



INFLUENCE OF THE CLUTCH MODEL IN THE VEHICLE LONGITUDINAL DYNAMICS

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Abstract. *The clutch transmits the engine's torque by means of the friction between two components called plate and flywheel and it connects the engine's crankshaft to the gearbox. The interaction between plate and flywheel is controlled by means of the clutch pedal. When this pedal is fully pulled, the plate is completely moved away from the flywheel characterizing a full disconnection. Although, when the clutch pedal is fully released, the plate is pressed against the flywheel characterizing a full connection. Between these stages there is a slip phenomenon. The most of the vehicle longitudinal dynamic models, described in the literature, neglects the clutch dynamic effect, assuming that there will be no further effect on the intended analysis. The objective of this paper is modeling the vehicle longitudinal dynamics performance as proposed by the literature and construct two other models, firstly inserting an analytical clutch dynamic model and secondly inserting a multibody clutch model. Finally the results are compared and it is told how relevant the clutch model for a vehicle longitudinal dynamic analysis is. The simulations are executed by MATLAB/Simulink™ which interacts with Adams™ for running the multibody clutch model.*

Keywords: *clutch 1, vehicle dynamics 2, longitudinal dynamics 3*

1. INTRODUCTION

The vehicular dynamic studies and analyzes the interactions between the vehicle, the driver and the environment as well as load reactions involved. The literature proposes to divide the vehicular dynamic into three areas: longitudinal, lateral and vertical.

The longitudinal dynamics is responsible for calculating the vehicle power consumption required so that it can fulfill a path, estimating by means of equations: the forces acting on the system, the aerodynamic drag and the tire-ground interaction, factors related to the inclination angle and driver behavior.

The internal combustion engine has less favorable performance characteristics, and can be used only with a suitable transmission. Despite this shortcoming, it has found the widest application in automotive vehicles to date because of its relatively high power to weight ratio, good fuel economy, low cost, and easiness to start (Wong, 2001).

The power train is the system responsible for supply the vehicle power demand. The manual transmission gear usually consists of the clutch, the gearbox, the propeller shaft, and the drive axle with a differential (to allow relative rotation of the driven tires during turning maneuvers) (Wong, 2001).

The gearbox provides a number of gear reduction ratios ranging from 3 to 5 for passenger cars. The number of gear ratios is selected to provide the vehicle with the tractive effort-speed characteristics as close to the ideal as possible (Wong, 2001).

The equation models proposed by the literature to calculate the vehicle longitudinal dynamics neglects the effect of the clutch, considering the engine directly coupled to the gearbox. However, the clutch is an important component in the powertrain and it has significant influence at the engine torque transmitted to the gearbox during the gearshift, affecting the engine speed during the process.

According to Genta (1997) apart from the action of the throttle control, the power supplied by the engine depends mainly on the rotational speed.

From a physical point of view, the dry clutch consists of two disks (the clutch disk connected to the mainshaft and the flywheel disk connected to the engine) covered with a high friction material and a mechanism which presses the disks against each other (clutch closed or engaged) or keeps them apart (clutch open or disengaged). During an engagement phase, the clutch disk is moved towards the flywheel disk until the friction due to their contact allows the torque transmission (Glielmo *et al.*, 2006).

The simulations are performed by a multibody dynamic analysis software Adams™ (Automated Dynamic Analysis of Mechanical Systems), with Matlab/Simulink™ where are implemented the equations proposed in the literature.

The scope of this study is evaluate the performance of the vehicle longitudinal according to the equations proposed by the literature and construct an analytical clutch model and also another one with a multibody clutch, comparing the results and the relevance of the clutch in vehicle longitudinal dynamics analysis.

2. VEHICLE LONGITUDINAL DYNAMICS

In this paper it will be used the longitudinal vehicle dynamics methodology proposed by Gillespie (1992) where the model is based on the acting forces on the vehicle travel direction as shown in Fig. 1.

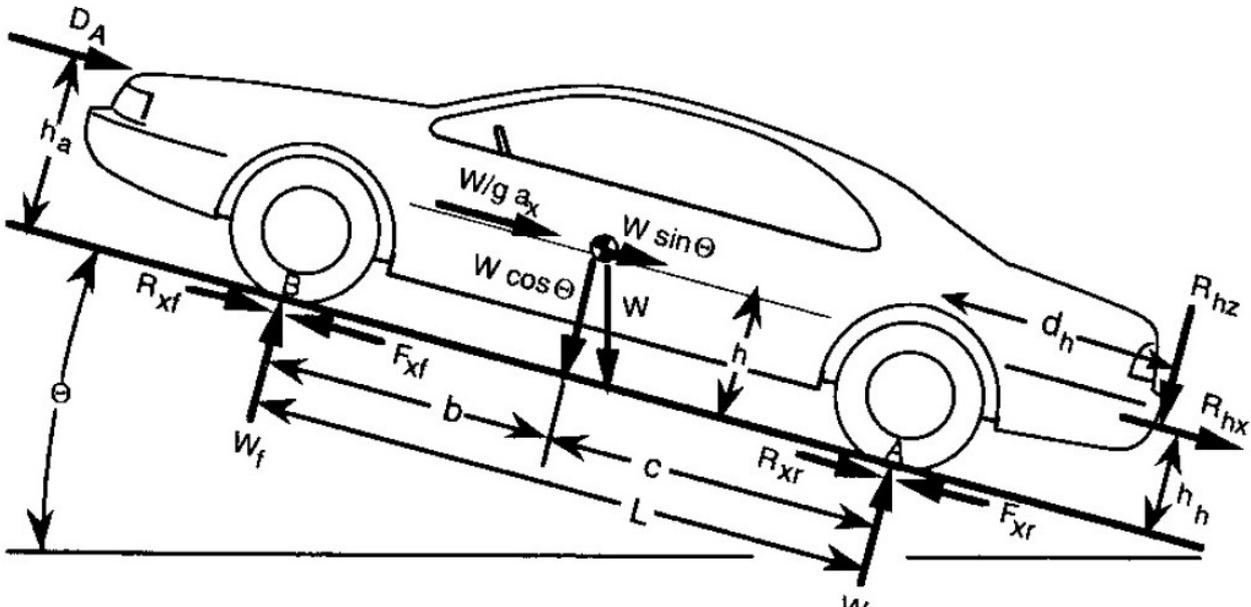


Figure 1. Arbitrary forces acting on a vehicle (Gillespie, 1992)

2.1 Aerodynamic Drag

The aerodynamic load (D_A) is the resistance imposed by the air during the vehicle passage, this effect is proportional to the square of the vehicle speed. According to Ehsani *et al.* (2009), a vehicle traveling at a particular speed in air, generates a resistance force of its motion. This force is known as aerodynamic drag and it is resultant from two components: shape drag and skin friction.

Due to the complexity of the airflow outside the vehicle, this load is based on empirical constant and a term known as drag coefficient, as shown in Eq. (1).

$$D_A = \frac{1}{2} \rho V^2 C_D A \quad (1)$$

Where ρ is the air density [kg/m^3], V is the vehicle speed [m/s], the term A refers to the frontal area of the vehicle and C_D is the coefficient drag.

2.2 Rolling Resistance

Rolling resistance is a result of energy loss in the tyre, which can be traced back to the deformation of the area of tyre contact and the damping properties of the rubber. These lead to the transformation of mechanical into thermal energy, contributing to warming of the tyre (Reimpell and Stoll, 1996).

At low speeds on hard pavement, rolling resistance (R_x) is the primary resistance load caused essentially by: the tire deformation, the pavement and the tire adhesion on the ground. This paper will consider hard surfaces, such as asphalt and concrete. In these cases, the ground stiffness is higher than the tires, therefore the road can be considered undeformable. The rolling resistance is shown by the Eq. (2).

$$R_x = f_r W \quad (2)$$

Where W is the vehicle weight [N] and f_r represents the rolling resistance coefficient calculated by the Eq. (3) in function of the vehicle speed V .

$$f_r = 0,01 \left(1 + \frac{0,62 V}{100} \right) \quad (3)$$

2.3 Road Grade Influence

This term refers to the weight force decomposition resulting from the road grade. In uphill, the weight force component acts retarding the vehicle movement, and in downhill, the weight force aids the movement.

The grade angle also results in a component of weight parallel to the ground which, as in the case of accelerating or braking on flat ground, results in a longitudinal weight transfer. The effects of grade and longitudinal acceleration can be combined in finding the changes in front and rear loads due to both (Milliken *et al.*, 1995).

2.4 Acceleration Performance

The vehicle acceleration generates resistance forces as the vehicle longitudinal displacement as the powertrain rotational inertia. The available traction force (F_x) in function of the engine torque and the transmission ratio is given by Eq. (4).

$$F_x = \frac{T_e N_{tf}}{r} - ((I_e + I_t)N_{tf}^2 + I_d N_f^2 + I_w) \frac{a_x}{r^2} \quad (4)$$

- T_e = Available engine torque [Nm];
- N_{tf} = Total gear ratio;
- η_{tf} = Transmission overall efficiency;
- r = Tire external radius [m];
- I_e = Gearbox inertia [kgm^2];
- I_t = Engine inertia [kgm^2];
- I_t = Gearbox inertia [kgm^2];
- I_d = Differential inertia [kgm^2];
- I_w = Wheels and tires inertia [kgm^2];
- N_f = Gearbox transmission ratio;
- a_x = Vehicle longitudinal acceleration [m/s^2].

The vehicle acceleration performance is given by the Eq. (5):

$$M a_x = \frac{W}{g} a_x = F_x - R_x - D_A - W \sin(\Theta) \quad (5)$$

Where M is the vehicle mass [kg] and Θ the road grade [rad].

Joining the Eq. (4) with Eq. (5) and isolating the engine torque (T_e) is possible to estimate the vehicle drive required torque in a predetermined situation.

$$T_e = \frac{\left(M a_x + \frac{((I_e + I_t)N_{tf}^2 + I_d N_f^2 + I_w) a_x}{r^2} + R_x + D_A + W \sin(\Theta) \right) r}{N_{tf} \eta_{tf}} \quad (6)$$

Maximum performance in longitudinal acceleration of an engine vehicle is determined by one of two limits-engine power or traction limits on the drive wheels. Which limit prevails may depend on vehicle speed. At low speeds tire traction may be the limiting factor, whereas at high speeds engine power may account for the limits (Gillespie, 1992).

The maximum contact force transmitted by the tire (F_{max}) is given by the Eq. (7), assuming a locked differential simplified model.

$$F_{max} = \frac{\mu \frac{W_c}{L}}{1 + \frac{h}{L} \mu} \quad (7)$$

- μ = Peak coefficient of friction;
- L = Wheelbase [m];
- W_c = Weight force acting on the front axle [N];
- h = Vehicle gravity center height [m].

3. VEHICLE POWERTRAIN

The power-torque-speed characteristics of the internal combustion engine are not suited for direct vehicle propulsion. A transmission, therefore, is required to provide the vehicle with the tractive effort speed characteristics that will satisfy the load demands under various operating conditions (Wong, 2001).

Traditionally, the term powertrain has been thought to include the engine, transmission, differential, and drive axle/wheel assemblies (Crolla, 2009). The manual gear transmission usually consists of a clutch, a gearbox, a propeller shaft, and a drive axle with a differential (to allow relative rotation of the driven tires during turning maneuvers) (Wong, 2001).

The engine is the power source in the driveline. The clutch connects and disconnects the engine to the rest of the driveline when the vehicle is equipped with a manual gearbox, used to change the transmission ratio between the engine and the drive wheels. The differential is a constant transmission ratio gearbox that allows the drive wheels to have different speeds (Jazar, 2008).

The internal combustion engine has a minimum speed (idle speed). To move-off from vehicle standstill, the speed gap between the lowest engine operating speed and the stationary transmission input shaft must be closed by means of a speed converter (Wong, 2001).

In automotive drivelines, the goal of the clutch is to smoothly connect two rotating masses, the flywheel and the transmission shaft, that rotate at different speeds, in order to allow the transfer of the torque generated by the engine to the wheels through the driveline (Glielmo *et al.*, 2006).

This clutch transmits the engine torque by force locking by means of friction forces on the transmission input shaft. It allows a rapid and complete separation and then jerk-free closing of the torque flow in both moving-off and shifting engagements (Lechner *et al.*, 1999).

In vehicles, there is a requirement for a device to provide a coupling from the engine crankshaft to the transmission. This allows the engine to be started and run without the vehicle moving and the vehicle to be started from the rest under control at various rates of acceleration (Crolla, 2009).

Clutches are associated with manual gearboxes and are normally operated by the driver, in other words, the driver has to change between one gear ratio and another, as the vehicle requires, when using this type of gearbox. The different gears have different ratios that allow different relative speeds between the engine and road wheels (Crolla, 2009).

The clutch makes use of friction to transmit the rotation of the engine crankshaft to the gearbox. When the clutch pedal is released the clutch spring forces the clutch plate against the flywheel, which is turned by the crankshaft (Rill, 2011).

When a vehicle with a manual gear transmission starts from the rest, in the initial period, it occurs a sliding between the driving and driven parts of the clutch, and the vehicle speed is not directly related to the engine speed (Wong, 2001).

During the clutch decoupling, the engine torque is not transmitted on, but when the contact between the clutch discs is resumed, the transmitted torque gradually returns by means of the slip between the clutch discs until system total coupling occurs.

It is also important to establish a dwell time between two subsequent gear shifts. This is required to avoid chattering and to satisfy comfort conditions (Casavola *et al.*, 2010).

The literature usually adopted the model where the engine speed is calculated according to the wheel rotation and the gear ratio chosen from the vehicle speed range at the time calculated. With the clutch model, it is possible to calculate the actual engine speed (ω_e) during the gear shift process by means of Eq. (8).

$$\omega_e = \omega_0 + \dot{\omega}_e dt \quad (8)$$

The engine acceleration ($\dot{\omega}_e$) is given by the Eq. (9).

$$I_e \dot{\omega}_e = T_e - T_{fr} - T_c \quad (9)$$

- T_{fr} = Friction losses torque [Nm];
- T_c = Resistance torque [Nm];
- dt = Simulation time step [s];
- ω_0 = Initial engine speed [rad/s];
- ω_e = Engine speed [rad/s];
- $\dot{\omega}_e$ = Engine angular acceleration [rad/s^2].

The maximum torque that the clutch can transfer to the vehicle transmission is usually slightly larger, from 10% to 20%, than the maximum engine torque (Genta, 1997). Being that the clutch transmitted torque depends of the clutch pedal position. When it's pressed, the clutch pedal reduces the normal force between the clutch plate and the flywheel and thus it reduces the maximum friction torque (Rill, 2011).

The clutch transmitted torque (T_{cl}) is given by the Eq. (10). as proposed in Kulkarni *et al.* (2007).

$$T_{cl} = \mu_{cl} F_n n \left(\frac{2}{3} \right) \left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2} \right) \quad (10)$$

- T_{cl} = Clutch transmitted torque [Nm];
- μ_{cl} = Clutch friction coefficient;
- F_n = Normal force applied between the discs [N];
- n = Clutch number of faces;
- R_o = Clutch external disk radius [m];
- R_i = Clutch inner disk radius [m].

4. DYNAMIC SIMULATION

The longitudinal dynamics simulation is normally used to compare the importance of energy balance characteristics for vehicles, without the need of building prototypes that can take much time and a high cost (Oliveira, 2005).

According to Oliveira (2005), the basic physical equations of the vehicle longitudinal motion are fundamental to understand the cause-effect relationships between driver, vehicle and ground. Some input variables are:

- Driving cycle;
- Vehicle specifications (rolling resistance, air resistance, weight etc.);
- Individual components specifications (performance loss of the gearbox).

The simulations implemented in this paper were made via the multibody dynamic analysis program AdamsTM (Automatic Dynamic Analysis of Mechanical Systems), where is implemented the vehicular model analyzed. The control of variables related to longitudinal dynamics, as described earlier, is done through the interface between Adams and MATLAB/SimulinkTM.

The simulated vehicle was based on a compact hatchback equipped with 1.0L engine. The implemented model was designed based on a dynamometer bench, to enable future experimental validations. The effects of vehicle suspension system were neglected to simplify the model, as these factors are disregarded by the current literature.

The model consists of two rolls set to simulate the longitudinal displacement inertia, in which four cylinders representing the vehicle's wheels are supported. The powertrain system is based on the rotate mass model that simulates the components inertia, as shown in Fig. 2.

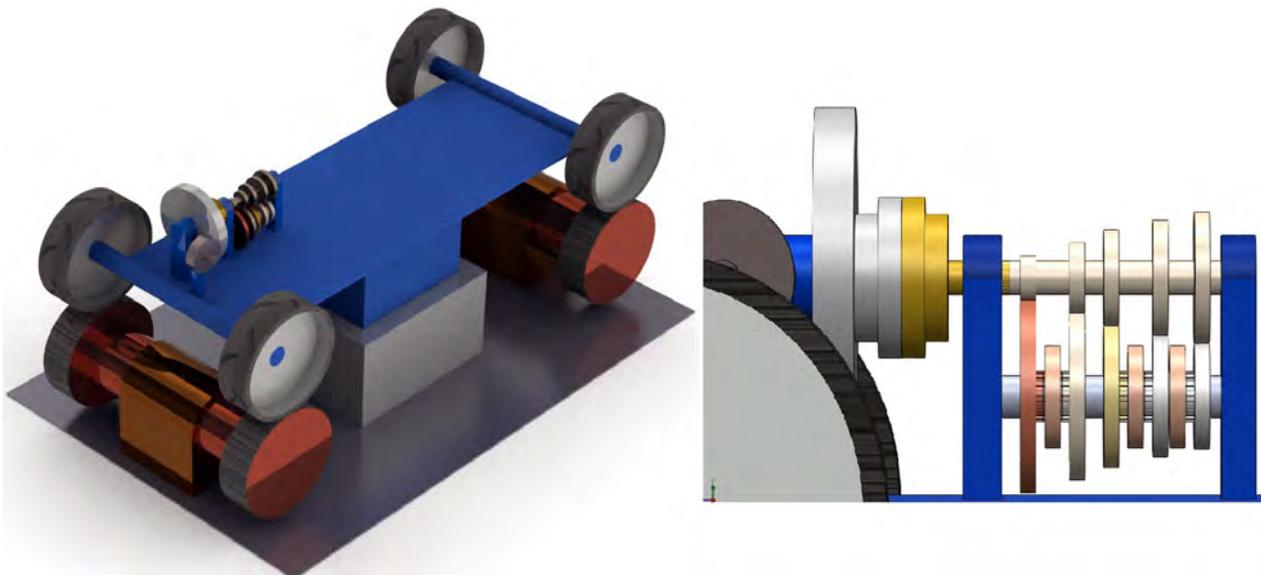


Figure 2. CAD model

The CAD model was exported to AdamsTM, where an appropriate revolution joints were created to allow the wheels movement and rotating masses. After, the engine torque is applied at the powertrain by means of the clutch contact model and the resultant torque is applied to the gear box where each transmission ratio has determined inertia value. After that, the available power in the gearbox output is sent to the differential where another reduction in the transmission ratio occurs and finally the differential torque is applied to the model front wheels.

On the wheels were applied torques related to the power supplied by means of the brake system, and in the rotating masses, were applied a movement resistance torque. In the model, the vehicle chassis was connected to the base to prevent longitudinal movement so that the wheels remain aligned with the rollers. The rotational movement between the rollers and the wheels are done by means of a joint, transmitting torques and acting speeds.

To facilitate the implementation of the vehicle dynamics equations, it was used a SimulinkTM / AdamsTM interface, generating a block of data from the dynamic model as shown in Fig. 3.

The SimulinkTM programmed algorithm works together with the AdamsTM solver. The SimulinkTM provides for AdamsTM torque values applied in the engine, the sleeves positioning in the gearbox, the rotating masses and also brake system resistance torques. The AdamsTM generates a response from an angular velocity of the wheels, which supplies the SimulinkTM algorithm to recalculate the required torque according to the new demand.

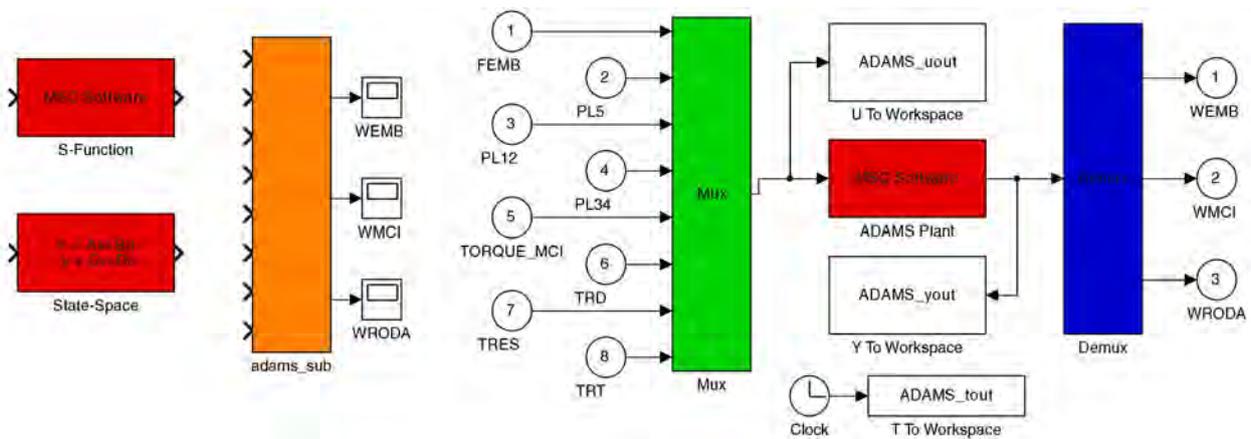


Figure 3. Adams™ generated block

4.1 Driving Cycle

The driver’s behavior is represented by calculating needed for the vehicle to achieve the required speed in a particular instant. As the interfacing between Simulink™ and Adams™ is implemented in fixed time steps, the required acceleration is calculated by the difference between the required speed and actual speed divided by the step of the simulation.

With the intention to establish a benchmark, standard cycles were utilized to determine the vehicle speed behavior, in a way that the mathematic model calculate the vehicle required power to follow the velocity profile predetermined by the cycle.

A driving cycle represents the way the vehicle is driven during a trip and the road characteristics. In the simplest case, it is defined as a sequence of vehicle speed (and therefore acceleration) and road grade (Corrêa *et al.*, 2011). These driving cycles are designed to be representative of urban and extra-urban driving conditions, and reproduce measures of vehicle speed in real roads. Some of them and the test procedures have been recently updated to better suit modern vehicles, following criticism towards the previous regulation (Serrao *et al.*, 2005).

In the simulations was used the US06 cycle that complements the U.S. test cycle FTP-75 because it represents high speeds and accelerations. This cycle represents a route of 12.8 km, with an average speed of 77.9 km/h, maximum speed of 129.2 km/h and duration of 596 seconds. Therefore, this cycle allows a better analysis of the vehicle performance in acceleration condition, to evaluate a critical operation in the proposed model.

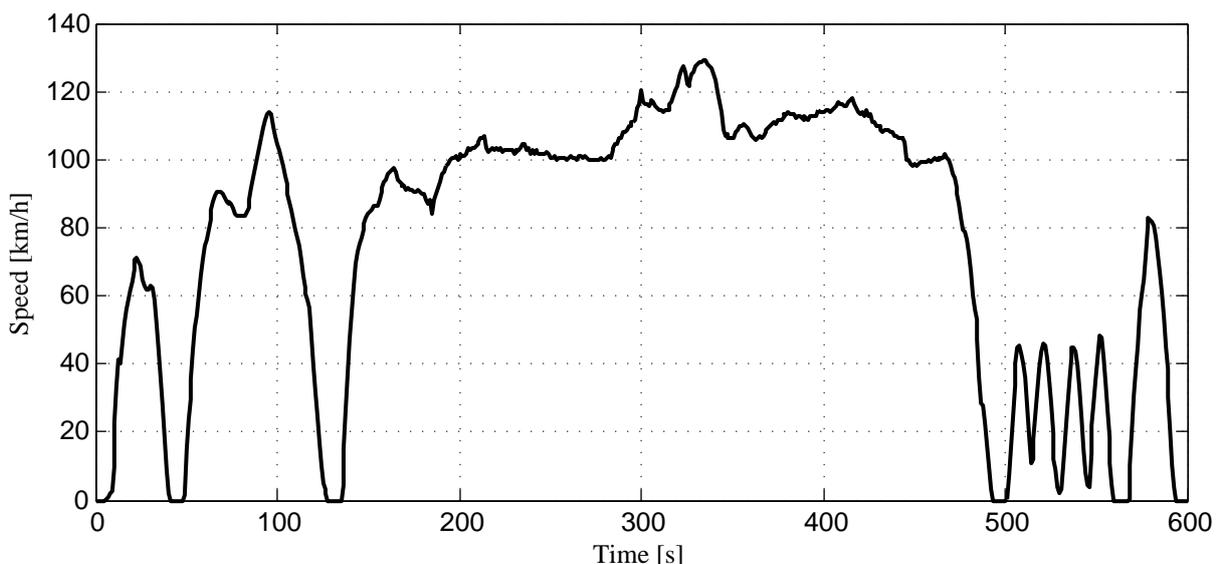


Figure 4. Velocity profile US06

This cycle was chosen for having accelerations and speed reductions where the influence of the clutch in the gear shifting can be observed. For a better presentation of the results it will be used the initial 130 cycle seconds, because after that it starts a long stretch of high speed where no gear shifting occurs.

4.2 Engine Torque Curves

The simulated model considers the engine torque curves Fig. 5, that is the previously described mathematic model compared with the available engine torque in function of the acceleration percentage and the engine speed. If the required torque exceeds the maximum torque available, there will be loss of performance.

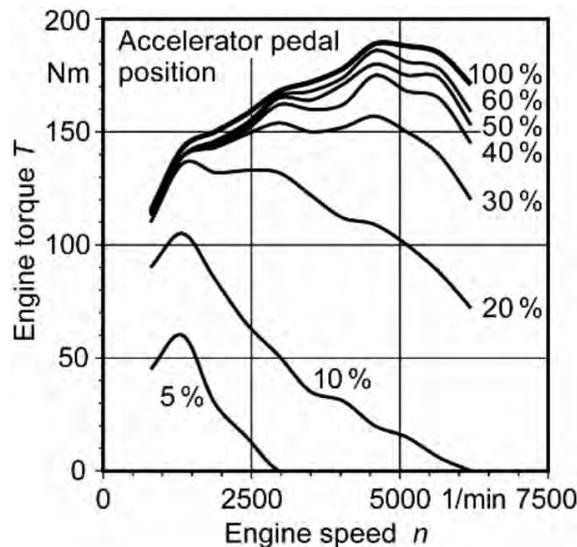


Figure 5. Engine torque curve (Lechner *et al.*, 1999)

5. LINEAR CORRELATION

To evaluate the difference between the simulation results, we used the linear correlation between the velocity profile and the standard cycle adopted. The correlation coefficient r is the intensity measure of the linear relationship between the two variables. The term r^2 is the square of the correlation coefficient, called determination coefficient, and consists of the sum of squares of prediction errors obtained as shown in Eq. (11), where x and y represent the curves values.

$$r^2 = \frac{(\sum (x_i - \bar{x})(y_i - \bar{y}))^2}{\sum (x_i - \bar{x})^2 \sum (y_i - \bar{y})^2} \quad (11)$$

The regression r^2 measures the variability proportion between the two curves, therefore, is a direct correlation function between the variables, showing the variance percentage of the variables. A value of r^2 close to 1 indicates a strong relationship between the two variables.

6. SIMULATION PARAMETERS

The geometric parameters used in the simulations were obtained based on a Compact Hatchback vehicle equipped with 1.0L engine and five ratios gearbox, as shown in Tab. 1.

Table 1. Vehicle parameters

Components	Units	Speed				
		1 st	2 nd	3 rd	4 th	5 th
Engine inertia	kgm^2	0.1367				
Transmission inertia	kgm^2	0.0017	0.0022	0.0029	0.0039	0.0054
Transmission ratio	-	4.27	2.35	1.48	1.05	0.8
Upshift speed	km/h	0	15	30	50	72
Diferencial inertia	kgm^2	9.22E-04				
Diferencial ratio	-	4.87				
Wheels + tires inertia	kgm^2	2				
Vehicle mass	kg	980				
Tires	-	175/70 R13				

Table 1 shows the gear shifting speeds as recommended by GM (2013), however to avoid frequent shifts, the vehicle speed at down shifting point should be 5-10 km/h lower than at up shifting point (Xi *et al.*, 2009).

The clutch model data were obtained from Haim (2011), as shown in the Tab. 2.

Table 2. Simulate clutch parameters

Clutch parameters	
Inner disk radius	67 (mm)
External disk radius	95 (mm)
Faces	2
Friction coefficient	0.27

The normal force applied to the clutch is calculated as proposed by Haim (2011), based on a graphical method that estimates the force applied in function of the clutch pedal position, the deformation of the clutch structure and the spring force that holds the coupled linked.

The driver's behavior simulation model is given by the activation of the clutch pedal, considering a 1 second up shift time. The graph in Fig. 6 shows the clutch fingers displacements in function of the foot switch position. Depending on the fingers position is possible identify the plate displacement as shown in Fig. 7.

During the first 0.3 seconds, the engine is disengaged from the transmission system. The clutch pedal is moving from the resting position (coupled system) to the limit (system fully decoupled).

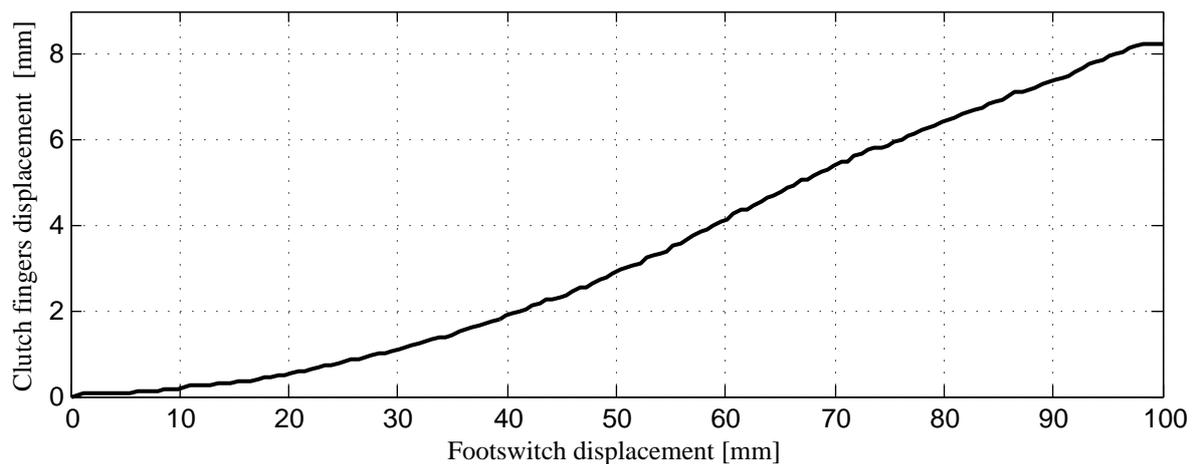


Figure 6. Clutch fingers displacement as a function of the footswitch displacement

After the total decoupling of the system, it occurs the shifting in an interval time of 0.2 seconds that changes the selector sleeve position in the Adams™ model gearbox.

During the remaining time, the pedal gradually returns from the end to the initial position, increasing the torque transmitted between the engine and gearbox to the complete coupling condition.

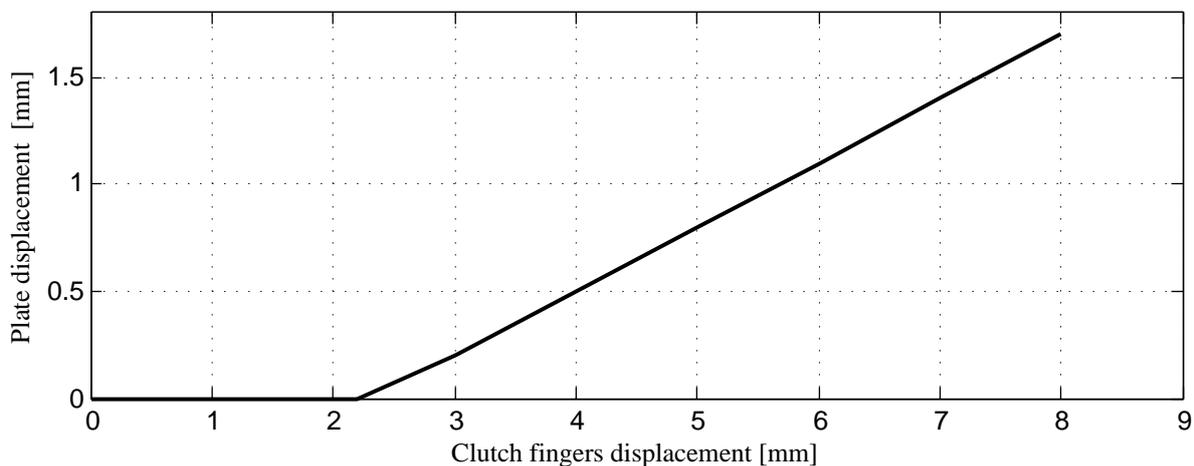


Figure 7. Plate displacement according to the clutch fingers displacement

In function of the plate displacement is possible to obtain the spring normal force applied in the clutch components as shown in Fig. 8.

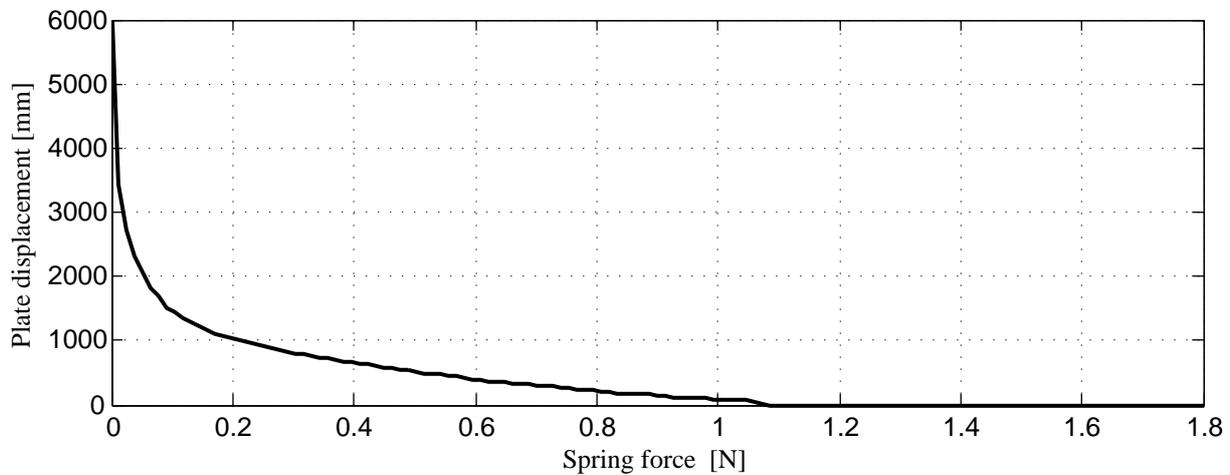


Figure 8. Spring normal force applied according to the plate displacement

7. RESULTS

Three simulations were performed, at first the clutch and the engine were coupled in the AdamsTM multibody model not allowing slip between the components. This first simulation used the modeling according to the literature where the clutch effect is neglected. At the second simulation the clutch was modeled according to the previously equation proposed, where the engine speed and the torque transmitted were calculated. In the third simulation, it was set up in AdamsTM multibody model the contact clutch engagement to simulate the slip between the components during the gear shifting, in this case the engine speed was obtained directly from Adams model.

The results found in the three simulations are shown in the Fig. 9, which presents the correlations between the results simulation velocity profiles and the velocity profile proposed by the standard US06.

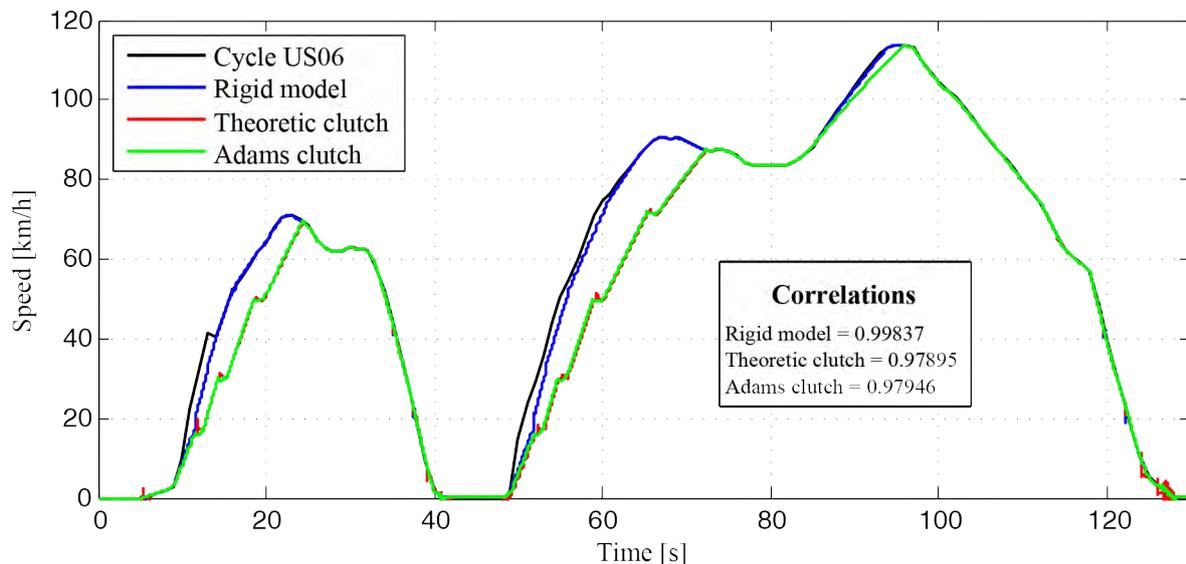


Figure 9. Correlations between the simulated models and the velocity profile US06

As the simulation objective is follow the standard velocity profile imposed, correlations are always close to 1, and the differences are found in the second decimal place. As it can be seen, in the Fig. 9, both the theoretical model and the clutch multibody showed very similar results.

One can notice a visible difference between the performance of the rigid model that neglects the clutch in comparison to the other models, this occurs because there's no interruption in the torque transmitted to the driveline during the gear shifting. Primarily due to the velocity profile in high accelerations, which cause a situation where the simulated model does not have enough power to fulfill the required velocity profile. For every gearshift, it occurs a transmitted torque

reduction to the vehicle wheel, causing a speed decrease in a situation where the vehicle has no spare power capacity to compensate the performance decrease.

The engine angular speed in the three simulated cases is shown in the Fig. 10. The rigid model achieves the up shifting speeds established before because it doesn't consider the decoupling clutch, getting better acceleration performance. The clutch theoretical model has a lower reduction in engine speed during recoupling, followed by a sudden reduction when the coupling is taken overall. Otherwise the multibody model by considering the contact between the clutch components, it represents the best slippage between the components generating a rotation profile closer to the real.

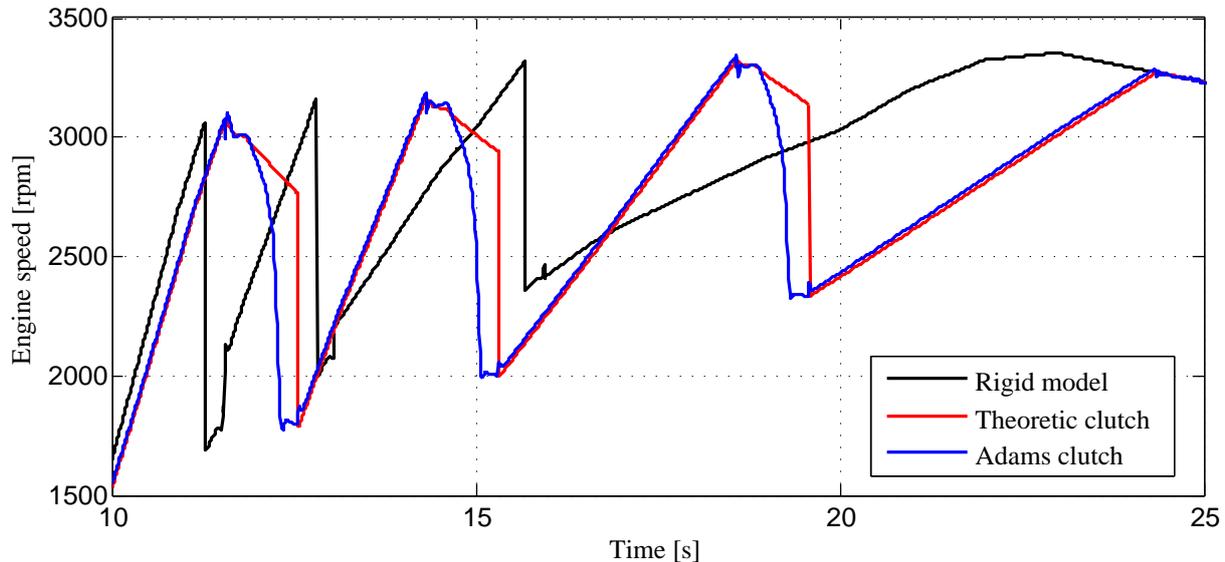


Figure 10. Differences between the engine speed regimes according to the simulated model

To better visualize the difference between the engine speeds found by the rigid and the multibody model, it was obtained the engine speed in function of the vehicle speed as simulated in the rigid model, during the multi-body model simulation. This to obtain results with the synchronized gear shifting, being made at the same speed, which it was not possible as shown in Fig. 10. Figure 11 illustrates the behavior between the two modelings.

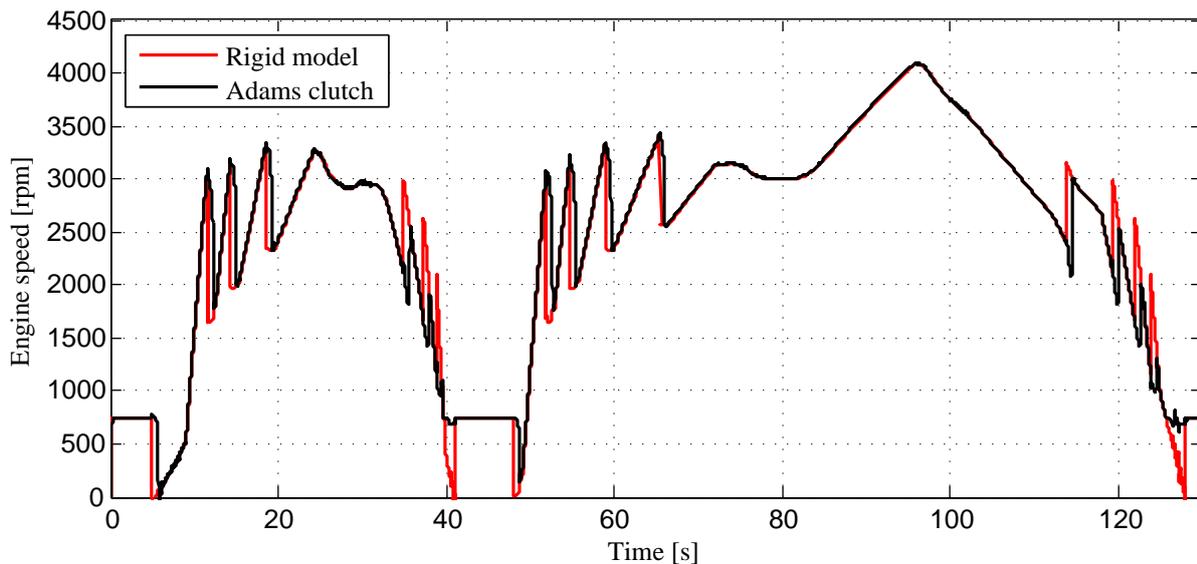


Figure 11. Engine speed comparative in models with and without clutch

Due to the scale factors, the results can be better observed by analyzing a smaller range of results shown in the Fig. 12.

As it can be seen in the Fig. 12, the engine rotation of the rigid model changes immediately after the gear shift, which does not occur in the true. The multibody model represents the gradual engine speed changing during the shift process considering the components slippage and the process accelerations.

The Fig. 12 also demonstrates the gear shifting stages, where it is divided into: coupled system, decoupling, gear shifting, and recoupling, and being therefore possible to observe the process dynamic behavior of each plot.

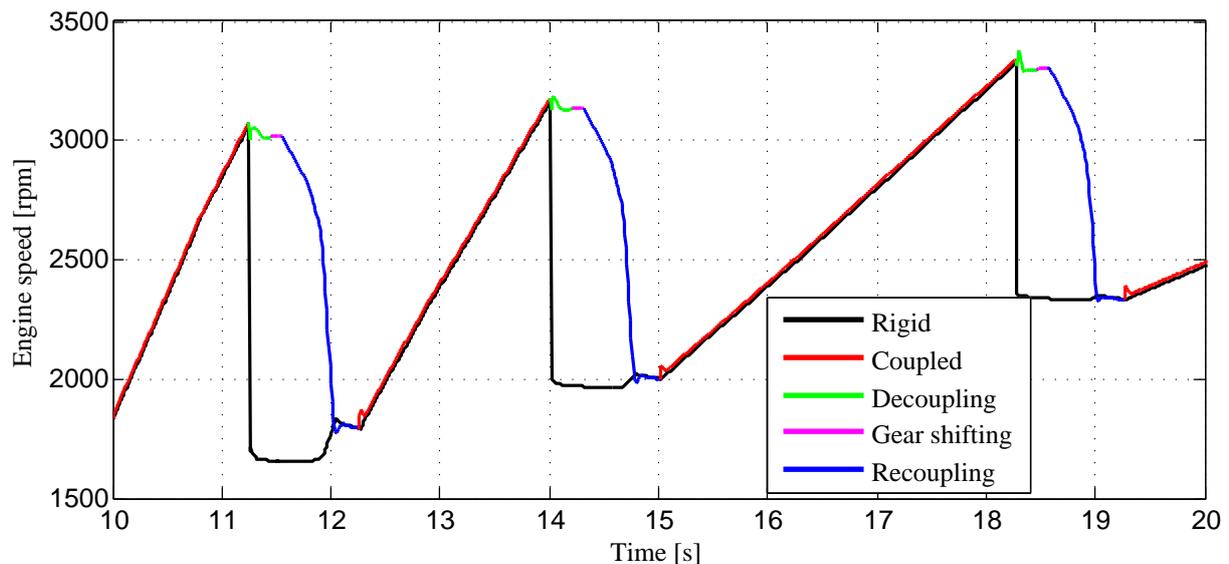


Figure 12. Engine speed comparative in models with and without clutch

To avoid engine shutting down, the vehicle speed of n gear at the point of downshift line must be higher than or equal to the minimum speed of n gear (Xi *et al.*, 2009).

In the results presented until this moment, when it occurs the vehicle starting from the rest (zero speed), the engine speed drops below the idle speed, condition which the engine cannot operate. To assuage the situation it is possible to increase the engine speed before departure, representing the driver behavior, besides increasing the clutch recoupling time and engine torque supplied in the first gear, thereby generating clutch slip until the vehicle reaches a certain speed and still preventing that the engine operates below the idle speed. In a braking situation, the gearbox is changed to neutral position when the engine speed becomes less than the idle speed. With the modification described, the engine speed in function of the gear ratio adopted is shown in the Fig. 13.

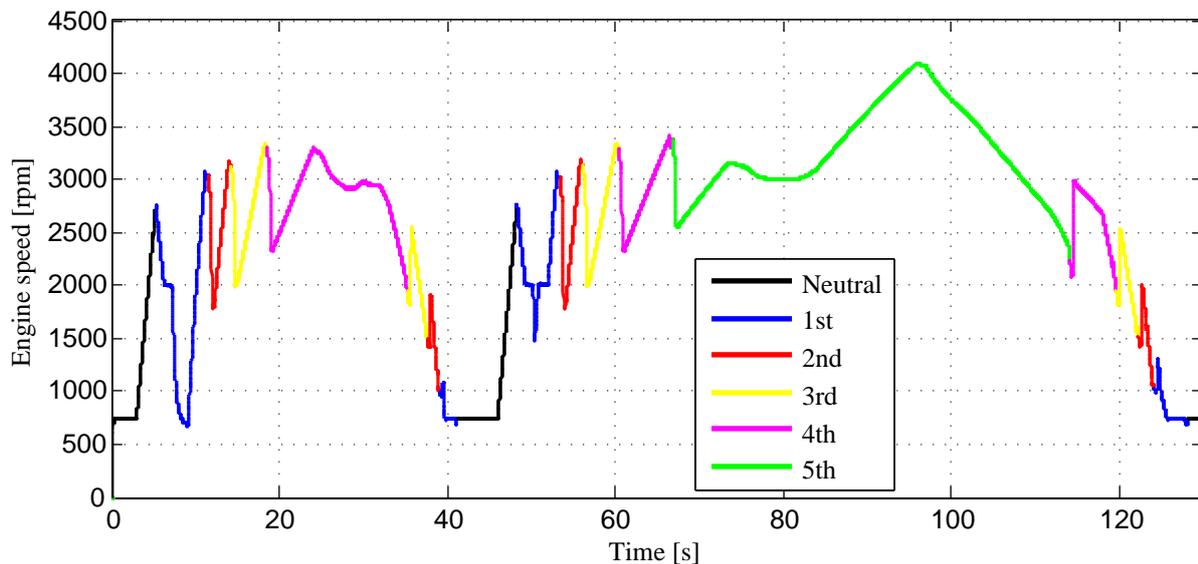


Figure 13. Engine speed according to the gear

8. CONCLUSION

In this study, we evaluated the influence of the clutch in the vehicle longitudinal dynamics calculation. The results were obtained by means of simulations in the multibody dynamic analysis program Adams™ together with Simulink/Matlab™.

The calculation methodology was described in the book Fundamentals of Vehicle Dynamics taking considering the system transmission inertia and geometric factors of a Compact Hatchback, equipped with 1.0L engine and also a reduction system with five ratios.

The simulations considered the initial 130 seconds the velocity profile proposed by the complement of the American

standard FTP-75 (US06 cycle), which represents two cycles of vehicle acceleration and braking.

It can be observed that with the inclusion of the clutch in the simulation, there is a vehicle performance reduction at the time of gear shifting due to engine uncoupling from the rest of the powertrain, generating a speed decrease due to transmission torque reduction/interrupting.

Due to the US06 cycle consists of a long and high acceleration stretches, the maximum vehicle power is required. It occurs a performance loss in each gear ratio change, because there is not surplus power to compensate the speed decrease after the gear shift.

Among the three models simulated, the rigid model which neglected the clutch effect had the better performance because it does not consider the torque supplied interruption to the powertrain during the gear shifting, but this situation does not match the true vehicle operating condition.

Both clutch models have similar performance, but with different behaviors in the recoupling process generating differences in the engine speed.

The multibody model allows a better representation of the engine speed, considering the slippage between the clutch components, thus allowing also a better driver behavior representation when the vehicle starts from the rest.

Finally, it is concluded that the clutch is a fundamental vehicle powertrain component, so its dynamic behavior cannot be disregarded in the longitudinal dynamics calculating, because it generates performance changes in vehicle acceleration and engine speed, in order to approximate the simulation to the actual vehicle behavior.

9. ACKNOWLEDGEMENTS

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