A Brief Study of Reducing Fuel Consumption in Hybrid Electric Vehicles

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Abstract: Automotive control has become a driving factor in automotive innovation over the last twenty five years. In order to meet the enhanced requirements for lower fuel consumption, lower exhaust emissions, improved safety as well as comfort and convenience functions, automotive control had to be applied. The Interest in hybrid electric vehicles derives from several technical and economical considerations. Control strategies for hybrid topologies are algorithms for selecting the power split between the engine and the motor in order to minimize the fuel consumption and the emissions. For hybrid vehicles it is possible to use different kinds of batteries characterized by capacity and weight increasing the autonomy, in terms of working hours or distance. The principal aim of this paper is to demonstrate that fuel consumption and emissions can be reduced by using different control strategies in the same or different topologies.

1. INTRODUCTION

The main objective of automobile industry is developing a vehicle with a lot of performance criteria such as acceleration, braking, comfort, consumption, emissions etc. Improvement of the overall energy efficiency is one of the most important subjects when developing new vehicle technologies. The hybrid electric vehicle (HEV) powertrain includes the advantages of conventional and electrical vehicles (long range of functioning, low emissions and improved consumption). A combined simulation and analytical approach enables analysis of energy flows and energy losses on different energy paths within the hybrid powertrain and evaluation of their influences on energy consumption of the vehicle.

For simulating and evaluating the vehicles it was used ADVISOR (Advanced Vehicle Simulator) that works interactively with Matlab and Simulink programming environment and contains a database with important types of vehicles, engines and electrical thermal batteries, mechanical transmission, etc. There are two main types of calculations that ADVISOR uses: backward-facing and forward-facing.

In backward-facing calculations, no driver behaviour is required. The user must input the driving pattern, a velocity profile, called the speed trace. The force required to accelerate the vehicle is calculated and translated into torque. This procedure is repeated at each stage from the vehicle/road interface through the transmission, drivetrain, etc., until the fuel use or energy use is calculated.

In forward-facing calculations, the user inputs the driver model, then the simulator generates throttle and brake commands that are changed into engine torque, which is passed to the transmission model and passed through the drivetrain until a tractive force is computed.

The majority calculations are done in the backward mode, although in order to keep the components from exceeding their physical limitations, some forward calculations are necessary [1].

1.1 Basics of Powertrain

An automotive powertrain consists of a power plant (engine or electric motor), a clutch in manual transmission or a torque converter in automatic transmission, a gearbox (transmission), final drive, differential, drive shaft, and driven wheels. The torque and rotating speed of the power plant output shaft are transmitted to the drive wheels through the clutch, gearbox, final drive, differential, and drive shaft. The required speed is limited to the motor's maximum speed. The required torque is limited to the difference between the motor's maximum torque at the limited speed and the torque required to overcome the rotor inertia. The limited torque and speed are then used to interpolate in the motor/controller's input power map. Finally, the interpolated input power is limited by the motor controller's maximum current limit. This behaviour is described in the following equations:

$$P_{mot.in.r} = \min(P_{mot.in.map}, I_{con.\max}, V_{bus.prev}) \qquad (1)$$

$$P_{mot.in.map} = f(T_{mot.\lim.r}, \omega_{mot.\lim.r}) \quad (2)$$

where *f is* the functional relationship described by the motor map, $P_{mot.in.r}$ is the required motor's input power, $T_{mot.\lim.r}$ is the limited motor torque required, $\omega_{mot.\lim.r}$ is the limited motor speed required, and, $P_{mot.in.map}$ is the input power required to power the motor at its maximum limited torque and speed.

$$\omega_{mot.\lim.r} = \min(\omega_{mot.r}, \omega_{mot.\max}) \quad (3)$$

$$T_{mot.\lim.r} = \min(f_1(\omega_{mot.\lim.r}), T_{mot.r} + J\left(\frac{\Delta\omega_{mot.\lim.r}}{\Delta t}\right)) \quad (4)$$

where f_1 is the functional relationship described by the motor torque envelope. The forward-facing part of the motor/controller model accepts as input the available input power, and produces as outputs the available rotor torque and speed. To compute the torque that can be produced by the motor/controller given the available input power, it is used the motor/controller efficiency computed during the backward facing calculations ($T_{mot.\lim.r} / P_{mot.\lim.r}$):

$$T_{mot.avail} = T_{mot.\lim.r}\left(\frac{P_{mot.avail}}{P_{mot.\lim.r}}\right) - J_{mot}\left(\frac{\Delta\omega_{mot.\lim.r}}{\Delta t}\right) \quad (5)$$

The engine defined by its moment of inertia is characterized by the developed drive torque T_{act_p}, internal friction torque, T_{frec-p}, and external load on clutch, T_{amb}:

$$J_p \frac{d\omega_p}{dt} = T_{act\_p} - T_{frec-p} - T_{amb} \quad (6)$$

If the clutch is fully engaged and is not alleged any internal friction, then:

$$T_{amb} = T_{cv} \quad (7)$$

$$\omega_p = \omega_{amb} \quad (8)$$

where T_{cv} is gearbox torque, and ω_{amb} is the clutch speed. The gearbox is described by the moment of inertia, J_{cv}, its viscous friction torque is characterized by coefficient of friction D_{cv}. If it is considered i_{cv} the gear ratio then the associated model is:

$$\omega_{amb} = \omega_{cv} i_{cv} \quad (9)$$

$$J_{cv} \frac{d\omega_{cv}}{dt} = T_{cv} i_{cv} - D_{cv}\omega_{cv} - T_{ap} \quad (10)$$

Although prop shaft is an elastic element to simplify the model it can be equated with a stiff shaft:

$$\omega_{ap} = \omega_{cv} \quad (11)$$

$$T_{dif} = T_{ap} \quad (12)$$

where ω_{ap} is the prop shaft speed, and T_{dif} is the differential torque and T_{ap} is the prop shaft torque. In the same way as the gearbox, the differential is shaped by the moment of inertia, J_{dif}:

$$\omega_{ap} = \omega_{dif} i_{dif} \quad (13)$$

where i_{dif} is the differential ratio.

$$J_{dif} \frac{d\omega_{dif}}{dt} = T_{dif} i_{dif} - D_{dif}\omega_{dif} - T_{aa} \quad (14)$$

if the wheels speed is the same. In (14) T_{aa} is the auxiliary loads torque, D_{dif} is the friction coefficient of the differential.

The internal combustion engine has a non-linear model with the rotation speed and the angle of the acceleration pedal as inputs and the torque of the crank shaft as output. In the SIMULINK model, a lookup table 2D is simulating the torque characteristic. The output is integrated and divided by the inertia moment (J_p), and the obtained signal is used as a reaction for the 2-D lookup table.

1.2 Description of Vehicle Movement

Power plant torque produces the tractive effort, *Ft*, which is found between tires of the driven wheels and the road surface and propels the vehicle forward. The resistance F_{tr} usually includes tire rolling resistance, aerodynamic drag, and uphill resistance. The vehicle acceleration can be written as:

$$\frac{dV}{dt} = a = \frac{F_t - F_{tr}}{\delta M_v} \quad (15)$$

where *V* is vehicle speed, F_t is the total tractive effort of the vehicle, F_{tr} is the total resistance, M_V is the total mass of the vehicle, and δ is the mass factor, which is an effect of rotating components of the powertrain. Resistance forces of the runway are:

$$F_{tr} = F_X + F_{air} + F_r \quad (16)$$

The grading resistance can be expressed as:

$$F_X = M_V g sin(\alpha) \quad (17)$$

where M_V is mass of vehicle, g is gravitational acceleration and the slope α is expressed as a percentage, positive when climbing and negative when descending.

$$\alpha = arctg \frac{slope}{100} \quad (18)$$

Aerodynamic drag is a function of vehicle speed, vehicle frontal area A_f, shape of the vehicle, and air density ρ_{air}

$$F_{air} = \mathrm{sgn}(V_d)\rho_{air}C_d A_f V_d^{\,2} \tag{19}$$

$$V_d = V - V_{wind} \tag{20}$$

where V_{wind} is the wind speed, A_f is equivalent front surface of the vehicle, and C_d is the aerodynamic drag coefficient. Usual values for C_d are in the range (0.2, 0.4).

The moment produced by the forward shift of the resultant ground reaction force is called the rolling resistant moment:

$$T_r = F_y \cdot a \tag{21}$$

$$F_r \cdot r_d = F_y \cdot a \tag{22}$$

$$F_r = F_y \frac{a}{r_d} = C_r \cdot F_y \tag{23}$$

$$F_r = (C_{r0} + C_{r1} \cdot v)\, F_y \tag{24}$$

C_r is the rolling resistant coefficient, *a* is the distance of displacement and r_d is the rolling radius. C_{r0} is usual between (0.004, 0.02) and $C_{r1} << C_{r0}$ [2]. The tire slip model relates weight on the tire, longitudinal force, vehicle speed, and slip in an equation or set of tables. Because of the limitations:

$$slip = \frac{\omega_{wr} r_{wh}}{v_r} - 1 \tag{25}$$

$$\omega_{wr} = \frac{1 + slip}{r_w} v_{\lim.req} \tag{26}$$

$$T_{wr} = F_{\lim.r} r_w + T_{w.loss} + J_w \left(\frac{\Delta\omega_{wr}}{\Delta t} \right) \tag{27}$$

where ω_{wr} is the required speed at wheel, r_{wh} is the wheel radius, T_{wr} is the required torque input to the axle, $F_{\lim.r}$ is the necessary average tractive force, $T_{w.loss}$ is the torque required to overcome bearing losses and brake drag and J_w is the rotational inertia.

To predict the maximum tractive effort that the tire–ground contact can support, the normal loads on the front and rear axles have to be determined by summing the moments of all the forces about point *R* (center of the tire–ground area). The normal load on the front axle F_{yf} can be determined as:

$$\begin{aligned} F_{yf} &= \frac{L_b}{L} M_v g \cos\alpha - \\ &- \frac{h_g}{L}(M_v g \sin\alpha + F_{air} + \delta \cdot M_v \frac{dv}{dt}) \end{aligned} \tag{28}$$

Similarly, the normal load acting on the rear axle can be expressed as:

$$\begin{aligned} F_{yr} &= \frac{L_a}{L} M_v g \cos\alpha + \\ &+ \frac{h_g}{L}(M_v g \sin\alpha + F_{aer} + \delta \cdot M_v \frac{dv}{dt}) \end{aligned} \tag{29}$$

Speed and traction effort are limited by traction characteristics of the tires.

$$F_{w.r} \le F_{w.\max} \; ; \; v_{w.r} \le v_{w.\max} \tag{30}$$

$$F_{w.r} = F_{t.r} - F_{brake.r} + F_{yloss} \tag{31}$$

1.3 Electric Vehicles

Dynamic performances of a vehicle are usually associated with acceleration time, maximum speed and gradeability. The design of these parameters depends mostly on mechanical characteristics (torque, power) of the electric traction motor. The most used in electric vehicles and hybrid electric vehicles are induction motors, synchronous permanent magnet and reluctance motors. In this paper it was used an induction motor model. The motor/controller model includes the effects of losses in the motor and controller, rotor inertia, and the motor's torque speed-dependent torque capability. Power losses are handled as a 2-D lookup table indexed by rotor speed and output torque. The motor maximum torque is given using a lookup table indexed by rotor speed. Motor control block ensures that the maximum current is not exceeded. Available torque is computed from available power by assuming that the ratio of rotor torque to input (electric) power is the same for the actual/achievable situation as was computed for the request. The motor model is based on the mechanical characteristic. In the stady-state, the motor is described by the equations:

$$\begin{gathered} U_a = R_a I_a + k\psi_e \omega = R_a I_a + E \\ T_e = k\psi_e I_a \end{gathered} \tag{32}$$

Maximum torque is obtained when a maximum current is given in the rotor circuit:

$$T_{emax} = k\psi_e I_{a\,max} \tag{33}$$

In this range of angular speeds, lower than the base angular speed ω_b, maintaining constant the magnetization flux, the EMF of the electric motor increases with the speed of rotation:

$$E = k\psi_e \omega \tag{34}$$

Mechanical power developed by motor has a linear variation dependent on the angular rotation speed:

$$P = T_{emax}\omega \tag{35}$$

$$P_{max} = T_{emax}\omega_b \tag{36}$$

Maximum intensity of the motor current is given by:

$$I_{a\,max} = \frac{U_{a\,max} - E_{max}}{R_a} = \frac{U_{a\,max} - k\psi_e\omega_b}{R_a} \quad (37)$$

To maintain the maximum current even when the angular speed is over ω_b, when the voltage value is limited by the power converter, it is imposed a constant value for EMF, independent of the speed variation. Maximum EMF, with (34) yields:

$$E_{max} = k\psi_e\omega_b = k\frac{\psi_e\omega_b}{\omega}\omega \quad (38)$$

This is the case when the flux is hyperbolic decreasing when the speed is increasing over ω_b. A consequence of the flux decrease over ω_b is the hyperbolic decrease of the torque:

$$T_e = k\frac{\psi_e\omega_b}{\omega}I_{a\max} \quad (39)$$

On the other hand, in this range of speed, the power of the motor is expressed as:

$$P = T_e\omega = k\frac{\psi_e\omega_b}{\omega}I_{a\max}\omega = k\psi_e\omega_b I_{a\max} = P_{\max} \quad (40)$$

which means the power is constant and at the maximum value.

The traction force can be evaluated with (2). For short periods of time, acceleration can be considered constant:

$$\frac{dv}{dt} = \frac{v_{k+1} - v_k}{t_{k+1} - t_k} \quad (41)$$

where v_{k+1} is the vehicle speed at time k+1 and v_k is vehicle speed at time k. Electricity consumption is calculated by integrating the battery pack supplied power:

$$P_{b\text{-}output} = \left[\frac{M_v g(\sin\alpha + (C_{r0} + C_{r1}\cdot v)\cos\alpha}{\eta_{ME}\eta_t} + \frac{0.5\rho_{aer}C_D A_f v^2 + \delta M_v\frac{dv}{dt}}{\eta_{ME}\eta_t}\right]v \quad (42)$$

Equations (41) and (42) allow to compute the tractive force and energy at any moment of a driving cycle. When regenerative braking is used the input power is:

$$P_{b\text{-}input} = \gamma\left[\frac{(M_v g(\sin\alpha + (C_{r0} + C_{r1}\cdot v)\cos\alpha)}{\eta_{ME}\eta_t} + \frac{0.5\rho_{aer}C_D A_f v^2 + \delta M_v\frac{dv}{dt}}{\eta_{ME}\eta_t}\right]v \quad (43)$$

where $\sin\alpha$ or $\frac{dv}{dt}$ or both terms have negative values and γ (subunit value), is called recuperative factor and indicate the percentage of available energy that an electric car can provide [3]. Net energy consumption from the battery pack is expressed as:

$$E_{iesire} = \int_{tractiune} P_{b-output}dt + \int_{franare} P_{b-input}dt \quad (44)$$

Electrochemical batteries are devices that convert electric energy in potential chemical energy during loading, and convert potential chemical energy in electric energy. Another important parameter of a battery is the state of charge (SOC) defined as the ratio between current capacity and total battery capacity. SOC changing during dt is given by:

$$S_{oc} = \frac{dQ}{Q(i)} = \frac{idt}{Q(i)} \quad (45)$$

where Q(i) is the battery capacity at discharge current i. In the charging mode the current has a negative value and in the discharging mode the current is positive. In this way the battery charge level can be expressed as:

$$S_{oc} = S_{oc0} - \int\frac{idt}{Q(i)} \quad (46)$$

where S_{oc0} is the initial value of charging level [4].

2. HYBRID ELECTRICAL VEHICLES

2.1 Architectures of Hev-s

A vehicle that has two or more energy sources and energy converters is called a hybrid vehicle. A lot of attention is focused on hybrid vehicle with an electrical powertrain (energy source energy converters) called HEV (hybrid electric vehicle). The architecture of this kind of vehicle is loosely defined as the connection between the components that define the energy flow routes and control ports. HEVs are classified into four topologies: series hybrid, parallel hybrid, series–parallel hybrid, and complex hybrid.

A series hybrid architecture is presented in Fig.1.

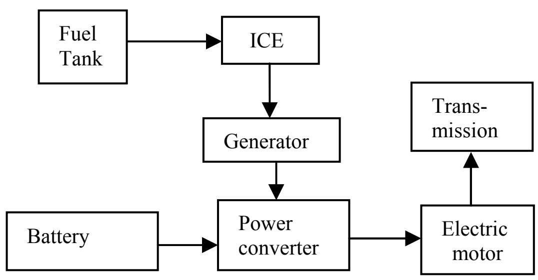

Fig.1. Series architecture

The main advantages are: engine is fully decoupled from the driven wheels so it can be operated at any point on its speed–torque characteristic map, and can potentially be operated within its maximum efficiency region. It doesn’t need multigear transmission. The main disadvantages are**:** the energy from the engine is converted twice, the inefficiencies of the generator and traction motor add up and the losses may be significant. The generator adds additional weight and cost.

A parallel hybrid drivetrain is a drivetrain in which the engine supplies its power mechanically to the wheels like in a conventional ICE-powered vehicle. The powers of the engine and electric motor are coupled together by mechanical coupling. The mechanical coupling of the engine and electric motor power leaves room for several different configurations.

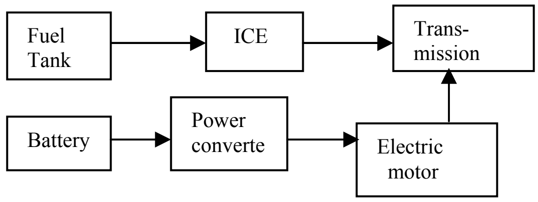

Fig.2. Parallel hybrid architecture

The advantages of this architecture are: single energy conversion for both electrical and mechanical, considerable design flexibility exists in selecting the size of the M/G. The electric motor can be positioned after the gearbox, motor shaft directly, making the energy losses through the gearbox to not exist.

A disadvantage of parallel arrangement is the various added powertrain parts such as added clutches and transmissions. Typically a three-shaft transmission is needed; two input shafts and one output shaft.

A short classification of parallel configurations is:

1) Parallel configurations with torque coupling devices:

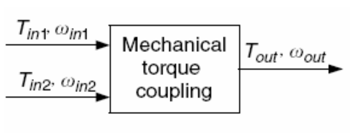

Fig.3. Torque coupling device

- configuration with two gearboxes: two multigear transmissions are used to produce many tractive effort profiles and high flexibility in the design and characteristics of both ICE and motor. These are provided with the possibility to operate in their optimum range

- configuration with one gearbox

- with one drive shaft: a) pretransmission;

b) posttransmission;

Only one transmission is used. Setting the torque coupling ratio allows different speed ratios between ICE and motor, therefore a high-speed motor can be used. In the "posttransmission configuration, the transmission can only modify the engine torque while the motor torque is directly delivered to the driven wheels.

- separate transmission on each axle: One axle is powered by the engine while the other axle is powered by the electric motor. The transmission of the electric motor may be single or multigear. This configuration keeps the original engine and transmission unaltered and adds an electrical traction system on the other axle. A disadvantage is that the batteries cannot be charged from the engine when vehicle is at standstill.

2) Parallel configurations with speed coupling devices.

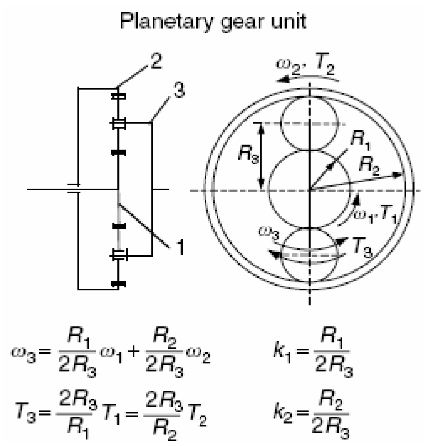

Fig.4. Speed coupling device

Operating modes:

Hybrid traction: When locks 1 and 2 are released the sun gear and ring gear can rotate and both the engine and electric machine supply positive speed and torque (positive power) to the driven wheels.

Engine-alone traction: When lock 2 locks the ring gear to the vehicle frame and lock 1 is released only the engine supplies power to the driven wheels.

Motor-alone traction: When lock 1 locks the sun gear to the vehicle frame (engine is shut off or clutch is disengaged) and lock 2 is released only the electric motor supplies its power to the driven wheels.

Regenerative braking: Lock 1 is set in locking state, the engine is shut off or clutch is disengaged, and the electric machine is controlled in regenerating operation (negative torque). The kinetic or potential energy of the vehicle can be absorbed by the electric system.

Battery charging from the engine: When the controller sets a negative speed for the electric machine, the electric machine absorbs energy from the engine.

2.2 Control Strategies

The main objective of control strategies is to manage as well as possible the energy of the vehicle, thus to extend the range of functionality with minimizing the fuel consumption and the emissions. Most important are:

- rules based control strategies:
  - i) logical rules: thermostat; power follower;
  - ii) Fuzzy rules: conventional; adaptive and predictive
- optimization control strategies:
  - i) global optimization;
  - ii) real time optimization

To minimize fuel consumption and emissions it is equally important to select an appropriate HEV topology as well as to develop the power flow control algorithm. There are few HEV control strategies (CS) mentioned in literature

[5]. The simplest strategy is the 'thermostat' or 'on/off' developed initially for series hybrid. The principle of this control strategy is to deplete the battery to a very low SOC and then trigger the internal combustion engine (ICE) to drive the generator to recharge the batteries while powering the electric motor (EM). Once the batteries are fully recharged, the ICE is shut off again until such needs arise again. During deceleration, some brake energy is recovered to help recharge the battery through regenerative braking. Propelling the vehicle entirely under pure electrical mode as often as possible gives the advantage of setting the ICE to operate at one point of torque and speed that is most efficient and least polluting. It also prevents the ICE from handling transient loads where the highest level of emissions is usually produced.

The ICE is set to run on only one fixed gear ratio that is either optimized for fuel economy or low emission when driving the generator. The lower limit on SOC is 0.52.

The Series Power Follower control strategy determines at what torque and speed the ICE should operate. Electrical power is generated, according to the given conditions of EM, batteries, ICE, and the power demanded by the vehicle. This CS is usually designed to maximize fuel economy, or minimize emission, or maximize battery life. The ICE may be turned off if the SOC gets too high, and turned on again if the power required reaches a certain threshold, or if the SOC hits the minimum level. This CS also incorporates regenerative braking to recycle some brake energy back into the batteries during vehicle deceleration. When the ICE is on, its power output tends to follow the power required, accounting for losses in the generator so that the generator output power converges with the power requirement. Therefore, in some instances, the ICE output power may be adjusted by the SOC, which tends to bring the SOC back to the middle of its operating range, or just keep the SOC above some minimum value. In general, when the SOC is low and the power demand is less than the power of the ICE at its maximum efficiency, the generator is run at a power as close as possible to the ICE's most efficient operation point, without exceeding the system voltage power constraint. The batteries are charged as much as possible to keep the ICE's efficiency as high as possible while maintaining a mid-level SOC. When the power demand is less than the power at maximum efficiency of the ICE, but greater than the battery charge power, the generator is set to run at a power equivalent to the ICE's most efficient operation point. The required power is used to propel the vehicle while excess power generated is used to charge the batteries.

The advantage of this control strategy is that the battery packs are relatively small and the SOC is always maintained around a mid-level. In general this allows the overall weight of the propulsion system to be lighter. The disadvantage is that the ICE is forced to operate at multiple points in its efficiency and emission maps to adjust for load changes. This causes the emission to increase as the operation of ICE moves away from its maximum efficiency point. However, changing the throttle slowly may compensate for this negative effect.

3. A COMPARATIVE STUDY OF HEV PERFORMANCES

To analyze the performance differences of different types of vehicles it was made a test in ADVISOR for every configuration. We have chosen for each configuration 5 ECE cycles meaning 5km of urban driving (a lot of starts, stops and breaks) and 5 EUDC cycles meaning 38,4 km highway driving.

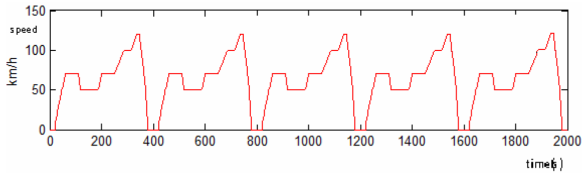

Fig.5. ECE cycles

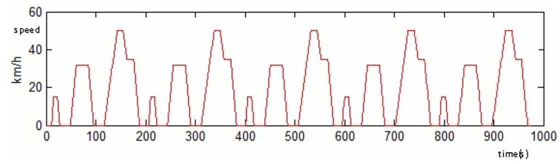

Fig.6. EUDC cycles

Firstly two conventional vehicles were tested. For the first one it was chosen a 41 kW gasoline engine and for the second one a 82 kW gasoline engine. There were used two different gasoline engines with different powers because the first engine it will be compared with an electrical vehicle with the same power. Performances of the 83 kW conventional engine are compared with a series and a parallel HEV, each of them using a 41 kW induction motor and a 41kW internal combustion engine.

1) The first case analysed is a conventional vehicle with a 41 kW power of internal combustion engine, and the weight 984 kg. The acceleration time is given by:

$$t_a = \int_{v_1}^{v_2} \frac{\delta \cdot M_v}{F_t - 0.5\rho_{aer} C_D A_f v^2 - (C_{r0} + C_{r1} \cdot v) M_v g} dv \quad (47)$$

For testing fuel efficiency it was considered 5 cycles ECE (equivalent 5km urban traffic), and 5 cycles EUDC (equivalent 38.5 km highway traffic) and the obtained results are:

- urban consumption 7.6l/100 km
- highway consumption. 5.1l/100 km

The dynamic performances are:

- acceleration time 0-100km/h: 18.4s;
- maximum acceleration-> 2.8m/ s ^{2} ;
- maximum speed: 155.8 km/h;

2) In the second case the tests were made over an 83 kW conventional vehicle with the weight 1104 kg.

Fuel consumption:

- urban 11.3l/100km
- highway 7 l/100km

As dynamic performance:

- acceleration time 0-100km/h: 9.5s
- maximum acceleration: 4.9m/s ^{2} ;
- maximum speed: 201.5 km/h;

3) The same tests were performed for an electrical vehicle with a 41 kW induction motor weight 1103 kg and MI-Pba batteries. The extra weight of the electrical vehicle compared with conventional vehicle is given by the energy storage system weight The result is given as the equivalent gasoline fuel consumption.

The equivalent urban consumption is 1.9 l/100 km and the remaining SOC at the end of the 5 ECE cycles is 91%. The urban consumption difference between the conventional car (ICE power 41 kw) and electrical car is a major advantage (5.7l/100 km).

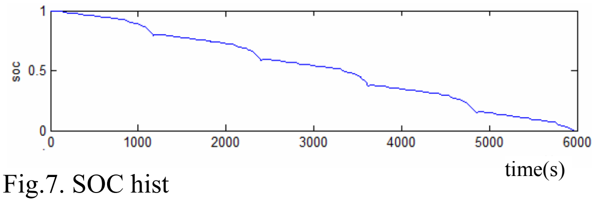

Fig.7. SOC hist

The equivalent highway consumption is 2 l/100 km, and the remaining SOC at the end of the 5 EUDC cycles is 34%. It can be observed a gain of 39% (3.1l/100km) in highway consumption compared with first case. The dynamic performance results:

-acceleration time 0-100km/h 17.6s

-maximum acceleration 5m/s ^{2} ;

-maximum speed 130.4 km/h;

4) For a better look the same tests were made over another electrical vehicle with MS-LI batteries 41 kW induction motor, and weight 856 kg. The weight difference compared with the previously case is given by the batteries production technology. Equivalent fuel consumption for 5 ECE cycles is 1.5 l/100 km and the remaining SOC at the end of the test is 67%. The result shows a lower consumption than the previously electric car, but a lower SOC at the end of the drive cycles. This has a negative impact on the vehicle operating range. The worse is that the batteries don't sustain the traction motor until the end of 5 EUDC cycles.

Overall System Efficiency over ECE cycles is 0.244.

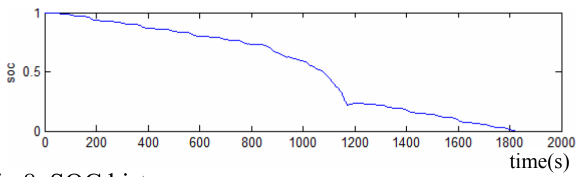

Fig.8. SOC history

The dynamic performances are:

- acceleration time 0-100km/h 14s;
- maximum acceleration 5m/s ^{2} ;
- maximum speed 124.3 km/h;

5) A hybrid electric vehicle may be the solution for less fuel consumption, less emissions and large range of functioning.

The ECE and EUDC test were performed for series configuration with a 41 kW induction motor, a 41 kW internal combustion engine, 1292 kg, MI-PbA batteries and thermostat strategy (50% hybridization) in order to compare with an 83 kW conventional vehicle. The results over ECE cycles were satisfactory. Fuel consumption starting the test with 50% SOC is 1.7l/100km and the level of SOC at the end of cycles is 42%. Starting the same test with initial SOC 100% the fuel consumption will be lower.

The highway consumption is 7.5l/100km and at the end of the tests, the SOC level is 67%. This result is at the satisfaction limit because of the starting SOC (50%) that determine the ICE engine to work more for recharging the batteries. Unfortunately it is worse than the test result of the electrical car but the major advantage is a higher level on SOC even that the initial level was 50% not 100% as in the electrical vehicle case.

The dynamic performance obtained is:

- acceleration time 0-100km/h 11.8s;
- maximum acceleration 4.5m/s ^{2} ;
- maximum speed 157.7 km/h;

6) For the same configuration as previously with Load Follower strategy the result of urban consumption test was not satisfactory compared with the other cases. Fuel consumption starting the test with 50% SOC is 13.6 l/100km and the level of SOC at the end of the drive cycles is better than the previous case: 59.6%. With the same initial SOC of 50% the highway consumption is 7l/100km and the final level of storage system is 55%. The consumption is higher than previously and SOC is the same and the dynamic performances are:

- acceleration time 0-100km/h 11.9s;
- maximum acceleration 4.5m/s ^{2} ;
- maximum speed 157.3 km/h;

7) For parallel configuration with an 41kW induction motor, an 41kW ICE, 1333 kg, MI-PbA batteries and thermostat strategy (50% hybridization) the results over ECE cycles were good to consider.

Urban fuel consumption starting the test with 50% SOC is 14.5l/100km and 56% final level for SOC. A major difference can be observed in fuel consumption if the starting level of SOC is 70% (6.9l/100 km).

The highway consumption (7.3l/100km) is not high even the starting SOC is 50% and 61% remaining SOC. With starting SOC 70% the fuel consumption is 5.5l/100 km.

As dynamic performance:

- Acceleration time 0-100km/h 9.2s;
- maximum acceleration 5m/s ^{2} ;
- maximum speed 196 km/h;

4. CONCLUSIONS

1. The conventional vehicle has a large operating range, good performance, but high fuel consumption especially at start, and high emissions. Electric vehicle with Pb batteries has no emissions but a small operating range.

2. The electric vehicle with LI batteries weighing less can provide a boost of power when needed but for short time, better performances compare to Pb electrical vehicle, but has a smaller operating range.

3. For series architecture the thermostat control strategy is better than the Load Follower.

4. The best configuration is parallel, and that's why many companies have chosen it for their models (Toyota, Honda).

REFERENCES

Aaron, B., Kristina, H., Terry H., Valerie J., Kenneth K., *ADVISOR Documentation*, NREL, 2002.

Mehrdad E., Yimin G., Sebastien E. G., *Modern Electric, Hybrid Electric, and Fuel Cell Vehicles*. Boca Raton, CRC Press, 2004.

Allen E.F., *Hybrid Vehicles and the future of personal transportation*. Boca Raton, CRC Press, 2009.

Iqbal H., *Electric and Hybrid Vehicles - Design Fundamentals*, Boca Raton, CRC Press, 2003.

Weng K.K.L., Saman K.H., and Harry C.W. *Drive and Control System for Hybrid Electric Vehicles"*, in *Handbook of Automotive Power Electronics and Motor Drives* ,(Ed**.** AliEmadi), Taylor & Francis Group, LLC, 2005.