

The Impact of Gear Meshing Nonlinearities on the Vehicle Launch Shudder

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ABSTRACT

During the launch of a car, severe torsional vibration sometimes may occur in its driveline due to somewhat the slipping of the clutch, its intuitive sense for an occupant is the longitudinal vibration of the vehicle, referred to as the launch shudder whose characteristic frequency is from 5 to 25 Hz generally. As the main vibration sources of the driveline and its crucial nonlinear components, the variable stiffness and backlash of the gear meshing are considered, their impacts on the launch shudder are analyzed in this paper.

Conformal mapping, finite element method and regression method etc. are the main approaches to calculate the variable meshing stiffness of a gear pair. If this stiffness is get, it can usually be substituted for its approximate analytical expression, just with finite harmonic terms, in Fourier Series form into Ordinary Differential Equations(ODEs) to calculate the vehicle responses with its nonlinearity considered.

There are two methods to model gear backlash, one is to introduce nonlinear backlash function to the stiffness matrix of the system, the other is to model and add restoring torques to its generalized force vector with the original stiffness matrix. The latter is more appropriate for the consideration of backlash as for the tandem gear pairs combination in multiple Degrees of Freedom(DOFs) system.

Aimed at analyzing launch shudder, a certain passenger sedan is taken as the research object, 12 DOFs branched torsional vibration model of the vehicle driveline is established, the nonlinearity for two-stage gear variable meshing stiffness and backlash are modeled, further their restoring torques deduced respectively. Vehicle responses, in both time and time-frequency domain are provided; meanwhile transmission errors of two-stage gear pairs with variable meshing stiffness considered and clunk phenomenon of loaded gears caused by the backlash are calculated under the typical launch operating scenario. Through the results of simulation, it is revealed that variable meshing stiffness has very small impact on the launch shudder and can be replaced by the averaged one for the corresponding gear pair in numerical analysis; moreover, the gear backlash may aggravate the shudder when it is larger, while in its rational range, its impact on launch shudder is also smaller.

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INTRODUCTION

Vehicle launch shudder phenomenon appears on the course of starting. Severe torsional vibration may occur in its driveline due to the slipping of the clutch and the tactile sense that passengers would mainly perceive is the longitudinal vibration of vehicle. Generally, the characteristic frequency of launch shudder is from 5 to 25 Hz. The gear meshing nonlinearities mainly consist of variable meshing stiffness and gear backlash. It is well known that variable meshing stiffness fluctuations and gear backlash effects are significant sources of vibration and instability in geared rotor systems. The former is the basic characteristics of gear drive, while for the lubrication between the meshing teeth profiles and avoidance of the expansion block due to gear tooth frictional heat, the latter between the meshing teeth profiles is also necessary[1].

Conformal mapping, finite element method and regression formula are the primary approaches to calculate variable meshing stiffness of a gear pair. After the variable meshing stiffness is obtained, it is usually put its approximate analytical expression in the Fourier series form into ODEs directly to calculate the responses considering this nonlinearity[2]. There are traditionally two methods to model gear backlash, one method is to introduce nonlinear backlash function to the stiffness matrix of system and the other is to model and add restoring torques to its generalized force vector with the original stiffness matrix.

Wan Zhiguo et al. presented a correction method of variable meshing stiffness considering misaligning between the root circle and base circle, built the multi-degree of freedom dynamic model for gearbox with cracked tooth. Variable meshing force was modeled in Fourier Series form. From the simulation results it is shown that vibration response of the system presents an obvious impact response characteristic[3]. Cui Yahui et al. modeled the variable meshing

stiffness by Fourier Series and created the dynamic model of geared rotor system to simulate the jump phenomenon. With the gradual increase of speed, it was found that the jump phenomenon's amplitude of one temporal frequency increased and the sidebands of mesh frequency and high order harmonic occurred[4]. The approximate analytical expression of variable meshing stiffness was put into ODEs directly to calculate the responses in both of the articles mentioned above, but the approximate analytical expression of the stiffness in Fourier series form only contains finite harmonic components. Cui Yahui et al. also considered the gear backlash and bearing stiffness, built the one level nonlinear dynamic model of gear-rotor system and examined the effect of gear backlash on the bifurcation and chaos of gear-rotor system. It is found that the vibration of the first order bending critical speed turned into chaos rapidly when clearance increased to a large value (230 μm)[5]. Li Yingming et al. created the one level mathematical model of gear pair nonlinear vibration and solved the model numerically based on Runge-Kutta Method to investigate the influence of backlash on gear vibration. It is discovered that chaos induced by gear backlash was easier to happen when the gear load is lighter[6]. Both of the articles mentioned above use the first method to model the gear backlash. This method is more common but not appropriate for the modeling of multiple levels gear backlash in multi-degree of freedom system. Ashley Crowther et al. set up the torsional vibration model of vehicle driveline and proposed deducing the restoring torque to model the gear backlash. The clunk phenomenon was simulated under the situation of torque changing in severe case and the correctness of the simulation was verified by the experiment[7], but the difference of restoring torque between the transmission error within and beyond the value range of gear backlash is ignored. The second method of modeling gear backlash is more appropriate for the modeling of multiple levels backlash in multi-degree of freedom system[8].

Taken launch shudder phenomenon of a certain passenger sedan as an object, 12 DOFs branched torsional vibration model of driveline is set up. The nonlinearity for two-stage gear variable meshing stiffness and backlash are modeled respectively by deducing the restoring torques. Vehicle responses for launch shudder, in both time and time-frequency domain are provided; meanwhile transmission errors of two-stage gear pairs with variable meshing stiffness considered and clunk phenomenon of loaded gears caused by the backlash are calculated under the typical launch operating conditions. Through the comparison of simulation, the impacts of gear meshing nonlinearities are analyzed.

A BRANCHED TORSIONAL VIBRATION MODEL OF THE VEHICLE DRIVELINE

A 12 DOFs branched torsional vibration model of vehicle driveline considered the nonlinear frictional torque characteristics of clutch, dry frictional damping of friction plate, the meshing stiffness and backlash of two-stage gear pairs, differential, the different torsional stiffness of left and right half axle, the longitudinal adhesion characteristics of tire and some other factors, which is illustrated in Figure 1.

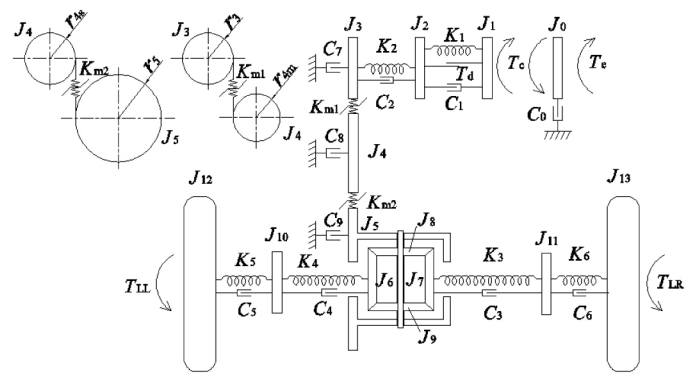


Figure 1. The branched torsional vibration model of vehicle driveline

In Figure 1, T_e is engine torque, T_c is clutch frictional torque, T_d is dry frictional damping torque of clutch friction plate. T_{LL} and T_{LR} are ground resistant torques of left and right driving wheels respectively.

The frictional torque transmitted by the clutch in the launching process is given in Equation (1), (2), (3). The friction coefficient of clutch frictional material is influenced by their relative slipping velocity and the temperature of the frictional surface[9].

$$T_c = \begin{cases} 2R\mu(v, T)F_n & \text{slipping} \\ T_{c_lock} & \text{engaged} \end{cases} \quad (1)$$

$$\begin{aligned} \mu(v, T) &= f(v)\delta(T) \\ &= \left(\mu_s + \frac{df(v)}{dv} v \right) \delta(T) \end{aligned} \quad (2)$$

$$\begin{aligned} T_{c_lock} &= \frac{J_1 T_e - J_1 C_0 \dot{\theta}_0 + J_0 K_1 (\theta_1 - \theta_2)}{J_0 + J_1} \\ &+ \frac{J_0 C_1 (\dot{\theta}_1 - \dot{\theta}_2) + J_0 T_d}{J_0 + J_1} \end{aligned} \quad (3)$$

Where μ_s is the coefficient of maximum static friction, v is relative slipping velocity between the friction surfaces, $f(v)$ is the function of friction coefficient varied with relative slipping velocity, $df(v)/dv$ is the slop of friction coefficient, T is the temperature of the friction surface, $\delta(T)$ is the temperature influence function of friction coefficient, R is the equivalent frictional radius and F_n is the applied normal force.

The dry frictional damping torque of the clutch is produced by the dry frictional damping washer in the friction plate. The dry frictional damping washer can rapidly attenuate the relative rotation between the friction plate and the platen hub when the torsional vibration occurs in driveline. Generally the normal force of the dry friction damping washer is ensured by disc spring, which can be considered as a constant. Thus, the dry friction damping torque in slipping state can be simplified as a constancy, which is usually 3~15Nm for the passenger cars. The nonlinear dry frictional damping torque is obtained, as derived in Equation (4) and (5).

$$T_d = \begin{cases} T_{d_slip} & \text{slipping} \\ T_{d_lock} & \text{locked} \end{cases} \quad (4)$$

$$T_{d_lock} = \frac{J_2 T_c + J_1 K_2 (\theta_2 - \theta_3) + J_1 C_2 (\dot{\theta}_2 - \dot{\theta}_3)}{J_1 + J_2} - K_1 (\theta_1 - \theta_2) - C_1 (\dot{\theta}_1 - \dot{\theta}_2) \quad (5)$$

The ground drag torques for the left and right driving wheels are respectively represented as Equation (6) and (7).

$$T_{LL} = F_{x1} r_d + T_{f1} \quad (6)$$

$$T_{LR} = F_{x2} r_d + T_{f2} \quad (7)$$

Where, F_{x1} and F_{x2} are the longitudinal forces between the driving wheels and road, T_{f1} and T_{f2} are the rolling resistance torques of the left and right driving wheels respectively, r_d is the dynamic radius of the wheel.

The average lumped LuGre model is used to simulate the longitudinal forces between the driving wheels and road[10], as given in Equation (8) where the bristle model is utilized.

$$F_{xi} = [\sigma_0 \bar{z}_i + \sigma_1 \dot{\bar{z}}_i + \sigma_2 v_i] \frac{F_{zf}}{2} \quad (8)$$

Where, $\bar{z}_i$ and $\dot{\bar{z}}_i$ are the average deformation and its rate respectively, σ_0 is the stiffness coefficient, σ_1 is the damping coefficient of the bristle, σ_2 is the viscous damping coefficient and v_i is the relative slipping velocity between tire and road($i=1,2$).

A half vehicle model under driving condition is shown in Figure 2, where T_l is the driving torque, F_{zf} and F_{zr} are vertical ground reaction forces of front and rear axle respectively, F_{xf} and F_{xr} are longitudinal forces of front and rear axle between tire and road, W_f and W_r are vertical loads of front and rear axle respectively and γ is the gradient of road. Because of the gradient of road and longitudinal acceleration of vehicle, vertical ground reaction forces of front and rear axle are given by Equation (9) and (10) respectively.

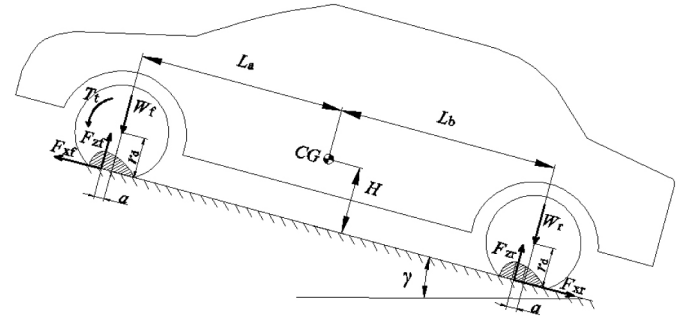


Figure 2. Half vehicle model on driving condition

$$F_{zf} = \frac{L_b m_v g \cos \gamma - m_v (g \sin \gamma + a_v) H}{L_a + L_b} \quad (9)$$

$$F_{zr} = \frac{L_a m_v g \cos \gamma + m_v (g \sin \gamma + a_v) H}{L_a + L_b} \quad (10)$$

Where, m_v is the Gross Vehicle Weight(GVW), which contains curb weight and two passengers of 75 kg, a_v is the longitudinal acceleration of the vehicle.

The meanings and values of the parameters in the branched torsional model of the vehicle driveline in 1st gear are listed in Table 1.

Table 1. The meanings and values of the parameters in the branched torsional model of vehicle driveline in 1st gear

Symbols	Meanings	Values	Units	Symbols	Meanings	Values	Units
J_0	Moment of inertia of the engine flywheel and pressure plate assembly	0.4000	kg·m ²	K_5	The torsional stiffness of the left front tire body	1.9572×10^4	Nm/rad
J_1	Moment of inertia of clutch friction plate, cushion plate and damping plate	0.0022	kg·m ²	K_6	The torsional stiffness of the right front tire body	1.9572×10^4	Nm/rad
J_2	Moment of inertia of clutch platen hub	0.0008	kg·m ²	C_0	Equivalent torsional viscous damping of the engine	0.01	Nms/rad
J_3	Moment of inertia of the transmission input shaft	0.0012	kg·m ²	C_1	Equivalent torsional viscous damping of the torsional vibration damper	0.20	Nms/rad
J_4	Moment of inertia of the transmission output shaft	0.0020	kg·m ²	C_2	Equivalent torsional viscous damping of the transmission input shaft	0.50	Nms/rad

Table 1. The meanings and values of the parameters in the branched torsional model of vehicle driveline in 1st gear (cont.)

J_5	Moment of inertia of the gear ring, shell, planetary gears and the pin shaft of the differential	0.0250	kg·m ²	C_3	Equivalent torsional viscous damping of the longer half axle	2.00	Nms/rad
J_6 、 J_7	Moment of inertia of a half axle gear about its own rotational axis	7.9600×10^{-5}	kg·m ²	C_4	Equivalent torsional viscous damping of the shorter half axle	2.00	Nms/rad
J_8 、 J_9	Moment of inertia of a planetary gear about its own rotational axis	2.7000×10^{-5}	kg·m ²	C_5	Equivalent torsional viscous damping of the left front tire body	10.00	Nms/rad
J_{10}	Moment of inertia of the short half axle, wheel hub, wheel rim and brake disc	0.7423	kg·m ²	C_6	Equivalent torsional viscous damping of the right front tire body	10.00	Nms/rad
J_{11}	Moment of inertia of the longer half axle, wheel hub, wheel rim and brake disc	0.7424	kg·m ²	C_7	Equivalent torsional viscous system damping of the transmission input shaft	0.02	Nms/rad
J_{12} 、 J_{13}	Moment of inertia of the left and right front tires	0.7740	kg·m ²	C_8	Equivalent torsional viscous system damping of the transmission output shaft	0.02	Nms/rad
K_2	Torsional stiffness of the transmission input shaft	8.7797×10^3	Nm/rad	C_9	Equivalent torsional viscous system damping of the differential	0.02	Nms/rad
K_3	Torsional stiffness of the longer half axle	5.9000×10^3	Nm/rad	r_3	The base circle radius of the pinion in 1 st gear pair	0.0133	m
K_4	Torsional stiffness of the shorter half axle	8.1000×10^3	Nm/rad	r_{4m}	The base circle radius of the wheel in 1 st gear pair	0.0461	m
K_{m1}	Averaged meshing stiffness of the 1 st gear pair in transmission	3.4080×10^8	Nm/rad	r_{4s}	The base circle radius of the pinion in final drive gear pair	0.0229	m
K_{m2}	Averaged meshing stiffness of the final drive gear pair in the transmission	5.9920×10^8	Nm/rad	r_5	The base circle radius of the wheel in the final drive gear pair	0.0944	m
H	CG height	0.5500	m	r_d	Dynamic radius of the wheel	0.2782	m
L_a	Distance for CG to front axle	0.9614	m	L_b	Distance between barycenter and rear axle	1.5036	m