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Engagement Model of Dry Friction Clutch with Diaphragm Spring

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Abstract*—*The duration of engagement of automotive clutch plays an important role in the driving comfort and smooth launching of the vehicle. It is a transient phenomenon controlled by many variables like dynamics of release bearing and linkage, relation between release bearing travel and pressure plate lift, the clamp load developed with respect to cushion deflection and inertia of driver and driven shafts. Modern automobiles employ diaphragm spring clutch, which is advantageous in terms of less overall height and weight, number of components, low release load and increased service life. The non-linear characteristics of the diaphragm spring can be exploited favorably in achieving smooth engagement process. In this paper a mathematical model of transient engagement dynamics is developed correlating the parameters like spring characteristics, clamp load characteristics, pressure plate lift and release bearing travel characteristics, clutch pedal kinematics during engagement, vehicle driveline dynamics during startup, etc. The engagement duration of the clutch can be simulated along with the clamp load build up and torque transmission to the driveline using this model. Results of simulation are also included here which were verified through actual tests. This analysis should be useful in design of release mechanism for achieving smooth clutch engagement and to compare various clutches on the duration of slippage.

KEYWORDS—Engagement model, Duration of slippage, Clutch, Load build-up

1. INTRODUCTION

Internal Combustion engines only provide useful power over a certain speed range. To be able to use this range for various driving conditions, vehicles must have a gearbox. The power from the engine is transmitted to the gearbox through the clutch.

The engagement of an automotive clutch depends on the pedal movement controlled by the operator. The pedal linkage movement causes displacement of the clutch release bearing which plays an important role in the engagement and disengagement of the clutch. The engagement dynamics depends on pedal characteristics, release bearing characteristics, pressure plate characteristics, clamp load characteristics, inertia of the drive line etc. In the course of over 100 years of automotive history, nearly all components have undergone enormous technological development. Modern four-wheeled automobiles employ a clutch with diaphragm spring. As a result of which the relation between release bearing travel and pressure plate lift is nonlinear. Morfordand Szadkowski[1], have presented a simulation of clutch engagement keeping the throttle at a constant position.Naruse[2] developed a minimum slip lock-up clutch control system with the aim to minimize slip loss in the torque converter, isolating the engine's fluctuation. Lam and Yang[3]analyzed the engagement of a wet friction clutch, to obtain the torque response during engagement phase of the clutch. Haj-Fraj and Pfeiffer[4] studied the dynamics and control of vehicle automatic transmissions in order to provide realistic predictions about the system behavior during the gearshift operations for electronically hydraulically controlled wet-clutches. Kraska, Ortmann and Wang[5] developed a control oriented solenoid and clutch model for a passenger car automatic transmission. Glielmo and Vasca[6] presented control method of the dry clutch engagement process for automotive applications.Morselli, Sandoni, Viscontit and Zanasi[7] presented the dynamic model of a car transmission system along with a simple control strategy for controlling the transmitted torque.Bemporad, Borrelli, Glielmo and Vasca[8]proposed a piecewise linear feedback control strategy for the automotive dry clutch engagement process.Garofalo, Glielmo, Iannelli and Vasca[9] presented piecewise linear time-invariant models of automotive driveline in which a slip control technique for the dry clutch engagement process is proposed, using crankshaft shaft speed as measured variables.Chen, Xi and Zhang[10] developed a nonlinear multi-rigid-body dynamic model of automated clutch system during engagement of clutch. Agarwal and Tripathi[11] developed the dynamic engagement model of automotive clutch with diaphragm spring considering the pedal characteristics and inertia of the driveline.

The dynamic models of clutch and driveline presented in the above mentioned work do not specifically define the spring characteristics as well as the role of cushion deflection for building up the clamp load. Moreover the relation between pressure plate lift and release bearing travel is not considered. Release Load is considered same as the Clamp Load. The model presented in this paper attempts to include− the nonlinear nature of diaphragm spring, the pedal kinematics during engagement, release bearing travel and pressure plate characteristics and the dynamics of the driveline and the overall vehicle. The model is capable of analyzing the vehicle dynamics completely during the vehicle launch. The model is tested on a vehicle and results of findings are included.

2. WORKING OF CONVENTIONAL CLUTCH

The main components of a clutch unit are: the clutch cover assembly consisting of the clutch housing (also clutch cover),

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the clutch pressure plate as the clutch disc friction partner on the clutch side, the diaphragm spring which generates the clamp load, the tangential leaf spring – a spring-loaded connection between the cover and pressure plate to provide pressure plate lift, the supportring and the spacer for positioning and providing a mounting for the diaphragm spring; the clutch driven plate which consists of the hub, torsion damper with friction device and stop pin, the segment cushion springs and the friction material riveted to them; the flywheel with the pilot bearing (also clutch guide bearing); the release mechanism with guide sleeve, release bearing and release fork.

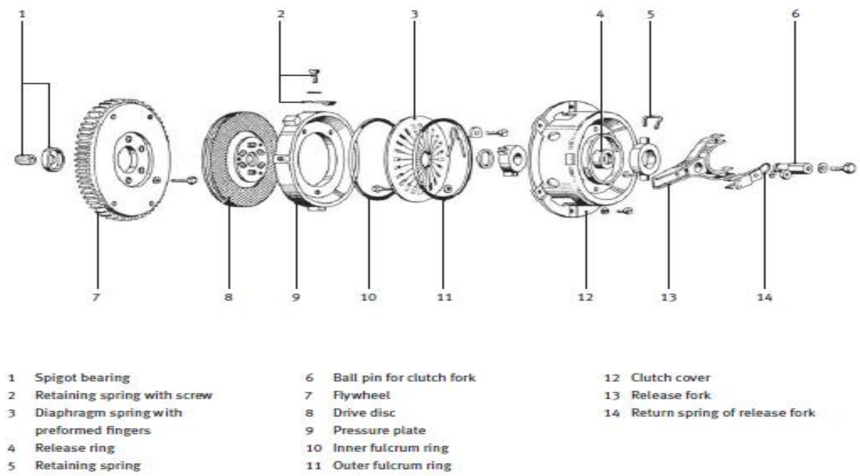

Figure I: Components of Clutch

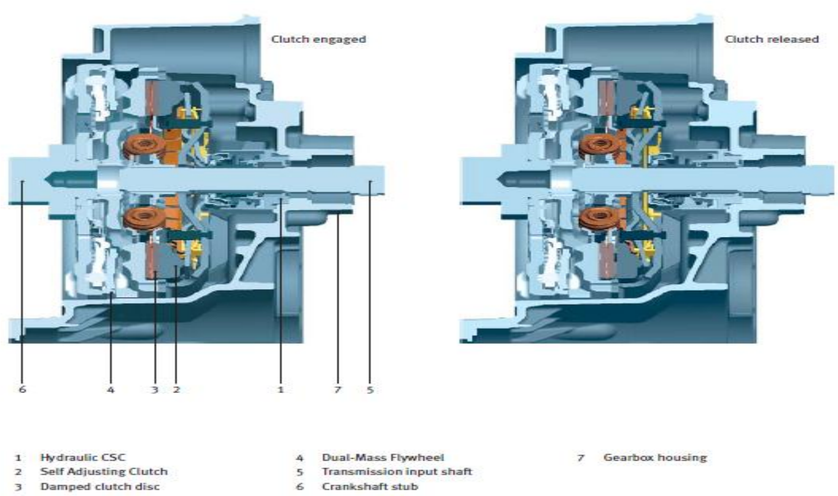

Figure II: Shows the engagement and disengagement of clutch

The two diagrams on the left detail the operating principle of a single-disc dry clutch with diaphragm spring. With the clutch engaged (left), the drive from the crankshaft is transmitted via the flywheel to the clutch pressure plate as shown in figure II. The positively engaged clutch driven plate transmits the drive via the hub assembly to the transmission input shaft. The diaphragm spring presses the axially variable pressure plate against the driven plate and flywheel. Thus the connection between engine and transmission is made. Depressing the clutch pedal disconnects the drive between engine and transmission. By actuating the release mechanism (rod link, cables or hydraulic system) the release fork and the release bearing connected to it moves toward the clutch cover assembly and depresses the diaphragm spring fingers. The diaphragm tips act as a lever. As further pressure is applied, the force direction is reversed by the diaphragm spring mounting; the pressure plate is relieved, and with the aid of the leaf springs moves away from the driven plate. The clutch disc is now able to rotate freely – engine and transmission are separated.

3. ENGAGEMENT MODELLING OF CLUTCH

The axial displacement of the release bearing $x_{rb}(t)$ depends on the dynamics of the release linkage mechanism as well as the clutch pedal movement controlled by the driver. This motion can be represented by a polynomial function of time,

$$x_{rb}(t) = a+bt+ct^2+dt^3+et^4+ft^5+gt^6 \qquad \text{Eq.(1)}$$

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This can be written as

$$x_{rb}(t) = [a \quad b \quad c \quad d \quad e \quad f \quad g] \left| \begin{matrix} 1 \\ t \\ t^2 \\ t^3 \\ t^4 \\ t^5 \\ t^6 \end{matrix} \right| \quad \text{Eq.(2)}$$

$$x_{rb}(t) = [\ A\ ] \left| \begin{matrix} 1 \\ t \\ t^2 \\ t^3 \\ t^4 \\ t^5 \\ t^6 \end{matrix} \right| \quad \text{Eq.(3)}$$

The above sixth degree polynomial is selected considering the fact that the movement of the manually operated clutch pedal, during release of pedal for clutch engagement, is most likely to be of splined nature, as shown is figure III.

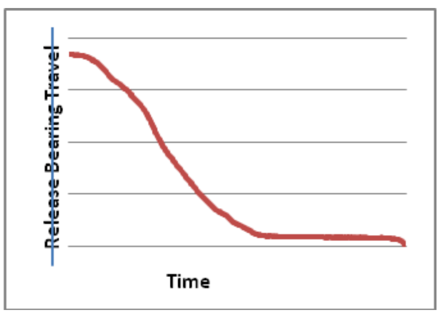


Figure III: Representation of release bearing travel with time

The release load is the load acting on the release bearing which initially rises until the operating point is reached, and then slowly drops again. The relation between release load and release bearing travel can be obtained from the experimental graph as shown in figure IV.

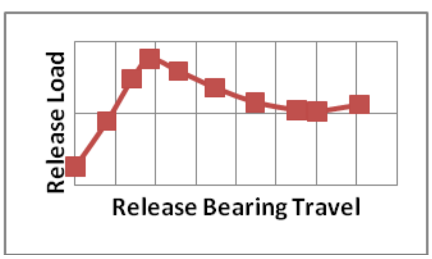


Figure IV: Representation of release load with release bearing travel

The displacement of the release bearing causes the pressure plate lift. The pressure plate is now at the maximum displaced position. Initially there is no movement of the pressure plate due to the deflection of the fingers of the diaphragm spring. As the release bearing moves further the pressure plate starts to rise. The relation of the pressure plate lift and release bearing travel without considering the cushion deflection is as shown in figure V.

Figure V: Representation of pressure plate lift with release bearing travel

The cushion deflection of the friction surfaces act as a spring which help in gentle clutch engagement and more favourable wear characteristics. Without a lining resilience system, the effective clamp load increases suddenly and relatively sharply during engagement. As the clutch is engaged, the clamp load slowly increases as the cushion springs must first be compressed.The clamp load is the axial load that is developed during the engagement of the clutch. It is responsible for engagement of the friction surface with the flywheel. As long as the pressure plate still makes contact with the clutch plate, the clamp load and cushion spring load correspond to one another. The relation of the clamp load and cushion deflection is as shown in figure VI.

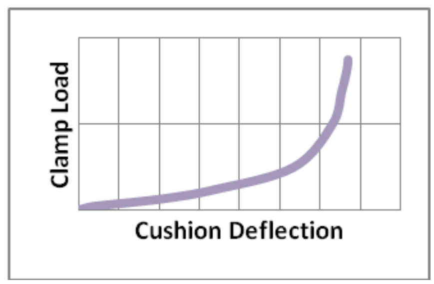


Figure VI: Representation of clamp load with cushion deflection

Assuming that the cushion deflection is equal to the pressure plate lift, the variation of clamp load with pressure plate lift and with release bearing travel can be found.

The motion of clamp load can be written as a polynomial function of degree 4 in terms of release bearing travel,

$$F_c\,(t) = p + q x_{rb}(t) + r x^2_{rb}(t) + s x^3_{rb}(t) + t x^4_{rb}(t) \quad \text{Eq.(4)}$$

This can be written as

$$F_c\,(t) = [p \quad q \quad r \quad s \quad t] \left| \begin{matrix} 1 \\ x_{rb}(t) \\ x^2_{rb}(t) \\ x^3_{rb}(t) \\ x^4_{rb}(t) \end{matrix} \right| \quad \text{Eq.(5)}$$

Thus,

$$F_c(t) = [\,P\,] \left| \begin{matrix} 1 \\ x_{rb}(t) \\ x^2{}_{rb}(t) \\ x^3{}_{rb}(t) \\ x^4{}_{rb}(t) \end{matrix} \right| \quad \text{Eq.(6)}$$

Equation(3) and Eq.(6) gives,

$$F_c(t) = [\,P\,] \left| \begin{matrix} 1 \\ [\,[\,A\,]\,[\,T\,]\,] \\ [\,[\,A\,]\,[\,T\,]\,]^2 \\ [\,[\,A\,]\,[\,T\,]\,]^3 \\ [\,[\,A\,]\,[\,T\,]\,]^4 \end{matrix} \right| \quad \text{Eq.(7)}$$

Thus the clamp load can be expressed as a function of time.

The transient friction torque transmitted by the clutch is given by

$T_f(t) = ¼\mu n F_c(t)[D_o+D_i] = HF_c(t)$ Eq.(8)

Using Eq.(7) and Eq.(8) we get,

$$T_f(t) = H\,[\,P\,] \left| \begin{matrix} 1 \\ [\,[\,A\,]\,[\,T\,]\,] \\ [\,[\,A\,]\,[\,T\,]\,]^2 \\ [\,[\,A\,]\,[\,T\,]\,]^3 \\ [\,[\,A\,]\,[\,T\,]\,]^4 \end{matrix} \right| \quad \text{Eq.(9)}$$

Figure VII shows the schematic diagram of the power transmission driveline of an automobile.

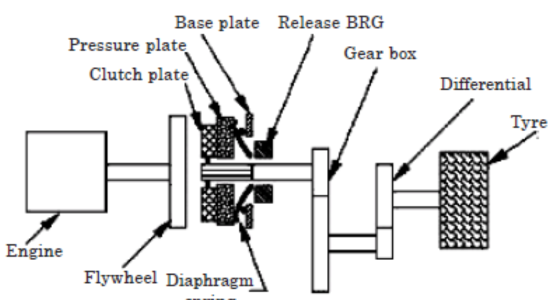


Figure VII: Driveline of an automobile

The dynamics of the engine shaft is given by

$T_e(t) - T_f(t) = I_e\ddot{\Theta}_e + C_e\dot{\Theta}_e + k_{te}\Theta_e$ Eq.(10)

Here, $T_e(t)$ is the engine torque, I_e, the engine rotary inertia, C_e , the damping coefficient of engine shaft journal bearing, and k_{te} is torsional stiffness of engine shaft. T_f (t) the friction torque transmitted by the clutch friction plate.

Similarly, the dynamics of the output shaft is given by,

$T_f(t) = I_R\ddot{\Theta}_I + C_I\dot{\Theta}_I + k_{tI}\Theta_I$ Eq.(11)

Here, $T_f(t)$ is the equivalent load torque referred to the output shaft of clutch. Also, I_R is the equivalent rotary inertia on load side, C_I , the damping coefficient of load shaft journal bearing and k_{tI} is torsional stiffness of load shaft.
The relationship between the load torque and wheel torque is given by

$T_f(t) = G_T T_W(t)$ Eq.(12)

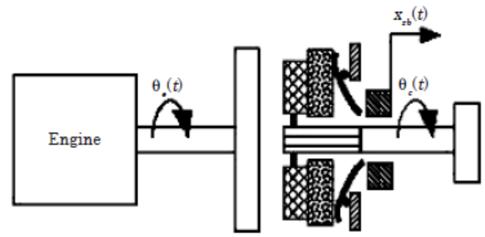


Figure VIII: The displacements in power transmission driveline of an automobile

Here, G_T is the combined gear ratio of differential and gearbox, and T_W , the load torque at the driving wheels. I_R is the equivalent load inertia referred to the clutch output shaft which includes the inertia of clutch disc, output shaft, gearbox, transmission shaft, differential gearbox, and wheel. At the time of vehicle launch the vehicle starts from a condition of standstill and the load inertia, connected down line the output shaft (carrying clutch friction plate) imposes a high load torque requirement. When the clutch is in engagement phase, the normal force provided by the pressure plate takes time to build up. Hence, during engagement phase, the instantaneous value of $T_f(t)$ is smaller than the instantaneous engine torque $T_e(t)$. Under such condition slippage occurs between clutch friction disc and engine flywheel, which continues till the clamp force on clutch friction plate $F_c(t)$ increases sufficiently, leading to the condition $T_f(t)=T_e(t)$. During this phase of clutch engagement, the rate of rise of $T_f(t)$ is very important for smooth engagement of the clutch. This rate depends on rate of rise of clamp force $F_c(t)$, which in turn depends upon the rate of change of pressure plate lift and the axial displacement of the release bearing. If the axial movement of release bearing is too sluggish, the building up of clutch friction torque will be slow, leading to unduly long engagement time, prolonged slippage, and consequently poor acceleration of the vehicle and excessive wear of the clutch friction plate. Hence, the motion trajectory of release bearing plays an important role in achieving adequate vehicle acceleration and long life of clutch plate friction lining.

4. DYNAMICS OF THE DRIVELINE

The friction torque T_f transmitted by the clutch causes angular acceleration of the rotating elements in the driveline, like the clutch disc, gears, and the wheel, as well as linear acceleration of the vehicle mass as a whole.

The inertia of the driveline having mainly the clutch, input shaft to transmission, gearbox, propeller shaft, differential, load shaft, wheel assembly etc. can be calculated by using the concept of reflected inertia.

A. Reflected Inertia

The reflected inertia is usually defined as the equivalentinertia of the transmission in neutral reflected (or referred) back to the clutch position.The reflected inertia of any gear train can be obtained if the inertia of all the individual elements and the relative speed of all the elements are known.

Let us consider two gears having number of teeth as Z_1 and Z_2 and radius as r_1 and $r_2$respectively.

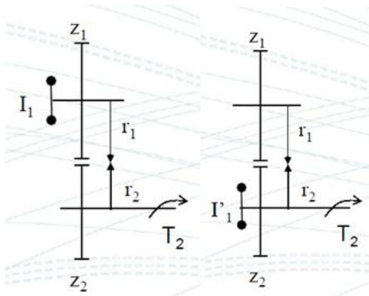


Figure IX: Concept of reflected inertia

Torque applied to shaft 2 = T

Torque applied to inertia I_1,

$T_1 = T_2(r_1/r_2)$. Eq.(13)

Speeds of shafts 1 and 2 are related by

$w_1 = w_2. (r_2/r_1)$ Eq.(14)

Now, Torque = Inertia * Angular Acceleration

$T_1 = I_1\, dw_1/dt$

Using Eq.(13) and Eq.(14)

$T_2. (r_1/r_2) = I_1\, dw_1/dt = I_1. (r_2/r_1). dw_2/dt$

Thus, $T_2 = I_1. (r_2/r_1)^2. dw_2/dt$ Eq.(15)

If $I_1$is replaced by a referred inertia I'_1, then we can write,

$T_2 = I'_1 dw_2/dt$

Therefore, $I'_1= I_1. (r_2/r_1)^2 = I_1. (Z_2/Z_1)^2$

The concept of referred inertia allows us to convert all the calculations on to a single shaft.

B. Reflected inertia of the driveline

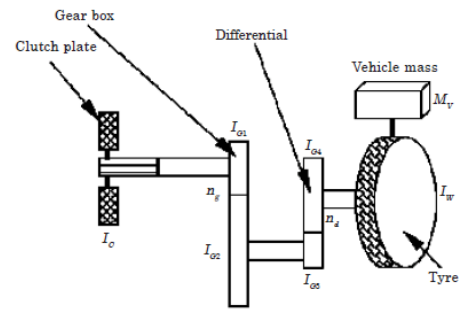


Figure X: Shows the inertia of various components, downside the driven shaft.

Applying the above discussed concept of reflected inertia the inertia of the driveline (I_R) can be calculated as,

$$I_R = I_C + I_{G1} + I_{G2}/n^2_g + I_{G3}/n^2_g + I_{G4}/n^2_g n^2_d + I_W/n^2_g n^2_d + M_V R^2_W/n^2_g n^2_d \quad \text{Eq.(16)}$$

5. MOTION OF THE CLUTCH SHAFT

Using Eq.(16), Eq.(11) can be written as,

$$T_f(t) = (I_C + I_{G1} + I_{G2}/n^2_g + I_{G3}/n^2_g + I_{G4}/n^2_g n^2_d + I_W/n^2_g n^2_d + M_V R^2_W/n^2_g n^2_d)\ddot{\Theta}_I + C\dot{\Theta}_I + k_{tI}\Theta_I$$

Eq.(17)

During a smooth clutch engagement, the clamp load and hence the torque will increase gradually. Therefore, the acceleration and angular velocity will also change gradually.The damping in the drive line is due to air–resistance and viscous effects in the bearings. At the time of vehicle launch the angular speed of clutch shaft will be low resulting into smaller air–resistance. Also, in modern drivelines, journal bearings have been replaced with anti–friction bearings. Therefore, the damping and stiffness terms in Eq.(17) may be dropped.

Hence Eq.(17) can be written as,

$$T_f(t) = (I_C + I_{G1} + I_{G2}/n^2_g + I_{G3}/n^2_g + I_{G4}/n^2_g n^2_d + I_W/n^2_g n^2_d + M_V R^2_W/n^2_g n^2_d)\ddot{\Theta}_I \quad \text{Eq.(18)}$$

Thus from Eq.(18) and Eq.(9) the angular acceleration of the clutch shaft can be written as,

$$\text{Where } [\ M\ ] = H\ [\ P\ ] / (I_C + I_{G1} + I_{G2}/n^2_g + I_{G3}/n^2_g + I_{G4}/n^2_g n^2_d + I_W/n^2_g n^2_d + M_V R^2_W/n^2_g n^2_d) \quad \text{Eq.(19)}$$

$$\ddot{\Theta}_I = [\ M\ ] \left| \begin{array}{c} 1 \\ [\ [\ A\ ]\ [\ T\ ]\ ] \\ [\ [\ A\ ]\ [\ T\ ]\ ]^2 \\ [\ [\ A\ ]\ [\ T\ ]\ ]^3 \\ [\ [\ A\ ]\ [\ T\ ]\ ]^4 \end{array} \right| \quad \text{Eq.(20)}$$

Using (20) the equation for angular velocity and angular displacement can be found by integration as,

$\dot{\Theta}_I = \int \ddot{\Theta}_I dt$ Eq.(21)

$\Theta_I = \int \dot{\Theta}_I dt$ Eq.(22)

The slipping of the clutch friction plate against the engine flywheel will continue till the angular velocities of the engine shaft and the clutch shaft become equal. The time duration for this event to occur can be determined by solving Eq.(21) and the total angular displacement under such slipping condition can be determined using Eq.(22). The coefficients of the matrix A and matrix P can be determined experimentally.

6. TEST RESULTS

The test is conducted on a light commercial vehicle with mass of vehicle as 1600 kg. The engine torque is 86 Nm and wheel radius is 0.261 m. The inner and the outer diameter of the clutch used are 0.134 m and 0.190 m. The moment of inertia of the clutch assembly is 0.0024 kgm^2. The numbers of friction surfaces on the clutch are 2 and co-efficient of friction between clutch surface and pressure plate is 0.27. The pedal load, pedal stroke with time are measured by using a combination of load cell and a string transducer.
The clutch release stroke is calculated by multiplying pedal stroke by lever ratio. An efficiency of 60% is considered in the experiment.

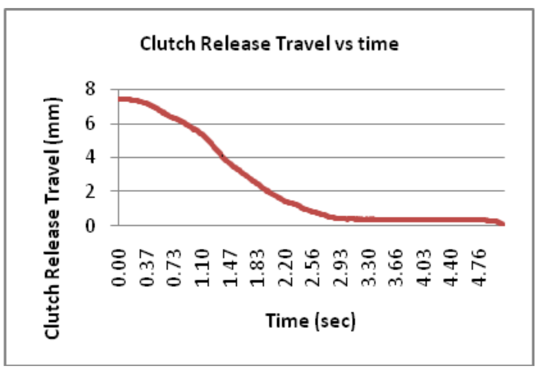
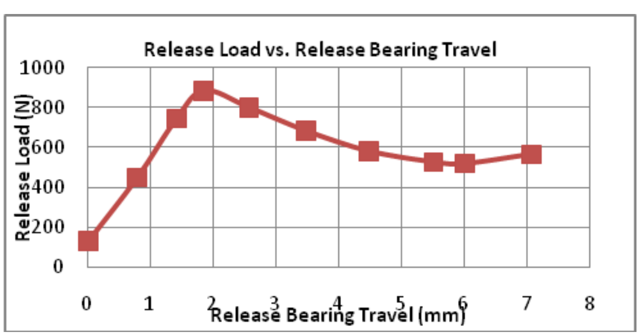
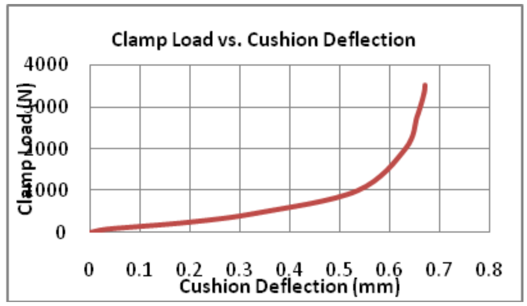


Figure XI: Variation of Clutch Release Bearing Stroke with Time.

By obtaining the graphs of clamp load with cushion deflection, pressure plate lift with release bearing travel, release travel with time, we can obtain the variation of clamp load with time.



FigureXII: Variation of Release Load with Release Travel



FigureXIII: Variation of Clamp Load with Cushion Deflection

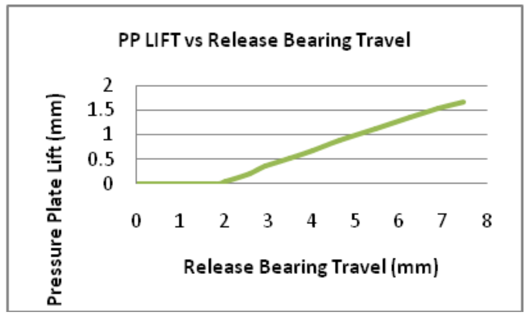
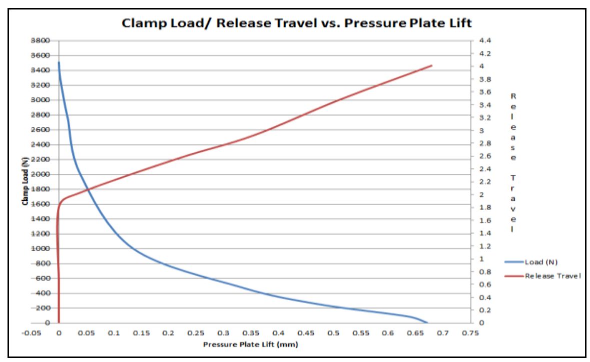


Figure XIV: Variation of Pressure Plate Lift with Release Travel



FigureXV: Variation of Clamp Load with Release Travel

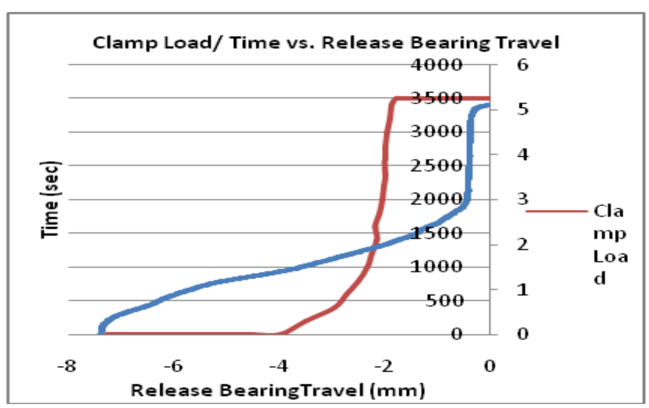


Figure XVI: Variation of Clamp Load and Time with Release Travel

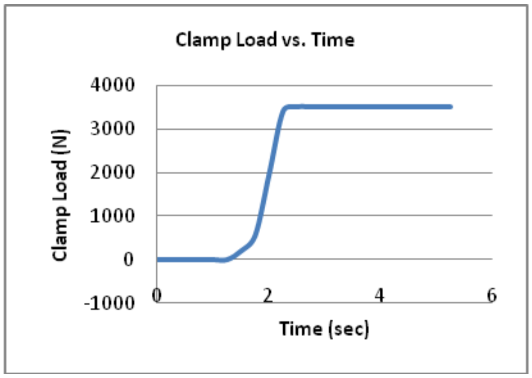


Figure XVII: Development of Clamp Load with Time

7. RESULTS

The clamp load is calculated to be 3500 N. The duration of slippage of the clutch is found to be 1.4 seconds.

8. CONCLUSION

An engagement model of transient dynamics has been presented, which can be used for analysis of driveline motion under different engagement conditions. Different release conditions can be simulated using this mathematical model. The model can be used to calculate the angular acceleration of the clutch shaft, based upon the behaviour of the non–linear disc spring, as well as, the motion trajectory imparted to the release bearing. The angular velocity and displacement of the clutch shaft, total slippage time and angle of slippage of the clutch plate can also be obtained using this model.

Using this model, complete simulation of the engagement dynamics of the clutch can be done for different diaphragm spring characteristic curves and clutch release curves. Results of simulation using relevant data corresponding to a typical small-size passenger car have been included.

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