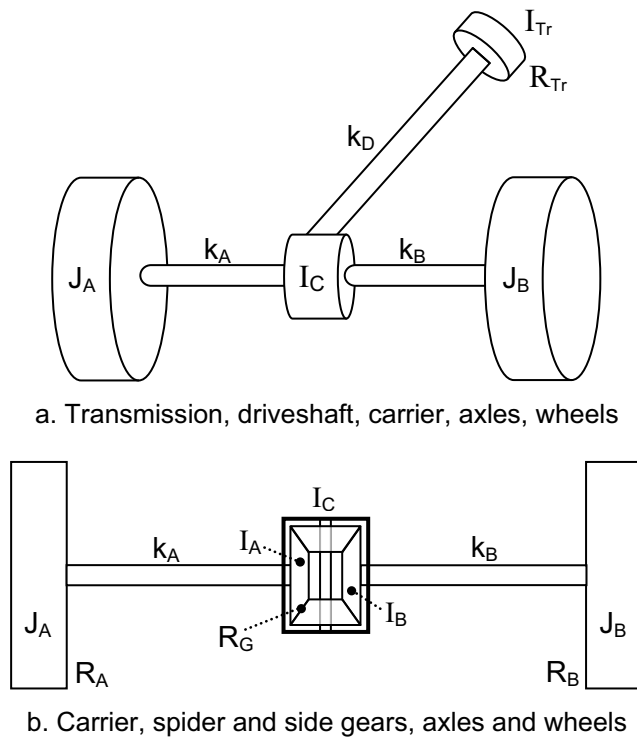


Figure 1. Rear Wheel Drive System Elements

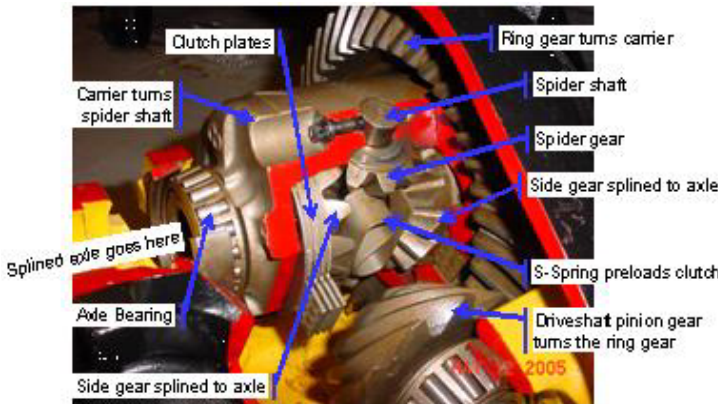


Figure 2. Cut-Away of a Limited Slip Differential

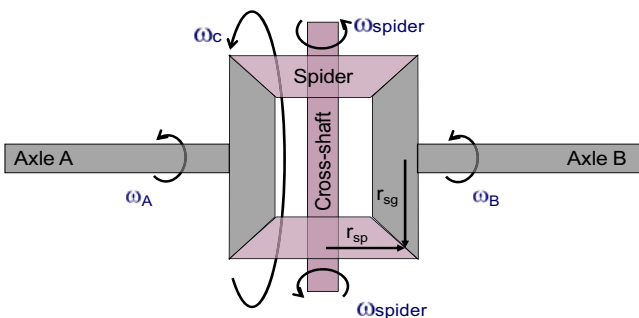


Figure 3. Schematic of an Open Differential

DIFFERENTIALS: OPEN AND LIMITED SLIP

Open Differentials

An *open differential* allows the driveshaft to deliver power to both axles while also allowing the axles to rotate at different speeds during cornering. Without a differential the driven wheels would slide or hop on the road surface while cornering, resulting in excess stress on the tires and vehicle, poor handling and possible loss of steering and braking control. The driveshaft pinion turns the ring gear (Figure 2) and their gear ratio is the *final drive ratio*. The ring gear is either integral to the carrier or rigidly connected to it, so the ring gear turns the carrier and the carrier turns the spider cross-shaft, which may be either a solid shaft (Figures 1b, 3) or a stub shaft for each spider gear (Figures 2, 4). The spider gears always rotate with the carrier around the axle axes, but during straight-ahead driving the spider gears do not rotate about the cross-shaft axis (Figure 3); all the elements rotate at the carrier speed, ω_C , about the axle axes. When the vehicle is turning the spider gears also rotate about the cross-shaft axis, which allows the outside axle (axle A) to rotate faster by $\Delta\omega$ and the inside axle (B) to rotate slower by same $\Delta\omega$, where

$$\Delta\omega = \frac{r_{sp}}{r_{sg}} \omega_{spider} \quad (1)$$

In equation (1), ω_{spider} is the angular velocity of the spider gears about the cross-shaft axis and r_{sp} and r_{sg} are the radii of the spider and side gears, respectively, measured from any common contact point on their pitch cones to their respective rotational axes (Figure 3). Consequently, in the absence of gear transmission error, the axle speeds are:

$$\begin{aligned} \omega_A &= \omega_C + \Delta\omega \\ \omega_B &= \omega_C - \Delta\omega \end{aligned} \quad (2)$$

so $\omega_A + \omega_B = 2\omega_C$. Any sliding between the spider and side gear teeth due to geometry imperfection or loading introduces slight deviations from the ideal relationships, and is accounted for in the model by the gear loss term R_G .

In the idealized case of no power loss due to friction, conservation of power requires the input power from the carrier to equal the output power to the axles:

$$T_C \omega_C = T_{sgA} \omega_A + T_{sgB} \omega_B \quad (3)$$

where T is torque (power = torque x angular velocity) and the subscripts are "C" for carrier, "sg" for side gear, and "A" and "B" for the two axles. During straight-ahead driving $\Delta\omega$ is zero and $\omega_A = \omega_B = \omega_C$, eqn. (2), so $T_C = T_{sgA} + T_{sgB}$, eqn. (3). Furthermore, T_{sgA} and T_{sgB} must be equal,

$$T_{sgA} = T_{sgB} = T_{sg} = T_C \quad (4)$$

where T_{sg} is the torque on the side gears from the spider gears, since any inequality would accelerate one of the axles, which is inconsistent with the requirement that the axle speeds are the same for straight-ahead driving, assuming the tires have the same rolling radii. The axle torques are not equal during acceleration and deceleration transients due to the variations required to accelerate and decelerate wheel and vehicle inertias. In a steady turn at constant speed, and for straight-ahead driving with unequal tire radii, the torque equality, eqn. 4, is still valid if friction losses are neglected since no inertias are accelerating or decelerating, but the power distribution between the axles is unequal. The faster axle gets a higher proportion of the available power.

Limited Slip Differentials

Equal axle torques, $T_{axleA} = T_{axleB}$, are typically sufficient to move the vehicle when both tires have good traction and the available torque is limited by the engine and transmission. On a low traction surface such as ice or gravel, however, the available torque is limited by the traction of the tires on the ground. If one tire loses traction, the torque to the other tire is also reduced since the differential maintains equal torque to both axles [cf. 1, 2]. With low torque at both wheels the vehicle may be unable to accelerate, or even move, even though one tire may have good traction. Adding clutches between the differential carrier and the axle side gears (cf., Figures 2 and 4) is a common way of providing an additional torque path to the tire with good traction to assist in accelerating the vehicle. A differential with these clutches is called a clutch-type limited slip differential (LSD). With the high level of friction in LSD clutches, the equal torques of the open differential, eqn. (4), no longer apply. Friction between the carrier and clutches opposes the high speed axle A, $T_{axleA} = T_{sg} - T_{clutchA}$, and promotes the low speed axle B, $T_{axleB} = T_{sg} + T_{clutchB}$. The reaction torques on the carrier are opposite to the axle torques, so $T_C = T_{final-drive} - 2T_{sg} + T_{clutchA} - T_{clutchB}$ during steady-state turning powered by the engine.

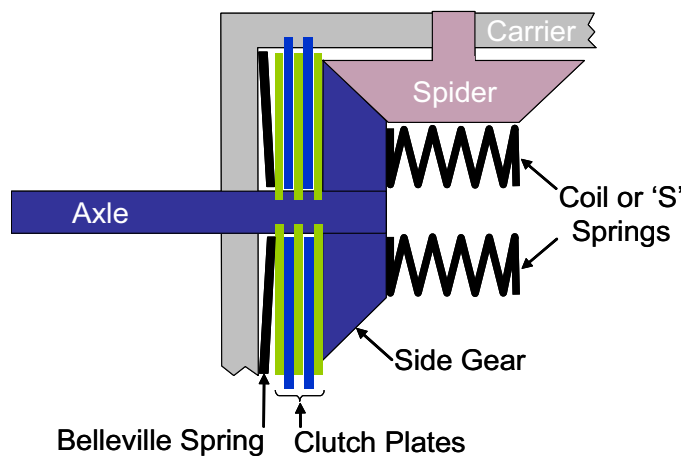


Figure 4. Cross-Section of a Limited Slip Differential

Under certain sliding conditions LSD clutches are susceptible to undesirable vibrations that can be a warranty issue and perceived quality problem. The vibrations are

often called shudder, judder, moan or chatter corresponding to whether the vibration frequency, or resulting noise, is low (under 10 Hz), high (over 400 Hz), in between, or distributed over a broad range of frequencies. The literature on limited slip differentials typically focuses on the torque distribution to the axles to determine torque bias [3] and ignores vibration. When axle vibration is considered it is typically for the case of one tire slipping and the other tire not slipping [4] since that is the condition for which the limited slip clutches are employed. Vibrations also arise, however, during normal driving when both tires have good traction.

CLUTCH FRICTION

The arrangement of the carrier, spider gears, side gears, axles, clutch plates and springs is shown in Figure 4. Only one axle, side gear and clutch are shown extending to left; the other axle, side gear and clutch that are not shown extend to the right symmetrically with respect to a centerline through the spider gear cross-shaft. Three possible spring arrangements are shown – Bellevilles on the outside of the clutch plates, or coil or S springs on the inside of the clutch plates – though only one of these is used in any particular LSD.

The limited slip clutch plates are splined to the axle and carrier in an alternating sequence (Fig. 4) so that when they are clamped tightly together the axle is forced to rotate at the same speed as the carrier. This eliminates the differential action but allows torque to be transferred to a tire with good traction. When the plates slip, the differential action allows the axles to rotate at different speeds and the clutch torques act to resist rotation of the faster axle and promote rotation of the slower axle. The sliding speed in each clutch is the difference between its axle's rotational speed and the speed of the carrier. During differentiation, when the spider gears rotate about the cross-shaft axis, one axle rotates faster than the carrier and the other rotates slower by the same amount, so the two clutches experience identical sliding speeds, $\Delta\omega$, but in opposite directions (equations 1 and 2).

The springs pre-load each clutch and a clutch does not slip until the torque on it exceeds the static breakaway torque, $T_{breakaway}$, due to the clamping force on the clutch, F . The clutch friction also varies with the sliding speed, $v = R_c \Delta\omega$, between the carrier and side gear, where R_c is the effective clutch radius. Consequently:

$$T_{clutch} = T_{breakaway} + FR_c^2 \frac{d\mu}{dv} \Delta\omega \quad (5)$$

where $d\mu/dv$ is the μ - v slope, which represents variation in the coefficient of friction with respect to variation in the sliding speed. Since friction always opposes relative sliding motion, if the sliding speed tries to reverse direction so does the clutch torque, as depicted in Figure 5, assuming equivalent friction behavior (anti-symmetric) in both directions.

The torque-speed slope represents the damping in the clutch and is particularly important for vibration stability. For constant clutch force, F , and constant radius, R_c , the