The study of the energetic performance in test cycles

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Abstract. One of the most important subsystems of modern passenger cars is the transmission. This paper aims to investigate the global performances of a modern transmission in different test cycles including the newly introduced WLTC (Worldwide Harmonized Light Vehicles Test Cycle). The study is done using a complex model developed in a performant simulation environment. Transmission efficiency calculation is emphasized, the efficiency being considered variable depending on engine torque, engine speed and gear ratio. The main important parameters (vehicle speed fluctuation, overall transmission efficiency, fuel consumption ratio) needed to compare test cycles and the transmissions performance are determined.

1. Introduction

Some of the main concerns for automotive manufacturers is to respect the mandatory emission reduction targets for new cars imposed by EU legislation. The fleet average to be achieved by all new cars by 2020-2021 is 95 g CO₂ / km, this means a fuel consumption of around 4.1 l/100 km of petrol or 3.6 l/100 km of diesel [1]. This target is based on NDEC (New European Driving Cycle) and is shown in [2] that this will translate in a WLTC (Worldwide Harmonized Light Vehicles Test Cycle) based target of 100 g CO₂ / km. The WLTC test cycle has as main objective the determination of the fuel consumption and pollutant emissions, close to the real driving. Unlike NEDC cycle, the WLTC test cycle is defined for several vehicle categories depending on the power/mass ratio and maximum speed declared by the manufacturer.

For emission prediction of newly developed vehicles the industry is heavily relying on modeling and simulation. Having such high targets it is essential to carefully choose the simplification hypothesis applied. One common simplification is the use of a constant transmission efficiency for every gear.

The transmission efficiency is dependent on several factors such as: transmitted torque, primary shaft speed, selected gear (or gear ratio), lubricant temperature, the level and quality of the lubricant and the constructive features of the transmission [3]. Therefore on different test cycles is functioning with different global efficiency. This paper aims to determine the effect of considering in simulations a variable transmission efficiency over the use of a constant value both at constant speed and in different test cycles. The efficiency is considered variable depending on transmitted moment, engine speed and coupled gear.

In order to simulate the performance, fuel consumption and emissions of a vehicle one can use LMS Imagine.Lab AMESim which is a programming environment for systems' modelling and simulation which is adequate for such studies [4], [5]. One of the main advantages is the availability of a library of components which can allow complex studies in most of the technical domains.

The fuel consumption in determined for a representative passenger car at constant velocity and for the following test cycles: US 06, Japanese 10-15, HWFET (Highway Fuel Economy Test Cycle), NEDC and WLTC. The vehicle speed fluctuation, overall transmission efficiency and fuel consumption ratio are determined and used to compare test cycles and the transmission performances.
2. Model development

Figure 1 presents the model that was developed in AMESim for the purpose of this study. The automobile's modelling for studying energetic performances needs the use of three libraries: IFP Drive, Mechanical and Signal & Control.

![Vehicle model diagram](image)

**Figure 1.** The vehicle model for energetic performances studies

The vehicle/powertrain model is a dynamic one and is composed by the following important submodels from IFP Drive library: internal combustion engine (DRVICE01D), engine control unit (DRVVECU1A), dual clutch transmission (DRVDCT01) and 1-D vehicle (DRVVEH4A). A driver submodel (DRVDRVA00A) is also needed to follow the velocity profile introduced with the mission profile and ambient data submodel (DRVMP2A). Because a single gear change law is used (for minimum fuel consumption) there is no need for a transmission electronic control unit and the mission profile submodel is used for computing the current gear. In this preliminary study the engine and transmission temperatures are considered constant and equal with the nominal ones.

The study is done for a middle class passenger car equipped with a 1.4 l turbocharged diesel engine and a 6 gears DCT (dual clutch transmission).

3. Transmission efficiency

In a general study on vehicles' fuel consumption [6], the powered axle efficiency is considered to be equal to the average value of 0.952, and the efficiency of the gearbox ($\eta_g$) is determined in each gear ($j$) using the relation (1).

$$\eta_g = \frac{1}{1 + a_j} - \frac{T_{aj}}{1 + a_j)T}$$

where: $T_g$ – resistant torque, $a_j$ – coefficient determined experimentally and $T$ – input torque.

The values obtained for the maximum input torque considering the given gearbox are centralized in table 1.

Taking into consideration the fact that will be performed also simulations where the variation of the transmission efficiency will be dependent on the transmitted torque, engine speed and coupled gear, the relation (2) is used, [6].
\[ \eta_t = \eta_{tm} - \frac{R_0 \cdot r_T}{i_c \cdot T} \]

where: \( \eta_{tm} \) – transmission efficiency with regards to the torque dependent losses; \( i_t \) – total transmission ratio; \( T \) – input torque; \( r_T \) – rolling radius; \( R_0 \) – equivalent resistance to idling losses in the transmission.

\[ \eta_{tm} = \eta_c^{N_c} \cdot \eta_{co}^{N_{co}} \cdot \eta_{ca}^{N_{ca}} \cdot \eta_{rul}^{N_{rul}} \]

where: \( \eta_c \) – efficiency of cylindrical gear; \( \eta_{co} \) – efficiency of conical gear; \( \eta_{ca} \) – efficiency of a universal joint; \( \eta_{rul} \) – efficiency of a rolling bearing; \( N_c \) – the number of pairs of cylindrical gears engaged under load; \( N_{co} \) – the number of pairs of conical gears engaged under load; \( N_{ca} \) – the number of universal joints; \( N_{rul} \) – the number of rolling bearings.

Table 1. Efficiency values in each gear for maximum input torque

| Gear | \( T_{ij} \) [Nm] | \( a_j \) [-] | \( \eta_{ij} \) |
|------|-----------------|----------|-------------|
| 1    | 0.5             | 0.060    | 0.941       |
| 2    | 0.5             | 0.056    | 0.945       |
| 3    | 0.5             | 0.047    | 0.953       |
| 4    | 0.5             | 0.035    | 0.964       |
| 5    | 0.5             | 0.02     | 0.978       |
| 6    | 0.5             | 0     | 0.998       |

The transmission efficiency afferent to torque dependent losses is determined as follow:

\[ \begin{align*}
R_0 &= R_{00} + a_1 \cdot v \\
\eta_{tm} &= \eta_{tij}^{N_i} \\
\eta_{tij} &= \eta_{tij}^{N_i} \cdot \eta_{rul}^{N_{rul}} \cdot \eta_{co}^{N_{co}} \cdot \eta_{ca}^{N_{ca}} \cdot \eta_c^{N_c}
\end{align*} \]

For the given transmission structure we have \( N_c = 2 \) and \( N_{rul} = 7 \). Considering from \([6]\) the values \( \eta_c = 0.97 \) and \( \eta_{rul} = 0.999 \) will result \( \eta_{tm} = 0.934 \).

The resistance equivalent to idling losses in the transmission depends on the velocity of the vehicle, expressed as linear function, as shown in relation (4), \([6]\).

\[ R_0 = R_{00} + a_1 \cdot v \]

where: \( R_{00} \) – low speed transmission resistance; \( a_1 \) – coefficient of velocity influence; \( v \) – vehicle velocity.

From relations (2), (4) and considering the dependency relation between transmission input speed and vehicle velocity results:

\[ \eta_t = \eta_{tm} - \left( R_{00} + a_1 \cdot \frac{\pi \cdot n_{in}}{30} \cdot \frac{r_T}{i_t} \cdot \frac{r_T}{i_t} \right) \cdot \left( i_t \cdot T \right)^{-1} \]

For front wheel drive with the engine of the vehicle transversely arranged we have: \( R_{00} = 10 \) N and \( a_1 = 0.3 \) N.m \( \cdot \frac{1}{m} \), \([6]\).

Figure 2 depicts the variation of transmission efficiency for gears 1 and 6, depending on the engine torque for two values of the engine speed. The transmission efficiency increases with the increasing engine torque for a constant speed value. It can be observed that for gear 6 the transmission efficiency has higher values and, when the engine torque increases, a bigger variation of the efficiency does not appear as it does for gear 1.

Figure 3 underlines the variation in transmission efficiency for gears 1 and 6, depending on the engine speed for two values of the engine torque. Transmission efficiency varies slightly with the increasing engine speed when the engine is working under full load.
4. Vehicle Speed Fluctuation
In order to quantify the fluctuation of vehicle speed in a test cycle, a specific parameter called VSF (Vehicle Speed Fluctuation) is defined in [7] as follow:

\[ VSF = \frac{\sigma}{m} \times 100 \% \] (6)

where: \( \sigma \) – standard deviation of vehicle speed; \( m \) – average vehicle speed.

Table 2 summarize the VSF values computed from the cycles velocity profiles ([8], [9]) using the relations for normal distribution from [10].

| Cycle  | \( m \) [km/h] | \( \sigma \) [km/h] | VSF  |
|--------|----------------|---------------------|------|
| US 06  | 83.4           | 34.2                | 41   |
| 10-15  | 33.6           | 18.2                | 54.1 |
| HWFET  | 78.2           | 15                  | 19.2 |
| NEDC   | 44.6           | 28.1                | 63   |
| WLTC   | 53.3           | 33.5                | 62.8 |
Figure 4 presents normal speed distribution and frequency of different speed values with a multiple of 10 km/h for the WLTC.

![Figure 4. Normal speed distribution for the WLTC](image)

For a speed of between 120 and 130 km/h the occurrence frequency is approximately 5%, which shows that the gearbox upper gears are used more often than in the NEDC.

5. Results

5.1 Transmission performance in test cycles

Using two power transducers to acquire the transmission input ($P_i$) and output ($P_o$) powers, the average transmission efficiency (including clutch losses) $\eta_{t\text{,cycle}}$ is calculated over the entire cycle with relation (7), and the values obtained are centralized in table 3. Only the traction regime is considered (positive power).

$$\eta_{t\text{,cycle}} = \frac{\int P_o \, dt}{\int P_i \, dt} \times 100 \, [%] \quad (7)$$

| No. | Cycle   | $\eta_{t\text{,cycle}}$ [%] | For $\eta_{t}$ constant | For $\eta_{t}$ variable |
|-----|---------|-----------------------------|-------------------------|------------------------|
| 1   | US 06   | 90.9                        | 89.5                    |                        |
| 2   | 10-15   | 81.2                        | 82.1                    |                        |
| 3   | HWFET   | 91.1                        | 89.5                    |                        |
| 4   | NEDC    | 84.9                        | 84.7                    |                        |
| 5   | WLTC    | 90.3                        | 89.4                    |                        |

One can observe an absolute difference between 0.2 % and 1.4 % between the values obtained with the two methods. In terms of CO₂ emissions the maximum difference is of 2 g CO₂ / km (a 1.3 % error).

For the cycle 10-15, the lower gear shifts are used, so the global transmission efficiency in this cycle is lower than that obtained in the other cycles. The absolute difference between maximum and minimum global efficiency is 9.9 % for $\eta_{t}$ constant and 7.4% $\eta_{t}$ variable.

Figure 5 presents the variations of transmission input and output power and of transmission efficiency in WLTC. For engine brake and idle regimes the calculated values are replaced by 1. It can be seen that for short periods the transmission efficiency decreases dramatically attaining a minimum of 35%.
5.2 Fuel consumption

In figure 6 is shown fuel consumption when driving at constant speed considering constant and variable transmission efficiency. Here, the speed at which the minimum fuel consumption is reached is around 53 km/h, which is the average speed corresponding to the WLTC cycle.

In figure 7 the fuel consumption obtained at constant speed is compared with the fuel consumption obtained in the considered test cycles (using cycle average speed as abscissa). As expected, important difference can be observed regardless of the cycle average speed.
To compare the influence on fuel consumption of different test cycles we use a parameter called FCV (Fuel Consumption Ratio) defined in [7] as follow:

$$FCR = \frac{FC_{cycle}}{FC_{ct.speed}} \cdot 100 \% \quad (8)$$

where: $FC_{cycle}$ – fuel consumption obtained in the test cycle; $FC_{ct.speed}$ – fuel consumption obtained at constant speed movement equal to the average speed of cycle;

In figure 8 is shown the correlation between FCR and VSF. A regression line having seat as intercept (imposed point) the corresponding point of the constant speed movement (0, 100) has a coefficient of determination $R^2 = 0.976$. 

**Figure 7.** Fuel consumption obtained for constant efficiency

**Figure 8.** Correlation between FCR and VSF
6. Conclusions
When the engine and transmission temperatures are considered constant, considering only the variation of transmission efficiency with the gear can be sufficient regardless of the test cycle used. The maximum error in terms of CO₂ emission for the given application was of 1.3 % when compared to the values computed using variable transmission efficiency with gear, input torque and input speed.

In terms of CO₂ emissions the type of test cycle used has a big influence with a maximum of 33 % relative difference between the minimum (in HWFET) and maximum (in 10-15). When compared using a relative parameter (FCR) the most disadvantageous cycles are NEDC and WLTC, the cycles with the highest VFS (around 63%). The linear correlation between FCR and VSF observed in [7] is also obtained for the current application and for the WLTC.

It is important to extend the study for a petrol engine application and to consider variable temperatures of engine and transmission.

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