VEHICLE GEAR SHIFTING CO-SIMULATION TO OPTIMIZE PERFORMANCE AND FUEL CONSUMPTION IN THE BRAZILIAN STANDARD URBAN DRIVING CYCLE

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ABSTRACT

The vehicle longitudinal dynamics is responsible for calculating the vehicle power consumption undergone a specific route, by means of the estimation of the forces acting on the system such as aerodynamic drag, rolling resistance, climbing resistance and the driver behavior. The gear shifting tactics influences the vehicle performance and fuel consumption because it changes the powertrain inertia and the engine speed. The literature presents gear-shifting strategies based on the engine power and torque as well as the fuel economy. The last tactics are difficult to determinate, because they depend on a large number of factors like vehicle speed, available transition ratios, engine efficiency, required acceleration, tire-ground traction limit and engine decoupling during the gear shifting process. This paper shows a study based on the Brazilian standard urban driving cycle NBR6601. As there are many factors involved in the vehicle behavior and also in the vehicle dynamics, it was developed an algorithm to optimize the gear shifting process: it makes a choice of the most adequate tactic for each cycle stretch. The analysis were performed by co-simulation between the multibody dynamics software Adams[TM] and Matlab/Simulink[TM], in which is defined the vehicle power demand.

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 vehicle power demand is a function of the aerodynamic drag, rolling and climbing resistance, powertrain inertia and the driving behavior. The vehicle longitudinal dynamics evaluates the effects of the motion resistance forces to define the engine-required power to reach the drive condition defined by the driver.

Brazil decreed the INOVAR-AUTO [1] that imposes a reduction of 12.08% vehicle fuel consumption reduction until 2017. The vehicle that increase the efficiency over 15.46% will have a 1% discount in the IPI, and the vehicles that increase the efficiency over 18.84% will receive 2% discount.

There are some strategies for the reduction of the vehicle fuel consumption for conventional powertrain system, reducing the external load (such as vehicle weight) or increasing the drive system efficiency including the engine and transmission systems [2]. Another way to reduce the fuel consumption is to put the engine to operate at optimum efficiency. In order to improve the powertrain efficiency, the gear shifting strategies in manual transmissions have been developed to satisfy vehicle requirements and to improve the engine efficiency [2].

The gear shifting process is an important event is the vehicle longitudinal dynamics, because it changes the powertrain inertia and the engine speed, influencing significantly in the vehicle acceleration performance and engine fuel consumption.

The gear shifting strategy it an alternative to reduce the vehicle fuel consumption, because it is possible to move the engine regime to the more efficient regions changing the transition ratio according to the situation.

There are some gear shifting strategies that aims the maximum vehicle performance based on the engine maximum torque or to maintain the vehicle at high speed using the engine maximum power [3]. Tactics based on fuel economy are more delicate, because they depend on parameters like vehicle speed, acceleration, transition ratio, engine and powertrain efficiency and the driver behavior. The longitudinal vehicle dynamics models proposed in the literature do not define a standard gear-shifting algorithm because this factor represents the driver behavior in function of the vehicle and road conditions.

According to [4] the efficiency and quality of the gear shifting are determined by the shift tactics, defining the exact time a shift event it executed. This represents significant effects over the vehicular performance and fuel consumption [5].

The shift control governs the transmission behavior, during the gearshift event, as a function of the motion resistance forces and the vehicle speed. As described by [6] the use of gear shifting indicators, composed of light and/or sound alarms connected to a software that manage the gear shifting process, results in 3.6% of fuel economy in a vehicle submitted to the New European Driving Cycle - NEDC. With the same purpose, [7] developed a driving assistance system to help drivers giving fuel efficiency guidelines according with the current vehicle situation.

To reduce the fuel consumption [8] evaluated experimentally the variability of driver behaviors, in relation with the engine speeds amplitude, and repeat the tests using ecodriving techniques to reduce the fuel consumption. Furthermore, [9] optimized the gearshift time in heavy vehicles considering the engine torque and clutch performance. In the same way [10] studied the predictive optimal gear shifting strategy, utilizing the route information from the vehicle navigation system and vehicle state in heavy duty trucks models and highway routes.

This paper aims to optimize the vehicle performance and fuel consumption by co-simulations between the multibody dynamics analysis software Adams^{TM} which contains the vehicle model based on a 1.0L Brazilian vehicle, and Matlab/Simulink^{TM} which supports the vehicle dynamics equations and the gear shifting control.

To create a comparison parameter it is used the Brazilian urban driving cycle NBR6601 [11] to represent the driver behavior by means of the inference of the acceleration in function of the standard velocity profile.

1. LONGITUDINAL VEHICLE DYNAMICS

In this paper it is used the longitudinal vehicle dynamics methodology proposed by [12] whose model is based on the motion resistances forces as shown in the Figure 1.
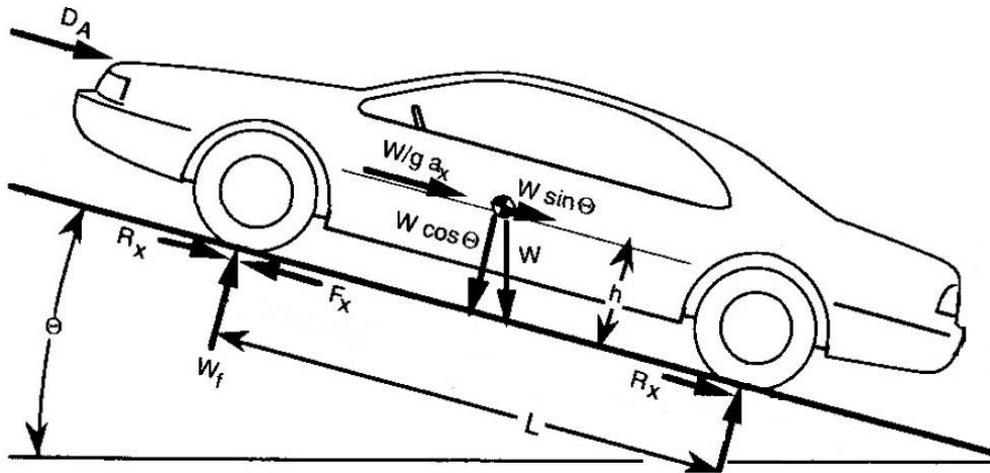
Figure 1 - Arbitrary forces acting on a vehicle adapted from [12]

1.1. Aerodynamic Drag

The aerodynamic drag (D_A) is the resistance imposed by the air during the vehicle passage, the Equation (1) shows the aerodynamic drag as a function of the air density ρ $[kg/m^3]$ the vehicle frontal area A $[m^2]$ and the vehicle speed V $[m/s]$.

$$D_A = \frac{1}{2}\rho V^2 C_d A \tag{1}$$

Due to the complexity of the airflow outside the vehicle, the drag coefficient (C_d) is based on empirical constant depending on the vehicle shape as Figure 2 shows.

	Vehicle type	Coefficient of aerodynamic resistance
	Open convertible	0.5…0.7
	Van body	0.5…0.7
	Ponton body	0.4…0.55
	Wedged-shaped body; headlamps and bumpers are integrated into the body, covered underbody, optimized cooling air flow	0.3…0.4
	Headlamp and all wheels in body, covered underbody	0.2…0.25
	K-shaped (small breakaway section)	0.23
	Optimum streamlined design	0.15…0.20
	Trucks, road trains	0.8…1.5
	Buses	0.6…0.7
	Streamlined buses	0.3…0.4
	Motorcycles	0.6…0.7
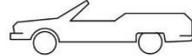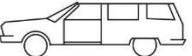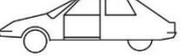
Figure 2 - Drag coefficients C_d for different vehicles [3]

1.2. Rolling Resistance

This resistance force corresponds to the energy loss in the tire, due to the contact area deformation and the rubber damping properties. At low speeds on hard pavement, the rolling resistance (R_x) is the primary resistance load caused essentially by: the tire deformation, and the tire adhesion on the ground phenomenon. The rolling resistance is estimate by the Equation (2) as a function of the vehicle weight (W)

$$R_x = f_r W \tag{2}$$

The rolling resistance coefficient (f_r) is calculate by the Equation (3) as a function of the vehicle speed.

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

1.3. Climbing Resistance

Correspond to the gravitational force component parallel to the road plane. When the car is moving uphill the climbing resistance acts against the movement. On the other hand, if the road angle is negative, the car is moving downhill and this force acts in favor of the movement [12]. The grade angle also results in a component of weight parallel to the ground which, in the case of accelerating or braking on flat ground, results in a longitudinal weight transfer [13].

1.4. Vehicle Traction Force

The vehicle longitudinal displacement and the powertrain rotational inertia acceleration generates resistance forces that change the engine power demand. The available traction force (F_x) at the vehicle wheels in function of the engine torque and the gearbox and differential transmission ratios is define by the Equation (4).

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

- T_e = Engine torque $[Nm]$
- N_{tf} = Total gear ratio
- η_{tf} = Transmission overall efficiency
- r = Tire external radius $[m]$
- I_e = 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 $[m/s^2]$

When the traction force (F_x) is positive, it is originated by the engine and powertrain systems. On the other hand, if this force becomes negative, the energy is dissipated by the vehicle brake system.

1.5. Vehicle Acceleration Performance

The vehicle acceleration is given by the Equation (5), disregarding the road grade influence because the NBR 6601 does not provide any information about the grade angle, so it is considered null during the simulations. To estimate the engine required torque in a predetermined situation, the Equations (4) and (5) are merge, and the engine torque (T_e) is isolate by the Equation (6).

$$Ma_x = F_x - R_x - D_A \tag{5}$$

$$T_e = \frac{\left(Ma_x + \frac{\left((I_e + I_t)N_{tf}^2 + I_d N_f^2 + I_w\right)a_x}{r^2} + R_x + D_A)\right)r}{N_{tf}\eta_{tf}} \tag{6}$$

1.6. Tire-Ground Traction Limits

The maximum longitudinal acceleration performance of the vehicle is determined by the engine available power or the traction limits on the drive wheels. The limit prevails depending on vehicle speed. At low speeds tire traction may be the limiting factor, whereas at high speeds engine power may account for the limits [12]. The maximum contact force transmitted by the tire (F_{max}) is given by the Equation (7), assuming a locked differential simplified model.

$$F_{max} = \frac{\mu\frac{W_c}{L}}{1 + \frac{h}{L}\mu} \tag{7}$$

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

1.7. Clutch Model

During the gear shifting process, the clutch decouples the engine from the rest of the powertrain. That results in a speed decrease because of the drive wheels power interruption. To return to the previous speed the vehicle needs to accelerate, which leads a fuel consumption increase. In this paper it is used the transmissions clutch torque model proposed by [14] as can be seen at the Equation (8).

$$T_{CL} = \mu_{cl}F_n n\left(\frac{2}{3}\right)\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right) \tag{8}$$

- T_{CL} = Clutch transmitted torque   [Nm]
- μ_{cl} = Clutch friction coefficient   [Nm]
- 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]

2. DRIVING CYCLE

To establish a benchmark, standard velocity profiles are utilized to determine the vehicle speed behavior at specific driving conditions in the real word. The driving cycles have been standardized as speed and elevation profiles, obtained statistically from real traffic situations, used as a common base for tests with different vehicles about pollutant emissions and fuel consumption [3].

A driving cycle represents the way the vehicle is driven during a trip seeing the road characteristics. In the simplest case, it is defined as a sequence of vehicle speed (and therefore acceleration) and road grade [15].

The driving cycles are designed to be representative of urban and extra-urban driving conditions, and they reproduce measures of vehicle speed in real roads, therefore these driving cycles can be used to estimate the required power to follow the cycle velocity profile. Some of them and the test procedures have been recently updated to better suit modern vehicles, following criticism towards the previous regulation [16].

In this paper it is used the standard Brazilian urban driving cycle NBR 6601 [11] as shown at the Figure 3. The cycle have total distance of 12 km, average speed of 32 km/h and 91.2 km/h maximum speed. The urban driving conditions are characterized by low speed, low engine load, and low exhaust gas temperature [17]. Moreover, the vehicle remains stationary for 17.2% of the time, and the cycle does not include road grade information.
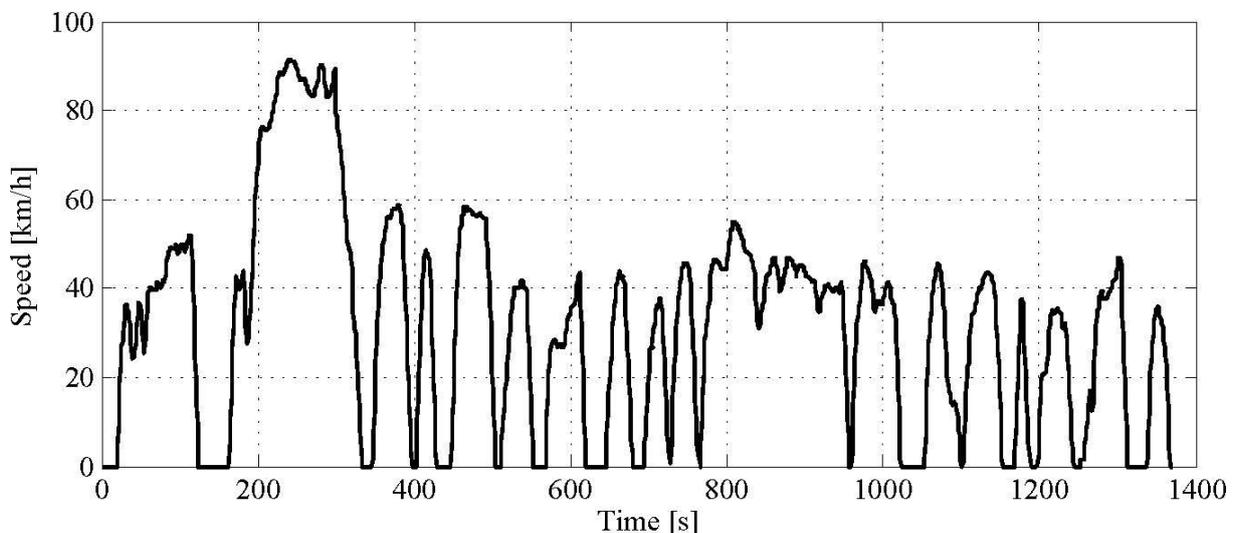
Figure 3 - Velocity profile NBR6601 [11]

Even with the current improvements, the regulatory cycles should be considered a comparison tool rather than a prediction tool. In fact, it is not possible to predict how a vehicle will be driven, since each vehicle has a different usage pattern and each driver his or her own driving style. In order to obtain more realistic estimations of real-world fuel consumption for a specific vehicle, vehicle manufacturers may develop their own testing cycles [18].

In real driving conditions, the vehicle stopped time can be greater than that shown in the standard velocity profile. It does not represent a real use conditions, anyway, the standard velocity profile can be used as a comparison between the available technological solutions [19].

3. ENGINE PARAMETERS

Due to the high complexity of the engines, for the sake of simplicity, the engine torque can be obtained from an experimental lookup table, as function of the engine speed and throttle angle without degrading the results of a longitudinal analysis. The engine's specific fuel consumption can be calculated in the same way [3]. The simulated model considers the engine torque curves at Figure 4 that represents the real engine torque in function of the acceleration percentage and the engine speed, based on experimental results for a vehicle similar to the simulated. The required torque is calculated by means of the Equation (6) then is compared with the available engine torque from the curves of Figure 4. If the required torque exceeds the maximum torque available, the simulation will use the maximum torque of the curves and there will be loss in the vehicle acceleration performance.
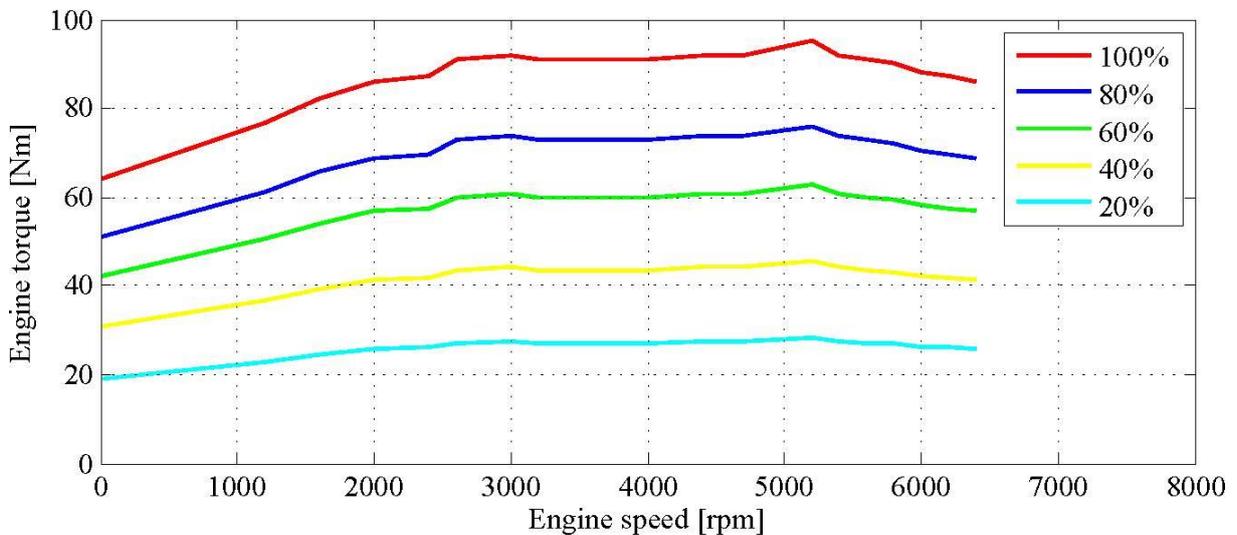
Figure 4 - Engine torque curves adapted from [20]

The inference of fuel consumption is based on three variables: engine speed, engine torque or power and engine fuel flow rate or specific fuel consumption [21]. In the simulations it is used a Otto cycle gasoline engine model, and the fuel consumption is estimated by the specific consumption map shown at the Figure 5 as a function of engine speed and torque.
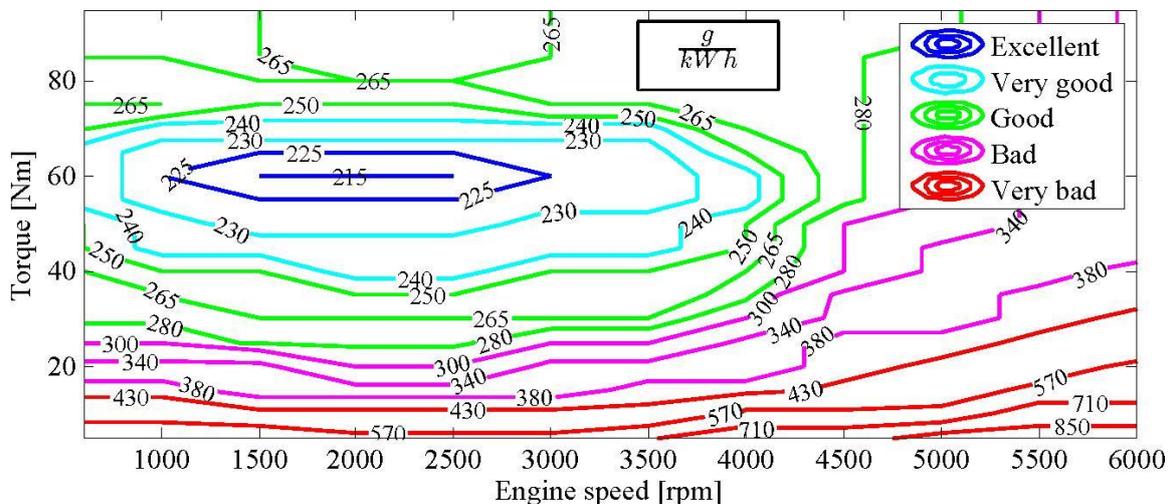
Figure 5 - Engine fuel consumption map adapted from [20]

The volumetric fuel consumption C_l for each simulation step dt is calculated by Equation (9), as a function of the engine power P_e, the fuel density ρ_f and the fuel specific consumption C_e obtained from the consumption map (Figure 5). The total fuel consumption is given by the sum of all the step consumptions.

$$C_l = C_e \frac{P_e\, dt}{\rho_f} \tag{9}$$

4. DYNAMIC CO-SIMULATION

According with [22] due to the increasing demand of quality and performance for the actual vehicles, the traditional design approach based on a sequential design of the components can no longer be applied, therefore engineers need to model and simulate the dynamic response of the whole system, taking into account the simultaneous interaction phenomena between components.

As discussed at [23] the co-simulation allows the development of a control strategy and the evaluation of various aspects of a vehicle system. This technique is used in the stage of project of a system when the physical or mathematical and mechatronic control system are designed [24].

The co-simulation platform must support the communication between two or more softwares [25]. The main technical difficulty is synchronizing the used programs in both directions. This kind of dynamic simulation allows a high detailed degree, keeping a good performance. However, to much detailed models tends to incur a high computational cost, so the simulation should provide only the needed information [26].

In this paper the simulations was performed between the multibody dynamic analysis program Adams[TM] (Automatic Dynamic Analysis of Mechanical Systems), that contains the vehicle model, and Matlab/Simulink[TM] in which it is implement the longitudinal vehicle dynamics equations described previously.

The simulations are base in a compact hatchback vehicle equipped with 1.0L engine as it can be seen at Table 1. The implemented model was designed based on a dynamometer bench in order to enable future experimental validations. The effects of vehicle suspension system were neglected to simplify the model because these factors are disregarded by the current literature.

Table 1 – Simulated Vehicle Parameters

Components	Units	Speed				
		1st	2nd	3rd	4th	5th
Engine inertia	kgm^2	0.1367				
Gearbox inertia	kgm^2	0.0017	0.0022	0.0029	0.0039	0.0054
Gearbox ratios	-	4.27	2.35	1.48	1.05	0.8
Differential inertia	kgm^2	9.22E-04				
Differential ratio	-	4.87				
Wheels + tires inertia	kgm^2	2				
Vehicle mass	kg	980				
Tires	-	175/70R13				

4.1. Adams^{TM} Model

4.1. Adams^{TM} Model

The model is based on two rolls set to simulate the vehicle longitudinal displacement inertia, in which four cylinders represent the vehicle wheels. The CAD model is exported to Adams^{TM}, where appropriate revolution joints are created to allow the wheels movement and rotating masses, as shown in Figure 6.
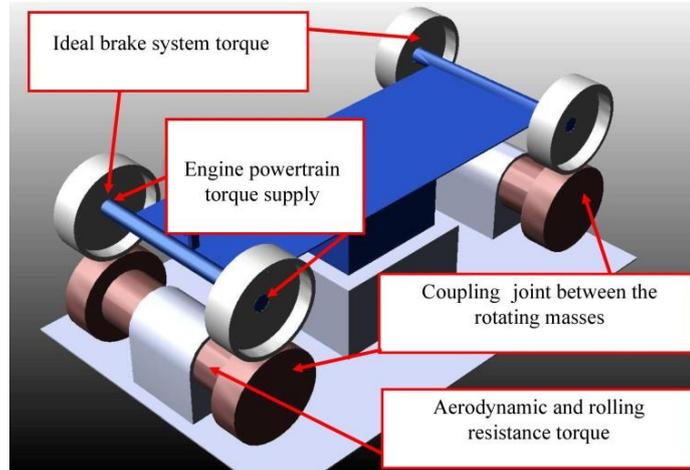
Figure 6 - Adams^{TM} model

On the front wheels are applied torques related to the power supplied by the powertrain. The brake system acts in the front and rear wheels of the model when the required torque becomes negative. In the rotating masses, movement resistance torques are applied referent to the aerodynamic drag, rolling resistance and powertrain inertia. In the model, the vehicle chassis is 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.

4.2. Matlab/Simulink^{TM} Model

To facilitate the implementation of the vehicle dynamics equations, it is used an interface between the Adams^{TM} model and the Matlab/Simulink^{TM} in which a block that contains the data from the dynamic model is generate as shown in Figure 7.
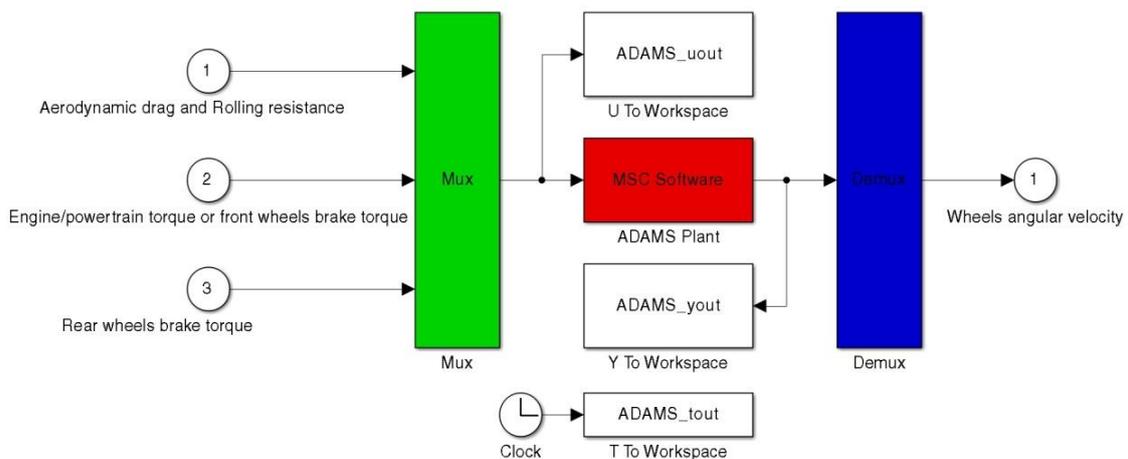
Figure 7 - AdamsTM generated block

The Simulink^{TM} programmed algorithm works together with Adams^{TM} solver. Simulink^{TM} provides for Adams^{TM} torque values applied in the wheels and in the rotating masses. Adams^{TM} generates a response from an angular velocity of the wheels, which supplies the Simulink^{TM} algorithm to recalculate the required torque according to the new demand.

5. GEAR SHIFTING STRATEGIES

To represent the vehicle behavior it is necessary to define some specific parameters to simulate the driver interaction with the system, to avoid gear-shifting instability. In the simulations performed was used a gear shifting time of 1 s as proposed by [27].

Adding a certain time between two subsequent gearshifts is also important for stabilization. This is required to avoid chattering and to satisfy comfort conditions [28]. The downshift occurs at 5 km/h below the upshift speed, as proposed by [29] to prevent gear shift instability.

The gear shifting strategies optimize the engine behavior to operate close to the optimum considering any of these three characteristics, maximum acceleration (maximum torque), higher speeds (maximum power) and fuel economy (minimum specific consumption). Usually the most used strategy is the best economy and efficiency [3].

The maximum power and torque tactics consist in increase the vehicle acceleration performance, usually this strategies act at engine higher speeds regimes, where the engine efficiency is lower, generating a fuel consumption increase.

To reduce the fuel consumption [8] evaluated experimentally the variability of driver behaviors, in relation with the engine speeds amplitude according to the driving style of 21 drivers. After that he reproduced the tests using ecodriving techniques. In situations where the fuel consumption is more important than the vehicle performance it is used a fuel economy strategy. Thereby this tactic is more difficult to define because it depends on a large number of factors like the engine regime, driver behavior, available gear ratios, required acceleration and fuel type.

Taking into in account these parameters, the fuel economy tactics consist in keeping the engine running at higher efficiency regions. In this paper it is used primarily the gear shifting speeds proposed by [30] for a similar vehicle that is simulated herein. The tactics consists in gear shifting when the vehicle reaches a predetermined velocity. In all the recommended gear shifting speeds the upshift occurs when the engine speed approches to 3000 rpm.

The maximum torque strategy consists in gear shifting when the engine reaches the maximum available torque. According to the Figure 4 the maximum engine torque is close to the 5300 rpm. To keep this engine regime it is defined the gear shifting speeds limits based on the maximum torque engine speed.

In the same way the maximum power tactics is at the 6400 rpm engine speed, and the gear shifting speeds limits are defined in function of this operational regime. Due to the high engine speed range available between the fuel economy strategy and the gear shifting at maximum engine torque, it is included two intermediate strategies to make gear shifting at 3500 and 4500 engine rpm.

6. RESULTS

To create a parameter to compare the simulated results and the standard velocity profile, it is used the linear correlation. 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 Equation (10), where x and y represent the curves values.

$$r^2 = \frac{\left(\Sigma(x_i - \bar{x})(y_i - \bar{y})\right)^2}{\Sigma(x_i - \bar{x})^2 \Sigma(y_i - \bar{y})^2} \tag{10}$$

The regression r^2 measures the variability proportion between the two curves, therefore, it 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.

Because the engine decoupling during the gear shifting process and the tire traction limit, the vehicle remains at speeds lower than the required at some stretches of the cycle, therefore the total distance traveled is affected by the vehicle performance. This causes differences in the results, because in the simulations it is used the fuel economy strategies for a shorter distance compared to the tactics focused on the vehicle acceleration performance, generating differences in the fuel consumption average per km traveled.

The results obtained from simulations of the gear shifting tactics described previously are shown in Table 2. As the simulation objective is follow the standard velocity profile imposed, correlations are always close to 1, and the differences are found in the third decimal place.

Table 2 - Correlations and fuel consumption according to the gear shifting strategy

Gear Shifting Tactics	Linear Correlation	Fuel Consumption (l)	Traveled Distance (km)	Consumption Average (km/l)
Fuel economy	0.9978	1.057	11.860	11.22
Gear Shifting at 3500 rpm	0.9980	1.126	11.875	10.55
Gear Shifting at 4500 rpm	0.9982	1.326	11.881	8.96
Maximum torque	0.9983	1.439	11.882	8.26
Maximum power	0.9984	1.687	11.890	7.05

The Table 2 results prove that the gear shifting tactic that improve the vehicle performance also increase the fuel consumption. Therefore, the distance traveled by the vehicle is above the distance travelled using the other strategies.

To optimize the fuel consumption without decrease the vehicle performance was develop a gear shifting optimization algorithm, that merge the tactics used in previous simulations to adjust them to the cycle stretches that they are more adequate.

Were developed two different optimizations algorithm. The first is utilized to keep a correlation similar to the maximum power and torque strategies, improving the fuel consumption. In addition, the second algorithm aims to keep the performance of the fuel consumption tactic and increase the average fuel consumption by changing the gear shifting profile.

7. GEAR SHIFTING OPTIMIZATION ALGORITHM

The optimization algorithm starts with the results of the strategies simulated previously, that are stored in a database. To expand the database with more information the optimization algorithm runs more similar to the previous simulation, which some gear shifting speeds are defined to control the vehicle gearbox. At every optimization step it is stipulated two engine speeds to create the standard gear shifting velocities. The first engine speed is randomly chosen between the engine speed range (1000 to 6500 rpm). The second engine speed is also randomly chosen between 1500 to 4000 rpm that represent the good efficiency region of the engine and presents results that do not decrease the vehicle performance.

After the optimization algorithm analyses the database results, it creates a vector to control the simulation gear shifting strategy. This vector is based on the engine speeds regimes used to calculate the gear shifting velocities as described previously.

With this vector, a new simulation is performed. The results found are stored at the database to create the new gear shifting control vector in the next optimization step.

Every simulation result stays at the database at least 30 steps, when this limit is reached; this result is eliminated if the performance or the fuel consumption is unsatisfactory according to the limit previously defined.

7.1. Performance Optimization

The best correlation found by the previous simulated strategies is the gear shifting at the engine maximum power point. This is due to the diminished number of gearshifts compared to another tactics as can see at the Figure 8.
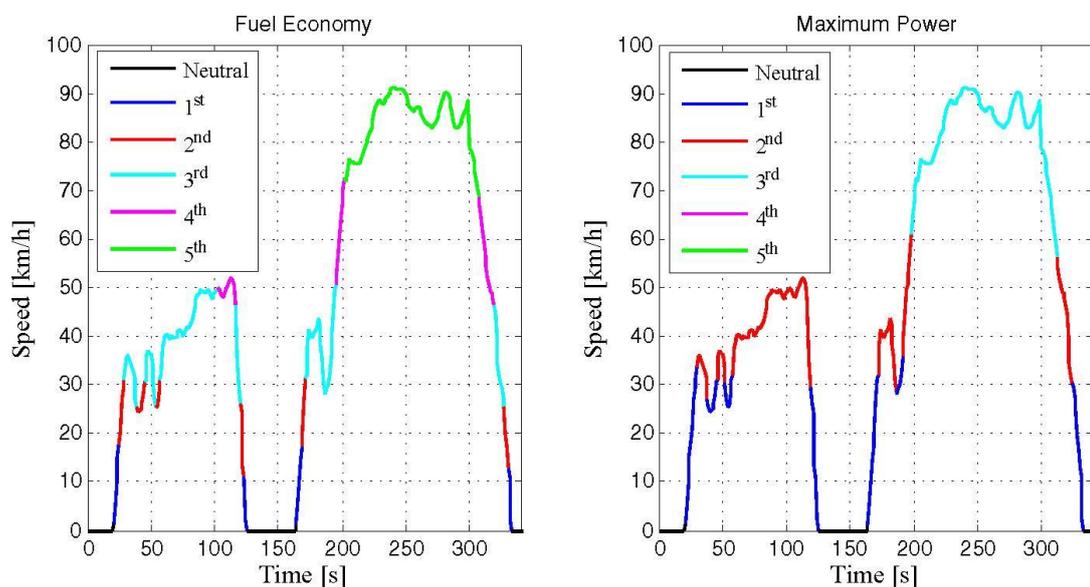
Figure 8 – Gear shifting profile using the fuel economy and maximum power tactics

For each gearshift there is a speed decrease that reduce the final traveled distance, and consequently the correlation as compared to the standard velocity profile. However it keeps the engine running at the maximum power regime that operates sometimes over the 5000 engine rpm that is the worst efficiency engine region as can be seen at the Figure 5.

On the other hand is not necessary to use the high acceleration strategies in the entire cycle. The performance optimization algorithm selects the tactics that has the best correlations compared to the standard cycle in a specific stretch of the velocity profile. If some tactics have similar correlation, the algorithm selects the tactics that has the best average fuel consumption.

After the analyses of the database and the choice of the tactics for each cycle stretched the control vector is generated and a new simulation is performed. After 47 interactions the algorithm found a tactics which the result is shown in the Table 3.

Table 3 - Performance optimization results

Gear Shifting Tactics	Linear Correlation	Fuel Consumption (*l*)	Traveled Distance (*km*)	Consumption Average (*km/l*)
Fuel economy	0.9978	1.057	11.860	11.22
Maximum torque	0.9983	1.439	11.882	8.26
Maximum power	0.9984	1.687	11.890	7.05
Performance optimized	0.9983	1.084	11.883	10.96

The optimized performance strategy present a similar correlation, but with 32.69% fuel economy compared to the maximum torque tactics. Even though running more interactions better result are not found. The gear shifting profile found by the performance-optimized strategy is shown at the Figure 9.
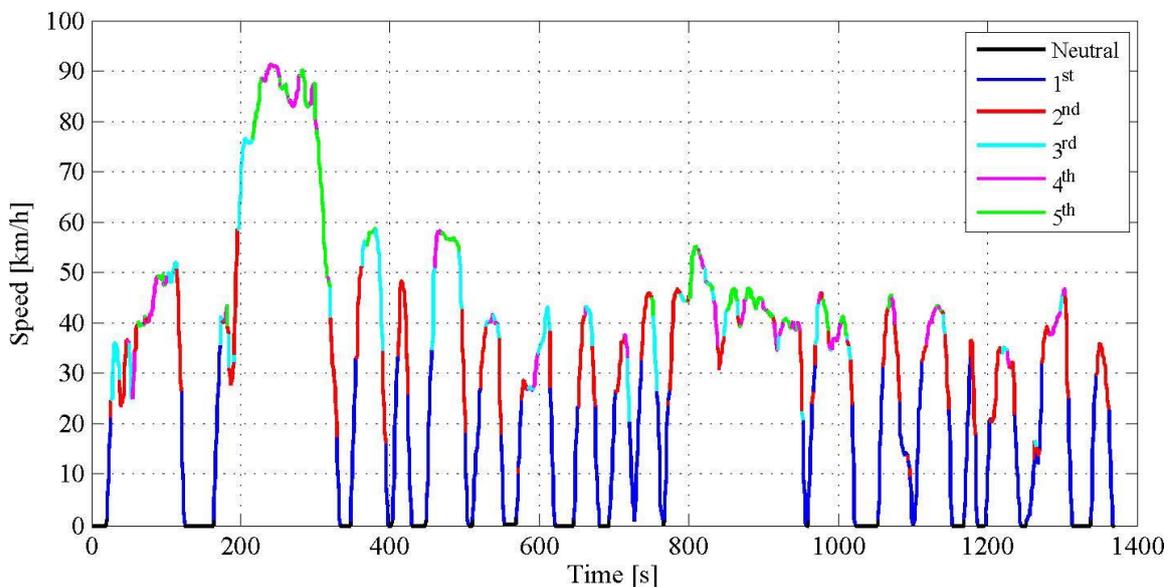
Figure 9 - Performance optimized gear-shifting profile

As can be seen at the Figure 9 the gear shifting profile is not uniform like to one performed by the tactics simulated previously (Figure 8). In some stretches of the cycle the gear-shifting algorithm keeps the same gear for a long period of acceleration to change the transition ratio when the power requested is low minimizing the speed decrease during the process. However when the power request is low, it is better to change the gear not to the next transition ratio but to skip a gearshift and change directly to the gear that represent the best average fuel consumption and can fulfill the power requested in that situation.

7.2. Average Fuel Consumption Optimization

To optimize the fuel consumption it is necessary to consider more factors than only gear shifting speeds. According to the gear shifting process it is possible to keep the engine running at the highest efficiency regions, but sometimes it reduces the vehicle performance.

In this case the algorithm selects the best average fuel consumption between the results off the database on a particular simulated stretch. If the strategies show a similar average, the choice will be made by the beast performance tactics in that condition. In the same way the control vector is generated and a new simulation is performed. The simulation runs 86 interactions to find the results show at the Table 4.

Table 4 - Average fuel consumption optimization results

Gear Shifting Tactics	Linear Correlation	Fuel Consumption (l)	Traveled Distance (km)	Consumption Average (km/l)
Fuel economy	0.9978	1.057	11.860	11.22
Average fuel consumption	0.9978	0.892	11.862	13.29

The average fuel consumption optimization presents similar performance than the standard fuel economy tactics, but with an improvement at the average fuel consumption of 18.45%.

The gear-shifting profile found by the simulation is shown at the Figure 10.
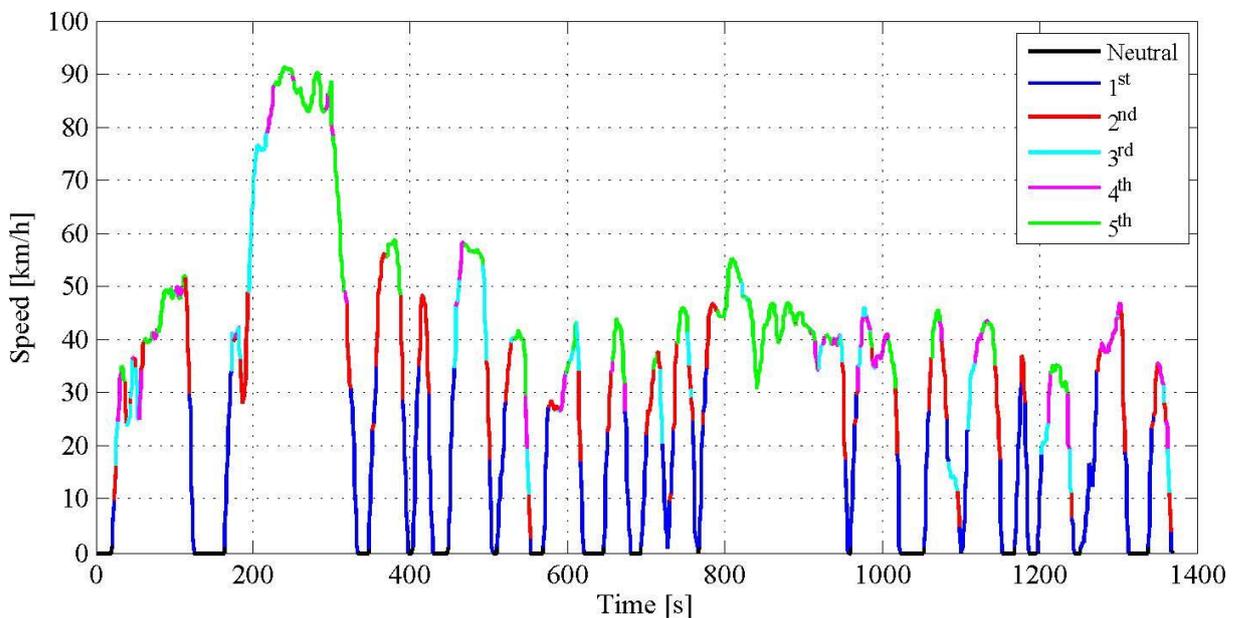
Figure 10 – Average fuel consumption optimized gear-shifting profile

The gear-shifting profile shows that in some stretches is better to keep the current gear to avoid the speed decrease caused by the engine decoupling than demanding more acceleration to return the vehicle to the cycle velocity. On the other hand, in other stretches it is better to upshift quickly or skip a gear in order to change the transition ratio directly to the gear that sets the engine regime to the best efficiency region.

CONCLUSION

In this paper it is studied the influence of the gear shifting strategy in the vehicle longitudinal dynamics at the Brazilian urban driven cycle NBR6601.

The results are obtained by co-simulations between the multibody dynamics analysis software Adams^{TM} and Matlab/Simulink^{TM} in which it is implemented the vehicle movement model and the gear shifting control.

The first simulations consider standards gear shifting strategies described at the literature. Some tactics improved the vehicle performance, but increases the fuel consumption. The fuel economy strategies present a good correlation and an acceptable performance.

The differences in the results are based in the number of gear shifting at the cycle. According to the gear shifting strategy used the engine runs at different regime. If the gear shifting occurs at high speeds of the engine, the vehicle performs less gearshifts that increase the performance, but also raise the fuel consumption when the engine operates over 4500 rpm.

When the gear shifts, the clutch decouples the engine and the gearbox occasioning a speed decrease. When the system recouples, the vehicle needs to accelerate to return to the cycle speed, but sometimes this acceleration is limited by the tire-ground contact and the vehicle stays below the cycle speed due to not using the total engine power.

To optimize the simulated vehicle performance and fuel consumption it is developed two algorithms that merge the gear shifting tactics. The algorithms compare the previous simulated results stored in a database and choose the best tactic for each cycle stretch. After the analysis, the algorithm creates a vector that defines the gear shifting control and runs a new simulation.

The performance optimization algorithm results in a correlation similar to the maximum torque strategy, but with average fuel consumption improve of 32.69% improve compared to the maximum torque tactics.

The average fuel consumption optimization algorithm results in a correlation similar to the standard fuel economy tactics, with a 18.45% improvement at the average fuel consumption.

The gear shifting profiles for the optimization results are different to the standard strategies that keep constants gear shifting velocities. The optimized algorithm changes the gear at the most suitable moment, to prevent gearshifts in situations of acceleration or to keep the engine running at the most efficient areas, sometimes skipping a gearshift.

Finally it is concluded that the knowledge of the gear shifting strategies and the vehicle parameters like the fuel consumption map and the torque curves are crucial to define in the vehicle dynamics model.

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