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# Tribodynamics of a new de-clutch mechanism aimed for engine downsizing in off-road heavy-duty vehicles

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## Abstract

Clutches are commonly utilised in passenger type and off-road heavy-duty vehicles to disconnect the engine from the driveline and other parasitic loads. In off-road heavy-duty vehicles, along with fuel efficiency start-up functionality at extended ambient conditions, such as low temperature and intake absolute pressure are crucial. Off-road vehicle manufacturers can overcome the parasitic loads in these conditions by oversizing the engine. Caterpillar Inc. as the pioneer in off-road technology has developed a novel clutch design to allow for engine downsizing while vehicle's performance is not affected. The tribological behaviour of the clutch will be crucial to start engagement promptly and reach the maximum clutch capacity in the shortest possible time and smoothest way in terms of dynamics. A multi-body dynamics model of the clutch system is developed in MSC ADAMS. The flywheel is introducing the same speed and torque as the engine (represents the engine input to the clutch). The hydraulic pressure is applied behind the piston to initiate the engagement. The angular motion of the plates is supported by friction torque between the plates and friction linings. The conjunctions between paper-based linings and steel plates are designed to be dry. Friction (the most significant tribological feature of the linings in torque transmission) is measured in a pin-on-disc tribometer and mapped into the dynamics model in MSC ADAMS. The pin-on-disc tribometer is able to capture the variation of friction coefficient with contact pressure and sliding velocity. The surface topography is obtained experimentally to examine the consistency of surface properties. The normal pressure and tribology of the contacting components determines the engagement time, clutch capacity and dynamic behaviour of the clutch.

**Key words:** off-road heavy-duty vehicle, clutch disconnect, dynamics, tribology.

## Introduction

Clutch systems have been in service since early 1900's [1]. The clutch disconnects the driver (e.g. IC engine) from the driven load (e.g. the drivetrain or parasitic loads). Clutches exploit the friction characteristics of the contacting surfaces to couple the driven to the driving system [2]. They can be categorised using different criteria. One category is the vehicular clutches for passenger cars, trucks, construction and farm equipment. The most common type is a mechanical clutch with friction lining interfaces. Other clutches operate hydraulically, pneumatically or electromechanically. The friction generated heat is removed either through air-cooling (dry) or oil-cooling (wet) [3].

The main objective of a clutch is the transfer of torque with reliable functionality, regardless of its design category. These objectives are achieved through a combined study of clutch dynamics and tribology

(i.e. tribo-dynamics). Analytical models for clutches have been presented in the literature to investigate their tribodynamics characteristics. Dynamics models typically comprise two degrees of freedom with lumped inertias in their most rudimentary form [4]. The accuracy of the model increases, when including the inertia of all rotating components inside the clutch pack [5]. A clutch model is occasionally integrated with other driveline components such as the gearbox and differential to investigate the effect of clutch dynamics on gear backlash, driveline NVH, drive quality and gear shifting control [6-10]. In these models, the torsional dynamics is the focus of investigations for torque transfer purposes. The effect of translational motion of the plates on the engagement time is usually neglected, except when dealing with noise and vibration issues in clutch engagement process, such as clutch take-up judder [5, 7, 8] and clutch in-cycle vibration [11, 12]. In heavy-duty vehicles, clutches comprise multi-discs, thus the translational motion of the plates influence the engagement initiation time.

Tribology is strongly coupled to dynamics in clutch applications. The driving and driven parts of the dynamics model are interfaced through the friction surfaces. The tribological characteristics of these are governed by the friction lining material and are usually measured experimentally [13]. The standards for assessing the friction properties include SAE standards for wet clutches (J286 and J2490) are detailed in [14, 15], and for dry friction materials (J661) [16] as well as in ASTM standards for friction and wear in dry contacts (G115 and G99) in [15, 16]. Pin-on-disc tribometer is frequently exploited to investigate the variation of friction coefficient with different parameters including contact pressure, velocity and temperature [17]. In the same work, it was shown that pin-on-disc tribometer results agree with those obtained from SAE standard J661. The variation of sliding velocity is considered quasi-statically in these studies. In a later publication [18] a pin-on-disc tribometer was used to study the effect of transient sliding velocity on the coefficient of friction.

Clutch tribodynamics models are inherently complex and non-linear. Addition of other driveline inertial components introduces further complexity, including further sources of non-linearity. The aim of this paper is to develop a simulation tool to investigate the tribodynamics of the new Cat® clutch system. The validated tool can then be used for extended NVH studies, such as those described in [5-8, 11, 12]. A detailed model of the clutch pack is developed, using the main influential parts. The effect of the remainder of the powertrain on clutch dynamics is included using indicative measured excitations. The dynamic model allows for the translational motion of components along the axis of torsional oscillations. Although friction material properties are reported by the manufacturer for wet clutch applications, the contact in the new design is considered as dry, necessitating measurements of the coefficient of friction. A pin-on-disc tribometer is utilised for this purpose, following ASTM instructions. The clutch (dis)engagement dynamics are compared with the experimental

measurements carried out at Perkins Engines. Potential extensions to the tribodynamics model are also discussed.

## Dynamics model

### Multi-body dynamics

The application of *Newton's laws* in large scale systems can be complex. In order to reduce these complexities, the system is modelled with discrete bodies, representing the main contributing components. These bodies are interconnected through joints which replicate the physical constraints of the actual system. This multi-body approach leads to a series of differential-algebraic equations (DAE system) using the Lagrangian dynamics for constrained systems in a series of generalised coordinates  $\{q\}$  [18]:

$$\frac{d}{dt} \left( \frac{\partial L}{\partial \dot{q}} \right) - \frac{\partial L}{\partial q} + \lambda^T \mathbf{C}_q = \mathbf{Q}_{nc}^T \quad (1)$$

$$\mathbf{C}(\mathbf{q}, t) = \mathbf{0}$$

where,  $L$  is the Lagrangian,  $\mathbf{Q}_{nc}$  is the vector of non-conservative generalised forces, such as damping and friction.  $\lambda$  is the vector of Lagrange multipliers,  $\mathbf{C}$  is the vector of constraint functions and  $\mathbf{C}_q$  is the constraint Jacobian matrix:

$$\{q\}_j = \{x, y, z, \psi, \theta, \phi\}_j, \quad j = 1, 2, \dots, n$$

### Clutch dynamics

The clutch components undergo translation and rotation. In order to correctly capture the system dynamics during clutch actuation, a multibody analysis is conducted in MSC ADAMS, which is subsequently validated against experimental measurements by Perkins Engines. The main influential inertial components are divided into driving and driven parts as:

- Flywheel and flywheel ring (1)
- Actuation piston (2)
- Pressure plate (3)
- Friction discs (4)
- Separator plate (5)
- End plate (6)
- Hub and coupler (not indicated)

Figure 1 shows the schematic of the clutch assembly. The flywheel and flywheel ring, actuation piston, pressure plate, separator plate and the end plate form the driving part of the clutch. These undergo translational motion, except for the flywheel, flywheel ring and the end plate. The friction discs, hub and coupler (output shaft) form the driven part of the clutch. The clearance zones are also indicated in Figure 1. The relative motion of the components (translation and/or rotation) is determined by the scalar constraint function,  $\mathbf{C}(\mathbf{q}, t) = \mathbf{0}$  in the model. Three types of joints are exploited in the tribodynamics model: fixed, revolute and translational. The flywheel is connected to the ground (external environment) through a revolute joint, thus allowing only rotation of the flywheel. The end plate is fixed to the flywheel. Other components in the driving part of the clutch are connected to the flywheel through translation joints, allowing for their displacement due to the applied actuation piston clamp load. On the driven side of the clutch model, the coupler (output shaft) is connected to the ground through a revolute joint, removing all the translational freedoms. The hub is fixed to the coupler in the actual system through bolts. The

friction discs are connected to the hub through splines. They undergo translation with respect to the hub. Thus, translational joints are applied between the hub and each friction disc.

The pressure plate (3) and end plate (6) are connected through ten retraction springs (7). The inertia of springs is neglected since its influence is negligible compared with the inertia of the other components. The contraction of the springs is restricted by the maximum clearance between the pressure plate and the end plate. When this clearance is taken-up, the engagement commences provided the clamp load overcomes the springs' retraction load. The clamp load is calculated from the oil pressure acting behind the actuation piston. The oil pressure is measured from the test-rig at the input of the clutch system. 1D CFD simulation is carried out to encapsulate the pressure loss through the pipes up to the inlet orifice behind the actuation piston. The rotating head pressure is calculated using the pressure at the inlet orifice (Equation 2) as:

$$F_{RH} = \rho \omega^2 \pi \left[ \frac{1}{4} R_{P_o}^4 - \frac{1}{2} R_{C_i}^2 (R_{P_o}^2 - R_{P_i}^2) - \frac{1}{4} R_{P_i}^4 \right] \quad (2)$$

where,  $\rho$  and  $\omega$  are the fluid density ( $\text{kg/m}^3$ ) and angular speed ( $\text{r/s}$ ), respectively.  $R_{P_i}$ ,  $R_{P_o}$  and  $R_{C_i}$  are: (i) the distance from the centre of rotation to the inner diameter of the piston chamber, (ii) the distance from the centre of rotation to the outer diameter of the piston chamber and (iii) the distance from the centre of rotation to the location of the inlet orifice. The rotational speed of flywheel builds up with time, leading to higher rotational head pressure acting behind the actuation piston.

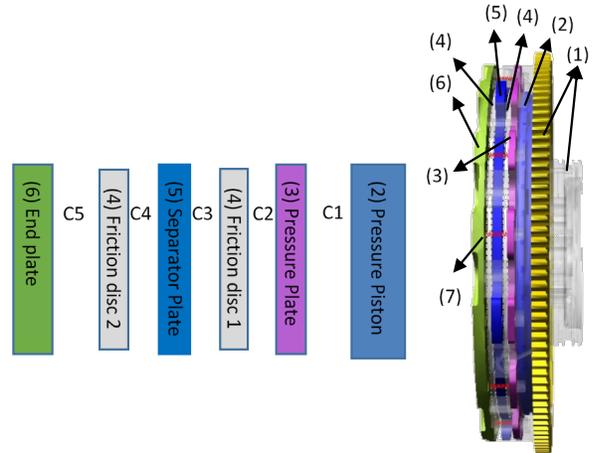


Figure 1. The schematic representation of the clutch system with clearances C1 to C5

The total clamp force on the actuation piston can be deduced by the integration of the pressure distribution acting on the piston area (Equation 3). The clamp load and hydraulic pressure are presented for one engagement-disengagement cycle in Figure 2. The maximum clamp load overcomes the retraction force of the springs at their fully deflected position and the surplus force generates the required normal force  $F_N$  for engagement.

$$F_{piston} = 2\pi \int P(r)rdr \quad (3)$$

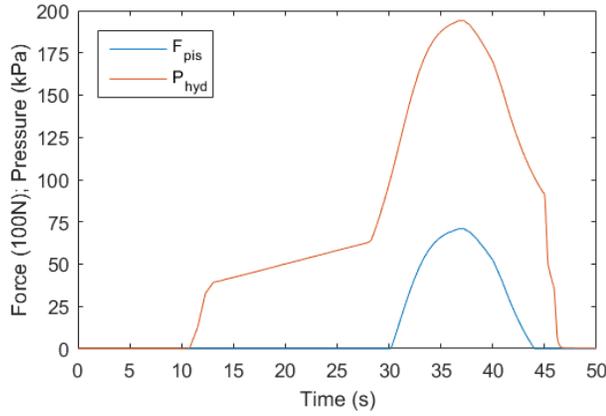


Figure 2. Total clamp force and hydraulic pressure on the actuation piston (CFD results provided by Cat®)

## Tribology of clutch linings

The clutch friction linings are made of different materials. Heavy-duty vehicles use paper-based friction materials in wet clutches. The recommended material is high energy paper, withstanding high flash temperatures with high temperature resistance. The friction coefficient is provided by the manufacturer. Its dynamic value varies between 0.11 and 0.14, when the contact flooded by a suitable lubricant. The static coefficient of friction varies between 0.16 and 0.20. For dry contact conditions, the friction properties of the material must be determined experimentally in order to ensure that the friction coefficient remains well above the recommended value by the manufacturer.

A pin-on-disc tribometer was used to measure the dry friction coefficient in accord with the ASTM G115 and G99 standards. During measurements, the temperature and humidity were recorded for the repeatable measurements (room temperature of  $20 \pm 2$  °C), with room humidity of 50 - 60%.

The measured coefficient of friction provides the characteristics for various sliding velocities and clamp load variation (slip speeds: 0.1 - 12 m/s, and clamp pressure: 31.25 - 574.40 kPa). Lower contact loads are not achievable due to tribometer limitations in stick-slip motions. The preliminary combination of the aforementioned conditions leads to four measurements. The lowest friction coefficient was obtained for 12 m/s and 574.4 kPa ( $\mu = 0.425$ ). This value is significantly greater than the reported value for wet conditions and it is applied to the tribo-dynamic model for simulation studies. The corresponding friction torque  $T_f$  is determined as:

$$T_f = \mu R_{eq} F_N \quad (4)$$

The current clutch design has four friction interfaces. The friction torque for each surface is defined separately due to different contact timing in the presence of translational motion of the plates.  $\mu$  is the friction coefficient and the equivalent radius  $R_{eq}$  is determined using uniform pressure assumption between the contacting surfaces (equation 5), thus [7, 8]:

$$R_{eq} = \frac{2R_o^3 - R_i^3}{3R_o^2 - R_i^2} \quad (5)$$

$R_o$  and  $R_i$  are the outer and inner radii of the friction surfaces, respectively.

## Resistive Torque

The friction torque during engagement should overcome the inertial resistance of the driven components and that of the output shaft. The latter is measured from the powertrain test-rig, mainly varying with angular velocity and contact temperature. In the current study, the dependency on temperature is neglected. Figure 3 depicts the variation of the measured transmission resistive torque with angular speed at the output shaft (circles). A cubic spline is fitted to the data points to interpolate the values during simulation studies (solid line).

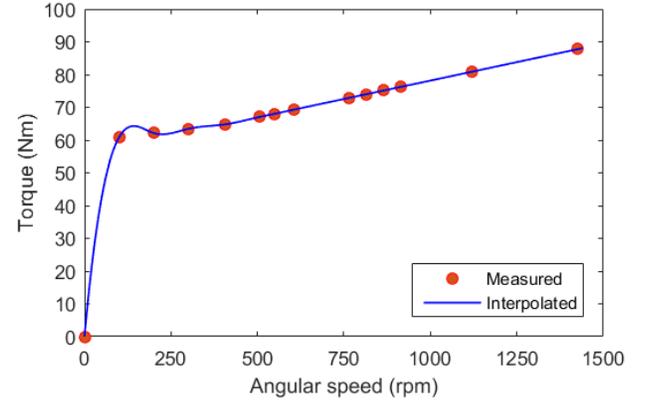


Figure 3. Resistive torque variation with angular speed at the clutch output

## Results

In this section, the dynamics of the clutch model are validated against experimental measurements taken from a test-rig at Perkins Engines. The engine (flywheel) angular velocity is measured for three clutch cycles (engagement and disengagement). The measured velocity is indicated as flywheel speed in Figures 4 and 5. The clutch is designed to disconnect the parasitic loads from the engine at the start-up. The parasitic loads are then connected back to engine, after the engine cranks at idle speed. Thus, the initial value of the engine speed equates zero. Thereafter, the engine speed rises to reach the idle condition. The hydraulic pressure builds up with the engine speed (Figure 2) and pushes the actuation piston towards the pressure plate (Figure 1). The piston force initially overcomes the retraction force of the springs. The separator plate and friction discs are clamped between the pressure plate and the end plate after the initial displacement phase of the pressure plate. A further increase in the clamp load leads to clutch engagement through an increase in the friction torque. At this instant, the clutch output speed rises to the engine idle speed. As the output speed increases, the resistive torque rises on the output shaft. The combined effect of resistive torque and flywheel deceleration causes a reduction in the output speed. As the contact load vanishes between the friction surfaces (cessation of sliding), the clutch output shaft is driven by the resistive torque. This process is shown in Figure 4 for the simulated clutch tribo-dynamics. Figure 5 shows the measured dynamics from the test rig.

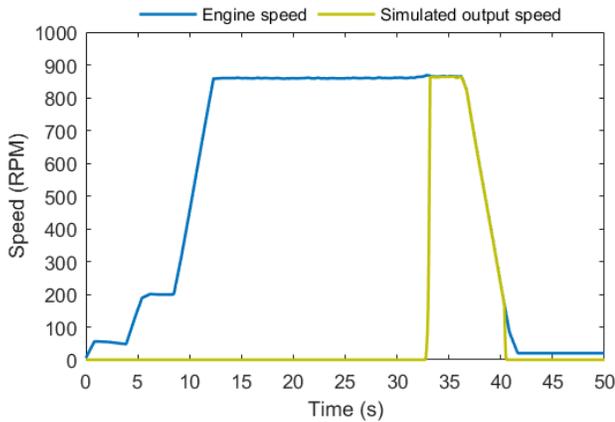


Figure 4. Flywheel speed (measured) and clutch output speed (simulated) time histories

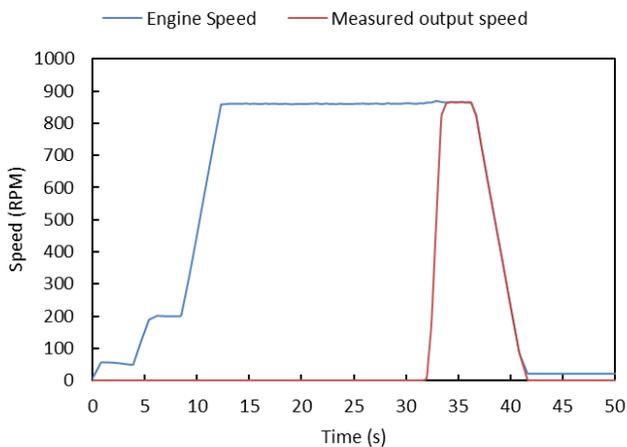


Figure 5. Flywheel and clutch output speed time histories (test rig)

The model solution is obtained through use of GSTIFF solver with integration time step size of  $50\mu\text{s}$ . The engagement initiation and duration are the most important factors in clutch design. The engagement initiates at about 31.9 s in the measured data (Figure 5) and continues for about 2 s. In the simulated results, the engagement initiates at 32.7 s. The clutch output velocity follows a sharper slope during engagement. The engagement completes at about 33.5 s. The comparison between the simulated and measured results indicates that the tribo-dynamics model closely follows the measured physical data. There are two discrepancies which should be addressed. The simulated take-up engagement takes place approximately 1s after in that measured and the same naturally applies to clutch disengagement. More experiments are required to resolve the ambiguities in pressure profile variations behind the piston and to include the translational resistance during the sliding motion of the components, for example by sliding friction of splines. Another deviation is observed in the completion of engagement. The simulated graph completes engagement at about 33.5 s and the output speed quickly increases. The completion phase in the measured result is smoother. The smooth variation in the speed suggests that friction coefficient may be low at the end of the engagement somewhat reduces as the relative sliding velocity decreases and clamp load rises. The tribo-dynamics model exploits a constant coefficient of friction, leading to a quicker completion of engagement.

Overall, the tribo-dynamics model replicates the actual clutch system dynamics. Such a model is useful for extended studies of clutch tribo-dynamics in conjunction with different engine and driveline conditions.

During engagement, torsional vibrations are expected to be transmitted to the driveline due to differences in the angular velocities of the clutch input and output. These vibrations are known as judder and appear in the frequency range 5-20 Hz [7, 8]. As the inertia of the vehicle reduces the judder frequency increases. In the dynamics of the current clutch design, take-up judder does not occur either in the experimental measurements or in the simulation results. Judder is influenced by the frictional properties of the friction linings and the driveline inertia. The large inertias in heavy-duty vehicles inhibit the propensity to judder. Centea [7] showed that a negative slope in friction coefficient variations with sliding velocity causes self-excited judder vibrations. As explained earlier, the abrupt increase in the clutch output velocity is related to high friction coefficient at high sliding velocity. As the sliding velocity decreases and contact pressure increases, the friction coefficient decreases accordingly. This behaviour demonstrates the positive slope of coefficient of friction variation with interfacial slip, thus the lack of take-up judder oscillations.

## Conclusion

Caterpillar Inc. has designed a new clutch system to downsize engines for their heavy-duty vehicles. A multi-body tribo-dynamic model replicating the actual clutch system is presented. Such a model can be used to study the performance of the clutch with different engine types and operating conditions. All the clutch inertial components are included in the multi-body analysis. Translational dynamics of the system is also considered for the engagement behaviour. A standard method is exploited to determine the minimum friction coefficient for the interfacial surfaces, whilst the contact is dry. The measured flywheel speed and oil pressure behind the actuation piston are applied as inputs to the clutch system. The model considers the effect of resistive torque at the clutch output. It is shown that the developed tribo-dynamic model successfully replicates the experimentally measured dynamic response using a test rig. Areas for potential modifications in the clutch model have been identified, aiming to improve the accuracy of the numerical results with respect to the measured experimental data.

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