

# Development of High Fidelity Dynamic Model with Thermal Response for Single Plate Dry Clutch

Amar Penta, Prasad Warule, Sanjay Patel, and Lohit Dhamija  
Tata Motors, Ltd.

## ABSTRACT

Single plate dry clutch is most commonly used in automotive transmission. This paper proposes a unique approach of modelling a single plate dry clutch in Simulink and Simscape simulation environment. Clutch model is divided into two subsystems as translational and rotational. The translational system is linear system of diaphragm and cushion spring as a two-degree freedom system. Nonlinearity of the diaphragm and cushion spring has been modelled based on experimental data. This enables to simulate friction torque variation during clutch engagement. In rotational system, frictional torque generation between flywheel-clutch disc and pressure plate-clutch disc has been modelled separately. This novel approach of developing separate friction models helps in understanding variation in torque carrying capacity due to rise in the temperature of the friction pads because of frictional and engine heat. The variation of friction coefficient with rise in temperature has been taken into account based on experimental data. The model is simulated for different conditions and results are discussed.

**CITATION:** Penta, A., Warule, P., Patel, S., and Dhamija, L., "Development of High Fidelity Dynamic Model with Thermal Response for Single Plate Dry Clutch," *SAE Int. J. Passeng. Cars - Mech. Syst.* 10(1):2017, doi:10.4271/2017-26-0260.

## INTRODUCTION

In automotive transmission, clutch plays vital role during gear shift. Quality of shift depends on clutch actuation and synchronization in synchromesh gearbox. The single plate diaphragm spring dry clutch is most commonly used in MT and AMT. It has several advantages with respect to ergonomics and operation. It reduces the pedal effort and improves clutch engagement, wear life, reliability etc. discussed in [1].

The mathematical model of clutch has several applications in clutch control design and development. For instance, in case of AMT development, [2] clutch control is optimized and proved with the high fidelity mathematical model. The mathematical model of clutch can also be used to come up with optimised clutch design. [3] builds virtual design system platform for clutch which saves the time, manpower and efforts in product development cycle. The proposed platform is based on numerical approach.

In order to study the effects of various parameters, which affect torque carrying capacity of the clutch, requires a model which includes all the required phenomena. [4,5] Includes dependence of the torque transmissibility on wear, slip speed, friction pads geometry. The influence of the diaphragm spring, flat/cushion spring and the torsional damper springs is also taken into account to model the system. This model also finds the dependence of the transmitted torque on contact friction and pressure distribution (uniform pressure or uniform wear). Moreover, transient engagement dynamics is

developed in [5] correlating the parameters like pressure plate lift and release bearing travel characteristics, clutch pedal kinematics during engagement and vehicle driveline dynamics during vehicle startup.

The coulomb friction model is used in [6] for rotational system of clutch. The diaphragm spring characteristic have been modelled as a combination of both linear and nonlinear system. While model in [7] follows approach for real time application, a fixed time step clutch model has been developed. The thermal characteristics influence the torque carrying capacity. Therefore [8,9] develop torque model considering thermal expansion of parts. This has been studied by simulating the torque transmission in a heavy-duty truck dry clutch at slipping conditions [8]. [1,10] include the wear of clutch which is responsible to reduce the thickness of friction pads. This modelling gives the flexibility to observe the temperature during slippage and after engagement.

This paper focuses on developing a high fidelity dynamic clutch model. The experiment based data is used to model nonlinearity of diaphragm and cushion spring. A different approach is undertaken to model the thermal effect of individual friction surfaces on torque carrying capacity.

## CLUTCH ENGAGEMENT DYNAMICS

Figure 1 shows the simplified schematic diagram of single plate dry friction clutch in disengaged condition. The schematic gives a brief idea about clutch parts and their positions in assembly. Flywheel is mounted on input shaft and it forms the friction pair with clutch disc.

The clutch is coupled to output shaft by means of spline. The clutch disc is free to move on output shaft within confined space. The clutch disc has two friction pads on either side and cushion spring is sandwiched between the friction pads by means of rivets. The intention of using cushion spring is to allow smooth engagement. The torsion damping springs are fitted on the central plane of the clutch disc in a circular fashion to damp the torsional oscillations. This entire subsystem of friction pads, cushion spring and torsion damping springs forms a clutch disc assembly. Pressure plate along with another friction pad of clutch disc forms second friction pair. The pressure plate, diaphragm spring and cover forms clutch cover assembly. Clutch cover assembly is usually bolted to flywheel and both together acts as a single unit. The diaphragm spring is a steel disc having hole at center, disc is slotted radially. This forms actuating release lever fingers and are operated by CRB. The outer ends of the slots are provided with enlarged blunting holes. These blunt holes help to distribute the concentrated pressure created during clutch actuation i.e. deflection of the fingers by CRB. The engagement and disengagement phenomena is governed by CRB travel. Figure 2 Shows, two clutch conditions (a) engaged condition and (b) disengaged condition. The clutch is said to be engaged when the cushion spring is compressed to flat disc by means of force applied by pressure plate. The compression force defines the torque carrying capacity of clutch.  $x_{bs}$  defines the CRB travel and  $x_{cs}$  cushion spring travel. The CRB travel and cushion spring are dependent such that  $x_{cs} = f(x_{bs})$ .

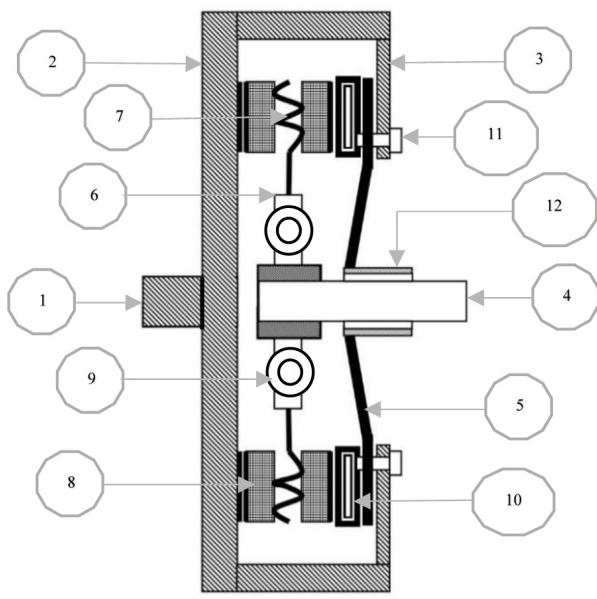


Figure 1. Schematic of single plate dry clutch (clutch is in open condition) : (1) Input Shaft, (2) Flywheel, (3) clutch cover assembly, (4) output shaft, (5) Diaphragm/Belleville spring, (6) Clutch disc assembly, (7) Cushion/Flat spring, (8) Friction pad, (9) Torsion damping spring, (10) Clutch pressure plate, (11) Fulcrum, (12) Clutch release bearing.

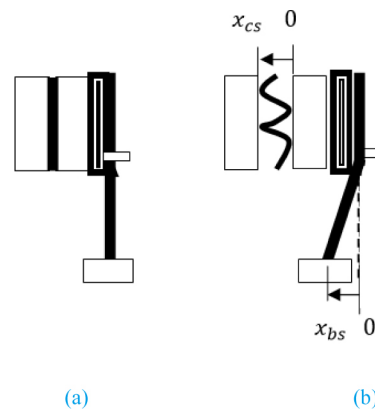


Figure 2. Axial displacement of diaphragm and cushion spring: (a) Engaged (unloaded), (b) Disengaged (loaded) condition

The mathematical model of clutch has been built by considering the physical systems involved and their dynamic interactions in multi domain environment. To simplify and to increase fidelity level, rotational and translational dynamics are modelled separately. Each sub system defines dynamic actions occurring at their subsystem level and their details are discussed in further subsections.

The considerations and assumptions made for clutch modelling are as follows

- Nonlinearity of both the diaphragm and cushion spring is considered.
- Variation of friction coefficient with respect to temperature is taken into account.
- Frictional torque generation on either side of clutch disc has been modelled separately.
- The temperature rises of friction pad due to frictional energy has been modelled.
- Effect of torsional damping spring has been ignored.
- There is no friction in translational motion of clutch disc.

### Translational Model

The translational system includes the interaction of clutch parts which exhibit the translational degree of freedom and plays role in transmitting the force applied at CRB to generate the friction torque. Let us assume subscript  $fp$ ,  $cp$  and  $bs$  represent friction plate, clutch pressure plate and bearing sleeve respectively. The symbols  $m$ ,  $C$ ,  $k$ ,  $x$  indicate mass, damping constant, stiffness and displacement respectively.

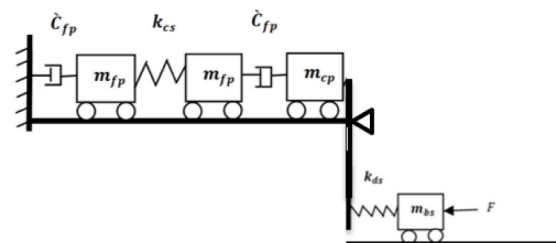


Figure 3. Translational FBD

The force applied at CRB is responsible for deflection of diaphragm, cushion spring and acceleration of friction pad, CRB. Figure 3 shows, the free body diagram of translational system.

The mass of friction pad is very small so its effect on the system is neglected. The ease of modelling of complex system is obtained by simplifying system without much effect on its principle operation. The simplified FBD for translational system can be drawn as follows,

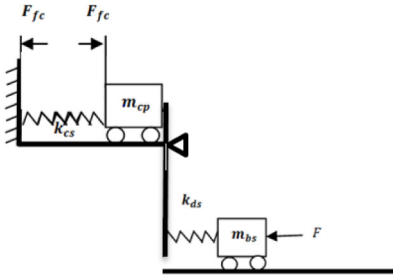


Figure 4. Simplified translational FBD

In figure 4,  $F$  is the force applied at CRB for clutch actuation and  $F_{fc}$  is the force responsible for friction torque generation. The stationary member which is left to the cushion spring represents flywheel and right is the clutch pressure plate. The  $F_{fc}$  applied at both these members is responsible for total friction torque generation.

The equation of motion of simplified translational system by applying D'Alembert's principle can be written as

$$\begin{aligned} m_{bs}\ddot{x}_{ds} &= -k_{ds}(x_{ds} - x_{cs}) + F \\ m_{cp}\ddot{x}_{cs} &= k_{ds}(x_{ds} - x_{cs}) - k_{cs}x_{cs} \end{aligned}$$

Rearranging above equation

$$m_{bs}\ddot{x}_{ds} + k_{ds}x_{ds} - k_{ds}x_{cs} = F \quad (1)$$

$$m_{cp}\ddot{x}_{cs} + (k_{cs} + k_{ds})x_{cs} - k_{ds}x_{ds} = 0 \quad (2)$$

In above equations  $\dot{x}$  and  $\ddot{x}$  represents linear velocity and acceleration. The diaphragm and cushion springs are nonlinear springs. The stiffness characteristic depends on their deflection i.e.  $k_{ds} = f(x_{ds})$  &  $k_{cs} = f(x_{cs})$ .

Above equations can be written in matrix form

$$\begin{bmatrix} m_{bs} & 0 \\ 0 & m_{cp} \end{bmatrix} \begin{bmatrix} \ddot{x}_{ds} \\ \ddot{x}_{cs} \end{bmatrix} + \begin{bmatrix} k_{ds} & -k_{ds} \\ -k_{ds} & (k_{cs} + k_{ds}) \end{bmatrix} \begin{bmatrix} x_{ds} \\ x_{cs} \end{bmatrix} = \begin{bmatrix} F \\ 0 \end{bmatrix}$$

Friction torque is function of cushion spring deflection  $x_{cs}$ . Cushion spring deflection is a function of diaphragm spring and diaphragm spring deflection is function of force applied at CRB.

### Rotational Model

The rotational system includes the systems of clutch which exhibits rotational degree of freedom and plays role in clutch engagement. Let us assume the subscript  $e, f, c$  and  $v$  represent engine, flywheel, clutch disc and vehicle respectively. The symbol  $J, T, \omega$  &  $\theta$  represent inertia, torque, angular speed and position.

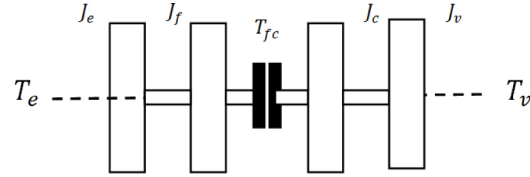


Figure 5. Rotational FBD

In figure 5,  $T_{fc}$  is the total friction torque generated due to compression of cushion spring and is the sum of friction torque generated between flywheel-friction pad and pressure plate-friction pad. The equation of motion of rotational system can be written as

$$(J_e + J_f)\dot{\omega}_f = T_e - T_{fc} \quad (3)$$

$$(J_c + J_v)\dot{\omega}_c = T_{fc} - T_v \quad (4)$$

In above equation  $T_v$  and  $J_v$  define vehicle torque due to road loads and vehicle effective inertia at clutch output respectively. The vehicle torque is function of vehicle mass ' $m$ ', coefficient of rolling resistance ' $\gamma$ ', road grade ' $\theta$ ', coefficient of drag ' $C_d$ ', air density ' $\rho$ ', vehicle frontal area ' $a$ ', vehicle velocity ' $v$ ', tire rolling radius ' $r_r$ ', gearbox ratio ' $N_{gb}$ ', differential ratio ' $N_{df}$ ' and gravitational constant ' $g$ '. It is given by

$$T_v = \frac{r_r}{N_{gb}N_{df}} (mg \sin \theta + \gamma mg \cos \theta + \frac{1}{2} C_d \rho a v^2)$$

The equations (3) and (4) hold good when there is slip i.e.  $(\omega_f - \omega_c) \neq 0$ . But Lockup condition of the clutch can be defined when as  $\dot{\omega}_f = \dot{\omega}_c$  i.e. slip is zero. Therefore, equation (3) and (4) can be equated and written as

$$\frac{(J_e + J_f)}{T_e - T_{fc}} = \frac{(J_c + J_v)}{T_{fc} - T_v} \quad (5)$$

Above equation can be written for  $T_{fc}$

$$T_{fc} = \frac{T_e(J_c + J_v) + T_v(J_e + J_f)}{(J_e + J_f) + (J_c + J_v)} \quad (6)$$

From above equation we can define lock up condition of clutch as,

$$(\omega_f - \omega_c) = 0 \text{ \&}$$

(Friction torque due to applied force  $F$ )  $\geq T_{fc}$