Analysis of the economic and ecological performances in the transient regimes of the European driving cycle for a midsize SUV equipped with a DHEP, using the simulation platforms

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Abstract. Currently, the tendency of the car manufacturers is to continue the expansion of the global production of SUVs (Sport Utility Vehicle), while observing the requirements imposed by the new pollution standards by developing new technologies like DHEP (Diesel Hybrid Electric Powertrain). Experience has shown that the transient regimes are the most difficult to control from an economic and ecological perspective. As a result, this paper will highlight the behaviour of such engines that are provided in a middle class SUV (Sport Utility Vehicle), which operates in such states. We selected the transient regimes characteristic to the NMVEG (New Motor Vehicle Emissions Group) cycle. The investigations using the modelling platform AMESim allowed for rigorous interpretations for the 16 acceleration and 18 deceleration states. The results obtained from the simulation will be validated by experiments.

1. Introduction

The continuous increase of the vehicle fleet at a global level, corroborated with the climatic changes of the last decades led to the development of the various antipollution norms, with a strictness of the criteria, becoming harder and harder to be respected by the car manufacturers, norms based on the driving of a certain number of kilometres for a specific representative cycle like, for example, the NMVEG cycle, undertaken on the roller testing bench after a prior maceration at a temperature between 20 - 30 °C for the norms Euro 1(1992)…Euro 6b (2014).

The NMVEG cycle (Figure 1) is composed of an urban part, 4 cycles ECE or UDC (Urban Driving Cycle) – (Figure 2), a cycle EUDC (Extra Urban Driving Cycle) – (Figure 3) and is characterized by 3 types of regimes:

- idle speed regimes;
- transient regimes for acceleration/deceleration;
- stabilized regimes (constant speed sequences).
Figure 1. Cycle NMVEG (New Motor Vehicle Emissions Group).

The severity of the acceleration regimes specific for a testing cycle is given by a parameter called relative positive acceleration (RPA) in [m/s²], having the formula [1]:

\[ RPA = \left( \frac{1}{d_{tot}} \right) \sum_{i=1}^{n} \left\{ \begin{array}{ll} \frac{a_i v_i}{3.6} & \text{if } (a > 0) \\ 0 & \text{(otherwise)} \end{array} \right. \]  

(1)

where \( d_{tot} \) is the total distance travelled, \( a_i \) the acceleration of the vehicle in [m/s²] for time step “i” if \( a_i > 0 \) and \( v_i \) the vehicle speed in [m/s] for time step “i”.

In table 1 is presented a summary of the main characteristics of the NMVEG cycle:

| Characteristics of the cycle | Unit | NMVEG |
|------------------------------|------|-------|
| Total distance travelled     | km   | 10.993|
| Time spent to idling speed   | s    | 267   |
| Average speed (with stop)    | km/h | 33.54 |
| Average speed (without stop) | km/h | 43.16 |
| Average acceleration         | m/s² | 0.59  |
| Maximum acceleration         | m/s² | 1.04  |
| RPA ECE [1]                  | m/s² | 0.147 |
| RPA EUDC [1]                 | m/s² | 0.0936|

Noting with “r” - the idle speed regimes, with “a” - the acceleration transient regimes, with “s” - the stabilized regimes and with “d” - the deceleration transient regimes, like in figures 2 and 3, for the NMVEG cycle we obtain:

- \((3x4+2) = 14\) idle speed regimes “r”;
- \((3x4+4) = 16\) acceleration regimes “a”;
- \((4x4+2) = 18\) deceleration regimes “d”;
- \((4x4+5) = 21\) stabilized regimes, constant speed regimes “s”.

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Analysing the transient regimes marked with “a” and “d”, we observe the following:

- the acceleration regimes, “a”, have a duration of 293s meaning 25% of the total duration of the cycle.
- the deceleration regimes, “d”, have a duration of 204s from the total of 1180 s, meaning 17% of the duration of the cycle.

This means that overall the transient regimes represent 42% of the duration of the cycle. As a result, the influence of these regimes on the economy and polluting emissions becomes significant, such influences being highlighted by this paper.

2. Presentation of the model
The model is specific to a SUV from the class Renault DUSTER equipped with a propulsion system diesel hybrid electric parallel (figure 4), [2], [3]. The study was achieved with the multi-domain simulation platform, AMESim, designed to simulate three battery loading states (82%, 41% and 2%) in the transient regimes of the NMVEG cycle from which only the transient acceleration regimes were selected.

The AMESim model proposed in figure 4 is composed from 8 large components, as follows:
- a driver component which is used to control the vehicle on the cycle;
- a battery of accumulators with a voltage of 310 V;
- an electric machine (motor/power generator) of approx. 36 kW;
a computer VCU-PH (Vehicle Control Unit – Parallel Hybrid) which manages the power and the operating mode for the combustion engine and the electric engine;

- compression combustion engine - Diesel (component based on schematics for torque, polluting emissions and fuel consumption identified on an ISO-N, for an engine of 1.5l dCI 81kW Euro 6);

- mechanical transmission with ratios parameterized according to the chosen vehicle;

- an exhaust line that includes the post-treatment systems for the exhaust gases specific to a Diesel vehicle (oxidation catalyst, NOx-trap, particle filter);

- a vehicle parametrized based on an SUV from the class Renault DUSTER.

Along with these components the model also presents elements like: type of cycle used for testing, two clutches, elements simulating the inertia and frictions and various sensors to recover information regarding the status, torque, temperatures and polluting emissions.

**Figure 5.** Model AMESim proposed for a vehicle equipped with DHEP

In the model used, a component with a major impact on the representativeness of the simulation from the economy and ecology perspective is the vehicle submodel (figure 6), [4], defined in AMESim as a vehicle with constant mass allowing the user to choose from two configurations for parameterization:

- road, in this configuration, the model is based on the rolling resistance of the vehicle, the aerodynamic resistance and the resistance of the slope, if present;

- roller testing bench, for this configuration, the user introduces the friction coefficients, similar to adapting the vehicle to the roller bench to reproduce the "curve of the road".

Also, depending on the longitudinal sliding of the tire, the user can chose between two configurations:

- no sliding between the tire and the carriageway, like in this paper;

- with sliding, the user defines the parameters of the longitudinal sliding of the tire.
Returning to figure 6 (vehicle sub-model), the ports 1 and 3 are signal ports receiving the front and rear braking commands, the braking torque for each axle being determined by the input signal, the maximum braking torque and the distribution between the two axles, front and rear, and ports 2 and 4 are rotating mechanical ports that receive the driving torque and return the angular velocity for each axle.

The port 5, being a mechanical translation port, receives the external forces applied to the car and returns its linear displacement, the speed, acceleration, and the ports 6 and 7 are signal ports receiving the wind velocity and the road slope.

Regarding the configuration chosen by the authors for the simulation presented in this paper, we chose the reproduction of the behavior of a SUV car from the Renault DUSTER class on the roller bench without longitudinal slide of the tire, using the following parameters: the distribution of the mass, characteristics of the brake, moment of inertia of the wheel plus the dimensions of the tire and the coefficients used to reproduce the “curve of the road”.

For the calculation formulas used by the model for the development of the simulation we can list:

1. wheel radius:
   \[ R_R = \frac{1}{2} \cdot D_r + 0.01 \cdot H \cdot l \]  
   \[[2]\]
   where \( D_r \) is the diameter of the wheel rim \([m]\), \( H \) the height of the tire \([\%]\) and \( l \) is the width of the tire \([m]\).

2. the total mass of the vehicle being considered and the effect of the moment of inertia of the wheel during the linear movement:
   \[ m_v = m + 4 \cdot \frac{J_R}{R_R^2} \]  
   \[[3]\]
   in which \( m \) is the total weight of the vehicle in \([kg]\), \( J_R \) is the moment of inertia of the wheel \([kgm^2]\) and \( R_R \) is the radius of the wheel \([m]\).

In the configuration without the longitudinal sliding of the tire, the driving force shall be calculated by the formula:

\[ F_m = \frac{T_2 + T_4}{R_d} \]  
\[[4]\]
where $T_2$ and $T_4$ are the torque inputs for the ports 2 and 4 (for the two axles front and rear) [Nm] and $R_d$ is the dynamic radius of the wheel [m].

For the case selecting the configuration with longitudinal slide of the tire, for each axis this slide $A_L$ is considered in the submodel and is defined as:

$$A_L = 100 \cdot \frac{(R_d \cdot \omega_r \cdot \frac{\pi}{30} - v)}{v}$$

in which $R_d$ is the dynamic radius of the wheel [m], $\omega_r$ the angular velocity of the wheel [rot/min] and $v$ is the translational speed of the vehicle [m/s].

The resistance forces simulated by the vehicle submodel are given by the following relationships:

- the force given by the resistance of the slope $R_p[N]$:
  $$R_p = m \cdot g \cdot \sin(\arctan(0.01 \cdot \alpha))$$
  where $m$ is the total weight of the vehicle in [kg], $g$ the gravitational acceleration: $g = 9.81$ [m/s$^2$] and $\alpha$ is the slope of the track [%].

- the aerodynamic resistance $R_a[N]$ is calculated using the following relationship:
  $$R_a = \frac{1}{2} \cdot \rho_a \cdot C_x \cdot S \cdot (v + v_w)^2$$
  in which $\rho_a$ is the air density defined in the submodel [kg/m$^3$], $C_x$ is the aerodynamic coefficient [], $S$ is the section of the vehicle [m$^2$], $v$ the speed of the vehicle [m/s] and $v_w$ is the wind speed [m/s] used as input for the port 6 (figure 6), [4].

- the force provided by the rolling resistance $R_r[N]$ provided by the formula:
  $$R_r = m \cdot g \cdot (f + k \cdot v + wind \cdot v^2)$$
  where $f$, $k$ and $wind$ are defined by the user as constants, [4].

Then, taking into account all this forces, the resistive forces applied to the vehicle [5], [6], [7] and [8], are obtained from:

$$\sum R = R_r + R_p + R_a$$

Because the roller bench will have to reproduce all these resistive forces, the force of total resistances, [4], becomes:

$$\sum R = R_p + (a + b \cdot v + c \cdot v^2)$$

in which: $a$ [N], $b$ [N/(m/s)], $c$[N/(m/s$^2$)] are the coefficients defined by the user as characteristic to the roller bench.
3. Results and discussions

The goal of the model used, described in chapter II, is to identify in the project development stage the behavior of the polluting emissions and the fuel consumption for the transient regimes for a SUV equipped with a powertrain diesel/hybrid for the 3 states of charging of the battery (SOC – State Of Charge), respectively for SOC = 82%, SOC = 41% and SOC = 2%.

Analyzing the global results of the instantaneous engine emissions, in the transient regimes for the whole NMVEG cycle (figures 7.a, 8.a, 9.a, 10.a, 11.a) we observe that they present similarities with the evolution of the engine torque provided by the internal combustion engine, figure 7, which indicates the presence or absence of the combustion process (disabling/enabled of the electric motor), and engine coolant temperature (figure 8).

![Figure 7. Engine torque on the NMVEG cycle](image)

![Figure 8. Coolant temperature on the NMVEG cycle](image)

Realizing also the cumul of the emissions from the figures (9.a, 10.a, 11.a, 12.a, 13.a), by integrating each polluting emission depending on time for each individual transient acceleration regimes we obtain the figures (9.b, 10.b, 11.b, 12.b, 13.b). The decelerations were not considered because the engine polluting emissions are almost null during the deceleration, due to the fact that for the modern internal combustion engines the deceleration is done with injection cut during deceleration, and even with energy recovery for the hybrid powertrain.

![Figure 9.a. Instantaneous HC emissions](image)

![Figure 9.b. Cumulated HC emissions](image)

It is known that the exit of the internal combustion engine from the thermal regime has negative consequences on both the fuel consumption and the pollutants [9], [10]. In the case of figure 9.b, the cumulated HC emissions present similarities compared to the CO emissions figure 10.b, when the engine is cold, the level of the HC emissions being up to 62% higher for the regime a1, 83% higher for the regime a2 and 75% for the regime a3.
For the configuration SOC = 2%, in figure 10.b is observed that the evolution of the cumulated CO emissions in the acceleration regimes for the 4 ECEs, presents an improvement depending on the evolution of the coolant temperature, figure 8. Thus, when the engine is cold (ECE 1), the CO emissions are up to 182% higher for the regime a1, 154% higher for the regime a2 and 130% for the regime a3.

The cumulation of the NOx emissions for the transient acceleration regimes in figure 11.b, does not present a significant evolution for the first 4 ECEs. It can be observed, also, that the torque contribution provided by the electric motor in the case SOC = 82% leads to an improvement of up to 76% for the regime a13, 67% for the regime a14, 55% for the regime a15 and 15% for the regime a16. In the case SOC = 42% the improvement of the NOx emissions is of 76% for the regime a13 and only 8% for the regime a14 and 1% for the regime a15.
When analyzing the cumulated emissions of mechanical particulates for the acceleration regimes, figure 12.b, is observed both an improvement of these emissions depending on the temperature increase for the engine coolant for the combustion engine and an improvement depending on the charging state of the battery. From this, was obtained a degrading of 48% for the regime a1, 35% for the regime a2 and 27% for the regime a3 for SOC = 2% as a result of the simulation of the engine start from cold on the cycle and an improvement of 74% for the regime a13, 63% for the regime a14, 43% for the regime a15 and 17% for the regime a16. In the case SOC = 41% we obtain a reduction of the emissions of 74% for the regime a13, 8% for the regime a14 and 2% for the regime a15.

Figure 13.a. Instantaneous CO2 emissions

Figure 13.b. Cumulated CO2 emissions

For the analysis of the fuel consumption were used the summed CO2 emissions during the acceleration regimes, figure 13.b. With regard to the 4 ECEs we can observe the same improvement tendency depending on the temperature of the cooling fluid. Thus, for SOC = 2% was obtained an increase of the CO2 emissions with 55% for a1, 63% for a2 and 56% for a3. On the EUDC sequence the results show a distinct reduction of the CO2 emissions, only for the case SOC = 82% thus obtaining an improvement of 56% for a13, 19% for a14, 5% for a15 and 1% for a16. In the case SOC = 41% was obtained an improvement of 56% only for a13, the other regimes presenting a degrading of 35% for a14, 8% for a15 and 2% for a16.

4. Conclusions

During the transient acceleration regimes corresponding to the 4 ECEs of the NMVEG cycle, urban use, the technology "Diesel Hybrid Total" without charging from the domestic electrical network can provide significant gains with regard to the polluting emissions HC/CO, NOx, PM and the fuel consumption. During the study were observed zero polluting emissions, zero CO2 emissions, for SOC = 82% respectively 41%.

During the urban use of a vehicle "Diesel Hybrid" with a discharged battery, in the case of the simulation conducted for SOC = 2%, for the transient acceleration regimes, the combustion engine being the main source of energy, the polluting emissions being generated exclusively by it, the most critical regimes being (a3, a6, a9 and a12), this means that a proper optimization of the calibration of the combustion engine is required.

In the case of using a vehicle with a discharged battery, the switching from an operating mode for the internal combustion engine to another (cold ~ 20 °C to hot ~85 °C) – figure 13, is accompanied by a reduction of the polluting emissions for the transient regimes as a result of the reduction of the internal friction of the engine. For the simulation conducted the polluting emissions were reduced by 100% to 150% depending on the pollutant, figures (7.b, 8.b, 9.b, 10.b, 11.b).
As a result of the activation of the internal combustion engine (vehicle speed > 55 km/h) for the extra urban use of the vehicle, with regard to the NOx and PM emissions, we observe that the SOC has a strong negative influence only for the regime a13 (first acceleration 0-70 km/h), the battery charging state depriving the internal combustion engine of the torque contribution of the electric motor.

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