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GEAR SHIFTING OPTIMIZATION STRATEGY FOR BRAZILIAN VEHICLES AND TRAFFIC

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Abstract. *The aim of this paper is demonstrate the effectiveness of gear shifting strategies on performance and fuel consumption of a typical Brazilian popular automobile. A collection of parameters and dynamic equations is described in order to reproduce a dynamic model compatible with a common national vehicle. Two technics are applied; the first is based on an ideal gear map in function of the vehicle velocity and torque, and the second performs an optimization loop above the Brazilian standard driving cycle. Finally it is proposed that the optimized management of gear shifting explores the whole performance and fuel economy potential of the actual popular car.*

Keywords: *Gear Shifting Strategy, Fuel Consumption, Traffic Performance, Longitudinal Vehicular Dynamics, Brazilian Popular Automobile*

1. INTRODUCTION

The vehicular fuel economy is one of the automobiles characteristic that influences directly on the cost of operation. It is not only important in the management of a fleet, but it is also relevant for a single owner of a car. Moreover, the automotive industry has been pressed by governmental regulations aiming the reduction of the fuel consumption, a relevant matter to the national energy management. As an example, the Brazilian government has proposed a target of reduction of 15% on fuel consumption for the vehicles to be sold in this country by 2017 in comparison to 2011. The companies which do not adapt their products will be penalized by a tax increase (FSP, 2012).

Gear shifting strategies produce really significant effect above the fuel consumption and its implementation is immediate in autonomous transmissions, however, in manual transmissions, it depends on an interaction with the driver. Prokop (2001) has suggested that the user has a partial comprehension about the vehicle dynamic, knowledge which is normally poor in details. Thus, the implementation of gear shifting strategies in manual transmission automobile must be assisted by an intelligent interactive system.

Vagg *et al.* (2012), describes that the use of gear shifting indicators, composed of light and/or sound alarms connected to management software installed in the Engine Control Unit (ECU), results in 3.6% of fuel economy in a vehicle submitted to the New European Driving Cycle. Furthermore, Guan and Frey (2012) study the impact of the implementation of a manufacturer independent gear shifting manager. This system is installed in a vehicle of a fleet and is composed of an intelligent control unit that uses heuristic technics for deducing the automobile condition (weight and engine map) and indicates the appropriated gear through a panel. The fleet had its fuel consumption reduced in 11%.

This paper analyses two proposes of gear shifting strategies, one based on an ideal gear map in function of the automobile velocity and the demanded torque, and the other obtained from global optimization.

2. LONGITUDINAL VEHICLE DYNAMICS

Vehicle Dynamics consists in a mathematical description of the interaction between human, automobile and surroundings. In a more strict approach, the Longitudinal Vehicle Dynamics comprehends the study of performance and energetic consumption of an automobile submitted to a driving cycle. Considering a vehicle with mass m_v, traveling with velocity v on tires with radius r_w through a flat plane inclined by an angle α in relation to the horizontal, under gravity g, it will be possible to draw the schematic proposed by (Guzzella and Sciarretta, 2005) on Fig. 1.

The sum of the longitudinal forces presented in Fig. 1 is described by Eq. (1):

$$m_v \frac{d}{dt} v(t) = F_t(t) - \left(F_a(t) + F_r(t) + F_g(t)\right) \quad (1)$$

where: F_t is the Traction Force, F_a is the Aerodynamic Drag, F_r is the Rolling Resistance and F_g is the Climbing Resistance. Each force will be characterized in the following sections.

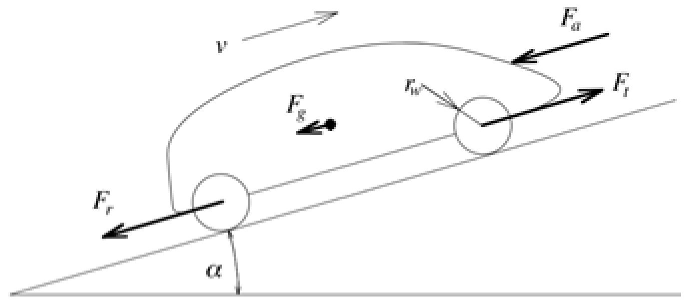

Figure 1. Longitudinal Forces acting on an automobile (Guzzella and Sciarretta, 2005)

2.1 Aerodynamic Drag

While the attacking frontal area penetrates through the air, high pressure zones are generated. As the streamlines proceed down the back side, zones of low pressure are created behind the car, effect that is boosted by the flow separation. This differential of pressure generates a longitudinal resulting force named Aerodynamic Drag (Gillespie, 1992).

The phenomenon of separation between the streamlines and the vehicle surface is a function of the automobile Reynolds Number which is linear to the vehicular shape and the speed of the air flow. As fast as the carcass moves against the air and as anti-streamline as its shape is, bigger are the turbulent region and the differential of pressure (Jazar, 2008).

$$F_a = \frac{1}{2}\rho C_d A(v + V_w)^2 \tag{2}$$

The Aerodynamic Drag is defined by the Eq. (2), where ρ is the air density, C_d is the drag coefficient, A is the vehicular frontal area and V_w is the velocity of the wind, which is positive against the direction of movement. The drag coefficients are obtained empirically and are constant in the range of velocities (and Reynolds Number) of common automobiles, as demonstrated in Fig. 2.

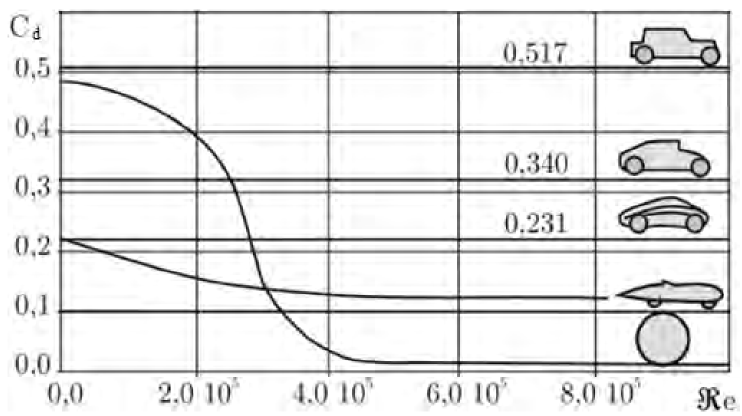

Figure 2. Values of drag coefficient of typical vehicular shapes as a function of Reynolds Number (Jazar, 2008).

2.2 Rolling Resistance

The Rolling Resistance is the dissipative effect caused by the tire carcass conformation in contact with the road and by the friction between the pavement and the rubber. It is treated in literature as the function of the rolling resistance coefficient (f) and the component of gravitational force normal to the road, defined by Eq. (3) (Ehsani *et al.*, 2010).

$$F_r = f.m_v.g.cos(\alpha) \tag{3}$$

The rolling resistance coefficient is a function of the velocity. At low speeds (typically smaller than 9 *m/s*), this function is approximately linear but at moderated speeds (between 9 and 40 *m/s*, the predominant operation mode) this function takes a parabolic shape. Finally, overtaking a critical speed (above 40 *m/s*), the coefficient increases sharply due to the excitation of stationary waves over the tire carcass (Jazar, 2008).

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Other aspects also affect the rolling resistance coefficient, like vehicle's weight, type of tire and its inflation pressure, soil stiffness, temperature and residual braking. Among these factors, the pressure is the most significant when the road is paved (Heißing and Ersoy, 2011). Figure 3 shows the shape of the rolling resistance coefficient curve and the influence of the inflation on it.

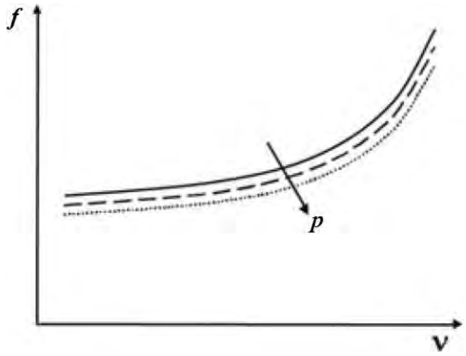

Figure 3. The rolling resistance coefficient curve and the inflation pressure effect.

Genta (1997) has proposed an equation for f, taking into account the vehicle's weight, tire type and inflation pressure, described by Eq. (4):

$$f = \frac{K}{1000}\left(5{,}1 + \frac{5{,}5*10^5 + 90 m_v g cos(\alpha)}{p} + \frac{1100 + 0{,}0388 m_v g cos(\alpha)}{p} v^2\right) \tag{4}$$

where K is a constant in function of the tire type (0,8 for radial and 1 for non-radial), and p is the tire inflation pressure.

2.3 Climbing Resistance

The component of the gravitational force parallel to the road plane is called Climbing Resistance. As expressed by Eq. (5), when α is positive, the car is moving uphill and Climbing Resistance acts against the movement. On the other hand, if α is negative, the car is moving downhill and this force acts in favor of the movement (Heißing and Ersoy, 2011).

$$F_g = m_v . g . sin(\alpha) \tag{5}$$

2.4 Tractive Force

The Traction Force is a system input when it is positive and originated by the powertrain. When this force is negative and the energy is dissipated by a conventional brake and/or by the powertrain, it becomes a system output. The component of Traction Force resulting from the engine, F_{te}, is expressed in Eq. (6), proposed by Gillespie (1992) which applied the Second Newton Law over a conventional powertrain composed by engine, clutch, transmission, differential and wheels:

$$F_{te} = \frac{T_e N_t N_d}{r_w} - [(I_e + I_t)(N_t N_d)^2 + I_d N_d^2 + I_w] \frac{d}{dt} \frac{v}{r_w^2} \tag{6}$$

where T_e is the engine's torque, N_t is the ratio of the transmission, N_d is the ratio of the differential, I_e, I_t, I_d and I_w are the rotational inertias of the engine, the transmission, the differential and the wheels, respectively. I_t and N_t are functions of the selected gear.

Due to the high complexity of the engines, for the sake of simplicity T_e 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 (Ehsani *et al.*, 2010).

The brakes are considered ideal since brake lock and tire slip are neglected because the drive cycle is based on trivial driving conditions. Thus, the components of Tractive Force resulting from them (F_{tb}) are written on Eq. (7):

$$F_{tb} = \frac{T_b}{r_w} \tag{7}$$

where T_b is the brake torque. Therefore, the full equation of the Traction Force is expressed by Eq. (8):

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$$F_t = \frac{T_e N_t N_d + T_b}{r_w} - [(I_e + I_t)(N_t N_d)^2 + I_d N_d^2 + I_w]\frac{d}{dt}\frac{v}{r_w^2} \tag{8}$$

3. DRIVING CYCLES

Various 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 (Ehsani *et al.*, 2010). These tests are made on chassis dynamometers inside a controlled room, where it is possible to replicate the same experimental conditions in all the tests. The chassis dynamometers also allow the input of specified loads at the wheels compatible with the automobile inertia and the movement resistances (Guzzella and Sciarretta, 2005).

4. GENETIC ALGORITHM

Genetic Algorithm (GA) is an optimization procedure which works with the parallel evolution of many points in a cost function, while the other traditional optimization procedures deal with only one point. Thus, it is possible to search in different regions of the domain, simultaneously, converging to a cloud of near optimal global solution (de Carvalho, 2009).

A basic GA starts from a population of points randomly chosen and then some steps are followed (Gen and Cheng, 2000):

a) Fitness: each point of the population is evaluated in the cost function;
b) Selection: individuals are selected for reproduction; as bigger is its fitness, grater are the chances of an individual be chosen;
c) Crossover: the parents chromosomes are combined;
d) Mutation: when an eventual probability prevails, one or more chromosomes are mutated;
e) Acceptance: the descendants are joined to the population and the points with worst fitness are eliminated.

This process is reiterated until a stop criterion is reached.

5. SIMULATION PARAMETERS

The situations analyzed are based on a typical 1.0*l* Brazilian popular automobile submitted to a standardized Brazilian urban driving cycle. The torque and fuel consumption curves of a 1.0*l* engine are presented in Fig. 4. Its respective transmission gear ratios, factory indicated gear shifting velocities and further data about the vehicle are available in Tab. 1.

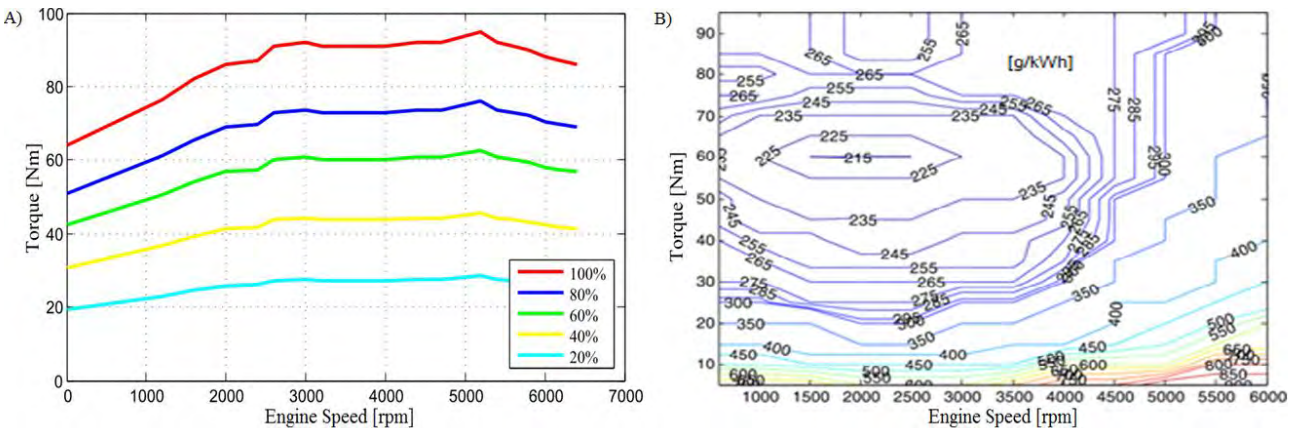

Figure 4. In "A", it is present a typical 1.0*l* torque curve in function of the throttle. The fuel consumption map is shown in "B" (Eckert, 2013)

The simulation driver logic loop rule is the following: the drive cycle velocity is compared with the actual velocity in each time step, the difference between them is divided by a delta of time, resulting in the objective acceleration. This delta of time is tuned as a function of the time of response desired; herein it was defined in 0.5s. The objective acceleration is then inputted in a reverse dynamic problem that returns the objective torque. If this torque is positive, it is attributed to engine torque (T_e), otherwise it is attributed to the brake torque (T_b), composing the Tractive Force. Finally the Tractive Force is inputted in the longitudinal vehicle dynamic system and the response is obtained. If the simulated automobile speedy exceeds a gear shifting velocity, the appropriated gear is selected according each simulation scope. Finally, the driver loop restarts.

Table 1. Simulation Parameters (Eckert, 2013; Haim, 2011)

Parameter	Magnitude	Parameter	Magnitude
Vehicle Mass $\boldsymbol{m_v}$ (loaded)	1206 [*kg*]	Clutch Inertia	3.90e-3 [*kgm²*]
Tire Geometry	205/70R15	Axes, Wheels, Tires, Brakes Inertia	1,724 [*kgm²*]
Tire Radius $\boldsymbol{r_w}$	0.334 [*m*]	Differential Ratio / Inertia	4.87 / 7.44e-4 [*kgm²*]
Tire Inflation Pressure $\boldsymbol{p}$	30 [*psi*]	1st Gear Ratio / Inertia	4.27 / 1,791e-3 [*kgm²*]
Tire Constant $\boldsymbol{K}$	0,8 (radial)	2nd Gear Ratio / Inertia	2.35 / 2,415e-3 [*kgm²*]
Frontal Area $\boldsymbol{A}$	1.8 [m^2]	3rd Gear Ratio / Inertia	1.48 / 3,421e-3 [*kgm²*]
Drag Coefficient $\boldsymbol{C_d}$	0.33	4th Gear Ratio / Inertia	1.05 / 4,782e-3 [*kgm²*]
Air Density $\boldsymbol{\rho}$	1.226 [kg/m^3]	5th Gear Ratio / Inertia	0.80 / 1,07e-03 [*kgm²*]
Gravity $\boldsymbol{g}$	9.81 [m/s^2]	1st / 2nd Gear Shifting Velocity	15 [*km/h*]
Road Inclination $\boldsymbol{\alpha}$	0 [*rad*]	2nd / 3rd Gear Shifting Velocity	30 [*km/h*]
Wind Velocity $\boldsymbol{V_w}$	0 [*m/s*]	3rd / 4th Gear Shifting Velocity	50 [*km/h*]
Engine Inertia	1.58e-1 [*kgm²*]	4th / 5th Gear Shifting Velocity	72 [*km/h*]

6. REFERENTIAL SIMULATION

The simulation of the vehicle with factory indicated gear shifting velocities undergone the Brazilian driving cycle is performed in this section. Its results concerning distance performed per liter of fuel and travel distance are the references for the improvements proposed by the following methods of gear shifting.

The speed profile of the standard cycle is satisfactorily traced by this simulated vehicle, as plotted in Fig. 5. However, there is a lack of power at 5th gear that delays the achievement of the grater velocities. Thus, while the Brazilian urban Cycle is 11990 *m* long, the simulated vehicle travels 11898 *m* and performs 14.056 *km/l*, which are referential for improvements proposed. The profile of used gears is also shown in Fig. 5.

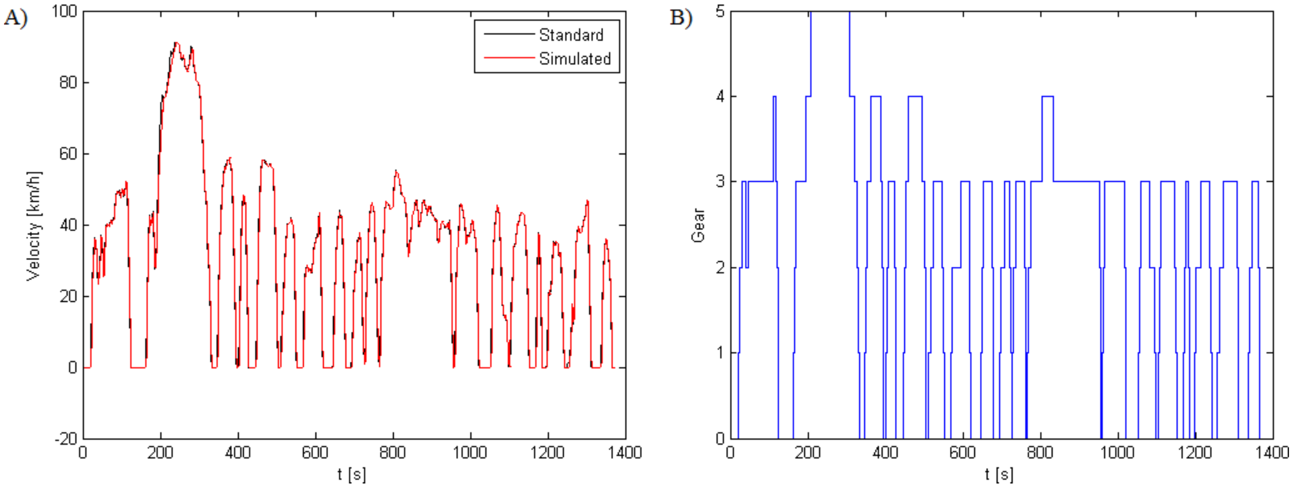

Figure 5. ("A") Velocity and ("B") gear profiles of the referential simulation

7. METHOD 1: IDEAL FUEL CONSUMPTION MAP

Once the powertrain is considered rigid, the engine speed will be proportional to the vehicle speed in function of the selected gear. Thus, it is possible to map which gear provides the minor engine fuel consumption in given automobile velocity and torque demand. Based on this logic and on the vehicle's characteristics (consumption map and gear ratios), the optimum gear map on Fig. 6 was built. It was avoided that the engine was submitted to speeds minor than 1000 *rpm* and major than 6000 *rpm*

This method proposes the substitution of the factory indicated gear shifting by the optimum gear map. However this map assumed an intricate shape what increased the total number of gear shiftings, as can be seen in Fig. 7. This fact and the premature use of the 5th gear, made the simulated vehicle take still longer to reach the higher speeds. Thus the travel distance and the performance per liter fall to 11810 *m* and 13.896 *km/h*, respectively.

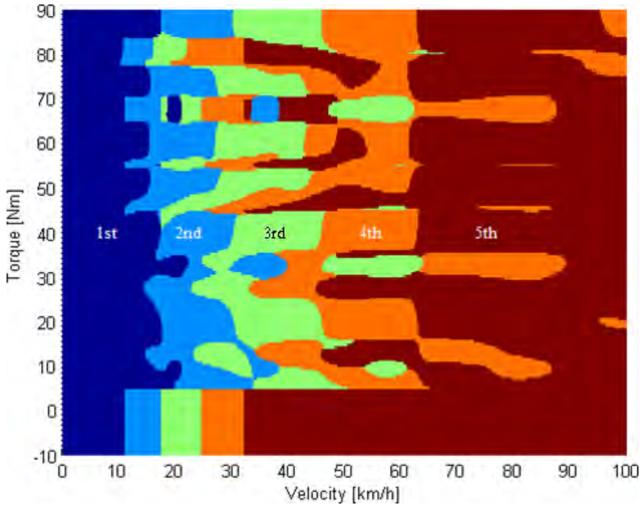

Figure 6. Optimum gear map.

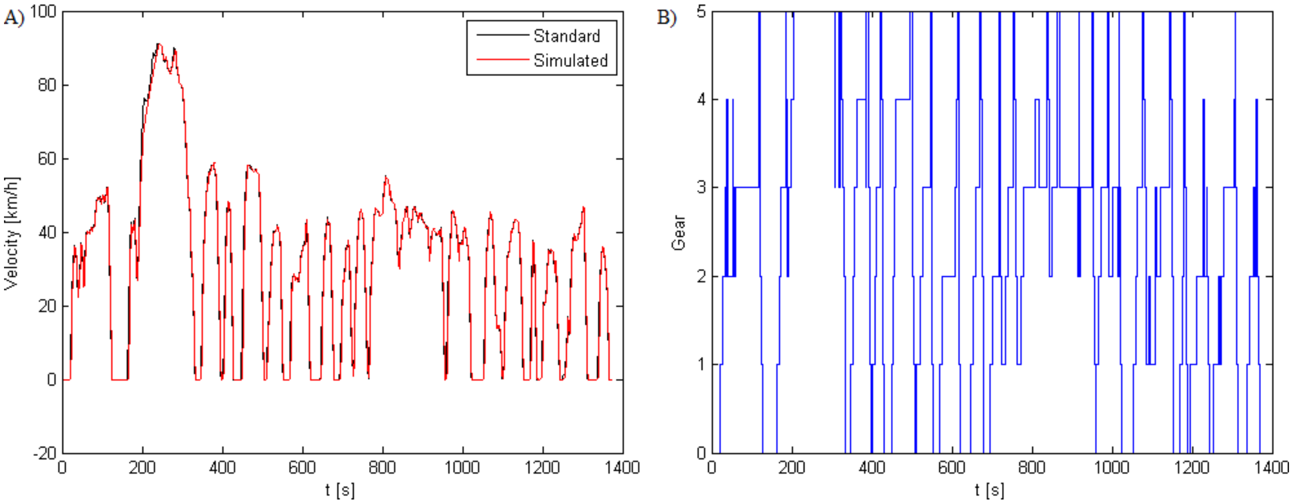

Figure 7. (“A”) Velocity and (“B”) gear profiles of the Method 1

8. METHOD 2: GENETIC ALGORITHM OPTIMIZATION

Table 2. Genetic Algorithm initial population

#	Base	1^{st} to 2^{nd} Gear Shifting Velocity [*km/h*]	2^{nd} to 3^{rd} Gear Shifting Velocity [*km/h*]	3^{rd} to 4^{th} Gear Shifting Velocity [*km/h*]	4^{th} to 5^{th} Gear Shifting Velocity [*km/h*]
1	Engine Speed: 2000 [*rpm*]	12.1	22.0	34.9	49.24
2	Engine Speed: 2500 [*rpm*]	15.1	27.5	43.7	61.6
3	Engine Speed: 3000 [*rpm*]	18.2	33.0	52.4	73.87
4	Engine Speed: 3500 [*rpm*]	21.2	38.5	61.1	86.2
5	Engine Speed: 4000 [*rpm*]	24.2	44.0	69.9	98.5
6	Engine Speed: 4500 [*rpm*]	27.2	49.5	78.6	110.1
7	Engine Speed: 5000 [*rpm*]	30.3	55.0	87.3	123.1
8	Engine Speed: 5500 [*rpm*]	33.3	60.5	96.1	135.4
9	Engine Speed: 6000 [*rpm*]	36.3	66.0	104.8	147.7
10	Factory Indicated Velocities	15.0	30.0	50.0	72.0

The fixed gear shifting velocities perform a gear shifting profile more stable than the one presented by the optimum gear map. However, the factory indicated gear shifting velocities assume a compromise between performance and fuel

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consumption in general conditions. Herein, it is used genetic algorithm to come up with the optimized gear shifting velocities focusing on (a) fuel economy, (b) performance and (c) best compromise under Brazilian urban cycle conditions.

The genetic algorithm proposed works with a 10 member population. A vector with the four gear shifting velocities is the chromosomes of each member. The first nine members of the initial population have their chromosomes built assuming engine gear shifting velocities from 2000 *rpm* to 6000 *rpm*, with 500 *rpm* steps. The tenth member's chromosome vector is based on the factory indicated gear shifting velocities. Table 2 contains the data about the initial population.

Each member is submitted to a fitness function what results in a ranking which is the base of the selection. The best ranked member receives 10 points and the score of each member falls linearly until the last member, which receives 1 point. Then, two different members are drawn for the crossover; the ones with bigger fitness scores have more chances of being picked up.

During the crossover, each parental chromosome has 50% of chances of compose the descendant gear shifting velocity vector, which has 10% of chances of being mutated. The mutation changes the value of the chromosome randomly between 70% and 130%.

The new descendant joins the population, while the last ranked is eliminated. Thus, the loop restarts from the evaluation of fitness of the new population, unless a stop condition is satisfied. Herein, this condition consists in the standard deviation of travel distance be shorter than 0.5 *m* and the standard deviation of performance per liter be shorter than 0.005 *km/l*.

The proposed fitness functions are built based on two ratios: the distance ratio (D_r, which is the proportion between the travel distance of the present simulation - D_p - in relation to the travel distance of the referential simulation - D_{ref}) and the economy ratio (E_r, which is the proportion between the travel distance per liter of fuel of the present simulation - Dpl_p - in relation to the travel distance per liter of fuel of the referential simulation - Dpl_{ref}). They are respectively presented by Eq. (9) and Eq. (10):

$$D_r = \frac{D_p}{D_{ref}} \tag{9}$$

$$E_r = \frac{Dpl_p}{Dpl_{ref}} \tag{10}$$

8.1 Best Fuel Economy

The target of this simulation is to find the gear shifting velocities that improve the distance per liter of the vehicle without reducing the total travel distance, in comparison with the referential simulation. For the attendance of this claim, the fitness function used is the sum of the economy ratio and the distance ratio multiplied by correction constant, as written in Eq. (11). Corrections on the ratios are necessary because they are not proportional, improving the algorithm performance.

$$Fitness = E_r + D_r * 3 \tag{11}$$

The population members that overdo 11905 *m* are eliminated in order to not degenerate the evolution in direction to the maximum E_r, once the travel distance and the distance per liter tend to be antagonistic.

8.2 Best Travel Distance

This simulation aims the search of the gear shifting velocities that optimize the travel distance, keeping the minimal distance per liter of fuel identical to the one performed by the vehicle in the referential simulation. For this purpose, the fitness function is the sum of the corrected distance ratio and the economy ratio, as described in Eq. (12).

$$Fitness = E_r + D_r * 10 \tag{12}$$

The population members that do not perform at least 14.0 *km/l* are eliminated in order to not degenerate the evolution in direction to the maximum D_r.

8.3 Compromise

Finally, this simulation intends to specify the gear shifting velocities which present the best compromise between the distance per liter of fuel and the travel distance. In this case, the fitness function is the minimum among the distance ratio and the corrected economy ratio, as explicit in Eq. (13).

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$$Fitness = \min(D_r, E_r * 0.99) \tag{13}$$

8.4 Results of GA simulations

The cloud of points obtained as result from the GA algorithms is exposed in Fig. 8. It is possible to notice that each simulation converged to their aims and all of them improved the performance of the vehicle within their scope. The points with the best fitness in each case are described in Tab. 3.

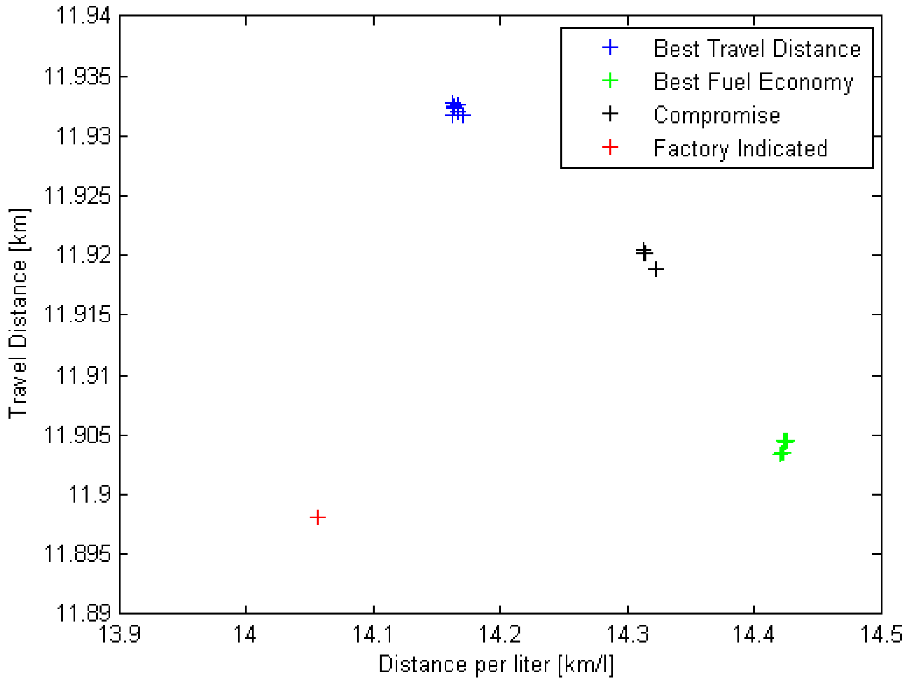

Figure 8. Clouds of results of the GA algorithms.

In all GA simulations, the velocity of shift from 4th to 5th gear is above the top speed of the Brazilian driving cycle, which is 91.2 *km/h*. As effect, the 5th gear is never used in this cycle. It does not mean that in steady state, popular Brazilian cars speeded about 90 *km/h* using 5th gear consume more fuel than the ones in 4th gear; but indicates that in conditions of normal urban traffic, the shift to 5th might not be profitable. Considering specifically the urban standard, this action is not indicated.

Table 3. Results of GA simulation with best fitness

	1st to 2nd Gear Shifting Velocity [*km/h*]	2nd to 3rd Gear Shifting Velocity [*km/h*]	3rd to 4th Gear Shifting Velocity [*km/h*]	4th to 5th Gear Shifting Velocity [*km/h*]	Travel Distance[(1)] [*m*]	Distance per liter of fuel [*km/l*]
Referential Simulation	15	30	50	72	11898	14.05
GA - Best Fuel Economy	18.59	21.65	44.35	96.00	11904	14.43
GA - Best Travel Distance	19.43	22.32	65.92	120.90	11932	14.17
GA - Compromise	18.87	21.81	52.57	93.30	11920	14.31

(1) standard urban cycle length: 11990m

The most economic case performed 14.43 km/l, 2.7% of fuel saving in relation to the referential simulation. The case “Best Travel Distance” also consumes less fuel than the referential simulation, what means that the optimization succeeds in finding more efficient gear shifting. This attribute is better demonstrated by the compromise case, in which the simulated vehicle approaches the major distance and the best fuel consumption, simultaneously.

All the cases demonstrate that the factory indicated that the shift to 2nd gear is premature and changes to 3rd and 4th are late. As the 5th was excluded from the optimization loop and its effect was not computed, nothing can be affirmed about this gear shifting.

9. CONCLUSIONS

The gear shifting strategy presents significant effects over the vehicular performance and fuel consumption. Among the technics implemented, the map of ideal gears, in function of the automobile velocity and torque, demonstrated operational instabilities which degrade both fuel economy and travel distance. Thus, even though this is a promising method, it has to be smoothed in future works.

The optimization of gear shifting strategies aided by Genetic Algorithm is effective on the improvement of the automobile performance and fuel economy (antagonistic characteristics) in a known driving profile. The driving cycle used was the Brazilian urban standard, but this study inspires the prospection of an intelligent system that recognizes the user routine driving pattern and, after an optimization based on the methods proposed, suggests to the driver the most opportune gear shifting timing. Such application demands no mechanical adaptations on Brazilian popular cars, and could be a short-term feasible alternative for reducing fuel consumption.

10. ACKNOWLEDGEMENTS

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