ANALYSIS OF FRICITION INDUCED VIBRATION DURING ENGAGEMENT OF CLUTCHES

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ABSTRACT

During engagement of clutches when vehicle is started or during engagement of clutch when vehicle is in motion, the friction clutch generates chatter or also called as Judder vibrations. The vibrational behaviour of friction clutches not only affects the dynamics of transmission system but also the vehicle as result of excitation transfer to body via suspensions and mountings. If excitation level is high then it may cause discomfort to passengers. In addition to those effects vibrations generated may cause wear of friction material and thereby reducing performance and life of clutches.

The paper present investigation of this judder vibration which is generated during engagement of clutches. While there are more than one reasons of judder to occur in clutches, our main aim is to analyse the effect of frictional lining material characteristics on propensity to judder. A two degree of model is presented in order to study effect of various clutch system components on its vibration performance. The model is used to study the behaviour of various hypothetical facing materials during engagement. Further paper also investigates the effect of clutch system and driveline components on clutch vocational performance. The mathematical model developed can predict the vibrations generated and can also predict relative velocity at which vibration amplitude will be high. Results of three friction lining material characteristics are compared to find best suitable characteristics which generates less vibration.

Key words: Clutch, Judder, Friction Lining Material Characteristics, Two Degree of Freedom.

1. **INTRODUCTION**

There are two different types of chatter vibrations that can occur:

- Self-induced chatters (friction vibrations)
- Pressure-induced chatters

The self-induced chatter is caused by a friction coefficient change with regard to the slip speed. Figure 1 shows a pseudo-model of this: a body is pressed on the belt by its own weight. Friction arises between the body and the belt. When the belt is set in motion, it takes the body with it because of the static friction, and deflects the spring. Above a certain spring deflection, the body remains still because the spring load corresponds to the static friction. There is a relative motion between the body and the running belt. If the dynamic friction coefficient of the contact becomes less than the static friction, the friction load suddenly decreases and the spring draws the body back over the belt until there is adhesion once more and the body is drawn forward once again. The process begins again from the beginning – the body vibrates.

A vibration can thus only occur when the dynamic friction coefficient is lower than the static friction coefficient or the dynamic friction coefficient drops with increasing slip speed, because otherwise a stationary balance develops.

![Figure 1: Pseudo-model for self-induced vibration](image1)

Pressure-induced chatter is the result of an outside impulse source with periodic excitation. The belt model can be useful here as well to understand the excitation mechanism (see Figure 2). A periodically changing normal force affects the body.

![Figure 2: Pseudo-model for forced vibration](image2)
shown in the diagram. The current spring load also changes due to the changing friction load between body and belt and thus the equilibrium of the body on the belt. The body vibrates on the belt with the excitation frequency. If this frequency is the same as the natural frequency of the body-spring system, it results in resonance magnification and thus in large body vibration amplitudes. Naturally the pressure-induced chatter can also occur with neutral friction coefficient behavior, because it is excited by outside force modulation. The damping effect of the friction coefficient increasing with the slip speed naturally occurs again, because it counteracts an increase of the vibration amplitudes near the resonance.

2. RELATED WORK

D.Centea and H.Rahnejat al. [1] this paper presents an investigation of the tensional vibration mode of the vehicular clutch system, referred to as judder. This phenomenon takes place during the clutch engagement process on light trucks with Diesel engines. A nonlinear multi-body dynamic model of the clutch mechanism is presented in order to study the effect of various clutch systems and driveline components on its vibration performance. The parameters include geometrical, physical, inertial and force data of the clutch system of vehicles. The paper demonstrates that propensity to judder is related to the various friction lining materials' characteristics. A Crowther and N Zhang al. [2] Clutch engagement judder and stick–slip are investigated analytically and numerically to examine the influencing factors on these phenomena. Models are developed for a four degree-of freedom (4DOF) torsional system with slipping clutch and for a powertrain with automatic transmission system. Stability analysis is performed to demonstrate that clutch judder is dependent on the slope of the friction coefficient and the analysis is verified with numerical simulations. An algorithm for modeling stick–slip is developed and is used in numerical simulations which show that the likelihood of stick–slip is increased by clutch pressure fluctuations, judder approaching engagement, and external torque fluctuations. Numerical simulations for second to third gear up shifts demonstrate that the likelihood of stick–slip to occur from clutch engagement is increased by clutch applied pressure fluctuations, judder approaching engagement, and external torque fluctuations and that the likelihood of stick–slip occurring is decreased dramatically by applied pressure ramps proximus to the engagement point. Agusmain Partogi O., Paul Sas al. [3] in this paper, a friction model appropriate for wet friction clutches based on the extension of the Generalized Maxwell Slip (GMS) friction model is integrated to a four-DOF lumped-mass-spring-damper system which represents a typical SAE#2 test setup. Degradation models expressing the evolutions of the friction model parameters are also proposed, where the structure of the degradation models is inspired from experimental results obtained in the earlier work. This way, the engagement dynamics of the clutch system during the useful lifetime can be simulated. It appears that the previously developed pre-and post lockup features extracted from the simulated signals obtained in this study are qualitatively in agreement with the experimental results. Those features show their predictive behaviors that confirm their feasibility to be used for clutch monitoring and prognostics. Furthermore, the models and simulation procedure discussed in this paper can be employed for developing and evaluating prognostics algorithms for wet friction clutch applications. D Centea and H Rahnejat al. [6] this paper presents an investigation of the driveline torsional vibration behaviour, referred to as judder, which takes place during the clutch engagement process, particularly on small trucks with diesel engines. A non-linear multibody dynamic model of the clutch mechanism is employed to study the effect of various clutch system and driveline components on
the clutch actuation performance. The paper demonstrates that judder is affected by driveline inertial changes, variation in the coefficient of friction $\mu$ of the friction disc linings with slip speed, $v$, and the loss of clamp load. The results of the simulations show that various friction materials with different $\mu-v$ characteristics produce torsional self-excited vibrations of the driveline. The results also show that loss of clamp load relating to the speed of clutch actuation also contributes to judder. Shoaib Iqbal, Farid Al-Bender al. [7] in this study to understand the clutch vibrational and dynamical behavior, an SAE2 test setup mathematical model based on extended reset-integrator friction model was developed. In order to take into account the different phases of fluid lubrication during engagement cycle, the model includes the experimentally determined Stribeck function. In addition the model considers the viscous effect and the delay in the actuation pressure signal. The model is validated with the experiments performed on the SAE2 test setup in both time and frequency domains. By analyzing the set of experimental results, we confirmed that the amplitude of shudder vibration is independent of the amplitude of applied contact pressure fluctuation.

3. THE PROBLEM

![Figure 3: Physical example system](image)

Figure 3 shows the physical example system. The facing dynamometer driven by an electric motor is a relatively simple mechanical system to model for torsional vibrations. The reason for using this dynamometer as opposed to a vehicle application is that many factors can be eliminated in favour of sole concentration on the friction coefficient variation effects. Modeling a vehicle application would introduce torsional vibrations, misalignments, excess clearances of components and various random factors that would complicate the evaluation of the results and correlation process.

The dynamometer used for modeling consists of an electric motor to generate the system’s motion, a torque converter, inertia discs, and a simulated clutch actuated by air pressure and consisting of a rigid facing disc.

A theoretical problem is created using fig 3. Where all system is first at rest.

- Now electric motor is rotated at 1000 rpm. A torque convertor which is connected to electric motor output shaft then brings the remaining system to 1000 rpm.
When the whole system is maintaining velocity of 1000 rpm, a clamping force will be applied to facing disc.

The two mating reaction plates are regular pressure plates used to simulate pressure plate and flywheel.

The clutch will be slowed down from 1000 rpm to 0 rpm by applying a force to facing disc.

**Figure 4:** Physical representation of Mathematical Model

**Figure 5:** Free body diagram

- $\varphi$ - Inertia Disc Displacement
- $\dot{\varphi}$ - Inertia Disc Velocity
- $\ddot{\varphi}$ - Inertia Disc Acceleration
- $\theta$ - Facing Disc Displacement
- $\dot{\theta}$ - Facing Disc Velocity
- $\ddot{\theta}$ - Facing Disc Acceleration
- $k$ - Equivalent System Stiffness
- $J_1$ - Moment of inertia of inertia discs
- $J_2$ - Moment of inertia of facing discs

Equations of motion of this model are

$$J_1\ddot{\varphi} + k(\varphi - \theta) = 0$$

$$J_2\ddot{\theta} - k(\varphi - \theta) = -T$$

Where $T$ is frictional torque which is given by

$$T = F \times r \times \mu$$
Where $F$ is clutch actuating force, $R$ is mean radius of friction facings.

$\mu$ is friction coefficient of friction which is function of relative speed.

**Friction Coefficient Models**

*Figure 5:* Friction Coefficient Vs Relative Velocity Relative Velocity Material A

*Figure 6:* Friction Coefficient Vs Material B

*Figure 7:* Friction Coefficient Vs Relative Velocity Material C
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Figure 8: Friction Coefficient Vs Relative Velocity Material D

Initial Conditions

\[
\begin{align*}
\varphi(0) &= 0 \\
\theta(0) &= 0 \\
\dot{\varphi}(0) &= 104.7 \text{ rad/s} \\
\dot{\theta}(0) &= 104.7 \text{ rad/s} \\
J_1 &= 9.60 \text{ Kg.m}^2 \\
J_2 &= 0.058 \text{ Kg.m}^2 \\
k &= 48409 \frac{\text{N.m}}{\text{rad}}
\end{align*}
\]

4. COMPUTER SIMULATION PROGRAM

Using the mathematical model developed above, a computer program was intended to analyze actions of system during one engagement cycle.

The software used for programming was MATLAB/SIMULINK which evaluated a system of nonlinear differential equations with initial conditions by ode45 (Dortmund-prince) solver.

The output of program was plots of velocities of both inertia and engagement torque during engagement cycle

5. CLUTCH CLAMPING FORCE APPLICATION

The clutch clamping force was first treated as an instantaneous one and modelled as step function. But applying such instantaneous force led to excessive vibration of facing disc at start of engagement fig. 24. So in order to nullify these excess vibrations clamping force was model as ramp instead of step function at start of engagement, so that clamping force starts from zero and reaches maximum value after 0.15 seconds and thereafter remains constant at maximum value. The vibrations of facing disc decreased due gradual application of clamping force.
6. RESULTS
For Material A

![Figure 9: Facing disc velocity Vs time](image)

![Figure 10: Coefficient of friction Vs time](image)

For Material B

![Figure 11: Facing disc velocity Vs time](image)
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**Figure 12:** coefficient of friction Vs time

For Material C

**Figure 13:** Facing disc velocity Vs time

**Figure 14:** coefficient of friction Vs time
For Material D

Figure 15: Facing disc velocity Vs time

Figure 16: Coefficient of friction Vs time

Effect of Inertia of Inertia Discs

Figure 17: Facing disc velocity Vs time
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Figure 18: Enlarge view of fig. 17 time 1.4 to 1.5 sec

Effect of Inertia of Facing Discs

Figure 19: Facing disc Velocity Vs Time

Figure 20: Enlarge view of figure 19 from time 1.9 to 2 sec
**Figure 21:** Enlarge view of figure 9 from time 1.9 to 2 sec

Effect of Stiffness

**Figure 22:** Facing disc velocity Vs time

**Figure 23:** Enlarge view of fig 21 time 1.9 to 2 sec
Figure 24: Facing disc velocity Vs time with clamping force as step function

Figure 8 shows effect of application of clamping load on vibrational performance during starting of engagement. The results shows that using step function for clamping load during start of engagement generates unexpected vibration at start while using ramp function at start decreased vibration at staring of engagement.

The results form Figure. 9 and Figure. 10 for material A shows that as coefficient of friction was decreasing with decreasing relative velocity amplitude of vibration was less but as coefficient of friction increased with decreasing speed at end of engagement amplitude of vibration increased drastically. Same results are shown from figure. 15 and Figure. 16 for material D

The results from figure. 11 and figure 13 shows that as positive slope for coefficient of friction and relative velocity increases amplitude of vibration goes on decreasing. Figure.11 shows amplitude of vibration is on higher side for material B while figure. 13 shows amplitude of vibrations are on lower side for material C.

Figure 17 and figure. 18 shows effect of effect of inertia of inertia discs on performance of clutch. With decreasing inertia of inertia discs, amplitude of vibration and engagement time decreased. Frequency of vibration remains constant hence it does not have any effect of inertia of inertia discs.

Figure 19 and figure. 20 shows effect of effect of inertia of facing discs on performance of clutch. Facing discs inertia has no effect on engagement time as engagement time remains same with increasing inertial of facing discs. On the other hand, increasing inertia of facing discs reduces frequency of vibration and also amplitude of vibrations.

Figure 21 and figure 22 shows effect of equivalent stiffness. Engagement time remains constant with change in stiffness. Frequency and amplitude of vibration decreased with decreased stiffness.

7. CONCLUSION

From mathematical model developed in this paper it was observed that relation between friction coefficient and relative velocity has big influence on vibrational performance of clutch. Friction material having negative rate of change of friction coefficient with relative velocity generates self exited vibrations in clutches. Further it was observed that clutch vibration performance is also dependent on clutch systems and driveline components.
REFERENCES


