Automotive vehicle clutch engagement characteristics for different types of force trajectories

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Abstract. Vehicle driveline of automobile, which transmit engine energy to the wheels, is complex system with nonlinear behavior. In manual transmission clutch engagement performance depends upon driver’s skill and is significantly affected by trajectory of clutch force during engagement process. In this work different types of clutch force trajectories are analyzed based on engagement performance which can be incorporated in automated manual clutch system. A driveline of a vehicle is modeled using multi body dynamics software and simulation were carried out for different type of force trajectories i.e. linear, parabolic and spline, which is followed by clutch force during engagement. The results of simulation were used to analyze the quality of clutch engagement and vehicle performance by looking into clutch lock-up phase and vehicle launch. The clutch lock time and clutch lock speed for various types of trajectories were calculated and compared. These results were also compared with analytical results reported by researchers. The results would be useful in selecting appropriate type of clutch actuating force trajectories for automation of clutch system of automobiles.

1. Introduction

Numerical simulation of various components of automobiles, e.g. of suspension system, transmission system etc., helps in studying the dynamics of the vehicle [1]. A clutch system is an integral part of manual transmission in vehicle driveline. A controlled engagement operation of clutch is required for smooth driving of vehicle which largely depends upon driver’s experience and skills. Accuracy in clutch engagement operation is essential to minimize driveline oscillations and vehicle body, heat generation and loss of power due to slippage, wear and enhance driving comfort. Since clutch is repetitively used for gear shift operation and vehicle starting, some variation may be expected due to human limitations [2].

One of the important factors which affect the engagement quality of clutch system is the trajectory of clutch force [2] which is defined as path followed to acquire the final clutch force during engagement phase and depends upon the skill of driver. The clutch engagement process has three phases due to stick-slip motion. Transitions between these phases introduce non-linearity in vehicle dynamics, which result in vehicle jerk [3]. In order to minimize driver intervention and make the clutch engagement or clutch locking process automatic and efficient, a control system with a predefined trajectory may be employed. This automated manual clutch will help in to improve the driving performance as well as fuel efficiency [2, 4, 5].
Various type of paths or trajectories have been discussed in literature from low order-linear type, which provide constant loading rate to high order spline type for the automation of clutch system. The engagement characteristics whether smooth or sudden during starting and end of engagement depends upon types of trajectory profile [6]. The type of path is also responsible for any overshooting of clutch force while engagement.

Literature shows that various parameters have been used for appraisal of engagement quality of clutch [7]. Gao et al. [8] suggested minimizing lockup time, friction losses and vehicle jerk as indicators for evaluation of clutch system. Chen et al. [3] proposed to minimize jerk, wear and frictional dissipation by smooth transition. Maximum engine speed, lockup time, Jerk and Shuffle were used by Tripathi [1] to evaluate engine quality.

The ride comfort will increase with selection of appropriate trajectory. This trajectory may be implemented by using embedded mechatronics system which consist of an actuator governed by control system to develop automated manual clutch. Researchers [4, 6, 14] have proposed automated manual transmission that uses electro-hydraulic and electro-mechanical system which are driven by control unit for gear shift. Gao et al. [8] conducted simulation for two-speed automotive manual transmission and proposed to use fitted trajectory for feed forward input and used a PID controller for feedback control against modelling error. Chen et al. [3] proposed model referenced adaptive control (MRAC) instead conventional control to control friction torque which reduce the effect of discontinuity in process of clutch engagement due to stick-slip.

Various researchers [4, 5, 9, 10, 11] reported mathematical model of driveline considering non linearity and carried out transient analysis. Couderc et al. [9] also compared simulation with experiment results. Centea et al. [10] used a nonlinear multi body dynamic model of drive line and clutch system to investigate the torsional vibration. Chen et al. [3] used Popov hyper-stability criterion to compensate the discontinuity in stick-slip process numerical simulation of clutch engagement. Duque et al. [12] presented mathematical model for clutch system while considering Coulomb, Viscous and Stribeck friction and found good correlation with Stribeck model in terms of energy as well as slipping time. Vasca et al. [5] discussed a model of torque transmissibility as function of friction and contact pressure.

In this work, the driveline of the vehicle is modeled using a multibody dynamic software MSC ADAMS and simulation was performed. Three different types of engagement trajectories to be followed by clutch force were used in simulation of clutch system to study and compare the dynamic response of driveline and clutch engagement performance.

2. Vehicle driveline model

![Diagram of Vehicle Driveline](image-url)

**Figure 1.** Model of vehicle drive line [1,3,6]
The vehicle driveline makes up of various components which transmit power from engine crankshaft to wheel through clutch and transmission system [13]. Figure 1 shows schematically the dynamic model of driveline of an automotive [2, 6]. The meaning of various symbols used in model are explained in list of symbols at the end of the paper. Many works have been proposed of dynamic models comprising of simultaneously algebraic differential equations along with their solution [2, 4, 7].

3. Modeling and simulation
A multibody dynamics simulation software ADAMS was used to model and characterize the vehicle driveline and clutch engagement. The vehicle driveline has been modelled in ADAMS software as per the schematic diagram shown in figure 1. Further simulation study has been carried out on the developed model using various input parameters. The numerical model considers lumped inertia for engine, clutch, gear and wheels. The numerical values of various parameters associated with medium size light motor vehicle were used to model the driveline as used by [7] and have been listed in table 1.

| Parameter                           | Value  | Unit    |
|-------------------------------------|--------|---------|
| Moment of Inertia (Engine)          | 0.16   | kg m²   |
| Moment of Inertia (Clutch Disk 1)   | 0.016  | kg m²   |
| Moment of Inertia (Clutch Disk 2)   | 0.016  | kg m²   |
| Moment of Inertia (Gear 1)          | 0.04   | kg m²   |
| Moment of Inertia (Gear 2)          | 0.04   | kg m²   |
| Moment of Inertia (Transmission and Wheels) | 0.24 kg m² |
| Torsional Stiffness (Engine-Clutch Shaft) | 32000 N⋅m⋅rad |
| Torsional Stiffness (Clutch-Gear Shaft) | 3200 N⋅m⋅rad |
| Torsional Stiffness (Counter Shaft and Axle) | 16000 N⋅m⋅rad |
| Damping Coefficient (Crank Shaft)   | 100 N⋅m⋅s⋅rad |
| Damping Coefficient (Clutch-Gear Shaft) | 4 N⋅m⋅s⋅rad |
| Damping Coefficient (Counter Shaft and Axle) | 90 N⋅m⋅s⋅rad |
| Damping Coefficient (Support Bearings) | 0.012 N⋅m⋅s⋅rad |
| Mass of Wheel                       | 5      | kg      |
| Mass of Vehicle                     | 1000   | kg      |
| Wheel Radius                        | 0.3    | m       |
| Maximum Engine Torque               | 120 N⋅m |
| Friction Coefficient (Clutch)       | 0.3    |         |
| Clutch Plate ID                     | 0.2    | m       |
| Clutch Plate OD                     | 0.3    | m       |
| Number of Friction Surfaces         | 2      |         |

Constrains were specified based on functional requirement of the system. The translational movement of clutch plate was specified according to profile to trajectory. Constant value of friction coefficient was used for friction torque function [12]. The gear ratio and loading condition were considered constant.

The linear and angular displacement and velocities at critical points were used as state variables in simulation. Figure 2 shows the isometric view of the complete driveline model in ADAMS/view. Simulation of this model has been done to calculate the parameters.

4. Simulation of vehicle driveline
Three different generic trajectories i.e. linear, parabolic and spline were selected to study their effect. These paths were specified in model by using appropriate function to see the effect on quality of engagement and the ride comfort output. The clutch release time was taken as 0.7 second, as it corresponds to small-size car. The maximum clutch force was taken as 2000 N and the coefficient of friction for the clutch is taken as 0.5. Simulation of vehicle driveline was carried out to obtain the dynamic behavior of drive line. An important element of dynamic response of driveline is the variation of flywheel shaft and clutch shaft speeds during engagement. A detailed analysis of variation of engine crank shaft and clutch shaft Speed was carried out in ADAMS. Figure 3 shows the effect of different trajectory functions (for the clutch engagement) on to the variation of angular velocities of flywheel and
clutch over the time. It can be shown that the trajectory path significantly affects the flywheel and clutch angular velocities.

5. Results and discussion
The simulations were conducted by considering three different type of paths for clutch force during engagement. The engagement quality of clutch system was evaluated by calculating clutch lock-up speed and clutch lock-up time which are significantly affected by nature of trajectories as shown in the figure 3.

The clutch lock-up speed is a speed corresponding to lock-up point, where engine crankshaft speed becomes equal to clutch shaft. The speed value corresponding to lock-up point depends upon engine
specification and varies from engine to engine. A too low lock-up speed may result into killing of the engine [2].

During engagement process clutch shaft speed increases and engine/ flywheel shaft speed decreases due to inertial of driven parts i.e. wheels and vehicle body, as shown in figure 3. The clutch lock-up speed was noted more or less same for spline type path (~160 rad/sec) which is around 10% higher than linear type trajectories. This profile also provide smooth starting and smooth ending of clutch engagement process and almost no oscillations were observed in driven shaft during engagement. However few oscillations can be seen just before lockup point in linear and parabolic profile, which may be attributed to sudden start and sudden end type of characteristics of the two profiles.

The lock-up time represents the delay in achieving the locking up of clutch system. The lockup time directly affects the heating and wear due to slip of friction lining. The results obtained from simulation are shown in graphical form in figure 4. The spline force trajectory showed overall better results in terms of lock-up speed and lock-up time. Similar results were also reported by [6], however they found slippage and friction work slightly high compared to linear profile but still in medium range.

![Figure 4. Clutch lock-up speed and lock-up time for different force trajectories](image)

6. Conclusion

The trajectory followed during clutch engagement significantly affects the dynamics of driveline and driving comfort. Following points were noted from simulation of driveline for three different types of trajectories i.e. linear, parabolic and spline based on engagement quality.

- Clutch lock-up speed is highest for spline force trajectory and lowest for linear force trajectory among three profiles for given input parameters
- Clutch lock-up time was more or less same for linear and spline force trajectories and lower than time taken by parabolic force trajectory for given input parameters.

Spline force trajectory showed overall better results in terms of lock-up speed and lock-up time based on engagement process however other factors like jerk, slippage and frictional heating also need consideration for trajectories in automated clutch system.

7. List of symbols used

| Symbol | Description |
|--------|-------------|
| $\theta_e$ | Angular rotation of flywheel |
| $\theta_c$ | Angular rotation of clutch friction disc |
| $\theta_{cl}$ | Angular rotation of clutch pressure plate assembly |
| $\theta_g$ | Angular rotation of input gear of gearbox |
| $J_{g1}$ | Moment of inertia of input gear |
| $J_{g2}$ | Moment of inertia of output gear |
| $B_{ce}$ | Torsional damping coefficient of crankshaft bearing |
| $B_{ce}$ | Torsional damping coefficient of clutch output shaft bearing |
\[ \theta_g \] Angular rotation of output gear of gearbox
\[ B_{gw} \] Torsional damping coefficient of transmission shaft bearing
\[ \theta_w \] Angular rotation of wheel
\[ K_{cg} \] Torsional stiffness of clutch output shaft
\[ J_{wc} \] Moment of inertia of wheel about its own axis
\[ K_{ec} \] Torsional stiffness of crankshaft
\[ J_{e} \] Moment of inertia of flywheel and crankshaft
\[ K_{gw} \] Torsional stiffness of transmission shaft
\[ J_{c1} \] Moment of inertia of clutch friction disc
\[ m_c \] Vehicle mass
\[ J_{c2} \] Moment of inertia of clutch pressure plate
\[ m_w \] Wheel mass

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