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NUMERICAL ONE DIMENSIONAL SIMULATION AND VALIDATION OF A VEHICLE USING NBR6601 U.S. FTP 75

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Abstract. *The numerical vehicle simulation is very important in engineering for validation of prototype concept, checking the results on different drive cycles and dynamic situations as drivability tests, before the construction phase. The main objective of such numerical model is to predict the performance, fuel consumption and to help aiding the creation of base criteria to choose the best powertrain configurations, using computational hours rather than engineer hours, reducing design costs and assembly work, as well as the project time. This time economy is further used to meet other important project targets and to deliver the best technology for each specific project. Furthermore, the literature presents different strategies based on dynamic and bench tests for the engine power and torque maps in order to improve the vehicle performance, because of the powertrain inertia, the engine operation and the calibration maps. The specific map strategies based on fuel economy and optimization are very complex to be established due to large number of factors that can influence the vehicle behavior, the engine state and the driver feeling, required acceleration and vehicle speed in different points of the cycle. In this paper, it was analyzed the importance of different engine/driver maps to optimize the model, with objective to show that utilizing the real data acquired from benchmark projects, it is possible to deliver a really validated model, basic and simple, with results less than 1% error. The control maps were verified using INCA program to get DATA from a real car. The FCA database was used to check the engine fuel consumption to the standard cycle U.S. FTP-75 NBR6601. The simulations were performed using GT-suite software where the model based on a conventional 1.3L vehicle was developed. The simulation results were stored in a database to compare them with the measured results obtained within dynamic tests on dynamometer of a similar vehicle configuration. The objective is to prove, that is possible during the development/calibration phases the creation/utilization of basic vehicle simulations and models, helping to meet and improve the calibration targets, increasing the software complexity.*

Keywords: *Powertrain, simulation, fuel consumption, calibration*

1. INTRODUCTION

The project consists in creating a very simple vehicle model using GT-suite database and compare the results with real DATA acquired from a similar car, the objective is to show that is possible to use real parameters to meet the targets of consumption and check the calibration using simulation models.

The effort to improve the fuel economy of vehicles has been an ongoing process in the automobile industry. Fundamentally, the techniques used include the following aspects: Reducing vehicle resistance, improving engine

operation efficiency (Improved with calibration), properly matched transmission (Hardware and improvement of calibration when with automatic/automated transmission) and advanced drive trains (Hybridization) (Ehsani, 2018). Several technologies have been developed since the 1990s, when worldwide emissions standards started to impose boundaries on the levels of acceptable emissions of vehicles, which resulted in ever-reducing levels of emissions, as well as fuel consumption (Andwari et al, 2013);(Teng et al, 2007). In order to develop a numerical model in order to meet the project targets, the vehicle need to be studied since the fuel combustion to the tires.

The vehicle acceleration performance is limited by two factors: in the sprint condition, the acceleration is limited by the tire limit traction, and in high speed, the vehicle performance is limited by power available in the engine (Gillespie, 1992); (Spanos, 2012). Other fact that influences the performance of a vehicle and the fuel consumption during the process of gear shifting is the clutch uncoupling. The clutch is responsible for transferring the engine torque to the gearbox (Naus, 2010) and, consequently, the gearbox transfers this torque to the vehicle wheels. When the clutch disengages, it permits the synchronism of the gears in the gearbox during the gear shifting process. After the synchronism process, the clutch is gradually coupled to the engine, re-establishing the torque transmission to the vehicle tires (Zhao, 2014). However, during the moment of gear shifting, the engine torque is not transmitted to the gearbox, decreasing the speed of the vehicle, fact that increases the acceleration asked after clutch re engagement. As a result, it is a power dissipation caused by torque interruption, the gear shifting time must be as short as possible (Chen et al, 2011); (Gao et al, 2011).

In this project, it will be presented a study of how a new HEV (Hybrid electric vehicle) concept can be used with simulation, to improve the project time, as well as on fuel consumption in a specific vehicle, when submitted to the standard cycle of Brazilian conduct NBR6601 (ABNT, 2005). In this specific study, it is considered a five-speed manual gearbox applied to a compact hatchback vehicle with a 1.3L engine. The aim of this project is to compare the consumption of fuel, the drivability feeling during shifts and the project advantage utilizing a simple vehicle model to create different new concepts. This project want to prove that is possible to calibration engineers use basic simulation in their personal computer, to check calibration parameters, before the dynamic tests, reducing the embed test time, improving the quality of calibration (it is possible to check and input new functions using this computational simulation) and the associated costs.

2. VEHICLE EQUATIONS

This section shows the main forces acting on the vehicle and how they act on the system.

2.1 Longitudinal Vehicle Model

Considering the real world, a vehicle not only travels on a level road but also up and down the slope of a roadway as well as around corners. In order to create a model to this motion, the description of the roadway can be simplified by considering a straight roadway with two-dimensional movement. This two dimensional model will focus on vehicle performance, including acceleration, speed, and gradient, as well as braking performance. (Emadi, 2014)

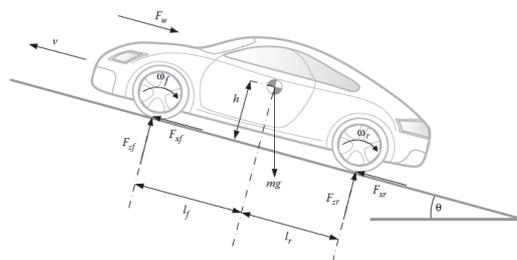


Figure 1: shows the forces acting on a vehicle as it travels at a given speed along a roadway with a specific grade.

Fundamental principles of mechanical systems can be used to express the relationship between the vehicle acceleration and the forces acting on the vehicle body as:

$$ma = F_t - F_w - F_g - F_r \quad (1)$$

Where m is the vehicle mass, a is the acceleration of the vehicle. F_t is the total tractive force acting upon the vehicle body, F_w is the aerodynamic drag force, F_g is the grading resistance force, and F_r is the rolling resistance force. (EMADI, 2014)

2.2 Rolling Resistance

The rolling resistance force is the force due to the moment, which opposes the motion of the wheel, and always assists in braking or retarding the motion of the vehicle. The equation for this force is a function of the normal load F_z and the rolling resistance coefficient f_r , which is derived by dividing the distance the normal force due to the road is shifted by the effective radius of the tire r_d (Emadi, 2014).

$$F_r = F_z f_r \cos(\theta) \quad (2)$$

2.3 Aerodynamic drag

According to (Emadi, 2014), the external aerodynamic resistance is comprised of two components, shape drag and skin friction. The shape drag arises from high-pressure areas in front of the vehicle and low-pressure areas behind the vehicle that are created as the vehicle propels itself through the air. These high-and low-pressure zones act against the motion of the vehicle, while the skin friction is due to the shear stress in the boundary layer on the surface of the body of the vehicle. In comparison, shape drag is much larger in magnitude than skin friction and constitutes more than 90% of the total external aerodynamic drag of a vehicle. Aerodynamic drag is a function of effective vehicle frontal area, A , and the aerodynamic drag coefficient, C_d , which are highly dependent on the design of the vehicle body:

$$F_w = \frac{1}{2} \rho A C_d (V + V_w)^2 \quad (3)$$

2.4 Grading Resistance

As a vehicle travels up or down an incline, gravity acting on the vehicle produces a force which is always directed downward, as shown in Figure 1. This force opposes the forward motion during grade climbing and aids in the forward motion during grade descending. In typical vehicle performance models, only uphill operation is considered as it resists the total tractive force. The equation for this force is a function of the road angle θ , vehicle mass m , and the gravitational acceleration g : (Emadi, 2014)

$$F_g = mg \sin(\theta) \quad (4)$$

For a relatively small angle of θ , $\tan \theta = \sin \theta$. Using this approximation, the grade resistance can be approximated by $mg \tan \theta$, or mgG , where G is the slope of the grade (Emadi, 2014). Once the driving cycle NBR6601 does not provide any information about the road degree, this factor is neglected in the current equating.

2.5 Vehicle Inertia

The vehicle longitudinal displacement and the acceleration of the rotational inertia of the vehicles drive unit (Powertrain) also influences on its power demand (Eckert et al, 2015). The equation (5) (Eckert et al, 2014) represents the inertia influence of the drive unit [N], according to the vehicle longitudinal acceleration (α_x) [m/s²] and transmission ratio.

$$I_p = ((I_e + I_p) N_t f^2 + I_d N_f^2 + I_w) \frac{\alpha_x}{r^2} \quad (5)$$

Where:

I_e is the engine inertia [kgm²]; I_p the gearbox inertia [kgm²]; N_t the total gear ratios [-]; I_d the differential inertia [kgm²]; N_f the differential gear ratio [-]; I_w the wheels and tires inertia [kgm²]; r the tire outer radius [m];

2.6 Acceleration Performance and Traction Force

The traction force available in the vehicle (F_t) [N] is calculated by the equation 6 (Gillespie, 1992) as a direct function of engine output torque (T_e) [N], overall transmission ratio ($N_t f$), efficiency ($\eta_t f$) and the tire outer radius (r) [m].

$$F_t = \frac{T_e N_t f \eta_t f}{r} - I_p \quad (6)$$

When the vehicle performance is limited by the available engine torque, the vehicle acceleration (αx) [m/s^2] can be calculated by the equation (7) (Gillespie, 1992) as a vehicle mass function (M) [kg], traction force available (Ft) [N], aerodynamic drag (Fw) [N] and the rolling resistance (Fr) [N].

$$\alpha x = \frac{Ft - Fw - Fr}{M} \quad (7)$$

Considering the acceleration of the vehicle as an input to simulate the driver's attitude based on the standard speed profile proposed in the NBR6601. The required acceleration (αr) [m/s^2] is calculated by the difference between the current speed (Vv) [m/s] and the speed proposed by the driving cycle (Vc) [m/s] as shown in equation (8) (Gillespie, 1992).

$$\alpha r = \frac{Vc - Vv}{dt} \quad (8)$$

Due to the use of a fixed simulation step (dt) [s] it is possible to calculate the required acceleration, without creating a request of higher acceleration caused by a possible short simulation step. The objective speed of the vehicle (Vc) [m/s] is also considered a step (dt) [s] forward to the simulation time, which allows the vehicle to reach the cycle speed requested synchronized with the driving cycle. By the union of equations (6), (7) and (8) is possible to estimate the required torque of the engine (Tr) [Nm], according to the required acceleration (αr) [m/s^2] and the forces of resistance to movement, as shown in the equation (9) (Gillespie, 1992).

$$Tr = \frac{\left(Mar + \frac{(Ie + Ip)Nt f^2 + IdNf^2 + Iw}{r^2} + Fw + Fr \right) r}{Nt f \eta t f} \quad (9)$$

2.7 Tire traction limit

When the vehicle is accelerating or braking on flat ground, which results in a longitudinal mass transfer. In vehicles propelled only by the front wheels, this mass transfer during acceleration of the vehicle can reduce the tire traction limit and decrease the performance of the vehicle as shown in (Eckert et al, 2014). Equation (10) shows the maximum grip ($Fmax$) [N], according to the geometry of the vehicle and tire coefficient of friction (μ).

$$Fmax = \mu Wf \quad (10)$$

The weight force acting on the front axle of the vehicle (Wf) [N] is calculated by the equation (11) proposed by (Jazar, 2008).

$$Wf = \frac{Wc}{2L} - \frac{Whax}{2Lg} \quad (11)$$

Where:

L is the length between the axis [m]; h is the gravity center of the vehicle height [m]; g is the gravity's acceleration [m/s^2]; c is the distance between the center of gravity and the rear axle of the vehicle [m];

If the performance is limited by the tire traction limit, the acceleration of the vehicle is calculated using the equation (7) replacing the traction force available (Ft) [N] for the maximum grip ($Fmax$) [N]. In this situation, the acceleration of the vehicle decreases and, consequently, the front weight force (Wf) [N] and the limit traction of the tire increases. Due to this, a loop is generated between the equations (7), (10) and (11) until the convergence acceleration of the vehicle and the traction limit.

2.8 Engine Fuel Consumption

To an engine which consumes real fuel and for which the instantaneous fuel consumption is given as:

$$\dot{m} f(t) = \frac{P_{eng}(t)}{\eta_{eng}(t) Q_{lhv}} \quad (12)$$

Where Q_{lhv} (MJ/kg) is the fuel lower heating value (energy content per unit of mass), $\eta_{eng}(t)$ is the engine efficiency, and $P_{eng}(t)$ is the power produced by the engine when it operates at a certain efficiency. (Eckert et al, 2014) Internal combustion Model

3. VEHICLE MODELING

This section shows how the main models used to simulate the vehicle using GT-Suite.

3.1 Engine Model

The engine model was developed using charts and tables containing the engine data, as soon as calibration, where you can change the parameters to achieve the efficiency and fuel consumption. It was based on the GT-suite model of efficiency.

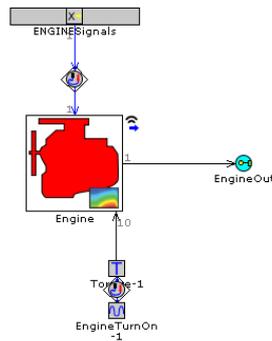


Figure 2: Engine model

In order to deliver to simulate the Start-Stop function was create a PID controller that send a torque signal to the engine.

3.2 Transmission Model

The transmission model was choose in order to perform the best comparison with the real car, it was based on charts containing the transmission data, where you can calibrate and change the parameters to achieve the efficiency and fuel consumption.

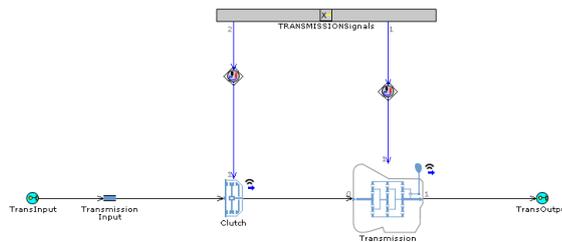


Figure 3: Transmission model

In the transmission model, was used a basic manual transmission from the GT-Suite Library, where was change the ratio and clutch efficiency to achieve the objective.

3.3 VEHICLE MODEL

The vehicle model was adapted from a vehicle of the GT-Suite Library, where was changed tires, brakes, vehicle mass, the differential and the signals of road.

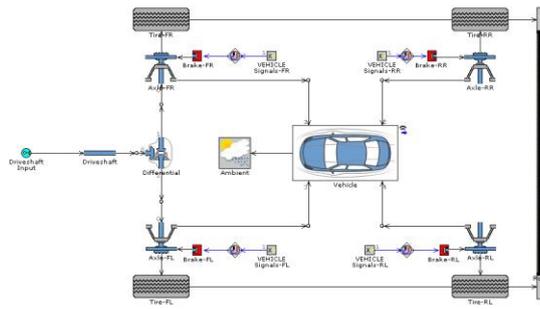


Figure 4: Vehicle model

3.4 DRIVER MODEL

The driver model was adapted from a vehicle of the GT-Suite Library, where was correlated the signals to the NBR6601 and was imposed the gear ratio, the clutch slip and shift points.

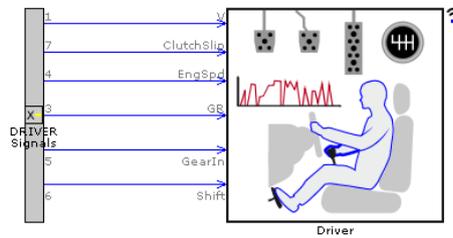


Figure 5: Driver model

3.5 ECU MODEL

The ECU model was adapted from a vehicle of the GT-Suite Library, where was correlated the signals to the NBR6601 as engine speed, driver pedal position and ignition.

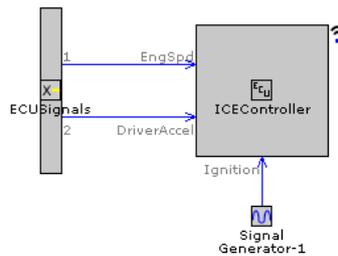


Figure 6: ECU model

3.6 Driving Cycle

The standard driving cycles are used to determine the behavior of the vehicle speed. Normally, the urban driving as characterized by low speed, low engine load, low temperature of exhaust gases, low pedal percentage (between 0-40% max), with the objective to simulate the real urban stile of drive. For this study, it was used the Brazilian NBR 6601 (ABNT, 2005), as show in figure 7.

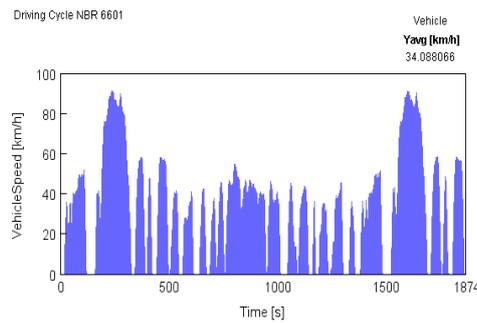


Figure 7: Speed profile in the NBR6601 cycle in GtSuite

4. SIMULATION

To simulate the vehicle dynamics, it was used the same software as mentioned before as modelling tool, and a vehicle available in the library base with some modifications to adaptation. Figure 8 demonstrates the first level of the deployed model. This model shows ECU (Engine Control Unit) used, the structure with engine/transmission/vehicle/driver and how it's disposed with specific force demand. Regarding that the equations of control are disposed in the intern structure of each component.

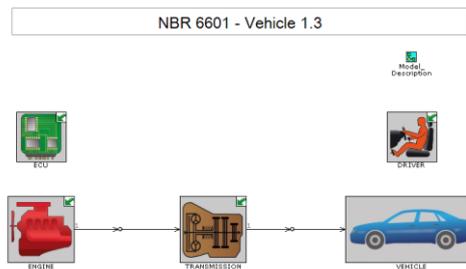


Figure 8: Vehicle architecture

The simulated vehicle is based on a compact hatchback equipped with 1,3L engine, as can be seen in table 1.

Table 1. Vehicle data

Displaced volume	1332 cc
Cylinder's number	4
Max Power (ABNT)	101 hp (Gasol) / 109 hp (Ethanol)
Max Torque (ABNT)	13.7kgfm (Gasol) / 14.2kgfm (Ethanol)
Compression ratio	13,2:1
Number of Valves	8
Ignition Order	1 - 3 - 4 - 2
Vehicle mass	1455 kg
Transmission ratio	1 - 4.273/2 - 2.316 /3 - 1.444 /4 - 1.029 /5 - 0.795

For the comparison between the real cycle and simulate cycle, as used a measurement of a similar car.

The results of Simulation were that the fuel consumption in grams was 2066, 66. Consider that the model is not using the loads of accessories, because of this, the fuel consumption will be better.

5. RESULTS

As described in the previous section, the calculations were performed for the working condition originally proposed in the computational model, the idea was to correlate the DATA of a similar car with same calibration parameters and in order to check the fuel consumption, the engine speed, the shifts and velocity profile. In contribution to validate the computational model constructed in this work.

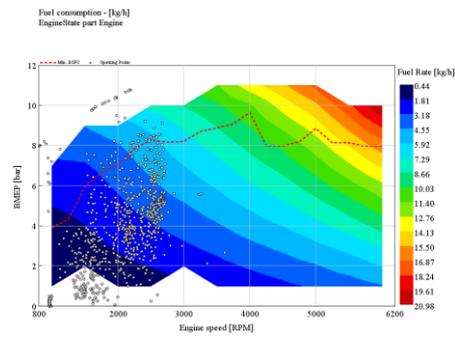


Figure 9: Engine state with the specific operational points

The figure 10, shows the Engine state of the engine in the simulation designed with the specific operational points. In the left side is the BMEP relative, the brake mean effective pressure and the engine speed in RPM and in the right side is the fuel rate relative.

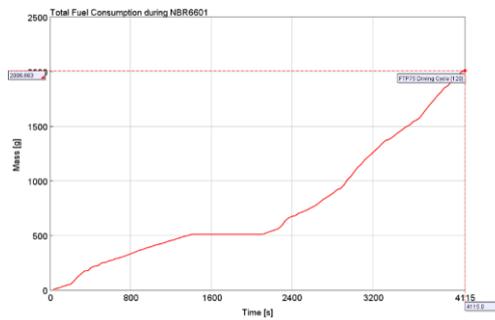


Figure 10: Fuel consumption of the simulation model

The figure 10 shows the total fuel consumption during the NBR6601 FTP 75, with the sum of 2006, 66 grams of gasoline during the cycle.

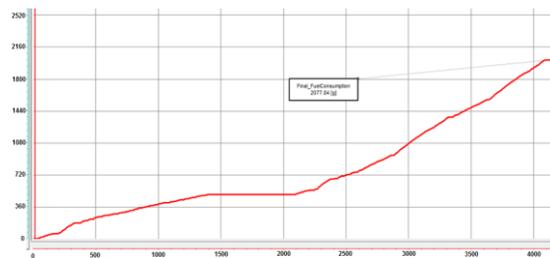


Figure 11: Fuel consumption measured from the similar car using ETAS MDA

The result of similar car was 2077, 88 grams, calculated using the DATA of the real vehicle. The measuring was made using INCA software and development ECU of the car. It was performed in the same conditions as the simulation. The axis are time in 'x' and fuel consumption in 'y'.



Figure 12: Velocity profile generated by the simulation model

The velocity profile calculated using the computational simulation is above. Below is the measured graph, where the time is in 'x' and vehicle velocity in 'y'. As soon as in the simulation.

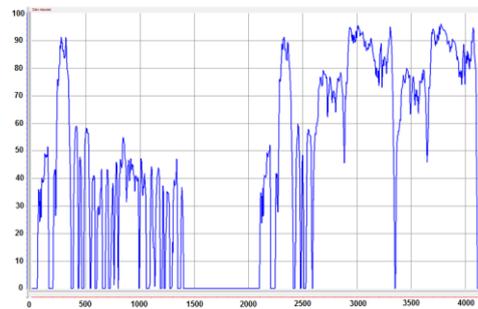


Figure 13: Velocity profile measured of the similar car

The gear profile from simulation is in figure 14, as soon as the real gear profile in figure 14. The 'x' axis is always time, and the 'y' is the gear selected or gear engaged at each different second time.

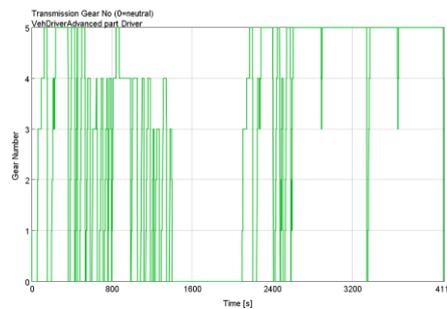


Figure 14: Gear profile generated by the simulation model

As possible to see, the gearshift points are respected, with a small difference between the second phase of NBR6601 FTP75 because of the measurement time that acquire the clutch engage and disengage.



Figure 15: Gear profile measured of the similar car

6. CONCLUSION

The simulation model is the first step to validate the structure of the future prototypes or projects and it can be always useful during the development phases. As presented in the results, the simulation was 3.42% better than the real car (2077.57 grams in the real car and 2066.88 grams in the simulation), if you consider that was not used the accessories electric loads (2% difference with load), it is possible to say that for a preliminary analyses, the model is very important. For calibration engineers, it means that, you can test and improve your calibration points using them in the model and checking the cycle instead of lose at minimum 1.5 hour in the real cycle in laboratory, just to check some small modification. The process time of the simulation was 4.2 minutes, using a Intel core i7 computer. However, it is possible to check more than one parameter and function in a workday. The development cost need to be discussed with these results, because it is possible to improve the time sheet with objective to reduce the delivery costs of releases or the new phases of calibration and hardware. Proving that, if you use the same calibration parameters, you can create a simple model, which you can use to improve your calibration, reducing validation phase and improving the quality, as soon as in SIL.

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