

Modelling, Analysis and Simulation of Clutch Engagement Judder and Stick-Slip

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ABSTRACT

Analytical and numerical study is carried out to study the behavior of stick-slip and judder phenomenon during engaging and disengaging of the automotive clutch. For this purpose, a four degree of freedom torsional power train lumped mass model is developed. This torsional vibration system includes engine-flywheel, clutch, gear box and vehicle drive line, which are connected to each other by shafts. Equation of motion of the system is developed and initially a stability analysis is carried out for various gradients of coefficient of friction using eigen value analysis. Later, a numerical simulation is carried out to analyze the judder and stick-slip phenomenon using commercially available mathematical tool MATLAB. It is observed that the clutch stick-slip is increased with increase in external torque and clutch pressure fluctuations.

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INTRODUCTION

Powertrain systems can be investigated for vibration and noise characteristics through dynamic modelling, analytical and numerical analysis and testing. These investigations lead to a better understanding of the causes and effects of vibration and noise and thus allow improvement in design of trucks, passenger cars and buses.

Clutches in buses, cars, trucks and other vehicles are used to gradually engage the engine to the drivetrain while avoiding unpleasant shocks, jerks and excessive drivetrain wear. A basic clutch (see [Figure.1](#)) has two plates that can be moved together by an actuator that exerts a force on one of the two plates. One is pressure plate that is connected by an axle to the gearbox and remaining part of the powertrain. The other is the friction plate that is connected to the crankshaft. As the clutch engages plates are pushed together by the actuator. When the plates touch, torque is transmitted from the engine to the drivetrain. The vehicle now starts to move. After some amount of time the speeds of two plates become equal. The plates are then sticking and the engine is directly connected to the drivetrain. To achieve a successful engagement, the right input force has to be applied by the actuator. This can be done by the driver through a foot pedal.

Judder

Judder is an abnormal vibration in a mechanical system, especially due to grabbing between two friction surfaces. i.e., vibration is produced instead of smooth gradual engagement. Judder in automotive clutches has been attributed to an increasing friction coefficient with decreasing slipping velocity, dubbed as a negative friction gradient, and is known to reduce with system damping ([Kani et al., \(1992\)](#)).

Stick-Slip

Stick-slip is the surfaces alternating between sticking to each other and sliding over each other. Some examples of stick-slip are music comes from bowed instruments, squeaking sound of balloon when rubbing with fingers, squeaking sound of chalk on a blackboard, squeaky shoes on floor, noise of stopping train, noise of car brake, clutch and tires. Stick-slip in clutch is a phenomenon with dependence on friction characteristics, system dynamics, and external tangential and normal forcing ([Martins et al.,\(2004\)](#)). Most studies concentrate on systems with sliding interface between one moving and one stationary surface or between one moving and a substrate moving at a predetermined velocity. A great deal of analytical work has been published for a various systems. In automotive clutches the sliding interface is between two masses with undetermined velocities in a non-linear, non-autonomous system, thus, investigation using numerical methods is appropriate.

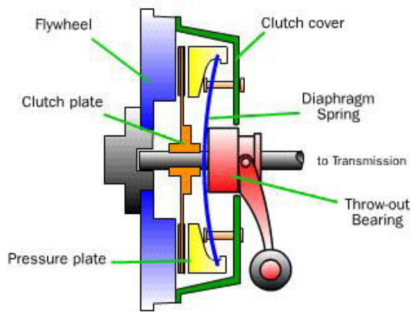


Figure 1. Automotive dry clutch [source: <http://www.ustudy.in/node/325>]

Reasons for Clutch Stick-Slip and Judder

Stick-slip motion may be regular or irregular. In the case of frictional sliding, stick-slip can have serious and often undesirable consequences, which resulting in noise, high energy loss (friction), surface damage (wear) and component failure.

Clutch Slip

Clutch slip is a condition where the full power of engine does not reach the transmission, i.e., the engine speeds up but the vehicle speed does not increase as it should. It occurs when the clutch disc is not gripped firmly and it slips between the flywheel and pressure plate as the member rotates.

Clutch slippage begins as a minor problem. At first, it will occur on initial and hard accelerations. As the vehicle moves on problem will occur on upshifts, downshifts and on any kind of acceleration. If it is left uncorrected, it becomes worse. Eventually, clutch discs becomes so badly worn that there is not enough friction present to move the vehicle.

Some of the reasons that the clutch may slip are overheating of pressure plate, deep grooves and traces of overheating on the pressure plate, facing contaminated on the inner portion, facing contaminated by grease or oil, facing carbonized, damaged diaphragm spring fingers, damaged idle damper, facing material worn down to rivets, facing scored on the flywheel side, wear marks on the release bearing inner bore and gearbox snout worn.

Clutch Stick

Sometimes problem is not with slipping but with sticking. The common reasons the clutch may stick are broken or stretched clutch cable, misadjusted linkage, air in hydraulic line, mismatched clutch components.

Broken or stretched clutch cable: Cable needs the right amount of tension to push or pull effectively.

Air in hydraulic line: Air affects the hydraulics by taking up space the fluid needs to build pressure.

Misadjusted linkage: When your foot hits the pedal, the linkage transmits wrong amount of force.

Mismatched clutch component: Not all aftermarket parts work with clutch. The right combination of components is essential for proper operations.

Clutch Judder

The reasons that clutch may judder are loose or worn out clutch facings, loose rivets, misalignment of pressure plate with flywheel, flywheel may be loose on the crankshaft flange and bent splined clutch shaft.

LITERATURE REVIEW

Rabeih and Crolla (1996) developed a torsional vibration model for a manual transmission truck coupled with a vehicle body longitudinal model and a vertical vibration model. They researched system response both in steady state running and transient running condition during and after clutch engagement. They varied the gradient of coefficient of friction versus stick-slip speed of clutch plates for the transient condition and compared results. For a system with zero or positive gradient of friction they found that the system acted with a transient damped response while for a negative gradient of friction the system acted with an unstable response. The final result they found was that the self-excited vibrations reduced with the high levels of system damping.

Bostwick and Szadkewski (1998) developed a model of a dry clutch tested with facing dynamometer. Like Centea et al., they also commented that clutch judder also occurs due to misalignments in the drivetrain that may produce fluctuating pressure between sliding surfaces, thermo elastic phenomena on contact surfaces and external torsional vibration in certain resonant conditions. They tested the friction material with negative sloped friction characteristics which produces torsional oscillations and compared the result with numerical model. Stick-slip was found numerically, but not validated with testing; a conclusion was drawn that it was associated with the self-excited vibration. This is reasonable as stick-slip is highly dependent on system dynamics.

Centea et al. (2001) researched dry clutch in manual transmission analytically. Through stability analysis they found that the negative friction slope produce clutch judder and that poor pedal acceleration could produce clutch judder even with a positively sloped friction characteristics. The judder oscillations were higher for more negative friction slopes. Stick-slip occurred in the simulations with the highest amplitude of oscillation before engagement.

In this paper a four degree of freedom torsional system model is used to represent a simplified powertrain system during clutch engagement. This powertrain system can be fitted with manual transmission. Clutch judder and stick-slip are researched qualitatively. First the system is defined, dynamic mathematical model provided and the equations of motion for the system with the clutch sticking and sliding are derived from the free body diagram. Then a stability analysis is performed for the linearized system, which demonstrates through Eigen values that how friction characteristics affect system stability, and therefore clutch judder.

Stick-slip algorithms are outlined and numerical results for nominal engagements for the research of contributing factors to judder and stick-slip are presented in the form of graphs

THE FOUR DEGREE OF FREEDOM (DOF) POWERTRAIN SYSTEM

The four degree-of-freedom (4DOF) torsional system is shown in figure 2. The free body diagram of 4DOF torsional system is shown in figure 3. The model has four rotating inertias (J_1 , J_2 , J_3 and J_4) connected by shaft elements (k_1 and k_2) or frictional contact (clutch engagement). The first inertia J_1 , represents the engine and flywheel, the second inertia J_2 , represents the one side of the clutch; they are connected by shaft stiffness k_1 . The third inertia J_3 , represents the other side of the clutch and transmission gear set. The second and third inertias J_2 and J_3 , are connected by frictional contact. The fourth inertia J_4 , represents the driveline inertia and vehicle mass; they are connected by shaft stiffness k_2 .

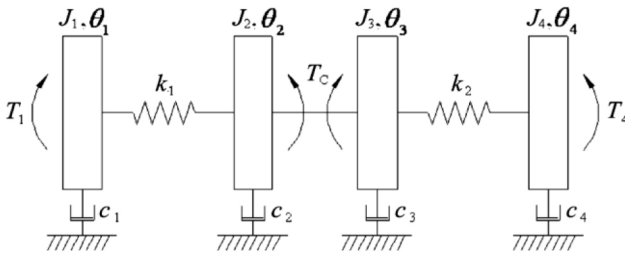


Figure 2. Four DOF powertrain system

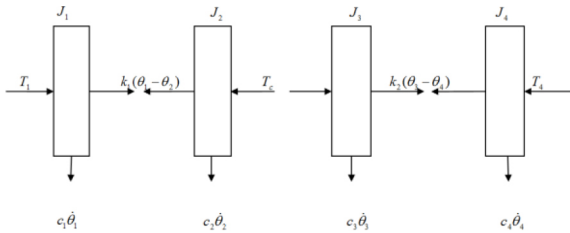


Figure 3. Free body diagram

The equations of motion for the system with the clutch (J_2 and J_3), either in stiction (sticking) or sliding are derived from the free body diagram shown in figure 3 and as follows

Sliding:

$$J_1 \ddot{\theta}_1 + k_1(\theta_1 - \theta_2) + c_1 \dot{\theta}_1 = T_1 \quad (1)$$

$$J_2 \ddot{\theta}_2 - k_1(\theta_1 - \theta_2) + c_2 \dot{\theta}_2 = -T_c \quad (2)$$

$$J_3 \ddot{\theta}_3 + k_2(\theta_3 - \theta_4) + c_3 \dot{\theta}_3 = T_c \quad (3)$$

$$J_4 \ddot{\theta}_4 - k_2(\theta_3 - \theta_4) + c_4 \dot{\theta}_4 = -T_4 \quad (4)$$

Stiction (Sticking):

$$J_1 \ddot{\theta}_1 + k_1(\theta_1 - \theta_{2/3}) + c_1 \dot{\theta}_1 = T_1 \quad (5)$$

$$(J_2 + J_3) \ddot{\theta}_{2/3} - k_1(\theta_1 - \theta_{2/3}) + c_2 \dot{\theta}_{2/3} + k_2(\theta_{2/3} - \theta_4) + c_3 \dot{\theta}_{2/3} = 0 \quad (6)$$

$$J_4 \ddot{\theta}_4 - k_2(\theta_{2/3} - \theta_4) + c_4 \dot{\theta}_4 = -T_4 \quad (7)$$

$\ddot{\theta}_{2/3}$ represents both coordinate 2 and 3; they are locked together and have the same solution in this state.

Nominal parameters used for the simplified powertrain system are given in Table 1.

Table 1. Nominal parameters used for simplified powertrain system.

Parameter	Value	Parameter	Value
J1	0.2 kg/m ²	c2	0 Nms/rad
J2	0.1 kg/m ²	c3	0 Nms/rad
J3	0.1 kg/m ²	c4	0 Nms/rad
J4	3 kg/m ²	Ns	10
k1	20600 Nm/rad	F	6000 N
k2	4840 Nm/rad	ro	0.0635 m
c1	0 Nms/rad	ri	0.051 m

Stability Analysis

Stability analysis for the slipping clutch can be performed for different friction characteristics. This is achieved by neglecting the non-autonomous or constant parts of the system of equations (1) to (4) and finding the eigen values of its system matrix (Centea et al.,(2001)).

In the 4DOF torsional system the engine torque T_1 the resistance torque, T_4 , are either non-autonomous or constant. The clutch torque T_c , is autonomous with a constant pressure. The only non-constant parameter in the clutch torque equation when sliding is the friction coefficient (μ_s). The coefficient of friction is dependent on the relative sliding velocity. Refer Figure. 4.

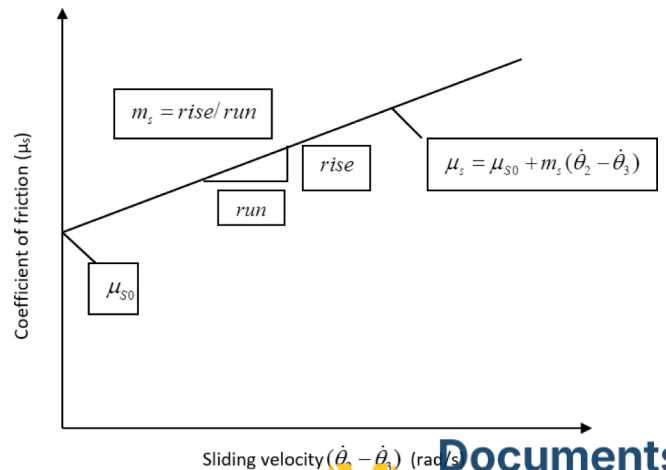


Figure 4. Variation of coefficient of friction with sliding velocity

Assuming constant pressure across the surface of the clutch plates, the equation for clutch torque is given by,

$$T_c = N_s R_m \mu_s F \quad (8)$$

where,

N_s - Number of friction surfaces

R_m - Clutch mean radius

μ_s - Coefficient of sliding friction

F - Clutch actuating force (friction force)

The mean clutch radius is determined from the clutch outside (r_o) and inside (r_i) radii

$$R_m = 2(r_o^3 - r_i^3) / 3(r_o^2 - r_i^2) \quad (9)$$

The coefficient of friction can be linearized to an initial condition value and gradient, m_s . Refer [figure. 4](#).

$$\mu_s = \mu_{s0} + m_s(\dot{\theta}_2 - \dot{\theta}_3) \quad (10)$$

Thus T_c can be written as

$$T_c = [m_s(\dot{\theta}_2 - \dot{\theta}_3)] N_s R_m F + \mu_{s0} N_s R_m F \quad (11)$$

Now the sliding clutch torque T is defined by two separate terms.

$[m_s(\dot{\theta}_2 - \dot{\theta}_3)] N_s R_m F$ is dependent on the relative velocity between the two sides of the clutch and $\mu_{s0} N_s R_m F$ is constant.

Coulomb friction model is the most commonly used friction model. It is also known as dry friction and it is being used in lubricated and contact boundaries modelling. The coulomb friction model is simplified as

$$F = F_c \text{sign}(v)$$

F_c is the coulomb friction sliding force and $v = \dot{\theta}$ is the sliding speed (velocity)

Coulomb sliding friction force is defined as

$$F_c = \mu_c N$$

where, N - normal load in the contact

Coulomb friction can cause problems in simulations due to the properties of sign function. The coulomb friction model is shown in the [Figure.5](#). It is noticed that F_c is independent of the sliding velocity between the surfaces under its action ([Edson Luciano Duque E. L. et al.,\(2012\)](#)).

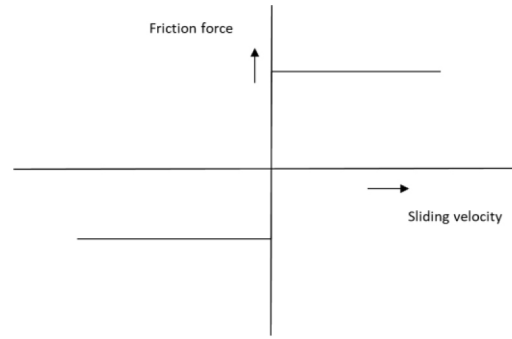


Figure.5. Coulomb friction model

A viscous friction model can be formulated as

$$F = k_v v$$

where k_v is the viscous coefficient

The viscous friction model is shown in the [Figure.6](#).

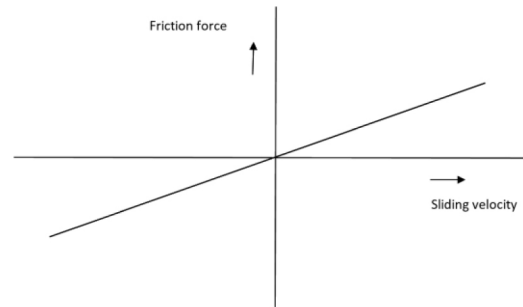


Figure.6. Viscous friction model

Since the coulomb friction model is problematic as regards both the analysis and simulation of a system behavior, a combination of the viscous friction model and the coulomb friction model could be advantageous. Using the combined Coulomb and viscous friction model eliminates the difficulty in determining the friction force at zero sliding speed both at start up and at direction change. The combined coulomb and viscous friction model is shown in the [Figure.7](#). ([Duque et al.,\(2012\)](#)).

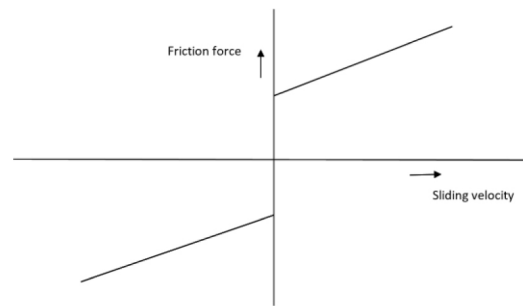


Figure.7. Combined coulomb and viscous friction model

The friction force F , value is taken as 6000 N in this work (refer [Table 1](#)).

The system, [equations \(1\) to \(4\)](#) can now be analyzed as the linearized homogeneous system as follows