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Modeling the clutch energy and clutch life of a heavy duty vehicle

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Abstract. Clutch energy is the thermal energy dissipated on the clutch disc, and it reaches its highest level during drive-off as a result of the difference between the angular speeds of the flywheel and clutch disc, and the torque transmitted. The thermal energy dissipated effects the clutch life. This study presents a new drive-off and thermal model to calculate the clutch energy for a rear wheel driven heavy-duty vehicle and to analyze the effects of clutch energy on temperatures of clutch pressure plate, flywheel and clutch housing. Three different driver profiles are used, based on the release of the clutch pedal in modulation zone: i) the pedal travels with the same speed all the way, ii) the travel speed of the pedal increases, iii) the travel speed of the pedal decreases. Vehicle test is performed to check the accuracy of the model. When compared to a simpler model that is widely used in the literature to calculate the clutch energy, the model used in this study calculates the clutch energy and angular speed behaviors of flywheel and transmission input shaft in better agreement with experimental results. Clutch wear and total clutch life are also estimated using the mean specific friction power.

Keywords: clutch energy; clutch life; slip time; clutch thermal model; heat distribution in clutch housing

1. Introduction

The life of clutch depends on the friction energy dissipated on clutch disc during the synchronization of angular speeds of flywheel and clutch disc. Maximum dissipated energy occurs during drive-off (when driver's misuse is neglected) because of high resistance torque and the maximum angular speed difference between engine flywheel and clutch disc. The factors affecting clutch energy during drive-off are clutch disc friction coefficient, engine torque, transmission and differential ratios, vehicle acceleration, tire radius, weight of the vehicle, the grade and road surface condition.

In literature, there is a simple model (Duque *et al*. 2009) used that simulates the clutch between engine inertia and reduced vehicle inertia; the model makes the following assumptions:

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• Engine torque is equal to maximum engine torque and is constant during slip time.

•Angular speed of the flywheel is equal to maximum engine speed and constant during slip time.

•Angular acceleration of the transmission input shaft is constant.

In this linear model, the clutch energy is calculated using the following Eq. (1)

$$Q = \frac{\dot{\theta}_E \cdot t_{slip} \cdot T_{e,\max}}{2} \tag{1}$$

where Q is the clutch energy, $\dot{\theta}_E$ is the maximum engine speed, t_{slip} is the slip time, $T_{e,max}$ is the maximum engine torque.

A model of engine and driveline of a passenger vehicle was developed by Duque *et al.* (2009) to calculate the heat generation on clutch friction disc during vehicle drive-off. The results were compared to the experimental results obtained on a flat road.

Coelho and Rabelo in (2011) presented effective parameters on clutch life of a heavy duty vehicle. The effect of mass of pressure plate on plate temperature after series of vehicle severe drive-offs was studied. The results proved the linkage between the mass of pressure plate and thermal absorption under the same energy level. The effect of clutch power on clutch life, which was derived from experimental study, was also presented.

For a heavy duty vehicle with automated transmission, a model was developed by Myklebust and Ericson in (2013) to analyze the transmitted torque under slipping condition. A real time simulation that could be applied to a vehicle with production sensors was developed and improved with a wear parameter.

A linear thermal model for clutch and clutch housing was developed and validated at test bench by Velardocchia *et al.* (2000). The different experimental parameters were developed for different tests such as gear shifting frequency, clutch housing temperature, clutch opened or closed. A finite element model of the clutch was developed by Sun *et al.* (2013) to calculate the thermal load under frequent drive-off conditions. The speed difference, overloading of clutch, heat capacity and conductivity, cooling structure were described as the main parameters to improve the clutch life.

The slipping clutch thermal model was developed to calculate friction coefficient and heat-transfer coefficient of dry clutch by Xiang and Kremer in (2001). A continuous temperature evaluation was also derived from the model for different slipping conditions.

Starting with the model of Duque *et al.* (2009) this work presents a detailed model to calculate the energy dissipated on the clutch disc of a heavy duty vehicle during drive-off. The results are then compared to the results of the simple linear model described above and those of the road tests. But instead of using the data taken from driver recorded throttle position, like in the case of Duque *et al.* (2009) for clutch pedal position, three different driver profiles based on the release of the clutch pedal and Wide Open Throttle (WOT) engine properties are used.

The model of the powertrain used in the present study is shown in Fig. 2. Using this model and following an iterative procedure, the slip time and angular speed of transmission input shaft are calculated. First, the slip time to the synchronization of angular speeds of engine and transmission input shaft is assumed and the angular speed of the transmission input shaft is calculated. Afterwards, the slip time is calculated using both angular speeds and the result is compared to the initial assumption; if the difference between the initial assumption and the calculated value is larger than 1% of the calculated value, iterations continue until the error is less than 1%. Once the slip time and angular speeds are determined, the clutch energy, specific dissipated energy, and mean specific friction power are calculated using the drive-off model.

Bench tests are performed to obtain overall heat transfer coefficients. Overall heat transfer coefficient includes all conductive and convective effects. A thermal model similar to one developed by Myklebust *et al.* (2015) is used to calculate the temperatures of clutch pressure plate and flywheel. The model of Myklebust *et al.* (2015) considers the heat transfer from engine coolant to flywheel and employs a single total mass for pressure plate and flywheel. However, the model developed in this paper neglects the heat transfer from engine coolant for simplicity and employs two separate masses for flywheel and pressure plate. The pressure plate temperature is the key parameter on friction coefficient of the clutch disc. As a result, both drive-off and thermal models run simultaneously and the method of Coelho *et al.* (2011) is used to calculate the clutch life.

An experiment that consists of 100 cycles of standing start at Ford Tatui Proving Ground is performed to check the accuracy of the results of the present model. The speed, wear rate and housing temperature data obtained from the tests are compared to the results obtained from calculations.

2. Modeling

2.1 Drive-off model

The driveline of the vehicle is shown in Fig. 1.

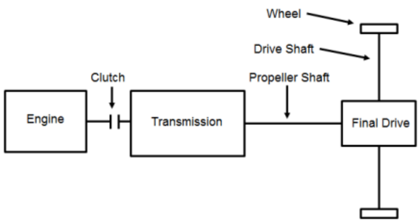


Fig. 1 Heavy duty vehicle driveline

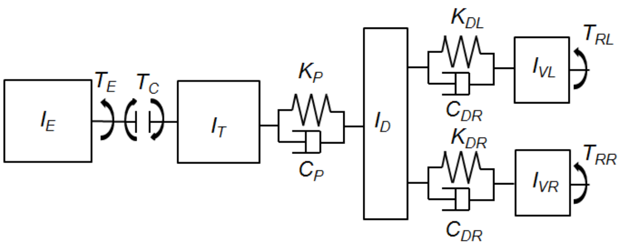


Fig. 2 Model of the driveline

To account for the effect of each driveline component on the torque transmitted through the clutch during drive-off and on change of angular velocity of transmission input shaft, the following model is developed; the model consists of engine, clutch, transmission, propeller shaft, differential, drive shafts, and is shown in Fig. 2.

Symmetry between right and left drive shafts is assumed, and all inertia, stiffness and damping of propeller shaft and drive shafts are reduced to the transmission input shaft. A simplified representation of the driveline model is redrawn in Fig. 3 where all torques and resistances as well are reduced to transmission input shaft.

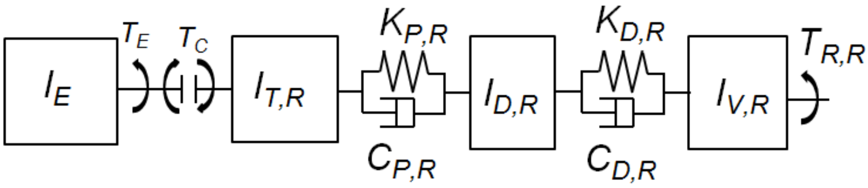


Fig. 3 Simplified model of the driveline

2.1.1 Engine model

The free body diagram of the proposed engine model is shown in Fig. 4 where T_E is engine torque, I_E is engine inertia, and T_C is clutch torque.

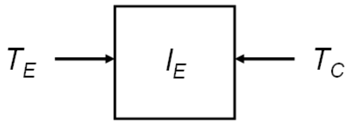


Fig. 4 Torque diagram of proposed engine model

The engine torque corresponds to WOT. Therefore, the engine speed depends on engine torque at WOT, engine inertia and clutch torque. The inertia of engine reduced to crankshaft axis is calculated from sum of inertia of all engine components and clutch cover assembly.

2.1.2 Clutch and clutch release system model

Clutch pedal movement during vehicle drive-off is different than clutch pedal travel during gear shifting. When gear is shifted, the driver pushes the clutch pedal all the way down and shifts the gear when pedal is at this position, and then leaves the pedal. However, the driver actuates clutch pedal from pressed position as theoretically (Dolcini *et al*. 2010) shown in Fig. 5 during drive-off. At the beginning of region *C*, a drop in engine speed starts and the modulation zone finishes at the end of region *D*. Therefore, regions *C* and *D* are modelled in the present model. The displacement of the pressure plate is assumed to change linearly with the displacement of the diaphragm spring during vehicle drive-off.

Three different driver profiles are used based on the release of the clutch pedal in regions *C* and *D*: i) the pedal travels with the same speed all the way, ii) the travel speed of the pedal increases, iii) the travel speed of the pedal decreases.

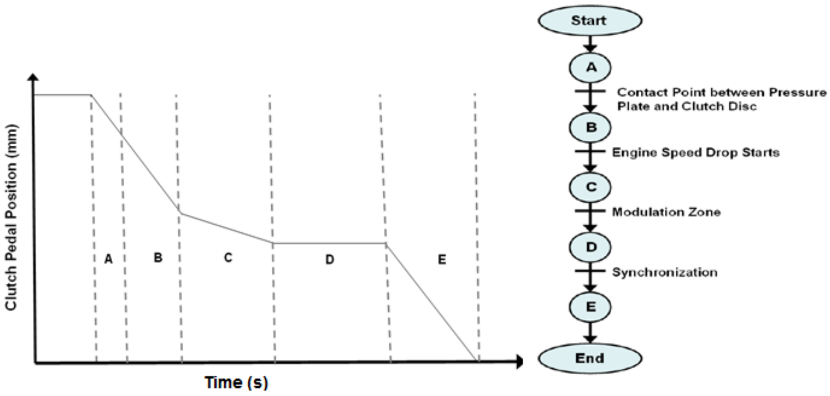


Fig. 5 Theoretical clutch pedal position during drive-off

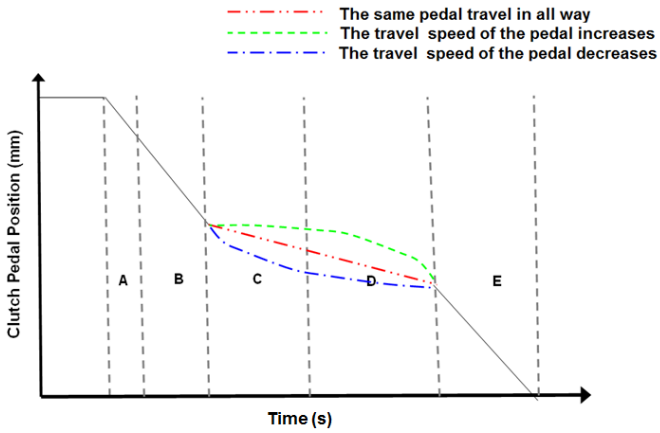


Fig. 6 Driver profiles for clutch pedal travel during slip time

The movement of the pressure plate is calculated for three different profiles, and the result is used in the following Eq. (8) to obtain the torque transmitted

$$T_C = F_{cushion} \cdot N \cdot \mu \cdot R_{mean} \tag{2}$$

where T_C is the torque transmitted through the clutch, $F_{cushion}$ is the cushion force on the clutch disc, μ is the coefficient of friction of organic facing material and is strictly dependent on temperature after a specific value, $N = 2$, and is the number of friction facing for single clutch disc, R_{mean} is the mean radius over which the clamp force actuates, and is calculated from the following Eq. (8)

$$R_{mean} = \int_{R_i}^{R_o} \int_0^{2\pi} rdAdr = \frac{2(R_o{}^3 - R^3{}_i)}{2(R_o{}^2 - R^2{}_i)} \tag{3}$$

Because the torque transmitted is also a function of slip time, to calculate the torque, an initial value for slip time is assumed and an iterative procedure is followed using the following equation for the new slip time at each iteration

$$t' = \frac{t_{assumed} - t_{calculated}}{2} \tag{4}$$

Once a slip time with an acceptable error (less than one percent) is obtained, the transmitted torque and speeds are calculated successively.

2.1.3 Propeller shaft model

The propeller shaft connects the output shaft of the transmission to the final drive. The propeller shaft of the vehicle is hollow with an internal diameter. The stiffness, K_P, of the propeller shaft is calculated with a length dependent stiffness formula (1) as shown below

$$K_P = \frac{I_P \cdot G}{L_P} \tag{5}$$

where I_P is the area moment of inertia of the cross section, L_P is the length and G is the modulus of rigidity. $K_{P,R}$ and $C_{P,R}$ which are the stiffness and the damping of the propeller shaft reduced to transmission input shaft respectively are calculated from the following formulas (Duque *et al*. 2009)

$$K_{P,R} = \frac{K_P}{i_t^{\,2}}, C_{P,R} = \frac{C_P}{i_t^{\,2}} \tag{6}$$

where i_t is the transmission ratio.

2.1.4 Transmission model

The transmission is a manual transmission, and the free body diagram is shown below.

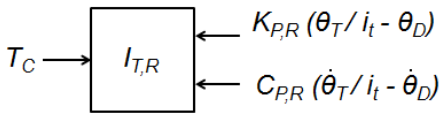


Fig. 7 Free body representation of transmission model

where i_f is the final drive ratio; T_c is the clutch torque and $K_{P,R} \cdot (\theta_T/i_t - \theta_D)$and $C_{P,R} \cdot (\dot{\theta}_T/i_t - \dot{\theta}_D)$ are stiffness and damping torques of the propeller shaft. From moment balance, the angular acceleration of transmission, $\ddot{\theta}_T$, can be found as shown below

$$\ddot{\theta}_T = \frac{T_C - K_{P,R} \cdot (\frac{\theta_T}{i_t} - \theta_D) - C_{P,R} \cdot (\frac{\dot{\theta}_T}{i_t} - \dot{\theta}_D)}{I_{T,R}} \tag{7}$$

where $I_{T,R}$ is the inertia of the transmission reduced to transmission input shaft and “*D*” stands for the final drive side of drive shafts.

2.1.5 Drive shafts model

The drive shafts transmit the motion to the drive wheels. It is assumed that both wheels have the same speed during drive-off and hence the two drive shafts are modelled as two springs that act in parallel with stiffnesses calculated using the length dependent stiffness formula (Duque *et al*. 2009) as shown below: The stiffnesses of the right and left drive shafts are

$$K_{DR} = \frac{I_D \cdot G}{L_{DR}}, K_{DL} = \frac{I_D \cdot G}{L_{DL}} \tag{8}$$

where L_{DR} and L_{DL} are the lengths of the drive shafts.

When the drive shafts are considered as two springs that act in parallel, the vehicle is assumed to be in straight line and the same angular displacement is assumed for both shafts. The total drive shaft stiffness is calculated from sum of the stiffness values of right and left drive shafts.

$$K_D = K_{DR} + K_{DL} \tag{9}$$

$K_{D,R}$ and $C_{D,R}$ which are the stiffness and the damping of the drive shafts reduced to transmission input shaft respectively are calculated from the following formulas (Duque *et al*. 2009)

$$K_{D,R} = \frac{K_D}{(i_t \cdot i_f)^2}, C_{D,R} = \frac{C_D}{(i_t \cdot i_f)^2} \tag{10}$$

2.1.6 Final drive model

The free body diagram of final drive is shown below.

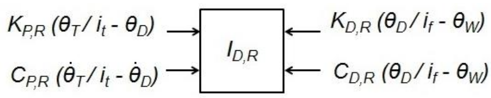


Fig. 8 Free body representation of final drive

where $K_{D,R} \cdot (\theta_D/i_f - \theta_W)$ and $C_{D,R} \cdot (\dot{\theta}_D/i_f - \dot{\theta}_W)$ are stiffness and damping torques of the drive shafts. From moment balance, $\ddot{\theta}_D$, angular acceleration of transmission input shaft, could be found

$$\ddot{\theta}_D = \frac{K_{P,R} \cdot (\frac{\theta_T}{i_t} - \theta_D) + C_{P,R} \cdot (\frac{\dot{\theta}_T}{i_t} - \dot{\theta}_D) - K_{D,R} \cdot (\frac{\theta_D}{i_f} - \theta_W) - C_{D,R} \cdot (\frac{\dot{\theta}_D}{i_f} - \dot{\theta}_W)}{I_{D,R}} \tag{11}$$

2.1.7 Vehicle model

The free body diagram of the vehicle reduced to transmission input shaft is shown below.

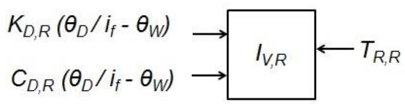


Fig. 9 Free body representation of vehicle model

where $I_{V,R}$ is vehicle reduced inertia and $T_{R,R}$ is resistance torque reduced to the transmission input shaft. The vehicle inertia, I_V is calculated from the following equation

$$I_V = I_{W,tot} + m \cdot r_{dyn}{}^2 \tag{12}$$

There are two tires for each end of drive shaft and each tire has mass moment of inertia, I_W. The total mass moment of inertia of wheels at the end of two drive shafts is sum of all inertia of tires.

Resistive forces are:

- Aerodynamic resistance F_{aero}
- Rolling resistance F_{roll}
- Gradient resistance $F_{gradient}$

The air resistance is made up of the pressure drag including induced drag (turbulence induced by differences in pressure) and surface resistance. The following formula is used to calculate aerodynamic drag (Gillespie 1992)

$$F_{aero} = \frac{C_D \cdot \rho \cdot A_f \cdot v^2}{2} \tag{13}$$

Neglecting slip, the velocity of the vehicle is expressed as follows

$$v = \dot{\theta}_W \cdot r_{dyn} \tag{14}$$

The dynamic radius of the tire is calculated from the formula (Gillespie 1992) where dimensionless constant K is 1.00 for heavy commercial tires. The tire with designation of 295/80 R 22.5 is used for modelled vehicle.

$$r_{dyn} = K \cdot (0.5 \cdot NRD + \phi \cdot NSW) \tag{15}$$

Air density (Gillespie 1992) is function of ambient pressure, P_{amb} and temperature, T_{amb}

$$\rho = 1.225 \cdot \frac{P_{amb}}{101.325} \cdot \frac{288.16}{(273.16 + T_{amb})} \tag{16}$$

Rolling resistance force is given below

$$F_{roll} = f_r \cdot W_{wheel} \tag{17}$$

where f_r is rolling resistance coefficient, and W_{wheel} is total weight on the wheel.

The following formula (Gillespie 1992) is used to estimate the rolling resistance coefficient of a radial heavy truck tire where v is speed in m/s and C_h is road surface coefficient (1.0 for smooth, dry asphalt surface).

$$f_r = (0.0041 + 0.000041 \cdot v) \cdot C_h \tag{18}$$

The gradient resistance force relates to the slope descending force. After calculating each resistance force, total resistance force is calculated as sum of all resistances (Sun *et al*. 2013)

$$F_{Rtot} = F_{aero} + F_{roll} + F_{gradient} \tag{19}$$

The total resistive force is calculated as the sum of all resistances (Sun *et al*. 2013)

$$T_R = r_{dyn} \cdot F_{Rtot} \tag{20}$$

From moment balance, $\ddot{\theta}_W$, the angular acceleration of vehicle could be calculated as follows

$$\ddot{\theta}_W = \frac{K_{D,R} \cdot (\frac{\theta_D}{i_f} - \theta_W) + C_{D,R} \cdot (\frac{\dot{\theta}_D}{i_f} - \dot{\theta}_W) - T_{R,R}}{I_{V,R}} \tag{21}$$

2.2 Thermal model

Isolated from the internal combustion engine and the driveline, the temperature diagram of the clutch pressure plate, the flywheel and the clutch housing can be drawn as in the figure below.

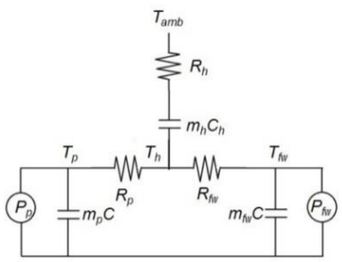


Fig. 10 Temperature diagram

The thermal resistance of the pressure plate is R_p, the thermal resistance of the flywheel is R_{fw} and the thermal resistance of housing is R_h. All of these resistances are calculated by matching experimentally measured cooling temperature profile with the one calculated using the cooling model. Results of an example calculation and an experimentally measured temperature of the pressure plate are shown in Fig. 11. The test spans the clutch engaged to the flywheel and rotating at constant 1500 rpm; clutch rotating at constant idle angular speed and clutch disengaged etc. Coefficient values for intermediate rpms are interpolated from experimental data.

Following the temperature diagram in Fig. 10, the model equations are as follows

$$m_p \cdot C \cdot \dot{T}_p = P_p + R_p \cdot (T_h - T_p) \tag{22}$$

$$m_{fw} \cdot C \cdot \dot{T}_{fw} = P_{fw} + R_{fw} \cdot (T_h - T_{fw}) \tag{23}$$

$$m_h \cdot C_h \cdot \dot{T}_h = R_p \cdot (T_h - T_p) + R_{fw} \cdot (T_h - T_{fw}) - R_h \cdot (T_h - T_{amb}) \tag{24}$$

where m_p is the mass of the clutch pressure plate, m_{fw} is the mass of the flywheel, m_h is the mass of the clutch housing, C is the thermal coefficient of the clutch and pressure plate, T_p is the temperature of the clutch pressure plate, T_{fw} is the temperature of the flywheel, T_h is the temperature of the clutch housing.

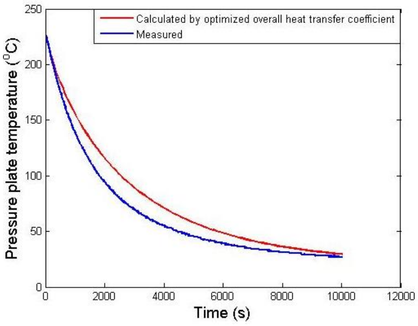


Fig. 11 Comparison of calculated by cooling model and experimentally measured pressure plate temperature

3. Calculation

3.1 Clutch energy

Once Eqs. (7), (11) and (21) are simplified, three main equations including all previous equations are obtained

$$\ddot{\theta}_T = \frac{T_C}{I_{T,R}} - \frac{K_{P,R}}{I_{T,R}\cdot i_t}\cdot\theta_T - \frac{C_{P,R}}{I_{T,R}\cdot i_t}\cdot\dot{\theta}_T + \frac{K_{P,R}}{I_{T,R}}\cdot\theta_D + \frac{C_{P,R}}{I_{T,R}}\cdot\dot{\theta}_D \tag{25}$$

$$\ddot{\theta}_D = \frac{K_{P,R}}{I_{D,R}\cdot i_t}\cdot\theta_T + \frac{C_{P,R}}{I_{D,R}\cdot i_t}\cdot\dot{\theta}_T - \frac{(K_{P,R}+\frac{K_{D,R}}{i_f})}{I_{D,R}}\cdot\theta_D - \frac{(C_{P,R}+\frac{C_{D,R}}{i_f})}{I_{D,R}}\cdot\dot{\theta}_D + \frac{K_{D,R}}{I_{D,R}}\cdot\theta_W + \frac{C_{D,R}}{I_{D,R}}\cdot\dot{\theta}_W \tag{26}$$

$$\ddot{\theta}_W = \frac{K_{D,R}}{I_{V,R}\cdot i_f}\cdot\theta_D + \frac{C_{D,R}}{I_{V,R}\cdot i_f}\cdot\dot{\theta}_D - \frac{K_{D,R}}{I_{V,R}}\cdot\theta_W - \frac{C_{D,R}}{I_{V,R}}\cdot\dot{\theta}_W - \frac{T_{R,R}}{I_{V,R}} \tag{27}$$

Once equations are written in the state-space form, the following equation consisting of first and second order differential equations is obtained.

$$\begin{bmatrix}\dot{\theta}_T\\ \ddot{\theta}_T\\ \dot{\theta}_D\\ \ddot{\theta}_D\\ \dot{\theta}_W\\ \ddot{\theta}_D\end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0\\ -\frac{K_{P,R}}{I_{T,R}\cdot i_t} & -\frac{C_{P,R}}{I_{T,R}\cdot i_t} & \frac{K_{P,R}}{I_{T,R}} & \frac{C_{P,R}}{I_{T,R}} & 0 & 0\\ 0 & 0 & 0 & 1 & 0 & 0\\ \frac{K_{P,R}}{I_{D,R}\cdot i_t} & \frac{C_{P,R}}{I_{D,R}\cdot i_t} & -\frac{(K_{P,R}+\frac{K_{D,R}}{i_f})}{I_{D,R}} & -\frac{(C_{P,R}+\frac{C_{D,R}}{i_f})}{I_{D,R}} & \frac{K_{D,R}}{I_{D,R}} & \frac{C_{D,R}}{I_{D,R}}\\ 0 & 0 & 0 & 0 & 0 & 1\\ 0 & 0 & \frac{K_{D,R}}{I_{V,R}\cdot i_f} & \frac{C_{D,R}}{I_{V,R}\cdot i_f} & -\frac{K_{D,R}}{I_{V,R}} & -\frac{C_{D,R}}{I_{V,R}} \end{bmatrix} \begin{bmatrix}\theta_T\\ \dot{\theta}_T\\ \theta_D\\ \dot{\theta}_D\\ \theta_W\\ \dot{\theta}_W\end{bmatrix} + \begin{bmatrix} 0 & 0\\ \frac{T_C}{I_{T,R}} & 0\\ 0 & 0\\ 0 & 0\\ 0 & 0\\ 0 & -\frac{T_{R,R}}{I_{V,R}}\end{bmatrix} \tag{28}$$

To solve the equations, initial conditions are required. The angular speeds of transmission input shaft and wheels are zero. Therefore, initial conditions of vector x is chosen as follows

$$x = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \tag{29}$$

3.2 Temperatures

Once Eqs. (22), (23) and (24) are simplified, three main equations including all previous equations are obtained

$$\dot{T}_p = -\frac{R_p}{m_p \cdot C} \cdot T_p + \frac{R_p}{m_p \cdot C} \cdot T_h + \frac{P_p}{m_p \cdot C} \tag{30}$$

$$\dot{T}_{fw} = -\frac{R_{fw}}{m_{fw} \cdot C} \cdot T_{fw} + \frac{R_{fw}}{m_{fw} \cdot C} \cdot T_h + \frac{P_{fw}}{m_{fw} \cdot C} \tag{31}$$

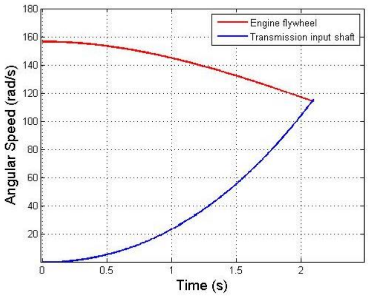
$$\dot{T}_h = -\frac{R_p}{m_h \cdot C_h} \cdot T_p - \frac{R_{fw}}{m_h \cdot C_h} \cdot T_{fw} + \frac{(R_{fw} + R_p - R_h)}{m_h \cdot C_h} \cdot T_h + \frac{R_h \cdot T_{amb}}{m_h \cdot C_h} \tag{32}$$

In the state-space form, they can be rewritten as

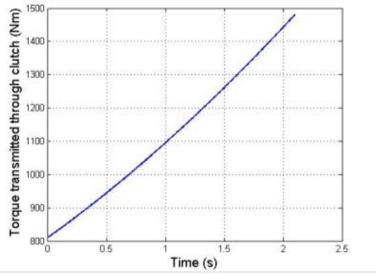
$$\begin{bmatrix} \dot{T}_p \\ \dot{T}_{fw} \\ \dot{T}_h \end{bmatrix} = \begin{bmatrix} -\frac{R_p}{m_p \cdot C} & 0 & \frac{P_p}{m_p \cdot C} \\ 0 & -\frac{R_{fw}}{m_{fw} \cdot C} & \frac{R_{fw}}{m_{fw} \cdot C} \\ -\frac{R_p}{m_h \cdot C_h} & -\frac{R_{fw}}{m_h \cdot C_h} \cdot T_{fw} & \frac{(R_{fw} + R_p - R_h)}{m_h \cdot C_h} \cdot T_h \end{bmatrix} \begin{bmatrix} T_p \\ T_{fw} \\ T_h \end{bmatrix} + \begin{bmatrix} \frac{P_{fw}}{m_{fw} \cdot C} \\ \frac{P_{fw}}{m_{fw} \cdot C} \\ \frac{R_h \cdot T_{amb}}{m_h \cdot C_h} \end{bmatrix} \tag{33}$$

The initial conditions of vector t_0 are

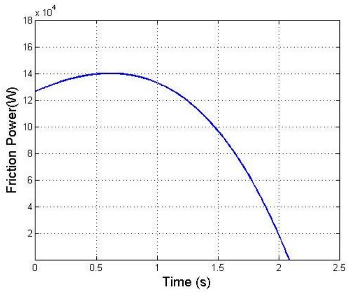
$$t_0 = \begin{bmatrix} 101 \\ 90 \\ 80 \end{bmatrix} \tag{34}$$



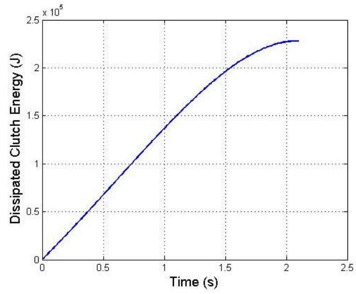
(a) The angular velocities of the engine flywheel and the transmission input shaft



(b) The torque transmitted through the clutch



(c) The friction power



(d) Dissipated energy on clutch disc

Fig. 12 The results of present model when the clutch pedal travels with the same speed all the way in modulation zone during drive-off

4. Results

4.1 Results of clutch energy

4.1.1 Results of linear model

The torque used is 1900 Nm (constant) and is equal to the maximum engine torque; the corresponding engine speed is equal to the maximum engine speed. In the linear model, it is assumed that the angular speed of the transmission input shaft increases linearly until reaching the maximum engine speed. The clutch energy is calculated from Eq. (1). The slip time and the specific energy, *e*, per unit area of clutch disc, are 1.07 s and 113.9 J/cm^2 respectively. The specific energy is calculated from the following

$$e = \frac{Q}{\pi \cdot (R_o^{\ 2} - R_i^{\ 2}) \cdot N} \tag{35}$$

where R_o and R_i are outer and inner diameters of the friction surface of the clutch disc.

4.1.2 Results of present model

Three different driver profiles are used based on the release of the clutch pedal in *C* and *D* regions of pedal travel as shown in Fig. 6: i) the pedal travels with the same speed all the way, ii) the travel speed of the pedal increases, iii) the travel speed of the pedal decreases.

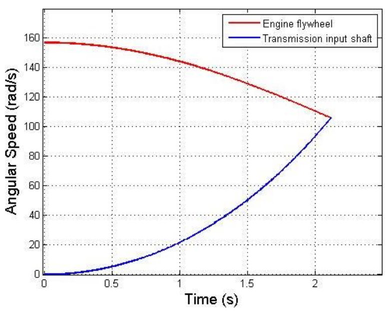
When the Pedal Travels with the Same Speed All the Way

The results of present model are shown in Figs. 12(a)-(d). Once the torque is determined, the change of friction power which is equal to multiplication of angular speed difference between the engine flywheel and the transmission input shaft and torque transmitted through the clutch during drive-off is calculated. The friction power drops to zero by the end of the synchronization time.

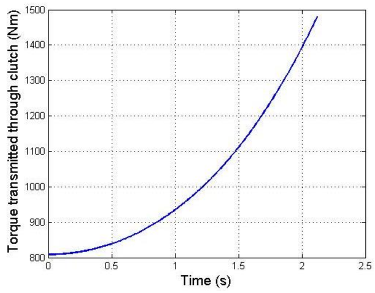
The dissipated energy on the clutch disc is equal to the area under the friction power curve up to the synchronization time, t_s = 2.10 s. The change of dissipated energy is plotted in Fig. 12(d). Using the maximum value, the specific energy is calculated to be 113.9 J/cm^2.

When the Travel Speed of the Pedal Increases

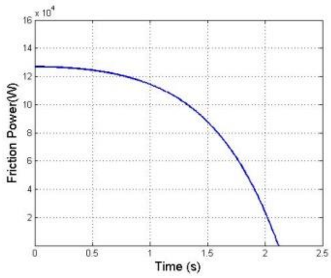
The results of present model for second driver profile are shown in Figs. 13(a)-(d). Once second driver profile of the present model is used, the specific energy is calculated as 103.3 J/cm^2.



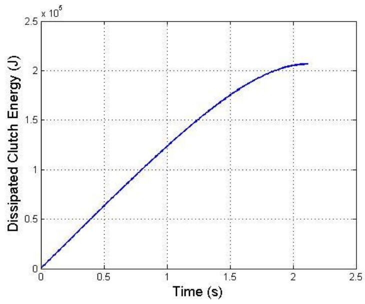
(a) The angular velocities of the engine flywheel and the transmission input shaft



(b) The torque transmitted through the clutch



(c) The friction power



(d) Dissipated energy on clutch disc

Fig. 13 The results of present model when the travel speed of the clutch pedal increases in modulation zone during drive-off

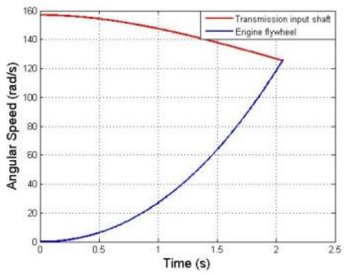
When the Travel Speed of the Pedal Decreases

The results of present model for third driver profile are shown in Figs. 14(a)-(d). Once second driver profile of the present model is used, the specific energy is calculated as 126.0 J/cm^2. The results in are shown in Figs. 14(a)-(d).

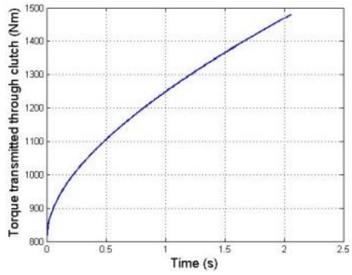
4.2 Results of thermal model

The clutch pressure plate, the flywheel and the clutch housing temperatures are calculated by the thermal model. Two cycles with different cooling times (15 s and 60 s) are explained in detail.

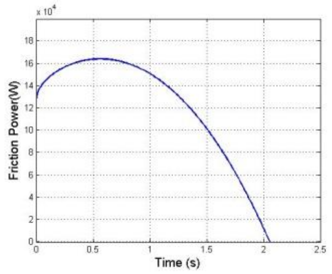
Each of them is heavy duty vehicle (fully laden) which is driven on a 12% grade ramp in first gear and contains 10 drive-offs. The drive-off model with the same speed all the way in modulation zone is used. The effect of mean friction power on the total clutch life (Coelho and Rabelo 2011) is shown in Fig. 15.



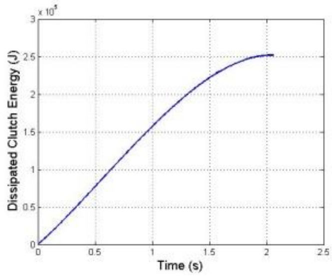
(a) The angular velocities of the engine flywheel and the transmission input shaft



(b) The torque transmitted through the clutch



(c) The friction power



(d) Dissipated energy on clutch disc

Fig. 14 The results of present model when the travel speed of the clutch pedal decreases in modulation zone during drive-off

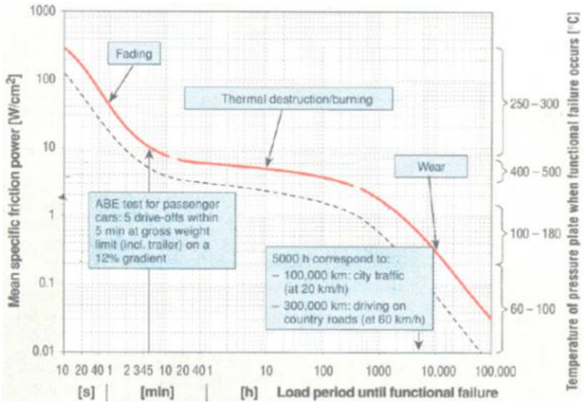
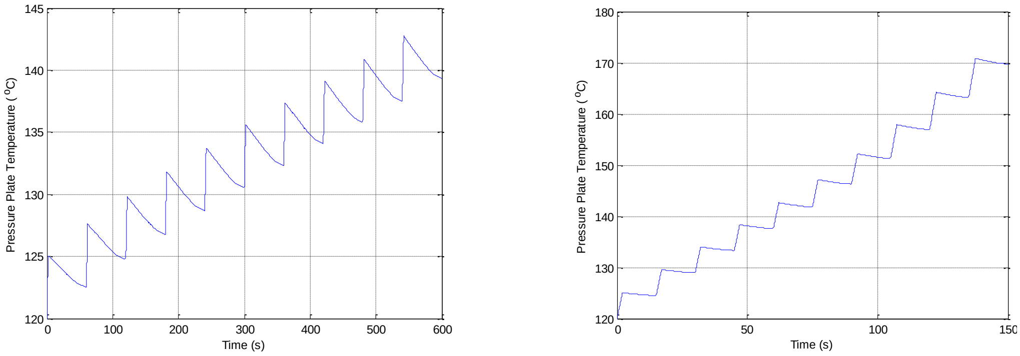


Fig. 15 The effect of mean friction power on the total clutch life (Coelho and Rabelo 2011)

The pressure plate temperature increases from 120°C to 139.3°C after 10 drive-offs when cooling time of 60 s as shown in Fig. 16(a). Clutch life of 53.9 hours for a clutch disc with 1.5 mm of rivet head on both sides occurs by using the data in Fig. 15 and this corresponds to wear of

927.3 mm^3 for a clutch disc with 1.5 mm of rivet head on both sides assuming the pressure plate cools down to 120°C after every 10 drive-offs.

The pressure plate temperature increases from 120°C to 169.8°C after 10 drive-offs when cooling time of 15 s as shown in Fig. 16(b). Clutch life of 859.5 s for a clutch disc with 1.5 mm of rivet head on both sides occurs by using the data in Fig. 15 and this corresponds to 52354.9 mm^3 of wear for a clutch disc with 1.5 mm of rivet head on both sides assuming the pressure plate cools down to 120°C after every 10 drive-offs.



(a) When cooling time of 60 s

(b) When cooling time of 15 s

Fig. 16 Temperature of pressure plate during drive-off when the pedal travels with the same speed all the way in modulation zone

The clutch housing temperature is calculated by present model when the travel speed of the pedal increases in modulation zone and shown in Fig. 17 to compare the final temperature with test data. Initial temperature is taken 80°C and cooling time is 60 s.

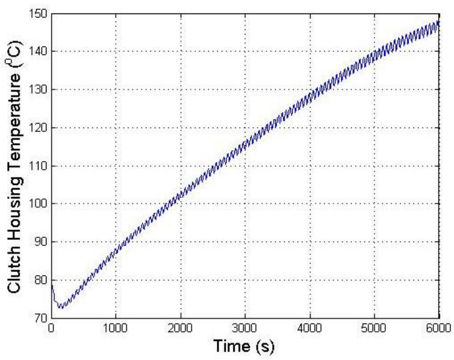


Fig. 17 Clutch housing temperature of present model

4.3 Experimental results

The heavy duty vehicle modelled here (fully laden) is driven on a 12% grade ramp in first gear and the same driver is used in all tests to minimize external effects as shown in Fig. 18. Angular speeds of engine flywheel and transmission input shaft are measured with 60 s of intervals (100 times). Each time the clutch housing temperature is also measured with thermocouples. The temperature of the clutch housing is checked during the test because friction coefficient dramatically drops with increasing temperature. The measured angular speed difference between the flywheel and the transmission input shaft and the torque read through ECU are used to calculate actual clutch energy during drive-off and to compare the results of the model with actual clutch energy and validate the model.

Fig. 18 A sample vehicle at the same experimental test

The change in angular velocities of the engine flywheel and the transmission input shaft obtained at first measurement is shown in Fig. 19. Resistance torque is equal to the gradient resistance torque at the beginning of slip time and increases with the contribution aerodynamic and rolling resistance torque as a function of angular velocity and acceleration. After obtaining the transmitted torque through the clutch and angular speed difference, the specific clutch energy is calculated as 107.0 J/cm^2.

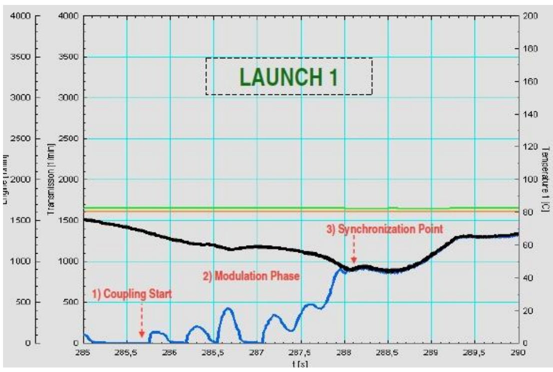
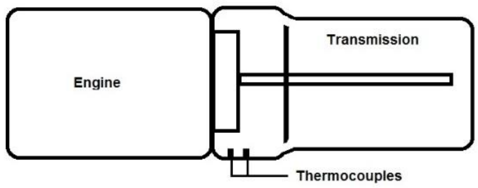


Fig. 19 Angular speeds of engine flywheel and transmission input shaft at 1st drive-off

The clutch housing temperature after each drive-off is measured by two thermocouples located in the clutch housing to control the clutch housing temperature as shown in Fig. 20 because friction coefficient of clutch disc is strictly dependent especially after 150°C. Besides, the measured temperature is used to validate the thermal model. It is observed clutch housing temperature increases from 80°C to 142.5°C. Finally, the wear rate was measured from the height difference of clutch rivets before and after the test to compare with the calculated wear rate for specific test maneuvers. All results for both the model and the experiment are tabulated in Table 1.



(a) The thermocouples on engine transmission assembly

(b) The thermocouples in the clutch housing

Fig. 20 The location of the thermocouples to record the clutch housing temperature

Table 1 Comparison of measured and calculated values

	Slip time at first drive-off (s)	Specific dissipated energy (J/cm^2)	Temperature of pressure plate after 100 cycles (ºC)	Temperature of clutch housing after 100 cycles (ºC)	Total wear after 100 cycles (mm^3)	Wear (%)
Simple Linear Model	1.07	85.54	-	-	-	-
Present model w/ 1st driver profile	2.10	113.9	230.3	156.2	15537.9	5.1
Present model w/ 2nd driver profile	2.12	103.3	213.8	145.6	10678.0	3.6
Present model w/ 3rd driver profile	2.06	126.0	250.1	168.8	23471.2	7.8
Experimental Results	2.37	107.0	-	142.5	19250.0	6.4

The measurements are repeated for 100 drive-offs to observe the behavior of angular velocities of the engine flywheel and the transmission input shaft. Most tests show the same behavior with the first test shown in Fig. 18 where the slip time is around 2.3 seconds. However, there are few extraordinary cycles that give slip times around 3.5 seconds as shown in Fig. 21.

While the third driver profile is close to the results of extraordinary cycles such as drive-off in 55th test, the specific energy results of first and second driver profiles of the present model (when the pedal travels with the same speed all the way and when the travel speed of the pedal increases) are closer to measured values. Once the present model (when the travel speed of the pedal increases) is used, absolute error of the specific energy decreases from 24.8%-absolute error of linear model-to 3.5%.

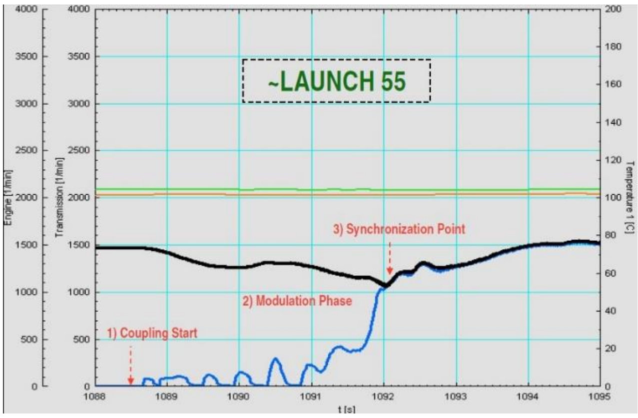


Fig. 21 Angular speeds of engine flywheel and transmission input shaft at 55th drive-off

5. Conclusions

In this paper, a rear wheel driven heavy duty vehicle was modeled for single and multiple drive-offs. The torque transmitted to driveline changes with clutch size, cushion deflection and friction coefficient of clutch disc. Cushion deflection was calculated by direct proportioning with clutch pedal release. Driver profiles were built based on the release of the clutch pedal in modulation zone. Clutch energy, slip time and mean friction power were calculated for drive-off models and used to obtain temperatures of pressure plate, flywheel and housing with the thermal model. The effect of cooling period on clutch life was presented. It is seen that an increase in pressure plate temperature causes a decrease in friction coefficient and an increase in slip time and clutch energy because of thermal heritage from previous cycles. Calculated clutch energies and housing temperatures were compared to experimental results. The results show that the present model gives more accurate results than the simple linear model in the literature. The driver profile when the travel speed of the pedal increases is matched with the driver tested. The driver profile when the travel speed of the pedal decreases gives closer results with test driver only for extraordinary cycles. Another observation is that repeated engagements in a short period affect the clutch wear exponentially. The experimental and simulated clutch wears are close; experimental results show more wear because of two main reasons: i) extraordinary cycles with high clutch energy and ii) heating until the test started is not modelled in the simulation. The present model allows optimization of clutch pressure plate mass, surface area and thickness of disc and driveline properties such as tire radius, final drive and transmission ratio for desired clutch life when general driving conditions are known at the design phase without need to a vehicle test.

Engine torque is calculated only from WOT engine characteristics but the model can be improved with a detailed study of engine torque. Different wears on internal and external rivet heads are observed in the tests. A 2D thermal model could be built to analyze the thermal load in the radial direction. Torque transmissibility also changes with the wear to rivet head and type of groove on clutch disc which may require more complex models which are not feasible for vehicle manufacturers.

Acknowledgments

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Note

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