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Vehicle gear shifting strategy optimization with respect to performance and fuel consumption

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ABSTRACT
Based on the movement resistance forces, the vehicle longitudinal dynamics is related to power demand for a specific route. The vehicle gear shifting influences significantly the acceleration performance and fuel consumption because it changes the engine operation point and the powertrain inertia. This paper presents a study based on the US06 velocity profile that involves high speed and high acceleration phases, where the vehicle performance is limited by both the engine power and the tire traction limit. For improving the vehicle performance without increasing fuel consumption, a genetic algorithm (GA) technique is used.

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1. Introduction

Gear shifting strategies are important in the analysis of the vehicle dynamics because they change the gearbox transmission ratio, the powertrain inertia, and the engine operation point, influencing both vehicle acceleration performance and fuel consumption. According to Ehsani et al. (2009), gear shifting strategies may optimize the engine operation point in three characteristics: maximum acceleration (maximum torque), higher speeds (maximum power), and fuel economy (minimum specific fuel consumption).

The clutch is responsible for transferring the engine torque to the rest of the powertrain, and it synchronizes the gears during gear shifting. After speed synchronization between the engine and the target gear, the contact between the clutch discs is gradually resumed and the transmittable torque increases to the driver demand level (Zhao et al., 2014). The vehicle acceleration performance is limited by restricted traction at low speed (especially in the start-up condition) and lack of available engine torque (Gillespie, 1992; Spanos et al., 2012).

Vehicles with combustion engines are probably the prominent fossil fuel consumers around the world (Delkhosh et al., 2014). An alternative to reduce the fuel consumption is to make the engine operate at optimum efficiency using a gear shifting strategy (Mashadi et al., 2014). According to Kahlbau and Bestle (2013), the shift quality is determined by the gear shifting strategy that defines the appropriate moment to execute the shift. However, it is difficult to determine the best condition to perform a gear shift due to the variety of factors that influence the vehicle dynamics. The implementation of gear shifting strategies in automotive manual transmission must be assisted by intelligent systems as proposed by Kang et al. (2012), to overcome the poor knowledge that a driver has about the engine and powertrain behavior. However, to develop this kind of gear shifting, a control system is necessary to understand the vehicle behavior according to each situation. Optimization algorithms are alternatives to analyze different gear shifting points during a driving cycle, according to performance limitations and resulting

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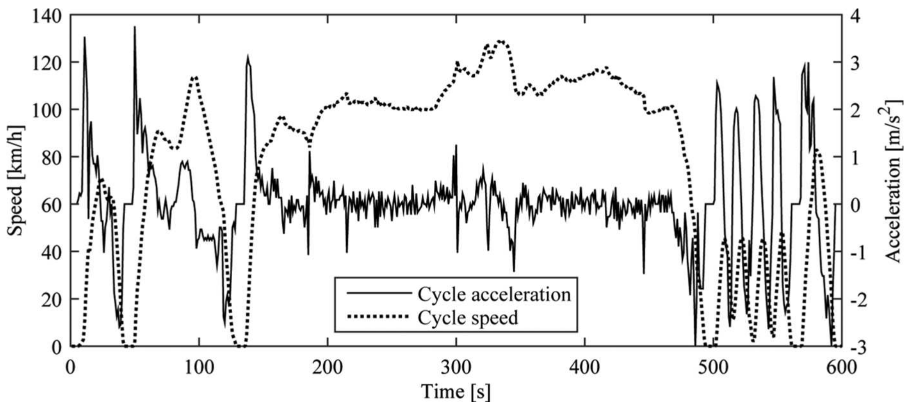

Figure 1. US06 driving cycle velocity and acceleration profile adapted from Barlow et al. (2009).

fuel consumption as proposed by Gao et al. (2011) with optimized gearshift time as function of the engine torque and clutch performance.

In a previous work, Eckert et al. (2014) developed a gear shifting optimization algorithm that decreases the calculated fuel consumption in a 1.0L Brazilian vehicle model keeping the same performance of the standard gear shifting strategy proposed by GM (2013) for the Brazilian urban drive cycle NBR6601 that presents a moderate driving condition where the vehicle is able to accomplish the cycle velocity profile. The present paper also aims at optimizing the gear shifting strategy, but this time applied to the US06 standard driving cycle containing high speeds and acceleration where maximum vehicle power is requested. The last represents a different vehicle dynamic behavior in comparison to the urban driving style. Previous simulations shown in Eckert et al. (2015) concluded that the simulated vehicle is not able to keep the speed requested by the US06 cycle in some acceleration phases because of both engine power and tire traction limitations. For this reason, the objective now is to increase the vehicle performance in order to get results closer to the requested velocity profile avoiding fuel consumption increases.

2. Driving cycle

According to Mashadi et al. (2014), driving cycles are used to analyze the vehicle performance in a specific area or city. These velocity profiles are designed to reproduce typical driving conditions on real roads. In this paper, the US06 standard driving cycle (Fig. 1) is adopted. This cycle contains high speeds and severe accelerations (Barlow et al., 2009), to allow analysis in critical operation conditions where maximum engine power or tire maximum traction limit are reached.

The driving cycle provides a target speed (V_c) with one step Δt ahead of the current simulation time, where the delay avoids algebraic loops. By comparing it to the actual vehicle speed V and dividing it by the constant time step Δt (which is the time given to the simulation to narrow the instant speed gap), we obtain the requested acceleration.

$$a_{\text{req}} = \frac{\Delta V}{\Delta t} = \frac{V_c - V}{\Delta t}. \tag{1}$$

3. Vehicle longitudinal dynamics

The vehicle power demand for achieving the requested acceleration is determined by the longitudinal vehicle dynamics proposed by Gillespie (1992) where the model is based on the movement resistance forces acting on the system, especially aerodynamic drag (D_A) and tire rolling resistance (R_x). The requested vehicle acceleration (a_{req}), the gearbox transmission ratio (N_t), differential transmission ratio

N_d, equivalent powertrain inertia, and mechanical efficiency (η_{td}) also influence the vehicle power demand. The engine requested torque (T_{req}) in a specific situation, when the climbing resistance is disregarded because of missing road grade information in the driving cycle is given as

$$T_{\text{req}} = \frac{\left(Ma_{\text{req}} + ((I_e + I_t)(N_t N_d)^2 + I_d N_d^2 + I_w)\frac{a_{\text{req}}}{r^2} + R_x + D_A\right) r}{N_t N_d \eta_{td}}, \tag{2}$$

where M is the vehicle mass, r is the tire effective radius, and I_e, I_t, I_d, and I_w are, respectively, the engine, gearbox, differential, and wheel inertias.

The aerodynamic drag is one of the most power demand factors at high speeds, and is defined by drag coefficient (C_D), vehicle frontal area (A), and air density (ρ):

$$D_A = \frac{1}{2}\rho V^2 C_D A. \tag{3}$$

Because of the energy lost by the tire deformation due to vehicle weight ($W = Mg$), the rolling resistance is the largest part of the requested power at low speeds:

$$R_x = 0.01\left(1 + \frac{0.62\ V}{100}\right) W. \tag{4}$$

3.1. Engine model

If the requested engine torque (2) is positive, it means that the vehicle is in an acceleration phase requiring engine power. The engine model is represented by look-up tables returning the torque and fuel consumption according to engine speed, throttle, and requested torque. The engine torque output (T_e) is defined by the curves shown in Fig. 2(a). According to the actual engine speed and the requested engine torque $T_e = T_{\text{req}}$, this look-up table determines the required throttle. If the requested torque (T_{req}) exceeds the maximum available torque, the engine output torque will be limited to the 100% throttle curve. The engine efficiency is represented by the specific fuel consumption curves in Fig. 2(b) as a function of engine speed and torque. The highest efficiency regions refer to the lowest specific fuel consumption values C_e $[g/(kWh)]$.

With the engine torque (T_e) and specific fuel consumption taken from Fig. 2(b), it is possible to define the volumetric fuel consumption Cl, where ρ_f is the fuel density and ω_e is the engine angular velocity:

$$Cl = \frac{C_e T_e \omega_e}{\rho_f}. \tag{5}$$

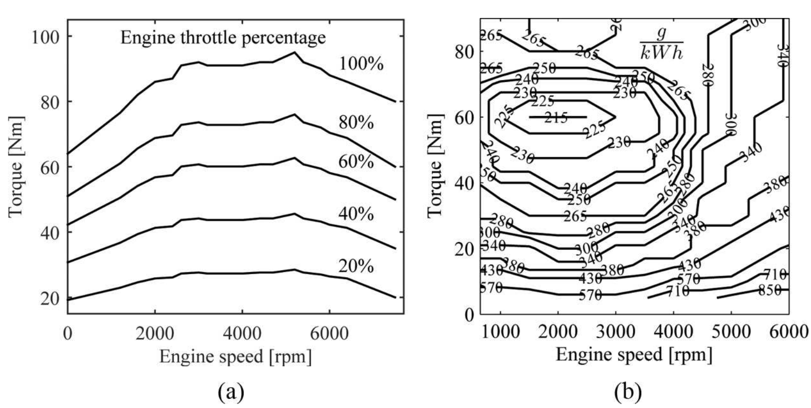

Figure 2. Engine torque (a) and specific fuel consumption map (b) (Adapted from Eckert et al., 2014).

When the cycle speed is zero and the requested torque vanishes, the simulated vehicle stops and its engine starts idling. However, the engine at idle still consumes fuel to run. This condition is modeled as a single operation point in the engine fuel consumption map at 800 rpm–10 Nm.

If the requested torque (2) is negative, this indicates that the vehicle is in a braking process. Here an ideal braking system is assumed, which applies the requested brake torque (T_{req}) calculated by Equation (2) as a movement resistance force to keep the vehicle at the given cycle speed. It does not influence the engine fuel consumption because it operates in a cutoff regime where fuel injection is stopped.

3.2. Clutch model

The clutch is modeled in two states and when it is completely coupled (normal driving simulation), the gearbox input torque (T_g) receives the total torque from the engine ($T_g = T_e$). In decoupling and recoupling states (gear shifting), the transmissible torque is given according to Kulkarni et al. (2007):

$$T_{cl} = \frac{2}{3}\mu_{cl}F_n n\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2} \tag{6}$$

where μ_{cl} is the friction coefficient, F_n is the normal force applied between the disks, n is the number of clutch faces, R_o and R_i are the clutch external and internal disk radius. If the engine torque (T_e) exceeds the maximum clutch torque (i.e., $T_e > T_{cl}$), the gearbox input torque will be the clutch transmissible torque ($T_g = T_{cl}$) only.

3.3. Vehicle acceleration

Due to the limitations by the engine and clutch, discussed in Sections 3.1 and 3.2, and the traction limit, the real vehicle acceleration (a_x) may not be identical with the requested acceleration (1). The available traction force (F_x) may be deduced from Equation (2) by substituting the requested torque (T_{req}) by the available torque (T_g) and setting $D_A = R_x = Ma_{\text{req}} = 0$:

$$F_x = \frac{T_g N_t N_d \eta_{td}}{r} - ((I_e + I_t)(N_t N_d)^2 + I_d\, N_d^2 + I_w)\frac{a_x}{r^2}. \tag{7}$$

This resulting traction force (F_x) will provide the necessary force to overcome the aerodynamic drag (D_A) and the tire rolling resistance (R_x). The rest of the available traction force will be used to accelerate the vehicle.

When the vehicle is leaving rest position, the longitudinal acceleration transfers a large load to the vehicle rear wheels decreasing the normal load applied to the frontal traction wheels and thus limiting the transmissible traction force between tire and ground, and consequently the acceleration performance. The maximum force transmissible by the tire is estimated from the vertical force applied to the frontal drive axle (W_f) and the tire-ground peak friction coefficient (μ) as

$$F_{\max} = \mu W_f. \tag{8}$$

The weight force W_f acting on the vehicle front axle is calculated as proposed by Jazar (2008), where L is the wheelbase, h is the vehicle gravity center, and c is the longitudinal distance between the vehicle rear axle and the gravity center:

$$W_f = \frac{Wc}{L} - \frac{Wha_x}{Lg}. \tag{9}$$

If the available traction force (F_x) is below the tire traction limit ($F_x < F_{\max}$), the vehicle acceleration is limited by the engine power ($F_L = F_x$) in Equation (7). If the available traction force exceeds the tire limit ($F_x \geq F_{\max}$), the acceleration is limited by the tire traction limit ($F_L = F_{\max}$) according to

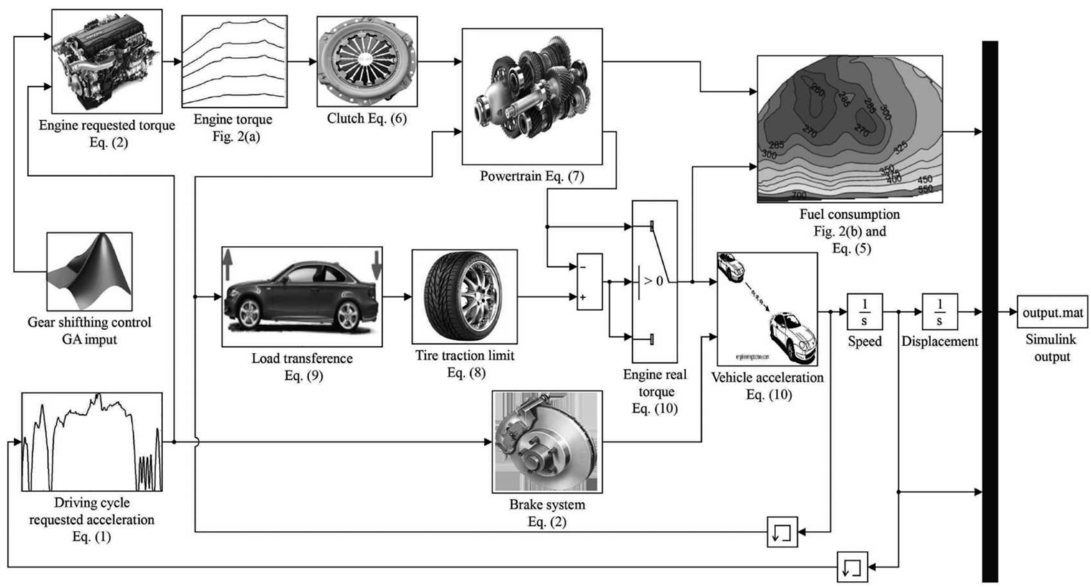

Figure 3. Simulink™ model for longitudinal vehicle dynamics.

Equation (8). In both cases, the real vehicle acceleration is

$$a_x = \frac{F_L - R_x - D_A}{M}. \tag{10}$$

If the traction force is limited by the tire, the actual engine torque (T_e) is lower than T_{req} and needs to be recalculated from Equations (2) and (10) as

$$T_e = \frac{F_{\max} r}{N_t N_d \eta_{td}} + ((I_e + I_t)(N_t N_d)^2 + I_d\, N_d^2 + I_w)\frac{a_x}{r}. \tag{11}$$

The final vehicle acceleration (a_x) is integrated by the ode113 to calculate the vehicle speed (V), Fig. 3. As mentioned in Section 2, the model uses delayed accelerations in Equations (7), (9), and (11) to avoid algebraic loops.

4. Simulation parameters

The Simulink™model is based on a compact hatchback equipped with 1.0L gasoline Otto engine and 5 gear ratios. As the optimization process needs to run many times, only the vehicle longitudinal dynamics is simulated. The simulated vehicle parameters are shown in Table 1.

5. Gear shifting optimization

The genetic algorithm (GA) is one of the heuristic-based optimization techniques that use a population of solutions in their search. This technique was used previously to optimize vehicle gear shifting process in Casavola et al. (2010) and Ahmed et al. (2012), and will be used here as well.

5.1. Gear shifting parameters

The gear shifting strategy has a significant influence on the vehicle desired performance (Ahmed et al., 2012) and it can decrease the engine fuel consumption if performed under appropriate conditions. In this paper, the gear shiftings are defined by velocities as design variables determining the gearbox upshifts

Table 1. Simulated vehicle parameters.

Components	Units	Speed 1st	2nd	3rd	4th	5th
Engine inertia (I_e)	kgm^2			0.1367		
Gearbox inertia (I_t)	kgm^2	0.0017	0.0022	0.0029	0.0039	0.0054
Differential inertia (I_d)	kgm^2			9.22E-04		
Wheels + tires inertia (I_w)	kgm^2			2		
Gearbox ratio (N_t)	–	4.27	2.35	1.48	1.05	0.8
Differential ratio (N_d)	–			4.87		
Powertrain efficiency (η_{td})	–			0.9		
Total vehicle mass (M)	kg			980		
Vehicle frontal area (A)	m^2			1.8		
Drag coefficient (C_D)	–			0.33		
Tires 175/70 R13 radius (r)	m			0.2876		
Tire peak friction coefficient (μ)	–			0.9		
Wheelbase (L)	m			2.443		
Gravity center height (h)	m			0.53		
Rear axle to gravity center (c)	m			1.460		
Clutch friction coefficient (μ_{cl})	–			0.27		
Clutch external disk radius (R_o)	mm			95		
Clutch internal disk radius (R_i)	mm			67		
Clutch faces (n)	–			2		

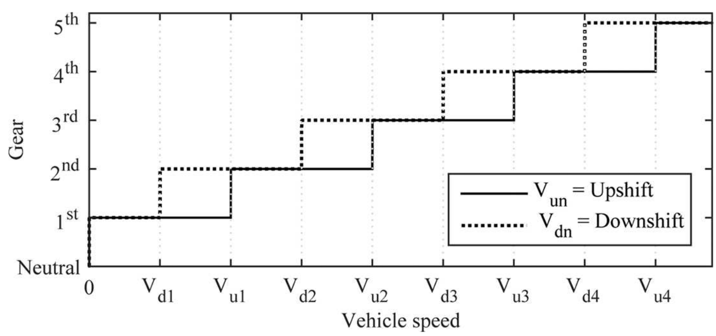

Figure 4. Gear shifting velocities.

(V_{un}) and downshifts (V_{dn}) for each cycle section, $n = 1, 2, 3, 4$. When the vehicle reaches one of these limit speeds, the control changes the gear as shown in Fig. 4. The gear shift time is limited to 1 sec as proposed by Yin et al. (2007). When the simulation detects gear shifting, the clutch decouples the engine from the gearbox during the first 0.3 sec, then it changes the transmission ratio in 0.2 sec, and finally restores the torque transmission during the remaining 0.5 sec.

The engine operation range must be considered in the choice of the gear shifting speeds V_n. The associated engine speed $\omega_n = \frac{V_n N_t(n) N_d}{r}$ cannot be below the idle speed (minimum engine speed $\omega_{\min} \approx 84$ rad/sec) or above the maximum speed limit ($\omega_{\max} \approx 680$ rad/sec).

Too many consecutive gear shiftings decrease the performance due to torque supply interruption by the clutch decoupling. Because the vehicle speed decreases during gear shift, the downshift speed must be lower than upshift speed ($V_{dn} = V_{un} - 1.39$) to avoid gear shifting instability as proposed by Xi et al. (2009). In order to avoid two subsequent gear shiftings, and consequently a lack of torque, a minimum interval of (0.83 m/sec) between two gear shifting speeds was stipulated. This result in the following constraints:

$$C = \begin{cases} \omega_{\min} \leq \omega_n \leq \omega_{\max} \\ V_{un} \leq V_{u(n+1)} - 0.83 \text{ for } n = 1, 2, 3 \\ V_{dn} = V_{un} - 1.39 \end{cases} \quad (12)$$

The use of a specific gear for a short time only, also can generate gear shift instability. To create a dwell time between two subsequent gear shifts, as suggested by Casavola et al. (2010), a filter is applied to replace gear shifting speeds V_n that are used for less than 3 sec by the previously applied values of V_n.

5.2. Problem formulation

The aim is to improve the vehicle performance for the US06 velocity profile. In order to compare the simulated vehicle speed with the standard driving cycle, the term R^2 as the square of the correlation coefficient may be used, Navidi (2008):

$$R^2 = \frac{\left(\sum\left(V_{ci} - \bar{V}_c\right)\left(V_i - \bar{V}\right)\right)^2}{\sum\left(V_{ci} - \bar{V}_c\right)^2 \sum\left(V_i - \bar{V}\right)^2}, \tag{13}$$

where V_{ci} represent the US06 cycle speeds at discrete time points, V_i are the simulated vehicle speeds, and $\bar{V}$ and $\bar{V}_c$ are associated mean values. The closer R^2 value is to 1, the higher is the similarity of the two data sets. The principal objective, therefore, is to minimize $(1 - R)$.

The second optimization criterion is to minimize the fuel consumption $(\int C_l dt \approx \sum Cl_i \Delta_t \sim \sum Cl_i)$ during the driving cycle range. The optimum result is the one that presents the lower engine fuel consumption among the better performance results according to the gear shifting strategy ($X = [V_{d1}\ V_{u1} \ldots\ V_{d4}\ V_{u4}]^T$) defined by the gear shifting velocities V_{dn} and V_{un}:

$$\min f_1(X) = 1 - R(X) \tag{14}$$

$$\min f_2(X) = \sum Cl_i \tag{15}$$

subject to: gear shifting constraints (12)

5.3. Selection, crossover, and mutation

The driving cycle range T_c [0 C_s] is divided into sections (k) of equal random size $(2 \le \Delta_t \le 20)$ [s] for each GA optimization. The number of cycle segments is then determined as $S = \left[\frac{C_s}{\Delta_t}\right] \in \mathbb{Z}$. The GA algorithm analyses each cycle segment $(1 \le k \le S)$ and creates a vector $x(k)$ with the gear shifting speeds used for the analysed cycle section:

$$x(k) = [V_{d1}\ V_{u1}\ V_{d2}\ V_{u2}\ V_{d3}\ V_{u3}\ V_{d4}\ V_{u4}] \tag{16}$$

The parameters $x(k)$ are encoded by a single gene. They are combined in a set of genes forming a chromosome that describe an individual solution $[X]$ (Figueredo and Sansen, 2014). Such a chromosome X represents the gear shifting strategy for the whole driving cycle.

$$[X]_{[8\times S]} = \left[[x(1)]^T\ [x(2)]^T \ldots\ [x(k)]^T \ldots\ [x(S)]^T\right] \tag{17}$$

In the genetic algorithm, the population performance values (1-$R(X)$) are rounded to the fourth decimal place, which is defined as the minimum significative improvement value. The population is ordered with respect to the objective function f_1 first and then with respect to f_2. The first ordering criterion (14) represents descending performance values $(1-R(X))$, where lower values represent better coincidence with the US06 cycle. If two or more members of the population have equal $1 - R(X)$ values (after rounding), the second criterion (15) is used to order this result by minimum engine fuel consumption.

Once the population is ordered, its members (j) that represent the results of a specify chromosome $[X]$ receive a fitness interval used in the selection process:

$$[L_{\min}(j)\ L_{\max}(j)] \text{ where } \begin{cases} L_{\max}(j) = L_{\min}(j-1),\ L_{\min}(0) = 1 \\ L_{\min}(j) = L_{\max}(j) - \frac{L_{\max}(j)}{0.68 {P_s}^{0.56}},\ 1 \le j \le P_s, \end{cases}$$

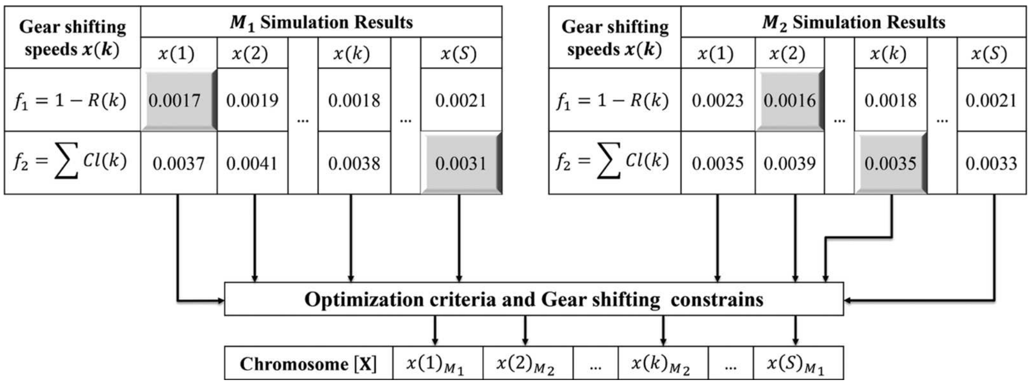

Figure 5. Crossover procedure example.

where (P_s) is the population size. The best result (first member of the population $j = 1$) receives a larger interval that increases the probability of being selected for the crossover process. For the other members smaller intervals are awarded, decreasing selection probability.

The crossover chooses two members (j) of the population randomly by means of the Matlab™function *rand* generating a P_C value. According to this P_C value interval j is selected ($L_{\max}(j) \leq P_C \leq L_{\min}(j)$). The two associated simulation results (M_1 and M_2) are compared with the US06 cycle in each cycle section (k). For each section, the crossover keeps the gear shifting speed values $x(k)$ with the better performance in this section. In case of identical f_1 values (after rounding), f_2 will make the difference. Finally, all shifting speeds $x(k)$ are combined in a new chromosome $[X]$ according to Equation(17), a new simulation is performed, and the result is stored increasing population size.

A numerical example is shown in Fig. 5. The values chosen by the optimization criteria are highlighted. For the first section ($k = 1$), the population result M_1 presents better performance than M_2, due to this the crossover keeps the gear shifting speeds $x(1)$ from M_1 to the first cycle section. In the second section ($k = 2$), M_2 presents the best performance and the crossover includes the gear shifting speeds $x(2)$ from M_2 in the chromosome $[X]$. In a generic section k and at the final section ($k = S$), the optimization criterion f_1 presents equal performance for both population members (according to rounded rules discussed previously) and the choice is made by f_2 criterion that keeps the $x(k)$ values from the member that presents the lower fuel consumption.

The mutation algorithm starts with selecting cycle sections (k) to mutate. For each cycle section (k), the mutation probability was defined as 50%. If mutation occurs, all the corresponding $x(k)$ shifting speeds are altered by adding a random value $-2.78 \leq V_{\text{mut}} \leq 2.78$ m/sec. If any of the mutated shifting speeds V_{dn} V_{un} $n = 1 \ldots 4$ are outside the constraints C, they are altered to the limit value. Similarly to the crossover process, the mutated chromosome is simulated and the new result is stored in the population.

5.4. Population and convergence criteria

To create benchmark results, the initial population is composed of standard gear shifting strategies proposed by Ehsani et al. (2009) (maximum torque (MT) and power (MP) strategies as well as fuel economy strategy (FE)); see Table 2. The maximum torque and power strategies are defined based on the respective engine maximum torque and power points (5300 rpm and 6400 rpm). The fuel economy strategy is defined by the gear shifting speeds proposed in GM (2013) for an urban driving behavior. Two intermediary strategies (gear shifting at the engine 3500 rpm (S35) and 4500 rpm (S45)) proposed by Eckert et al. (2015) are also considered. All this strategies keep the same gear shifting speeds (V_{dn} and V_{un}) over all cycle sections (k). The corresponding fuel consumption and vehicle performance are shown in Table 3.

Table 2. Standard gear shifting speeds according to the engine rpm.

Gear Shifting Strategy	Gear Shifting Speeds [km/h] V_{d1}	V_{u1}	V_{d2}	V_{u2}	V_{d3}	V_{u3}	V_{d4}	V_{u4}
Fuel economy (2800 rpm to 3300 rpm)	12	15	27	30	47	50	69	72
Shifting at 3500 rpm	16	19	32	35	52	55	77	80
Shifting at 4500 rpm	22	25	42	45	62	65	97	100
Maximum torque 5300 rpm	25	28	50	53	80	83	116	119
Maximum power 6400 rpm	31	34	59	62	95	98	136	139

Table 3. Comparison between standard and optimized gear shifting strategies.

	Gear Shifting Strategy	Fuel Consumption $f_2(X)$ [ml]	Vehicle Performance $f_1(X)$ (1-$R(X)$)
FE	Fuel economy	817.0	0.0261
S35	Shifting at 3500 rpm	847.7	0.0244
S45	Shifting at 4500 rpm	917.9	0.0225
MT	Maximum torque	1130.7	0.0210
MP	Maximum power	1294.6	0.0204
OS	Optimized strategy	1029.1	0.0164

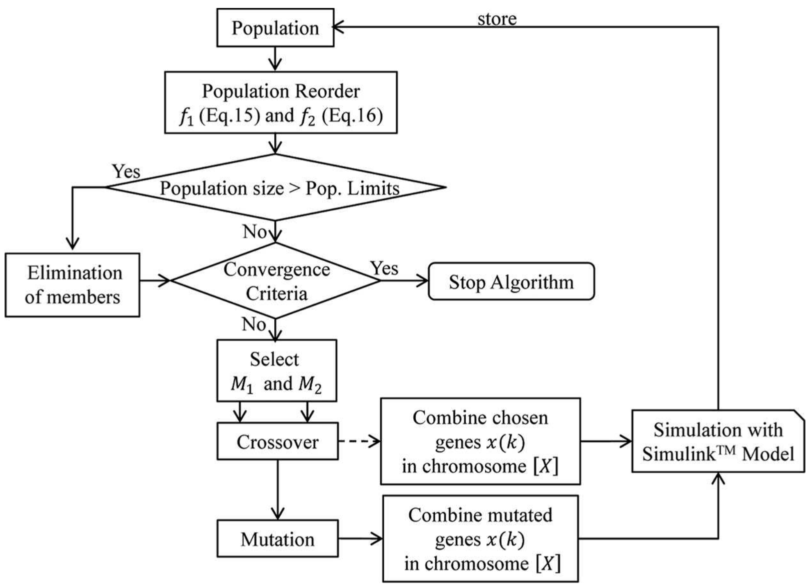

Figure 6. Flowchart of the GA optimization process.

Each of the above operators enlarges the population by new chromosomes X. Two elimination procedures control the population size. The first starts when the population size exceeds 50: all population members with a performance value f_1 higher than the population average, which were used in at least one crossover/mutation process, are eliminated. The second procedure replaces the worst performance result by the last simulation result, if the population size reaches 100.

Based on previous results presented by Eckert et al. (2015), the current simulated vehicle is unable to follow the US06 velocity profile in some acceleration sections. Thus, a convergence criterion is used, based on the repeatability of the best result in the population. If the best 10 results have equal performance and fuel consumption values (according to rounding rules), the GA algorithm finishes.

The optimization algorithm flowchart is shown in Fig. 6.

6. Results

The lower-left border of the results cloud shown in Fig. 7 represents the non-dominated solutions, which present the best compromise between the objective functions. All nondominated solutions including the S45 S35 and FE strategies could be considered as an optimum strategy. However, the principal aim of the optimization process is to improve the vehicle performance without the high fuel consumption of the MT and MP strategies. Due to this, the GA population hierarchy prioritizes the vehicle performance (f_1) as lower fuel consumption (f_2). Thus, the results are targeted to performance improvement more than fuel economy.

After the convergence of the algorithm, the MT and PM strategies were clearly dominated, and the optimized gear shifting strategy was selected among the nondominated solutions. Even with the performance improvement, the vehicle was unable to reach the desired speed in the high acceleration phases (Fig. 8), due to tire traction and engine power limitations. By this reason, the selected optimized gear shifting strategy was the one that reached the best performance among the nondominated solutions. A comparison of the optimized strategy and some standard gear shifting strategies is shown in Table 3.

The optimized gear shifting strategy results in a set of gear shifting speeds (V_{dn} and V_{un}) that change according to the US06 cycle. The gear shifting behavior according to the upshift speeds V_{un} is shown in Fig. 8 and the downshift speeds are defined according to (12) as $V_{dn} = V_{un} - 1.39$ [m/sec]. Every time that the vehicle reaches one of these limit speeds, the gearbox changes the gear ratio. The optimized gear shifting strategy shows 19.61% improvement in the vehicle performance compared with the maximum power strategy, which has the best performance of all standard gear shifting strategies.

The principal difference among the optimized gear shifting speeds and the standard strategies (Table 2) is the large interval between the V_{u1} and the V_{u2}. This generates better performance results avoiding gear shifts during the high acceleration phases, by performing earlier upshifts to the 2nd gear, when the requested torque is lower, and then postponing the next upshift using the whole engine speed range.

The engine fuel saving is limited by the performance request, which is the primary optimization criterion. In other words, the GA only reduces the fuel consumption if the performance of the analyzed result is equal according to rounded criteria. Therefore, in Table 3 the optimized strategy only demonstrates a fuel consumption decrease compared with the maximum power strategy, whereas other standard gear shifting strategies behave better as shown in Fig. 9.

The fuel economy strategy upshifts closer to the engine speed of 3000 rpm to keep the engine running in good efficiency region, but it decreases the vehicle performance at the acceleration sections dramatically. The maximum power strategy always shifts gears at high engine speeds to increase the acceleration performance, and consequently increases the engine fuel consumption. However, it is not necessary to operate the engine at such conditions in all cycle sections. When the vehicle leaves from zero speed, it is submitted to a high acceleration condition where the performance is limited by the tire traction that makes the use of the full engine power unfeasible. The optimized strategy upshifts to the 2nd gear earlier and extends the use of this transmission ratio during the engine speed range that contains the good efficiency region (high torque and 1000 rpm–4000 rpm. According to (Fig. 2b), it upshifts again only when the engine is close to the speed operation limit (6500 rpm) that postpones the upshift due to the lack of torque during the gear shifting process that would decrease vehicle speed. For the same reason, the optimized strategy eliminates gear shifts close to the final section (500 sec to 570 sec of Fig. 8) when the simulated vehicle keeps the 2nd gear to increase the vehicle performance. Once the vehicle has reached the requested speed, the power demand decreases and the optimized gear shifting chooses the available transmission ratio matching the requested torque with the minimum fuel consumption.

Figure 10 compares the optimized strategy behavior with two standard strategies by means of the primary objective function $f_1(X) = 1 - R(X)$ measuring the vehicle performance. This cycle section was chosen because it contains a high acceleration stretch followed by a small braking section and an acceleration stretch at high speeds.

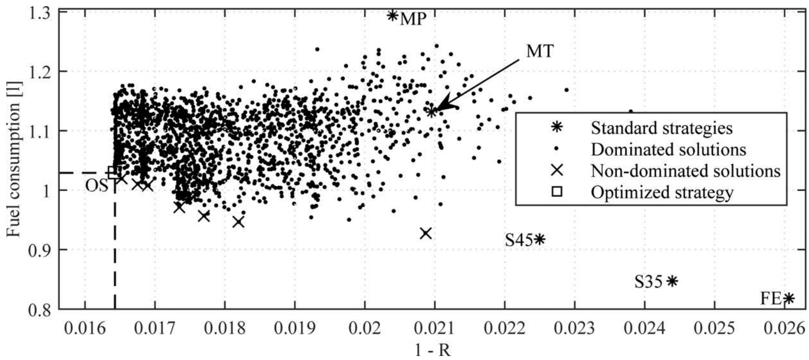

Figure 7. GA results.

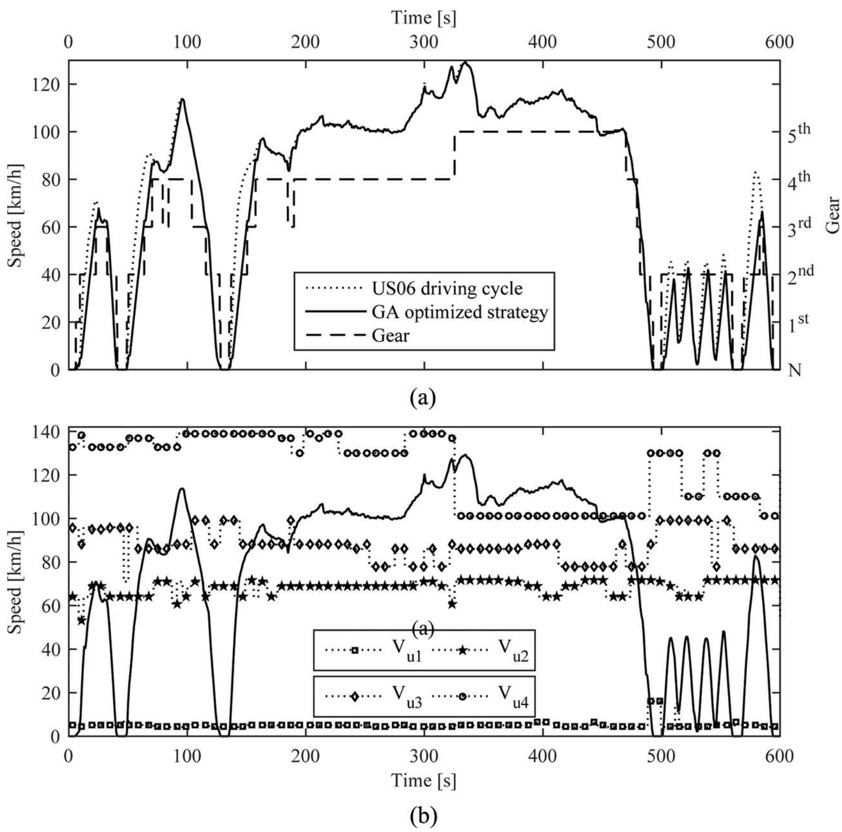

Figure 8. Comparison between the simulated optimal shifting behavior and US06 cycle.

The vehicle performance decreases for each gear shift due to engine decoupling generating a torque supply interruption and consequently a speed decrease. This effect is observed especially when the vehicle uses the fuel economy strategy. On the other hand, the maximum power strategy upshifts at the higher engine speeds, extending the use of each gear ratio. This increases the available torque at the wheels and the vehicle performance becomes limited only by the tire traction. However, when operating the engine in this condition, a substantial increase of fuel consumption occurs. To save fuel it is essential to perform gear shifts when the vehicle power demand is low, and starting acceleration stretches with the most appropriate transmission ratio to avoid premature upshift during the acceleration process.

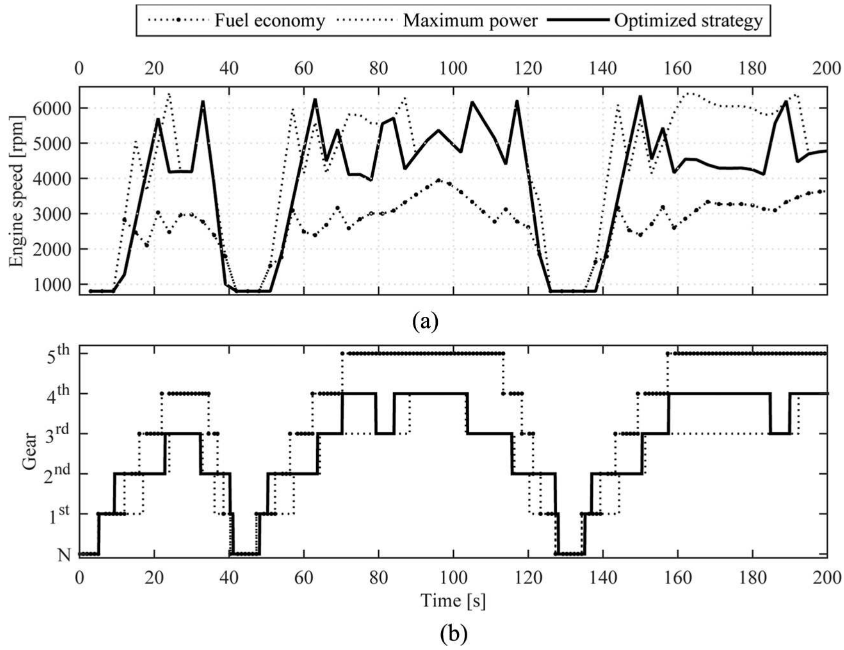

Figure 9. Comparison between three shifting strategies with regard to engine speed and chosen the gear.

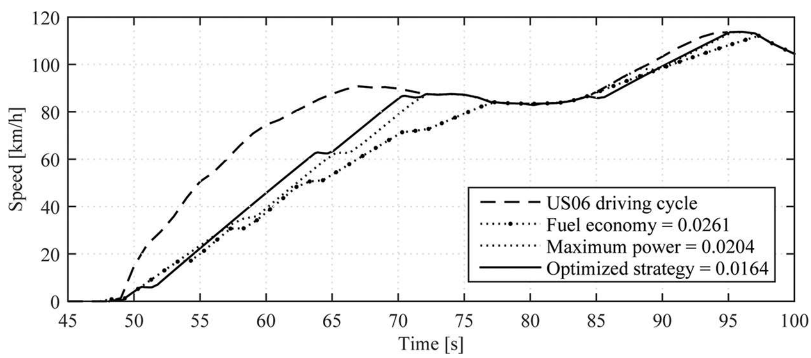

Figure 10. Comparison between simulated shifting strategies in a cycle stretch.

This can be observed in Fig. 10 between the 70 sec and 95 sec of the cycle. The fuel economy strategy upshifts until the top gear during the first acceleration stretch and then it keeps this gear at the second acceleration stretch. Thus, the available torque at the vehicle wheels is too low to follow the US06 cycle. The maximum power strategy utilizes the 3rd gear increasing the available torque at the wheels, but it performs an upshift during the acceleration which decreases the vehicle speed at the end of the stretch. The optimized strategy reaches the cycle request speed at 70 sec using the 3rd gear and upshifts to the 4th gear at the beginning of the breaking section. Already in the beginning of the second acceleration stretch, the vehicle downshifts to 3rd gear again to increase the available torque and quickly upshifts to the 4th gear in order to anticipate the engine decoupling, which improves the final reached speed at 95 sec.

In the implemented optimization process, the gear shifting speeds change during the cycle, allowing the vehicle to perform gear shifts at different driving conditions, to find the best gear shifting points according to the cycle section. However, a comparative could be made by an alternatively procedure to optimize constant gear shifting speeds to be used over the full cycle as the standard strategies.

7. Conclusion

In this paper, the influence of the gear shifting strategy on vehicle longitudinal dynamics is studied for the US06 cycle where the vehicle is submitted to high speed and acceleration phases. Previous results showed that the simulated vehicle is partly unable to reach the requested velocity at high speed stretches, where maximum engine power is required, and at low speeds, where the vehicle performance is restricted by the tire traction limit. To arrive at the best acceleration performance without increasing the fuel consumption significantly, a genetic algorithm technique is used. The optimized gear shifting strategy presents a better correlation compared with the analyzed US06 cycle than the standard strategies. The fuel consumption of the optimized strategy is between the results found with a maximum engine torque and 4500 rpm gear shifting strategies, and the standard maximum power strategy. Fuel savings of about 20% are achieved with a simultaneous improvement in the vehicle performance. Especially in high acceleration phases, the optimization process found a different gear shifting profile to be used by a gear shifting controller.

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