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MODELLING THE WLTP DRIVING CYCLE OF AN AUTOMOTIVE VEHICLE POWERED BY AN INTERNAL COMBUSTION ENGINE

Dissertation within the scope of the Master’s in Mechanical Engineering, in the specialty of Energy and Environment, supervised by Professor Pedro de Figueiredo Vieira Carvalheira and presented to the Department of Mechanical Engineering.

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Modelling the WLTP Driving Cycle of an Automotive Vehicle Powered by an Internal Combustion Engine

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Modelação do Ciclo de Condução WLTP de um Veículo Automóvel Propulsionado por um Motor de Combustão Interna

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Abstract

Cars play an essential role today as means of transport. Despite the growth of the hybrid and electric vehicle market, there are still millions of vehicles powered by internal combustion engine on the road.

The main objective of this work was the development of a spreadsheet in MS Excel that simulates a Worldwide Harmonized Light Vehicle Test Procedure (WLTP) driving cycle and allows the calculation of fuel consumption and CO_2 emissions of a vehicle powered by an internal combustion engine. This spreadsheet was used to study the impact of some vehicle variables on the fuel consumption and CO_2 emissions of a light passenger vehicle.

This modelling was developed using the WLTP 3b driving cycle and a Peugeot 308 1.2 Puretech 130 cv 2018 vehicle.

One of the first problems to be solved was determining the gear to use for each instant of the test. A lot of requirements needed to be fulfilled and was the development part that took up the largest percentage of this work's time.

Additionally, it was calculated the oil temperature evolution over time, since it is also a factor that has influence on fuel consumption as well as on the friction mean effective pressure. With all the calculations made, it is then possible to determine the fuel consumption and the carbon dioxide emissions.

The parameters that were varied in this study and to compare results were: Vehicle mass, differential gear ratio, tire rolling resistance coefficient, aerodynamic drag coefficient. This work is intended to be used as support for future research and guidance.

Keywords: Peugeot 308, CO_2 emissions, fuel consumption, WLTP driving cycle.

Resumo

Os automóveis desempenham hoje um papel essencial como meio de transporte. Apesar do crescimento do mercado de veículos híbridos e elétricos, ainda existem milhões de veículos movidos a motor de combustão interna nas estradas.

O principal objetivo deste trabalho foi o desenvolvimento de uma folha de cálculo em MS Excel que simule um ciclo de condução Worldwide Harmonized Light Vehicle Test Procedure (WLTP) e permita o cálculo do consumo de combustível e das emissões de CO_2 de um veículo automóvel propulsionado por um motor de combustão interna. Esta folha de cálculo foi utilizada para estudar o impacto de algumas variáveis do veículo no consumo de combustível e nas emissões de CO_2 de um veículo automóvel ligeiro de passageiros.

Esta modelação foi desenvolvida utilizando o ciclo de condução WLTP 3b e um veículo Peugeot 308 1.2 Puretech 130 cv 2018.

Um dos primeiros problemas a serem resolvidos foi determinar a mudança engrenada para cada instante do teste. Muitos requisitos precisaram de ser satisfeitos e foi a parte do desenvolvimento que ocupou a maior percentagem do tempo deste trabalho.

Adicionalmente, foi calculada a temperatura do óleo do motor ao longo do tempo, uma vez que também é um fator que influencia tanto o consumo de combustível como a pressão média efetiva de atrito. Com todos os cálculos feitos, foi então possível determinar o consumo de combustível e as emissões de dióxido de carbono.

Os parâmetros que foram variados neste estudo e para comparação dos resultados foram: Massa do veículo, relação de transmissão do diferencial, coeficiente de resistência ao rolamento dos pneus, coeficiente de arrasto aerodinâmico. Este trabalho pretende ser usado como suporte para futuras pesquisas e orientações.

Palavras-chave: Peugeot 308, emissão de CO_2, consumo de combustível, ciclo de condução WLTP.

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List of Symbols

a_{engine} – Engine surface area (m^2)

A_f – Frontal area of the vehicle (m^2)

$a_{\text{fins radiator}}$ – Area of the radiator fins (m^2)

a_{gearbox} – Gearbox surface area (m^2)

a_j - Vehicle's acceleration at instant j (m/s^2)

b – ordinate in the origin

bmep – Brake mean effective pressure (kPa)

bsfc – Brake specific fuel consumption (g/kW·h)

$C_{p,e,\text{oil}}$ – Specific heat of the engine oil (J/kg·K)

$C_{p,e,\text{lc}}$ – Specific heat of the engine liquid coolant (J/kg·K)

$C_{p,g,\text{oil}}$ – Specific heat of the gearbox oil (J/kg·K)

C_{rr} – Tire rolling resistance coefficient

C_x – Aerodynamic drag coefficient

d_{cycle} – Total cycle distance (km)

E_{CO_2} – Total CO_2 vehicle emissions (gCO_2/km)

$E_{\text{CO}_2,stopped}$ – CO_2 emissions when the vehicle is stopped (gCO_2/km)

$E_{\text{CO}_2,moving}$ – CO_2 emissions when the vehicle is moving (gCO_2/km)

fmep – Friction mean effective pressure (kPa)

F_c – Total volumetric fuel consumption (l/100 km)

$F_{c_{\text{stopped}}}$ – Volumetric fuel consumption when the vehicle is stopped (l/100 km)

$F_{c_{\text{moving}}}$ – Volumetric fuel consumption when the vehicle is moving (l/100 km)

g – Gravity acceleration (m/s^2)

h – Convection heat transfer coefficient ($W/m^2 \cdot K$)

h_e – Engine convection heat transfer coefficient ($W/m^2 \cdot K$)

h_r - Radiator convection heat transfer coefficient ($W/m^2 \cdot K$)

k_r - Factor taking into account the inertial resistances of the drivetrain during acceleration

m – Test mass [kg]

m – slope

m_e – engine mass (kg)

m_g – gearbox mass (kg)

$m_{e,\mathrm{oil}}$ – engine oil mass (kg)

$m_{g,\mathrm{oil}}$ – gearbox oil mass (kg)

$m_{e,\mathrm{lc}}$ – engine liquid coolant mass (kg)

$\dot{m}_f$ – Fuel mass flow rate (kg/s)

$m_{f_{\mathrm{stopped}}}$ – Mass of fuel consumed when the vehicle is stopped (kg)

$m_{f_{\mathrm{moving}}}$ – Mass of fuel consumed when the vehicle is moving (kg)

n – Rotational speed (rpm)

$n_{i,j}$ – Rotational speed at gear i and instant j (rpm)

ndv_i – Ratio between engine speed in rpm and vehicle speed in km/h in gear i

n_{idle} – Idle engine rotational speed (rpm)

n_{max} – Maximum engine rotational speed (rpm)

n_{min} – Minimum engine rotational speed (rpm)

$n_{\mathrm{min\ drive}}$ – Engine lowest rotational speed for the driving cycle (rpm)

$\eta_{f,b}$ – Engine brake fuel conversion efficiency

n_{norm} – Normalised engine speed (rpm)

$P_{\mathrm{available}}$ – Available power (kW)

PMR – Power Mass Ratio (W/kg)

$P_{required}$ – Required power (kW)

$P_{\mathrm{norm,\ wot}}$ – Normalised power at wide open throttle (kW)

P_{rated} – Rated power (kW)

$\dot{Q}_c$ – Heat power transferred by convection (W)

$\dot{Q}_g$ – Heat power generated by the engine (W)

Q_{HV} – Heating Value (MJ/kg)

s – Rated engine rotational speed (rpm)

SM – Safety Margin

t_{air} – Air temperature (ºC)

T_b – Engine brake torque (N·m)

t_i – Time (s)

t_{oil} – Oil temperature (ºC)

t_{tot} – Thermostat opening temperature (ºC)

V_d – Engine displacement volume (dm^3)

v_j - Vehicle speed at instant j (km/h)

ρ_{air} – Air density (kg/m^3)

ρ_{fuel} – Fuel density (kg/m^3)

Acronyms/Abbreviations

CO_2 – Carbon dioxide

DEM – Departamento de Engenharia Mecânica

FCTUC – Faculdade de Ciências e Tecnologia da Universidade de Coimbra

GRPE – Working Party on Pollution and Energy

NEDC – New European Driving Cycle

RPM – Rotations per minute

UN-ECE – United Nations Economic Commission for Europe

WLTP – Worldwide harmonized Light vehicle Test Procedure

1. INTRODUCTION

Cars play an essential role today as means of transport. Despite the growth of the hybrid and electric vehicle market, there are still millions of vehicles powered by internal combustion engine on the road. To adapt to today's markets, vehicles with internal combustion engine must be increasingly efficient, economical, and environmentally friendly, otherwise the public will resort to other types of vehicles, and in some cases, there is even state support to buy more environmentally friendly means of transport. Before being launched on the market, these vehicles will first have to go through numerous evaluations with standardize test procedures.

The standard procedure to evaluate the fuel consumption and CO_2 emissions of automotive vehicles in the EC is the Worldwide harmonized Light vehicle Test Procedure (WLTP). There is the need to develop simulation tools to predict the fuel consumption and CO_2 emissions of automotive light vehicles in the WLTP driving cycle to be used in the design phase, because they allow to make better decisions among several design options. Another concern that motivates this study is the difference between the official fuel consumption and CO_2 emissions values and the real values perceived by vehicle users.

In this work was developed a simulation model of the WLTP driving cycle, using MS Excel. The model allows to calculate the fuel consumption and CO_2 emissions in each phase of the driving cycle and in the total driving cycle as well as when the vehicle is stopped or moving. It also allows to calculate the evolution of the engine oil temperature during the driving cycle. The model considers the vehicle physical characteristics and the brake specific fuel conversion efficiency of the vehicle engine as a function of the engine rotational speed and brake torque.

The simulations were performed with a Peugeot 308 1.2 Puretech 130 cv 2018 and compared with experimental values for the fuel consumption and CO_2 emissions, available in the literature, in the total cycle and in the cycle phases. The maximum difference between the simulation and the experimental results are less than 15%.

In this work simulations were performed to study the influence of the vehicle mass, the differential gear ratio, the tire rolling resistance coefficient and the aerodynamic drag coefficient on the fuel consumption and CO_2 emissions.

2. LITERATURE REVIEW

Driving cycle tests are developed to assess the performance of vehicles in various ways, including fuel consumption and pollutant emissions.

In the European Union (EU) the first driving cycle was designated ECE R15, established in 1970 by Council Directive 70/220/EEC, and was mandatory for vehicle type approval. The ECE R15 driving cycle was performed on a chassis dynamometer and corresponded to urban driving with a duration of 195 seconds, a total distance of 1.013 km and a maximum speed of 50 km/h, as presented in Figure 1.

In 1990 the EEC Directive 90/C81/01 introduced the Extra Urban Driving Cycle (EUDC). The EUDC has a duration of 400 seconds, a total distance of 6.955 km and a maximum speed of 120 km/h, as presented in Figure 2, and for low-powered vehicles the speed was limited to 90 km/h [1].

In the same directive, EEC Directive 90/C81/01, was also established the ECE+EUDC driving cycle, also known as MVEG-A [2]. It was performed on a chassis dynamometer, including four ECE R15 segments repeated without interruption followed by a EUDC segment as presented in Figure 3. Before the test began the vehicle is allowed to soak up for 6 hours at a test temperature between 20°C and 30°C. It was then started and allowed to idle for 40 seconds before starting the measurements of fuel consumption and pollutant emissions [3].

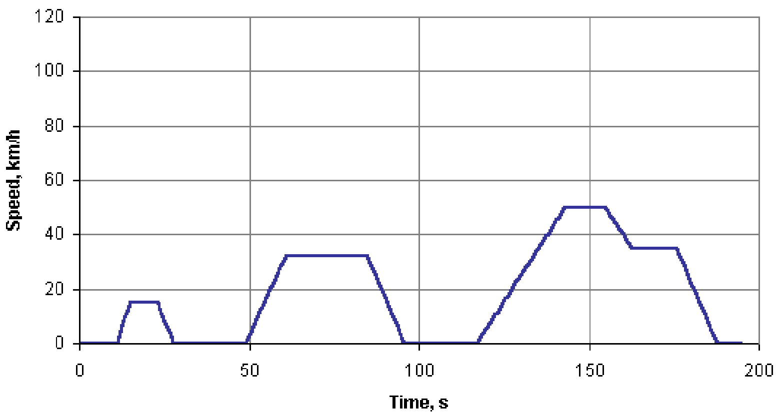

Figure 1 ECE R15 driving cycle

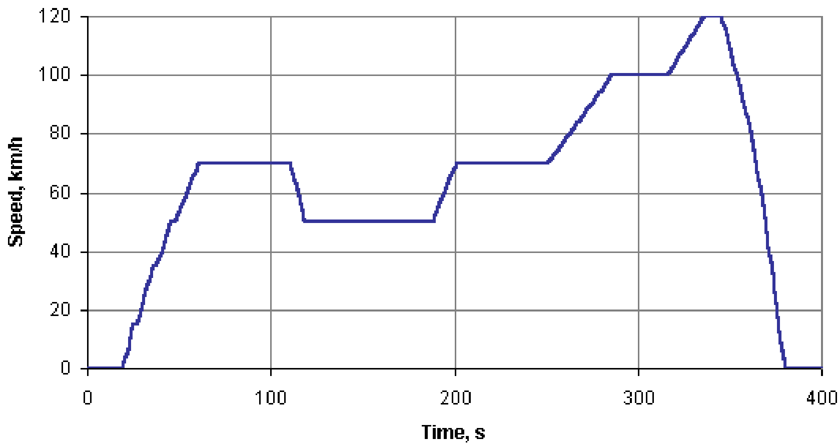

Figure 2 EUDC driving cycle

In the 00's it was implemented the New European Driving Cycle (NEDC), also known as MVEG-B, presented in Figure 3, and would bring some changes in comparison with the MVEG-A test. The idling period of 40 seconds in the beginning of the test had been eliminated, so the engine fuel consumption measurement would start at exactly 0 seconds as well as the emission samplings. This new test would start with four repetitions of the ECE cycle and then the EUDC segment to account for more aggressive high speed driving modes.

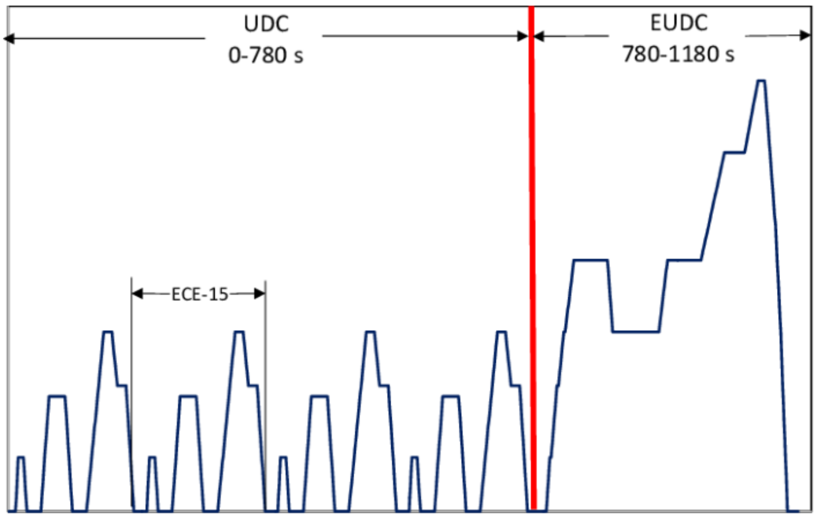

Figure 3 NEDC driving cycle

For the last update for the driving cycles until now we have the Worldwide harmonized Light vehicle Test Procedure (WLTP), the most precise driving cycle nowadays, developed by the World Forum for the Harmonization of Vehicle Regulations (WP.29) of the United Nations Economic Commission for Europe (UN-ECE) through the social unit Working Party on Pollution and Energy transport program (GRPE) [4].

It is a driving cycle meant to be used as a global test cycle across the globe, so pollutant emissions including CO_2 as well as fuel consumption values can be compared worldwidely. Note that even though the WLTP has a common global core, the European Union and other regions may apply this test in different ways depending on their road traffic laws and needs [4].

In Europe, it replaces the previous emissions test cycle, the New European Driving Cycle (NEDC), which was in use since the 00's, and it is now in replacement since 2017 [5].

The main updates from this driving cycle to the old one are the more realistic driving behaviour, greater range of driving situations (urban, suburban, main road, motorway), longer test distances, more dynamic and representative accelerations and decelerations, higher average and maximum speeds, more realistic ambient temperatures closer to the European average, shorter stops, influence of optional equipment as the CO_2 values and fuel consumption are provided for individual vehicles as built and stricter car set-up measurement conditions [5].

3. VEHICLE

In this section will be discussed the vehicle selected for the study, its technical specifications and class will be presented.

3.1. Selected Vehicle

The vehicle chosen for this study was the Peugeot 308 1.2 PureTech 130 cv 2018, since it was one of the few with its specifications available online and the most updated. It is classified as a medium size compact car which is most common among manufacturers and users, has good comfort, luggage space and an affordable price for the medium class.

At the year this model was launched, there were already standards to be met to achieve in 2020 regarding the emissions of nitrogen oxides which requires that these emissions under real conditions do not exceed 1.5 times those recorded during tests on the trials. It was the first vehicle in the Groupe PSA to debut three new engines that already comply with this requirement known as the Euro 6.2 standard.

This Peugeot 308 has a new generation of the PureTech gasoline engine with 130 cv and gasoline direct injection. This new version features performance gains, better dynamics, and lower fuel consumption and all with an impact on efficiency. The engine has evolved to allow an even more efficient reduction in pollutant emissions, regardless of the conditions of use. Figure 4 presents a visualization of the car.

Figure 4 Peugeot 308 1.2 PureTech 130 cv 2018

3.2. Vehicle Technical Specifications

Now the technical specifications of the car under study will be presented in more detail. Only the characteristics which are relevant will be presented so that we can carry out some necessary calculations and find the fuel consumption with the greatest possible precision. It will be needed to get values at the decimal of a second in each phase of the tests. Table 1 presents a summary of the required technical specifications to perform this work [6][7].

Table 1 Peugeot 308 1.2 Puretech 130 cv 2018 technical specifications

Size, Dimensions, Aerodynamics and Weight	
Wheelbase [cm]	262
Length [cm]	425.3
Width [cm]	180.4
Height [cm]	145.7
Front Axle [cm]	155.9
Rear Axle [cm]	155.3
Front Tyres Dimensions	205/55 R16
Rear Tyres Dimensions	205/55 R16
Curb Weight [kg]	1178
Power Mass Ratio [W/kg]	81.2
Aerodynamic Drag Coefficient - C_x	0.29
Tire Rolling Resistance Coefficient - C_{rr}	0.008
Performance	
Top speed [km/h]	188
Acceleration 0 to 100 km/h [s]	9.7
Gearbox	
Transmission Gearbox	6 Speed Manual
Gear Ratio [1^{st}]	0.28947
Gear Ratio [2^{nd}]	0.48837
Gear Ratio [3^{rd}]	0.77500
Gear Ratio [4^{th}]	1.02564
Gear Ratio [5^{th}]	1.31429
Gear Ratio [6^{th}]	1.65625
Gear Ratio [Reverse]	0.38710
Gear Ratio [Differential]	0.22078

Consumption	
Fuel Consumption-Open Road [l/100 km]	4.3
Fuel Consumption - City [l/100 km]	6.3
Fuel Consumption - Combined [l/100 km]	5.0
Fuel Consumption - Low WLTP [l/100 km]	7.1
Fuel Consumption - Medium WLTP [l/100 km]	5.8
Fuel Consumption - High WLTP [l/100 km]	5.1
Fuel Consumption - Extra High WLTP [l/100 km]	6.1
Fuel Consumption - Combined WLTP [l/100 km]	5.8
Range [km]	1060
CO_2 emissions [g/km]	116
CO_2 emissions WLTP [g/km]	132
Engine Technical Data	
Engine Type	Inline 3
Engine Code	EB2DTS
Fuel Type	Petrol
Fuel System	Direct Injection
Compression Ratio	10.5
Maximum Brake Power [kW]	96
Engine Speed at Maximum Brake Power [rpm]	5500
Maximum Brake Torque [N·m]	230
Engine Speed at Maximum Brake Torque [rpm]	1750
Traction	Front Wheel Drive

3.3. Vehicle’s Class

The operation of this driving cycle will be described below. It is divided into four distinct phases, Low, Medium, High and Extra High with a total time of 1800 seconds.

It is also important to consider the vehicle’s classification for the process, because for each of them we will have different speeds during the driving cycle. This classification is based on the Power to Mass ratio (PMR). In Figure 5 and Figure 6 is presented the corresponding category and their respective maximum speeds in each of the phases [4].

Category	PMR, W/kg	v_max, km/h	Speed Phase Sequence
Class 3b	PMR > 34	v_max ≥ 120	Low 3 + Medium 3-2 + High 3-2 + Extra High 3
Class 3a		v_max < 120	Low 3 + Medium 3-1 + High 3-1 + Extra High 3
Class 2	34 ≥ PMR > 22	-	Low 2 + Medium 2 + High 2 + Extra High 2
Class 1	PMR ≤ 22	-	Low 1 + Medium 1 + Low 1

Figure 5 WLTP driving cycle for each category [8]

Phase	Duration	Stop Duration	Distance	p_stop	v_max	v_ave w/o stops	v_ave w/ stops	a_min	a_max
	s	s	m		km/h	km/h	km/h	m/s²	m/s²
Class 3b (v_max ≥ 120 km/h)									
Low 3	589	156	3095	26.5%	56.5	25.7	18.9	-1.47	1.47
Medium 3-2	433	48	4756	11.1%	76.6	44.5	39.5	-1.49	1.57
High 3-2	455	31	7162	6.8%	97.4	60.8	56.7	-1.49	1.58
Extra-High 3	323	7	8254	2.2%	131.3	94.0	92.0	-1.21	1.03
Total	**1800**	**242**	**23266**						
Class 3a (v_max < 120 km/h)									
Low 3	589	156	3095	26.5%	56.5	25.7	18.9	-1.47	1.47
Medium 3-1	433	48	4721	11.1%	76.6	44.1	39.3	-1.47	1.28
High 3-1	455	31	7124	6.8%	97.4	60.5	56.4	-1.49	1.58
Extra-High 3	323	7	8254	2.2%	131.3	94.0	92.0	-1.21	1.03
Total	**1800**	**242**	**23194**						

Figure 6 WLTP Class 3 cycles: selected parameters [8]

Each of these driving phases represents a specific driving style, the Low phase being classified as urban driving and therefore will register a maximum speed of 56.5 km/h. The Medium phase and High phase represent extra-urban driving, reaching speed of 76.6 km/h and 97.4 km/h respectively. Finally, we will have the Extra High phase, which represents driving on the motorway where the speed of 131.3 km/h will be reached. In Figure 7 we can see both the velocity and acceleration profile for a Class 3b WLTP driving cycle.

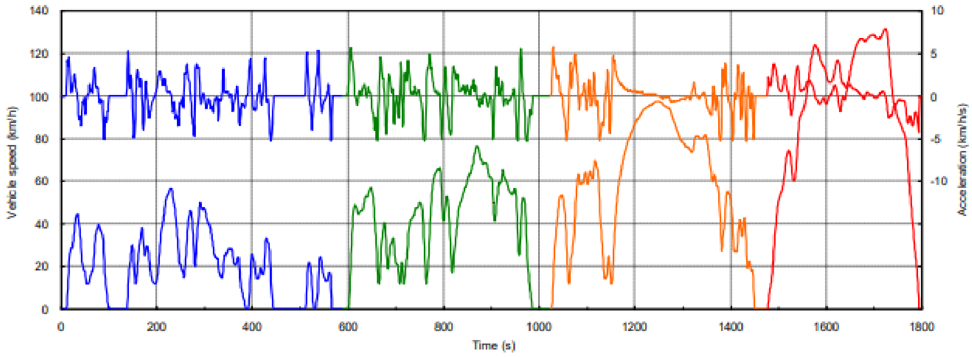

Figure 7 The speed and acceleration profile of the WLTP 3b driving cycle [4]

4. GEARSHIFT CALCULATION

In this chapter, the calculation method developed for this project will be presented.

In the first section 4.1, we will have a brief explanation of the model used to carry out this calculation as well as which parameters to consider when carrying out the calculations of the driving cycle for the chosen vehicle.

Section 4.2 shows the method used to determine the gear selection, as well as the criteria and requirements that will need to be considered. After this process, the table with the most suitable gears for each instant of this driving cycle can be developed using programming in Excel.

This last part of the Excel programming for the determination of the gears will not be presented here in table due to its extension, therefore, in the following chapter, regarding the simulations and experimental tests, will already be carried out with the values of the gears determined in this chapter.

4.1. Model Explanation

The instrument used to perform all calculations in the course of this work was the Microsoft Spreadsheet Excel, developed by Carvalheira (2018), created to model CO_2 emissions and fuel consumption of a Diesel engine powered vehicle for the NEDC [1].

This numerical model allows the calculation of the fuel mass flow at any working point as well as brake torque, engine rotational speed and oil temperature. It is also considered the vehicle’s aerodynamic drag, tire rolling resistance and powertrain. The vehicle speed is already given in the WLTP technical report tables [4] and with a timestep of 0.10 seconds between each segment of the calculation.

4.2. Gear Selection and Shift Point Determination

For the vehicle's engine, the EB2DTS, it is our goal to determine the mass fuel flow rate at each working point defined by a pair of values of engine rotational speed and bmep. To obtain these data was used Figure 8, which represents the engine brake fuel conversion efficiency map as a function of engine brake torque and rotational speed. The points selected to obtain these data were the intersections of the isolines of brake fuel conversion efficiency with the lines of constant engine rotational speed. For each point selected was calculated the mass fuel flow rate and bmep using equations (1), (2) and (3) and the engine displacement volume. For each selected engine rotational speed was made a plot of the mass fuel flow rate as a function of bmep for the selected working points at this engine rotational speed. For each selected engine rotational speed, a straight line was fitted to the plot of the mass fuel flow rate as a function of bmep. For each plot was determined the slope and ordinate in the origin of the straight line fitted. A plot was made of the slope of the straight line as a function of the engine rotational speed and a polynomial of second degree was fitted to the data, as presented in Figure 9. Then a plot was made of the ordinate in the origin of the straight line as a function of the engine rotational speed and a polynomial of second degree was fitted to the data. With these data was possible to determine the mass fuel flow rate of the engine at any working point of the engine defined by an engine rotational speed and bmep using the following procedure. With the engine rotational speed of the working point was possible to determine the slope and ordinate of the origin of the straight line using the polynomial of second degree that give the slope and the ordinate of the origin in the straight line for each engine rotational speed. With this slope and ordinate of the origin it is possible to calculate the mass fuel flow rate using the equation of the straight line and the bmep of the working point.

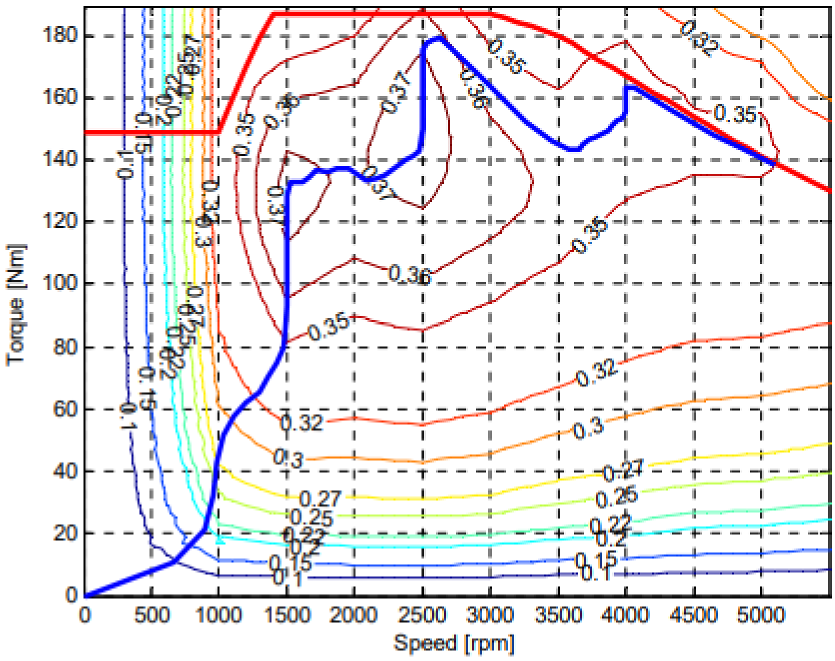

Figure 8 Efficiency Map, Full-Load Line (blue line) & OOL (red line) of the EB2DT engine [9]

For each working point the brake specific fuel consumption, bsfc, is given by Eq. (1) where $\eta_{f,b}$ represents the brake fuel conversion efficiency of the EB2DT engine in the working point, obtained from Figure 8, and Q_{HV} is the heating value of the fuel. In Eq. (2), T_b stands for engine brake torque and V_d for displacement volume of the engine. The fuel mass flow rate, $\dot{m}_f$, is calculated using Eq. (3).

$$\text{bsfc[g/kWh]} = \frac{3600}{\eta_{f,b} \times Q_{\text{HV}}[\text{MJ/kg}]} \tag{1}$$

$$\text{bmep[kPa]} = \frac{4 \times \pi \times T_b[\text{N} \cdot \text{m}]}{V_d[\text{dm}^3]} \tag{2}$$

$$\dot{m}_f[\text{kg/s}] = \frac{\text{bsfc[g/kWh]} \times T_b[\text{N} \cdot \text{m}] \times 2\pi \times n[\text{rpm}]}{60 \times 3.6 \times 10^9} \tag{3}$$

In Appendix 1 it is possible to check the tables and corresponding graphics used for these calculations. After these it was possible to formulate a set of tables, each for a different engine rotational speed beginning at 750 rpm, then 1000 rpm and up to 5500 rpm with increments of 500 rpm of the fuel mass flow rate as a function of bmep, presented in Appendix 1.

Table 2 Peugeot EB2DT engine torque, bsfc, normalized power and fmep

n (rpm)	T_b (N.m)	P_b (kW)	bsfc_$T_{b\max}$ (g/kW.h)	n_{norm}	P_{norm}	m	b	fmep (kPa)
750	96.50	7.58	335.66	0.0000	0.07880	5.82E-07	1.15E-04	-1.97E+02
1000	135.00	14.14	250.50	0.0526	0.14698	6.25E-07	9.70E-05	-1.55E+02
1500	212.00	33.30	258.20	0.1579	0.34622	8.20E-07	1.03E-04	-1.25E+02
1750	230.00	42.15	260.22	0.2105	0.43821			
2000	230.00	48.17	262.24	0.2632	0.50082	1.15E-06	1.98E-04	-1.72E+02
2500	230.00	60.21	254.29	0.3684	0.62602	1.40E-06	2.54E-04	-1.81E+02
3000	230.00	72.26	262.24	0.4737	0.75122	1.71E-06	3.12E-04	-1.82E+02
3500	227.00	83.20	279.72	0.5789	0.86500	2.05E-06	3.95E-04	-1.93E+02
4000	212.50	89.01	279.72	0.6842	0.92542	2.36E-06	4.81E-04	-2.04E+02
4500	197.00	92.83	281.60	0.7895	0.96516	2.59E-06	6.14E-04	-2.37E+02
5000	182.00	95.29	285.43	0.8947	0.99075	2.84E-06	7.24E-04	-2.55E+02
5500	167.00	96.19	289.37	1.0000	1.00000	3.42E-06	7.39E-04	-2.16E+02
6000	153.00	96.13	294.00	1.1053	0.99946			
6250	146.00	95.56	297.00	1.1579	0.99347			

Figure 9 presents the evolution of the slope of the straight line of fuel mass flow rate as a function of bmep, with engine rotational speed.

Figure 10 presents the evolution of the maximum brake torque and of the brake specific fuel consumption at maximum brake torque with the engine rotational speed.

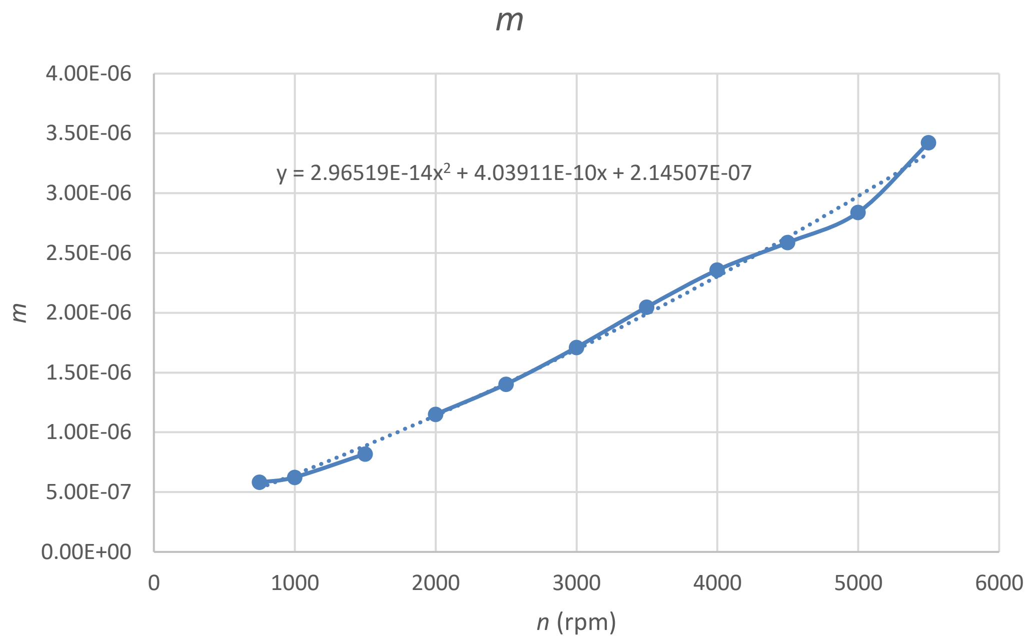

Figure 9 Evolution of the slope of the straight line of fuel mass flow rate as a function of bmep, with engine rotational speed

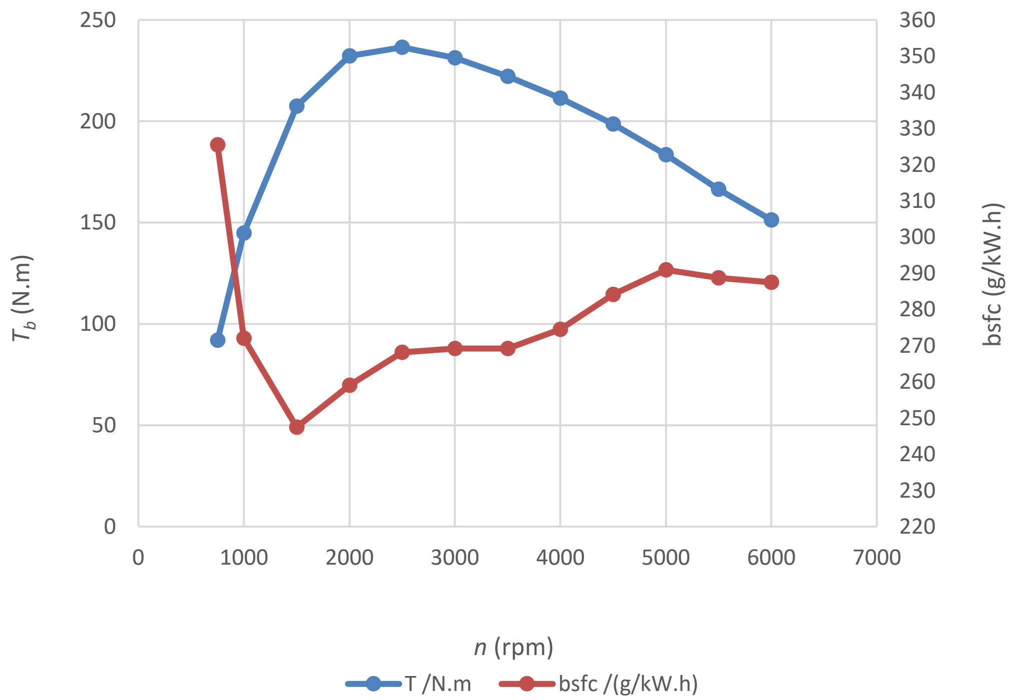

Figure 10 Evolution of maximum engine brake torque and engine brake specific fuel consumption at maximum brake torque with engine rotational speed

4.2.1 Calculation of Required Power

We will then start by determining the power required at each instant to overcome the driving resistance and to accelerate, $P_{\text{required},j}$, given by Eq. (4), where k_r represents the factor taking into account the inertial resistances of the drivetrain during acceleration and is set to 1.1, a_j is the vehicle's acceleration at instant *j* and v_j is the vehicle speed at instant *j* [10].

$$P_{\text{required},j}[\text{kW}] = \left(\frac{f_0 \times v_j + f_1 \times v_j^2 + f_2 \times v_j^3}{3600}\right) + \frac{k_r \times a_j \times v_j \times m[\text{kg}]}{3600} \quad (4)$$

The values for f_0, f_1 and f_2 are obtained using the following equations respectively:

$$f_0 = C_{rr} \times m[\text{kg}] \times g[\text{m/s}^2] \quad (5)$$

$$f_1 = 0 \quad (6)$$

$$f_2 = \frac{C_x \times A_f[\text{m}^2] \times 0.5 \times \rho_{\text{air}}[\text{kg/m}^3]}{3.6^2} \quad (7)$$

For the equations above, C_{rr} represents the tire rolling resistance coefficient, *m* is the test mass, g is the gravity acceleration, C_x the drag coefficient, A_f the frontal area of the vehicle and ρ_{air} the air density.

s

4.2.2 Determination of Engine Speeds

Regarding the determination of engine speeds, the following requirements must be met:

- For each instant where the speed (in km/h) is inferior to 1, the engine speed must be in idle and the gear in neutral with the clutch engaged.

- For speeds higher than 1, from first to sixth gears, the engine speed should be calculated using Eq. (8), where ndv_i is the ratio between engine speed in rpm and vehicle speed in km/h in gear *i* and v_j corresponds to the speed in km/h:

$$n_{i,j}[\mathrm{rpm}] = ndv_i \times v_j[\mathrm{km/h}] \tag{8}$$

- Only gears that result in engine rotational speeds within the range $n_{\min} \leq n_{i,j} \leq n_{\max}$, are usable.

If the gear is above the second, $n_{\max}$ is given by Eq. (9) and $n_{\min}$ by Eq. (10), where s corresponds to the rated engine rotational speed in rpm (at rated power) and $n_{\min\,drive}$ given by Eq. **(**11**)** is the minimum engine speed for gears above the 2nd and the vehicle in motion.

$$n_{\max}[\mathrm{rpm}] = 1.2 \times (s[\mathrm{rpm}] - n_{\mathrm{idle}}[\mathrm{rpm}]) + n_{\mathrm{idle}}[\mathrm{rpm}] \tag{9}$$

$$n_{\min}[\mathrm{rpm}] = n_{\min\,drive}[\mathrm{rpm}] \tag{10}$$

$$n_{\min\,drive}[\mathrm{rpm}] = n_{\mathrm{idle}}[\mathrm{rpm}] + 0.125 \times (\mathrm{s} - n_{\mathrm{idle}}[\mathrm{rpm}]) \tag{11}$$

- If vehicle is in the second gear and $ndv_2 \times v_j \geq 0.9 \times n_{\mathrm{idle}}$, $n_{\min}$ is given by Eq. (12) where ndv_2 is the ratio between engine speed in rpm and vehicle speed in km/h in gear 2 and *s* is the rated engine rotational speed in rpm (at rated power) .

$$n_{\min}[\mathrm{rpm}] = \max(1.15 \times n_{\mathrm{idle}}; 0.03 \times (s - n_{\mathrm{idle}}) + n_{\mathrm{idle}}) \tag{12}$$

- In the specific case where $ndv_2 \times v_j < \max(1.15 \times n_{idle}; 0.03 \times (s - n_{\text{idle}}) + n_{\text{idle}})$, the clutch shall be disengaged.

- For the first gear:

$$n_{\text{min}}[\text{rpm}] = n_{\text{idle}}[\text{rpm}] \tag{13}$$

4.2.3 Calculation of Available Power

The possible gears to use shall comply with the following condition, $P_{\text{available},ij} \geq P_{\text{required},j}$. The available power must be calculated using Eq. (14).

$$P_{\text{available},ij}[\text{kW}] = P_{\text{norm}_{\text{wot}}}(n_{\text{norm},ij}) \times P_{\text{rated}}[\text{kW}] \times \text{SM} \tag{14}$$

The rated power, P_{rated}, is already given in the vehicle's specifications and the SM (Safety Margin) for this test is set to 0.9. It is only necessary to calculate the normalised power at wide open throttle for which we will first need to use Eq. (15) to calculate the normalised engine rotational speed at instant j and gear i, $n_{\text{norm}\ i,j}$.

$$n_{\text{norm},ij} = \frac{ndv_i[\text{rpm/km/h}] \times v_j[\text{km/h}] - n_{\text{idle}}[\text{rpm}]}{s[\text{rpm}] - n_{\text{idle}}[\text{rpm}]} \tag{15}$$

Therefore, the normalised power and consequently the available power for each instant can be calculated using the equation presented in Figure 11 for the evolution of normalised power with normalised engine rotational speed.

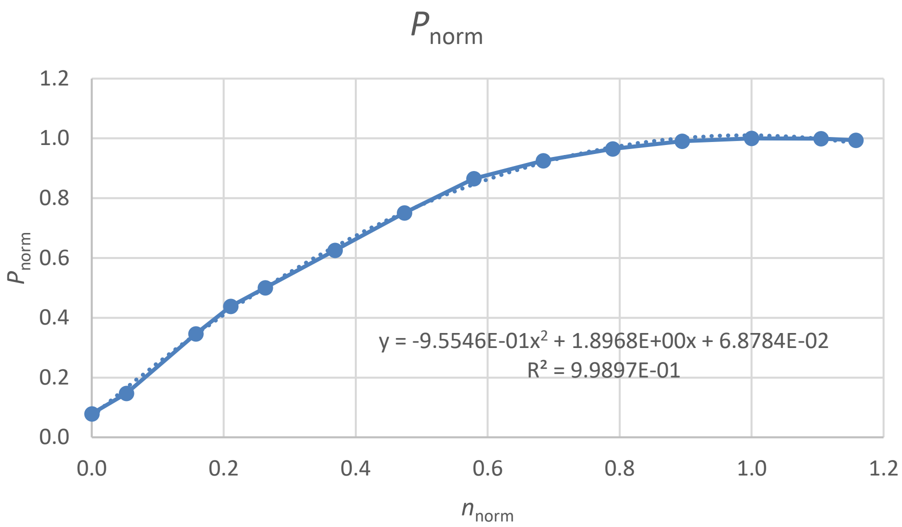

Figure 11 Normalised power as a function of normalised engine rotational speed

4.2.4 Additional Requirements for Gear Use

There are more requirements to be fulfilled when selecting the gear that do not require any type of calculation, however they must necessarily be considered when programming in MS Excel.

To see the remaining details that need checking for the rules of this driving cycle, consult Annex 1.

5. OTHER FACTORS THAT INFLUENCE FUEL CONSUMPTION AND CO_2 EMISSIONS

Even though data presented in engine fuel consumption maps already allows a good calculation of fuel mass flow rate for a certain brake torque and engine rotational speed, this is only accurate when the engine brake torque is positive. For null or negative torque, it is necessary a better method to accurately calculate the fuel mass flow rate.

To do so, in section 5.1, first it will be determined the engine oil temperature at each instant. Another factor that will impact the fuel consumption is the oil temperature. It initially begins at the room temperature, and with the course of time it will increase until it stabilizes at a certain value. With the oil at the initial low temperature, it will slightly increase the fuel consumption.

In section 5.2 it will be determined the fmep, necessary for the calculations in chapter 6.

5.1. fmep

The fmep values of the engine in each point of the cycle are function of the engine rotational speed and the oil temperature. Reference [11] presents graphs of the evolution of fmep with engine rotational speed for oil temperatures of 25 ºC, 50 ºC, 75 ºC and 100 ºC as it is shown respectively in Figure 12, Figure 13, Figure 14 and Figure 15.

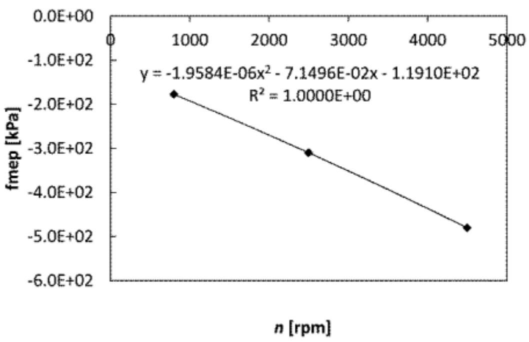

Figure 12 Evolution of **f**mep as function of engine speed for an oil at 25 °C [11]

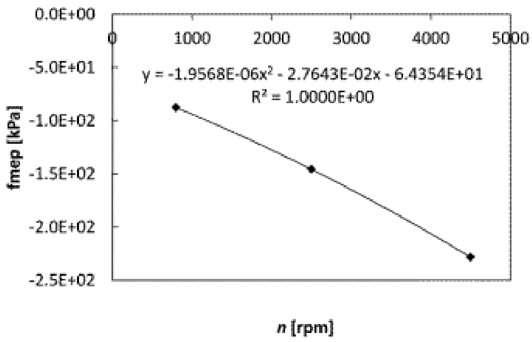

Figure 13 Evolution of fmep as function of engine speed for an oil at 50 °C [11]

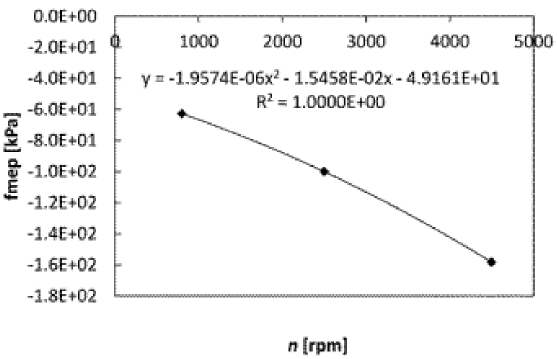

Figure 14 Evolution of fmep as function of engine speed for an oil at 75 °C [11]

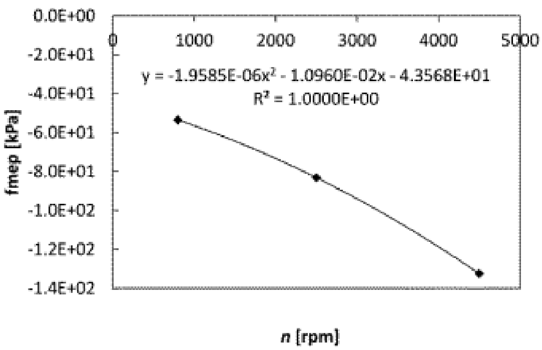

Figure 15 Evolution of fmep as function of engine speed for an oil at 100 °C [11]

Based on Figure 12, Figure 13, Figure 14 and Figure 15 for each engine oil lubricant temperature, fmep can be calculated as a function of engine rotational speed, n, by Eq. (16), where the coefficients of the second order polynomial are functions of the oil temperature t_{oil} in ºC.

$$\text{fmep}(t_{\text{oil}}, n)[\text{kPa}] = a_f(t_{\text{oil}}) \times n^2[\text{rpm}] + b_f(t_{\text{oil}}) \times n[\text{rpm}] + c_f(t_{\text{oil}}) \qquad (16)$$

The evolution of a_f, b_f and c_f with the engine oil temperature and the respective polynomial equation of third order are presented respectively in Figure 16, Figure 17 and Figure 18.

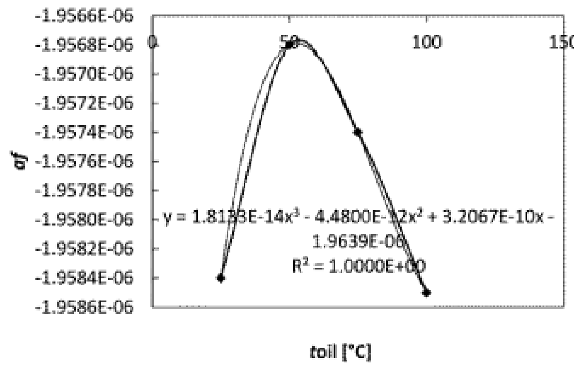

Figure 16 Evolution of a_f with the engine oil temperature t_{oil} in °C [11]

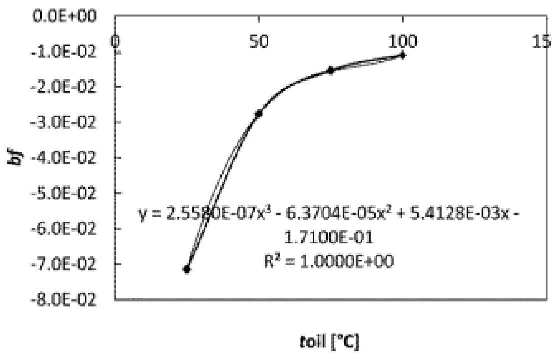

Figure 17 Evolution of b_f with the engine oil temperature t_{oil} in °C [11]

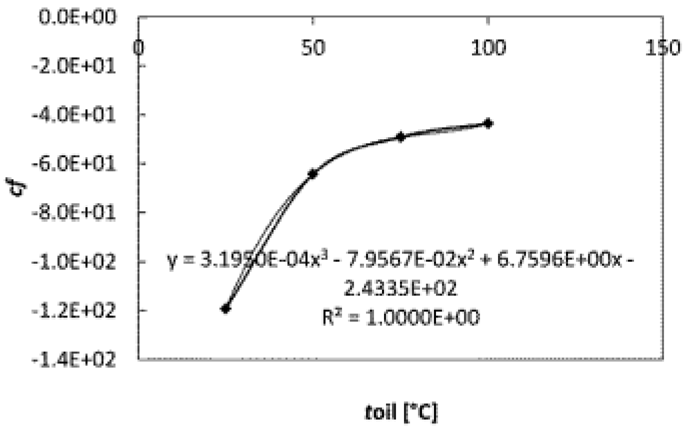

Figure 18 Evolution of c_f with the engine oil temperature t_{oil} in °C [11]

The equations consequently used to calculate $a_f(t_{\mathrm{oil}})$, $b_f(t_{\mathrm{oil}})$ and $c_f(t_{\mathrm{oil}})$ are Eq. (17), (18) and (19) respectively.

$$a_f(t_{\mathrm{oil}}) = a_{3af} \times t_{oil}^3[°\mathrm{C}] + a_{2af} \times t_{oil}^2[°\mathrm{C}] + a_{1af} \times t_{\mathrm{oil}}[°\mathrm{C}] + a_{0af} \tag{17}$$

$$b_f(t_{\mathrm{oil}}) = a_{3bf} \times t_{\mathrm{oil}}^3[°\mathrm{C}] + a_{2bf} \times t_{\mathrm{oil}}^2[°\mathrm{C}] + a_{1bf} \times t_{\mathrm{oil}}[°\mathrm{C}] + a_{0bf} \tag{18}$$

$$c_f(t_{\mathrm{oil}}) = a_{3cf} \times t_{\mathrm{oil}}^3[°\mathrm{C}] + a_{2cf} \times t_{\mathrm{oil}}^2[°\mathrm{C}] + a_{1cf} \times t_{\mathrm{oil}}[°\mathrm{C}] + a_{0cf} \tag{19}$$

With the values of a_f, b_f and c_f presented in Table 3, which apply to the Volkswagen 2.0 TDI CBDB engine code, were calculated the fmep as a function of engine rotational speed for different engine oil temperatures, respectively, 100 °C, 75 °C, 50 °C and 25 °C. The fmep for 100 °C for the PSA EB2DT engine was calculated using Eq. (20), where m is the slope and b the ordinate in the origin of the straight line fit to the evolution of mass fuel flow rate with bmep for a given engine rotational speed.

$$\text{fmep[kPa]} = \frac{-b}{m} \tag{20}$$

After that was calculated the difference between fmep for 100 °C and 75 °C, between fmep for 100 °C and 50 °C and between fmep for 100 °C and 25 °C for the Volkswagen 2.0 TDI CBDB engine. It was assumed that the difference between fmep for 100 °C and 75 °C, between fmep for 100 °C and 50 °C and between fmep for 100 °C and 25 °C was equal for the PSA EB2DT engine and Volkswagen 2.0 TDI CBDB engine. The fmep for the PSA engine at 75 °C and a given engine rotational speed was calculated adding the fmep of the PSA EB2DT engine at 100°C and the same engine rotational speed with a difference of fmep between 75 °C and 100 °C for the Volkswagen 2.0 TDI CBDB engine at the same engine rotational speed. The fmep for the PSA EB2DT engine at 50 °C and a given engine rotational speed was calculated adding the fmep of the PSA EB2DT engine at 100 °C and the same engine rotational speed with a difference of fmep between 50 °C and 100 °C for the Volkswagen 2.0 TDI CBDB engine at the same engine rotational speed. The fmep for the PSA EB2DT engine at 25 °C and a given engine rotational speed was calculated adding the fmep of the PSA EB2DT engine at 100 °C and the same engine rotational speed with a difference of fmep between 25 °C and 100 °C for the Volkswagen 2.0 TDI CBDB engine at the same engine rotational speed.

The values of fmep as a function of engine rotational speed for engine oil temperature of 100 °C ,75 °C, 50 °C and 25 °C for the PSA EB2DT engine are plotted respectively in Figure 19, Figure 20, Figure 21 and Figure 22.

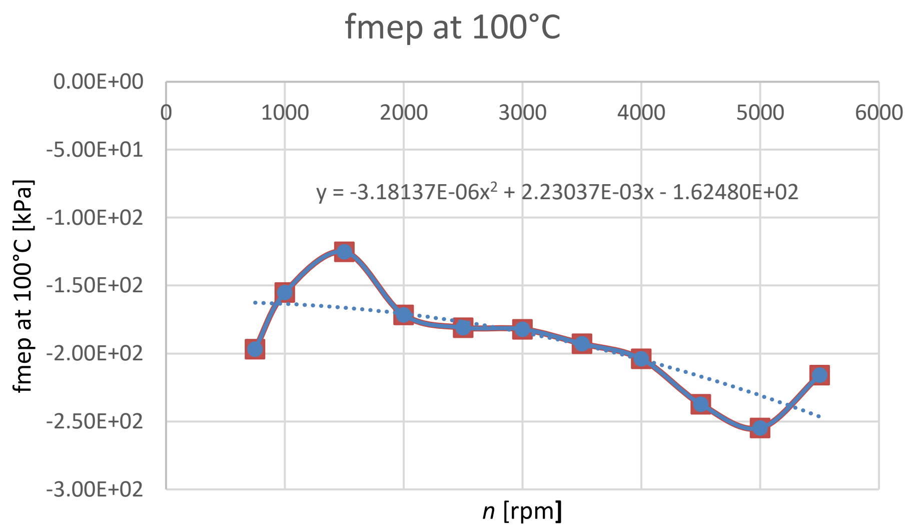

Figure 19 fmep as a function of engine speed with t_{oil} = 100°C for PSA EB2DT engine

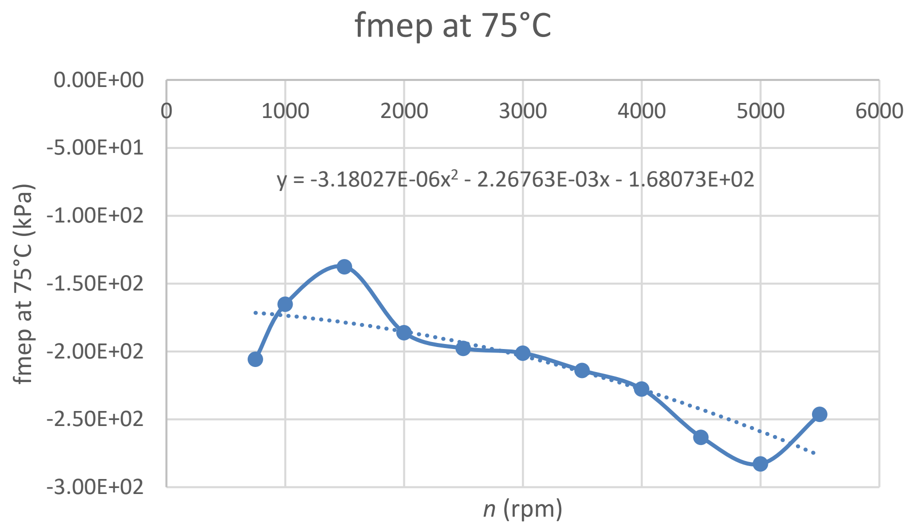

Figure 20 fmep as a function of engine speed with t_{oil} = 75°C for PSA EB2DT engine

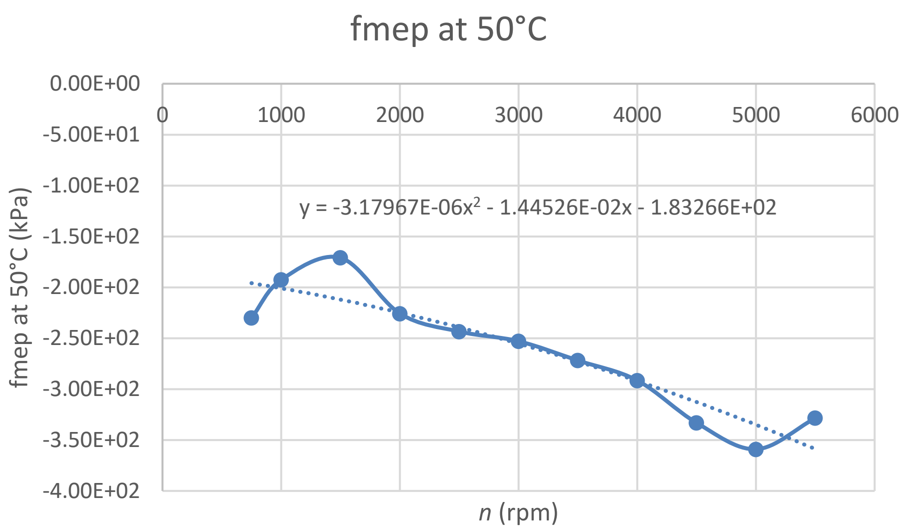

Figure 21 fmep as a function of engine speed with t_{oil} = 50°C for PSA EB2DT engine

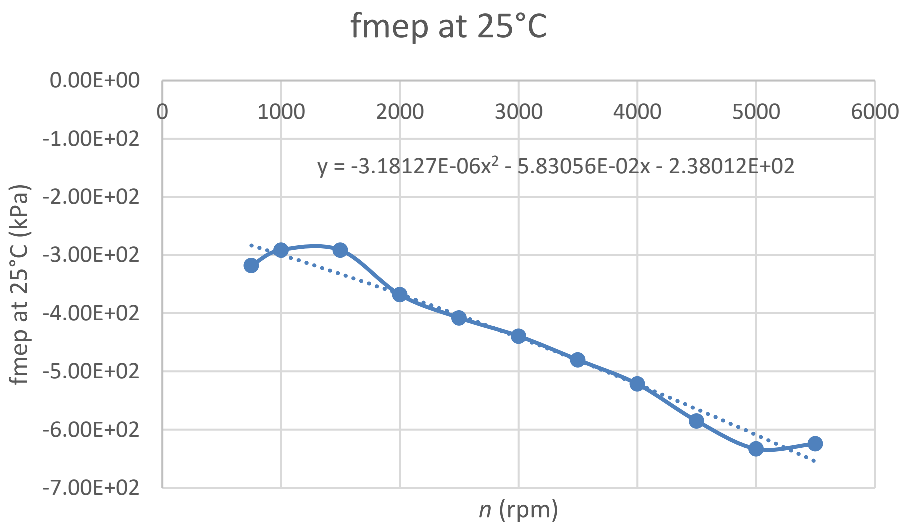

Figure 22 fmep as a function of engine speed with t_{oil} = 25°C for PSA EB2DT engine

Table 3 Coefficients a_f, b_f and c_f for different engine oil temperatures for the VW 2.0 TDI, engine code CBDB [11].

t [°C]	100	75	50	25
a_f	-1.9585E-06	-1.9574E-06	-1.9568E-06	-1.9584E-06
b_f	-1.0960E-02	-1.5458E-02	-2.7643E-02	-7.1496E-02
c_f	-4.3568E+01	-4.9161E+01	-6.4354E+01	-1.1910E+02

Table 4 Coefficients a_f, b_f and c_f for different engine oil temperatures for the PSA EB2DT engine.

t [°C]	100	75	50	25
a_f	-3.1814E-06	-3.1803E-06	-3.1797E-06	-3.1813E-06
b_f	2.2304E-03	-2.2676E-03	-1.4453E-02	-5.8306E-02
c_f	-1.6248E+02	-1.6807E+02	-1.8327E+02	-2.3801E+02

Plots were then made for the values of a_f, b_f and c_f as a function of the engine oil temperature and the polynomial of the third degree was fitted to the data for each of them. The plots of a_f, b_f and c_f as a function of engine oil temperature and the polynomial of third degree fitted to the data are presented respectively in Figure 23, Figure 24 and Figure 25.

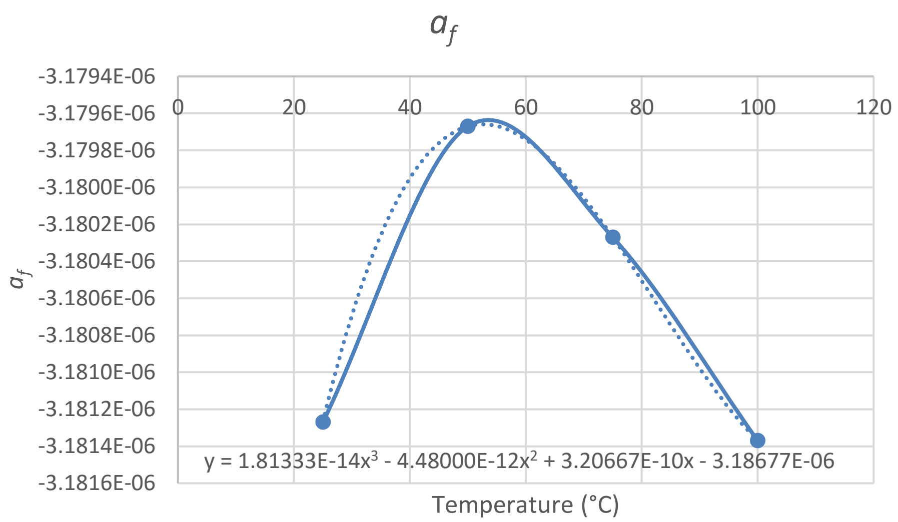

Figure 23 Evolution of coefficient a_f with engine oil temperature for the PSA EB2DT engine

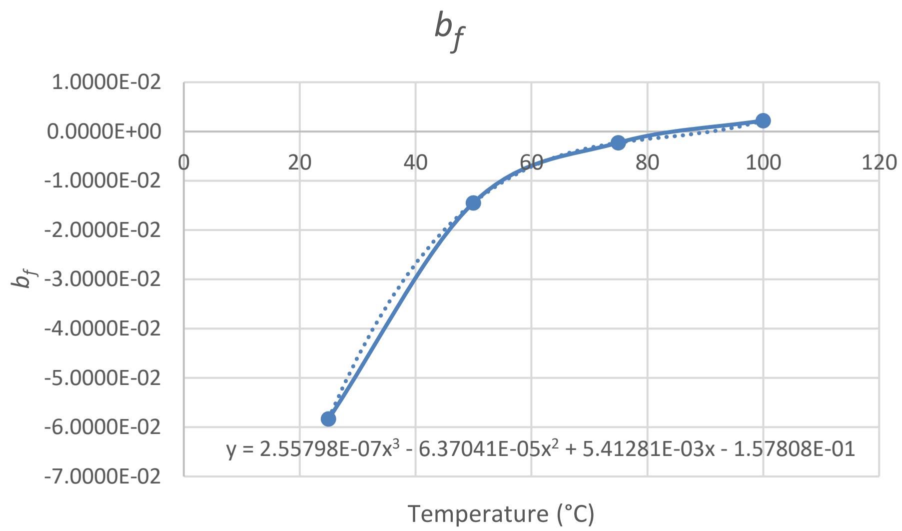

Figure 24 Evolution of coefficient b_f with engine oil temperature for the PSA EB2DT engine

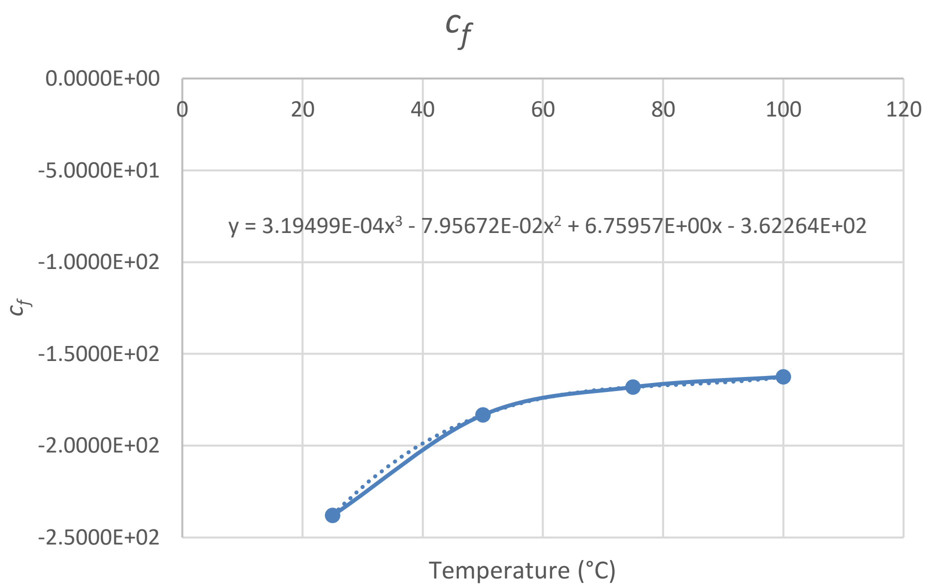

Figure 25 Evolution of coefficient c_f with engine oil temperature for the PSA EB2DT engine

During the driving cycle when the engine is at a given oil temperature and rotational speed the values of a_f, b_f and c_f are calculated respectively using Eq. (17), (18) and (19) for the given engine oil temperature. With this a_f, b_f and c_f values for the engine oil temperature, the engine fmep is calculated using Eq. (16) with given engine rotational speed.

5.2. Engine Oil Temperature

Knowing now the values of the necessary coefficients, we will finally determine the oil temperature evolution during the driving cycle. In order to proceed, we need to know some data about the geometry of the engine, which is given in Table 5, data about the geometry of the radiator in Table 6, data about the geometry of the gearbox in Table 7 and data about the specific heat and mass of the engine, gearbox, engine coolant liquid and engine and gearbox lubricants in Table 8.

Table 5 Engine dimensions

Engine	
Length [m]	0.525
Height [m]	0.687
Width [m]	0.569
Surface area [m^2]	2.101
h [$W/m^2 \cdot K$]	10.00

Table 6 Radiator dimensions

Radiator	
Length [m]	0.58
Width [m]	0.30
Fins surface area/projected area	10.0
Fins surface area [m^2]	1.74
Fin length [m]	0.05
h [$W/m^2 \cdot K$]	200

Table 7 Gearbox dimensions

Gearbox	
Length [m]	0.4442
Height [m]	0.40
Width [m]	0.35
Surface area [m^2]	0.946
h [$W/m^2 \cdot K$]	10.0

Table 8 Specific heat and mass of engine, gearbox, engine coolant liquid and engine and gearbox lubricants

	m [kg]	c_p [J/kg·K]	$m*c_p$ [J/K]
Engine	100	630	63000
Gearbox	36.228	607.5	22008.51
Engine coolant liquid	7	4186	29302
Engine lubricant oil	3.52	1900	6688
Gearbox lubricant oil	1.672	1900	3176.8
Total	148.42		124175.31
Thermostat temperature [ºC]	82		
V_d [dm^3]	1.198		

Using the data presented in Figure 9, Figure 23, Figure 24 and Figure 25 with the given equations we have the data to find the values of fmep and bmep overtime, using consequently Eq. (16) and Eq. (2).

When bmep is larger or equal to fmep, the fuel mass flow rate as function of the engine speed and engine oil temperature is given by Eq. (21).

$$\begin{gathered}\text{if bmep[kPa]} \geq \text{fmep[kPa]} \\ \dot{m}_f[\text{kg/s}] = m[\text{kg/(s}\cdot\text{kPa)}] \times (\text{bmep[kPa]} - \text{fmep[kPa]})\end{gathered} \tag{21}$$

For values of bmep lower than fmep the fuel mass flow rate was null has given by Eq. (22).

$$\begin{gathered}\text{if bmep[kPa]} < \text{fmep[kPa]} \\ \dot{m}_f[\text{kg/s}] = 0\end{gathered} \tag{22}$$

Finally, before proceeding to the tests and simulation, we will then calculate the evolution of the engine oil temperature during the test. To do so and knowing that the oil temperature is initially at 23 ºC, which is around the room temperature, we will begin by calculating the power of the heat transferred by convection by the engine gearbox and radiator to the ambient air, $\dot{Q}_c$ using Eq. (23) and Eq. (24), where h_e and h_r are respectively the engine convection heat transfer coefficient and the radiator convection heat transfer coefficient, t_{oil} is the engine oil temperature, t_{air} is the ambient air temperature, t_{tot} is the thermostat opening temperature and a_{engine}, a_{gearbox} and $a_{\text{fins radiator}}$ are respectively the surface areas of the engine, the gearbox and the radiator fins.

For $t_{\text{oil}} > t_{\text{tot}}$:

$$\begin{aligned}\dot{Q}_c[\text{W}] = {} & h_e[\text{W/m}^2 \cdot \text{K}] \times a_{\text{engine}}[\text{m}^2] \times (t_{\text{oil}}[°\text{C}] - t_{\text{air}}[°\text{C}]) \\ & + h_e[\text{W/m}^2 \cdot \text{K}] \times a_{\text{gearbox}}\,[\text{m}^2] \times (t_{\text{oil}}[°\text{C}] - t_{\text{air}}[°\text{C}]) \\ & + h_r[\text{W/m}^2 \cdot \text{K}] \times a_{\text{fins radiator}}[\text{m}^2] \times (t_{\text{oil}}[°\text{C}] - t_{\text{air}}\text{C}])\end{aligned} \tag{23}$$

For $t_{\text{oil}} \leq t_{\text{tot}}$:

$$\begin{aligned}\dot{Q}_c[\text{W}] = {} & h_e[\text{W/m}^2 \cdot \text{K}] \times a_{\text{engine}}[\text{m}^2] \times (t_{\text{oil}}[°\text{C}] - t_{\text{air}}[°\text{C}]) \\ & + h_e[\text{W/m}^2 \cdot \text{K}] \times a_{\text{gearbox}}[\text{m}^2] \times (t_{\text{oil}}[°\text{C}] - t_{\text{air}}[°\text{C}])\end{aligned} \tag{24}$$

It is now necessary to calculate the heat power generated by the engine, $\dot{Q}_g$. When the values for bsfc are null, generated power is also null, otherwise it can be calculated using Eq. (25).

$$\begin{aligned}\dot{Q}_g[\text{W}] = {} & \dot{m}_f[\text{kg/s}] \times Q_{\text{HV}}[\text{MJ/kg}] \times 10^6 \\ & \times \left(1 - \left(\frac{3600}{\text{bsfc}[\text{g/kW} \cdot \text{h}] \times Q_{\text{HV}}[\text{MJ/kg}]}\right) - 0.30\right)\end{aligned} \tag{25}$$

Having all the required data to calculate the temperature of the oil of the engine, t_{oil}, we will then determine it using Eq. (26) where $(t_{j+1} - t_j)$ corresponds to the timestep of 0.1 seconds, the mass of the engine, gearbox, engine oil, gearbox oil and engine liquid coolant are respectively m_e, m_g, $m_{e,\text{oil}}$, $m_{g,oil}$ and $m_{e,\text{lc}}$ in kg, the specific heat from the engine, the engine, gearbox, engine oil, gearbox oil and engine liquid coolant are respectively $c_{p,e}$, $c_{p,g}$, $c_{p,e,\text{oil}}$, $c_{p,g,\text{oil}}$ and $c_{p,e,\text{lc}}$ in J/kg·K. In this analysis it is assumed that the engine oil temperature is equal to the temperatures of the engine, gearbox, engine coolant liquid and gearbox oil lubricant.

$$t_{\mathrm{oil}_j}[°\mathrm{C}] = t_{\mathrm{oil}_{j-1}}[°\mathrm{C}] + \frac{(\dot{Q}_g[\mathrm{W}] - \dot{Q}_c[\mathrm{W}]) \times (t_{j+1} - t_j)[\mathrm{s}]}{m_e \times c_{p,e} + m_g \times c_{p,g} + m_{e,\mathrm{oil}} \times c_{p,e,\mathrm{oil}} + m_{g,\mathrm{oil}} \times c_{p,g,\mathrm{oil}} + m_{e,\mathrm{lc}} \times c_{p,e,\mathrm{lc}}} \tag{26}$$

6. FUEL CONSUMPTION AND CO_2 EMISSIONS

Section 6.1 is dedicated to present the calculations of the main objective of this work, finding the fuel consumption and in section 6.2 the CO_2 emissions of the vehicle.

6.1. Fuel Consumption

It is known that there is fuel consumption while the car is moving as well as when it is stopped. Obviously when it is not moving the fuel consumption will be nearly null, nevertheless it will be calculated for a better precision of the results.

The vehicle is considered stopped when the speed is equal or lower than 1 km/h [10]. When the vehicle is stopped the mass of fuel consumed in the time interval $(t_{j+1} - t_j)$ of the cycle, $m_{f_{\text{stopped},j}}$, is calculated using Eq. (27), where ρ_{fuel} represents the density of the fuel, $idle\ consumption$, is the engine fuel consumption when idling and is set to be 0.7 l/h and $(t_{j+1} - t_j)$ corresponds to the timestep of the test which is 0.1 seconds.

$$m_{f_{\text{stopped},j}}[\text{kg}] = \frac{\rho_{\text{fuel}}[\text{kg/m}^3] \times idle\ consumption[\text{l/h}] \times (t_{j+1} - t_j)[\text{s}]}{3600 \times 1000} \tag{27}$$

To calculate the total fuel consumption of the cycle when the vehicle is stopped, $F_{c_{\text{stopped}}}$, it was used Eq. (28), where d_{cycle} stands for the total distance of the cycle in km and the results are in litre per 100 km. The total fuel consumption when the vehicle is stopped can be calculated for each of the phases of the cycle using an equation like Eq. (28) making the summatory of the fuel consumption for the vehicle stopped for each phase and with the distance of each phase.

$$F_{c_{\text{stopped}}}[\text{l/100km}] = \frac{\sum m_{f_{\text{stopped},j}}[\text{kg}] \times 10^5}{\rho_{\text{fuel}}[\text{kg/m}^3] \times d_{\text{cycle}}[\text{km}]} \tag{28}$$

When the vehicle is moving at a speed higher than 1 km/h [10] the engine fuel consumption is calculated using the equation from Figure 9 and Eq. (29) to calculate the slope of the straight line of the mass fuel flow rate consumed by the engine as a function of the engine rotational speed. After this Eq. (30) was used to determine the fuel mass consumed by the engine between instant t_j and t_{j+1}. When bmep < fmep, the fuel mass consumed by the engine between instant t_j and t_{j+1} is equal to 0, according to Eq. (31).

Knowing the fuel mass consumed by the engine for the vehicle in motion in all instants of the cycle it is possible to calculate the mass of fuel consumed in the cycle summing up the mass of fuel consumed in every time step of the cycle when the vehicle is in motion as expressed by Eq. (32).

The total fuel consumption when the vehicle is moving, $F_{c_{\text{moving},j}}$, can be calculated for each of the phases of the cycle using an equation like Eq. **(**33**)** making the summatory of the fuel consumption for the vehicle moving for each phase and with the distance of each phase.

$$m_j[\text{kg}/(\text{s}\cdot\text{kPa})] = a_{m2} \times n_{ij}^2[\text{rpm}] + a_{m1} \times n_{ij}[\text{rpm}] + a_{m0} \tag{29}$$

$$\begin{cases} \text{if bmep} \geq \text{fmep} \rightarrow m_{f_{\text{moving},j}}[\text{kg}] \\ m_{f_{\text{moving},j}}[\text{kg}] = m_j\left[\dfrac{\text{kg}}{\text{s}\cdot\text{kPa}}\right] \times (\text{bmep}[\text{kPa}] - \text{fmep}[\text{kPa}]) \times (t_{j+1} - t_j)[\text{s}] \end{cases} \tag{30}$$

$$\text{if bmep} < \text{fmep} \rightarrow m_{f_{\text{moving},j}}[\text{kg}] = 0 \tag{31}$$

$$m_{f_{\text{moving}}}[\text{kg}] = \sum m_{f_{\text{moving},j}}[\text{kg}] \tag{32}$$

$$F_{c_{\text{moving}}}[\text{l}/100\text{km}] = \frac{\sum m_{f_{\text{moving},j}}[\text{kg}] \times 10^5}{\rho_{\text{fuel}}[\text{kg}/\text{m}^3] \times d_{\text{cycle}}[\text{km}]} \tag{33}$$

The total engine fuel consumption is calculated by Eq. **(**34**)** adding the fuel consumed by the vehicle when it is stopped with the fuel consumed by the vehicle when it is moving. This can be made for the total cycle and for each of the cycle phases.

$$F_c[\mathrm{l/100km}] = F_{c_{\mathrm{stopped}}}[\mathrm{l/100km}] + F_{c_{\mathrm{moving}}}[\mathrm{l/100km}] \tag{34}$$

6.2. CO_2 Emissions

As it was for the fuel consumption, there are CO_2 emissions when the vehicle is stopped and when it is moving.

To calculate CO_2 emissions when the vehicle is stopped, $E_{\mathrm{CO_2,stopped}}$, with the fuel mass already calculated using Eq. (27) it is now possible do determine the emissions, by Eq. **(**35**)**, where M_C is the molar mass of Carbon, M_H is the molar mass of Hydrogen, M_O is the molar mass of Oxygen and the $(\mathrm{H/C})$ is the Hydrogen to Carbon molar ratio of the fuel which is set to 1.876.

$$E_{\mathrm{CO_2,stopped}}\left[\mathrm{g_{CO_2}/km}\right] = \sum m_{f_{\mathrm{stopped},j}}\,[\mathrm{kg}] \times \frac{\left(\dfrac{M_\mathrm{C}[\mathrm{kg/mol}] + 2 \times M_\mathrm{O}[\mathrm{kg/mol}]}{M_\mathrm{C}[\mathrm{kg/mol}] + (\mathrm{H/C}) \times M_\mathrm{H}[\mathrm{kg/mol}]}\right) \times 1000}{d_{\mathrm{cycle}}[\mathrm{km}]} \tag{35}$$

The CO_2 emissions for the moving vehicle, $E_{\mathrm{CO_2,moving}}$, are estimated using Eq. **(**36**)**, using the fuel mass already calculated in Eq. (29).

$$E_{\mathrm{CO_2,moving}}\left[\mathrm{g_{CO_2}/km}\right] = \sum m_{f_{\mathrm{moving},j}}\,[\mathrm{kg}] \times \frac{\left(\dfrac{M_\mathrm{C}[\mathrm{kg/mol}] + 2 \times M_\mathrm{O}[\mathrm{kg/mol}]}{M_\mathrm{C}[\mathrm{kg/mol}] + (\mathrm{H/C}) \times M_\mathrm{H}[\mathrm{kg/mol}]}\right) \times 1000}{d_{\mathrm{cycle}}[\mathrm{km}]} \tag{36}$$

The total CO_2 emissions are calculated by Eq. **(**37**)** adding the CO_2 emissions by the vehicle when it is stopped with the CO_2 emissions by the vehicle when it is moving. This can be made for the total cycle and for each of the cycle phases.

$$E_{\mathrm{CO_2}}[\mathrm{g_{CO_2}/km}] = E_{\mathrm{CO_2,stopped}}[\mathrm{g_{CO_2}/km}] + E_{\mathrm{CO_2,moving}}[\mathrm{g_{CO_2}/km}] \tag{37}$$

7. TESTS AND SIMULATIONS

In this chapter we will proceed to the accomplishment of numerical simulations determining the engine fuel consumption and emissions of CO_2 of the vehicle. A more extensive study will be carried out, varying certain parameters such as the vehicle mass, tire rolling resistance coefficient, drag coefficient and differential gear ratio to evaluate their impact in comparison with the original vehicle configuration results.

7.1. Simulation With the Original Characteristics

The simulation was performed using Microsoft Excel and it is possible to view some of the results below.

In Figure 26 it is presented the engine rotational speed as a function of distance, during the cycle. The engine rotational speed is kept at a moderate level, only reaching higher values in the fourth phase meaning it was the phase where it was being pushed the most, but not being near to its maximum. By analysing the peaks, it is also possible to understand in which phases there was acceleration or deceleration. In the first and second phase it is not so visible because there were short periods between acceleration and braking, showing this way the dynamics of this driving cycle.

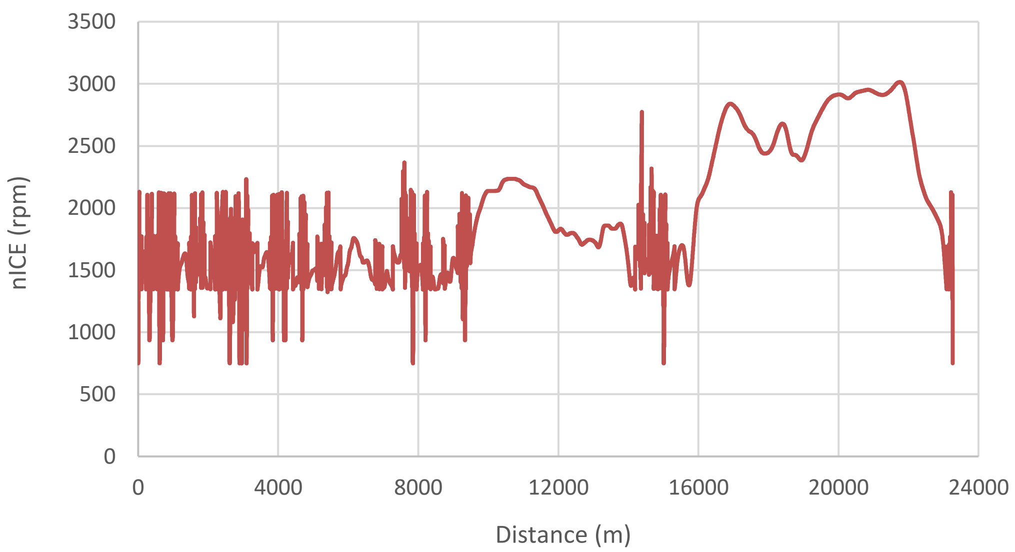

Figure 26 Engine rotational speed as a function of distance during the driving cycle

Figure 27 shows the intensity of the aerodynamic drag force as a function of distance during the cycle. It increases as the vehicle moves from the first phase to the fourth phase because the vehicle speed is increasing, and this force is proportional to the square of the vehicle speed.

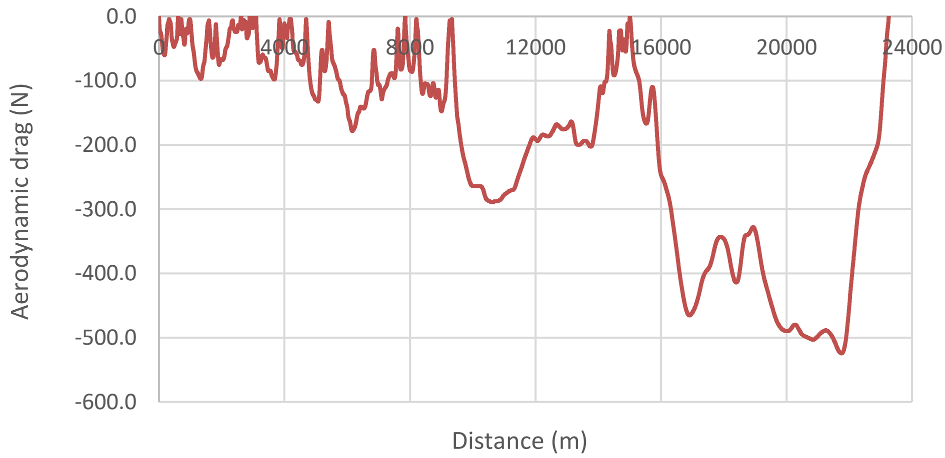

Figure 27 Drag force as a function of distance during the driving cycle

In Figure 28, are presented the engine working points during the cycle defined by pairs of the engine brake torque and engine rotational speed (red line). It is also presented the engine maximum brake torque as a function of the engine rotational speed (blue line). Notice that in any point of the driving cycle the engine reaches the maximum brake torque correspondent to its engine rotational speed and the engine rotational speed normally stays between 750 rpm and 2100 rpm so the fuel consumption would be kept low.

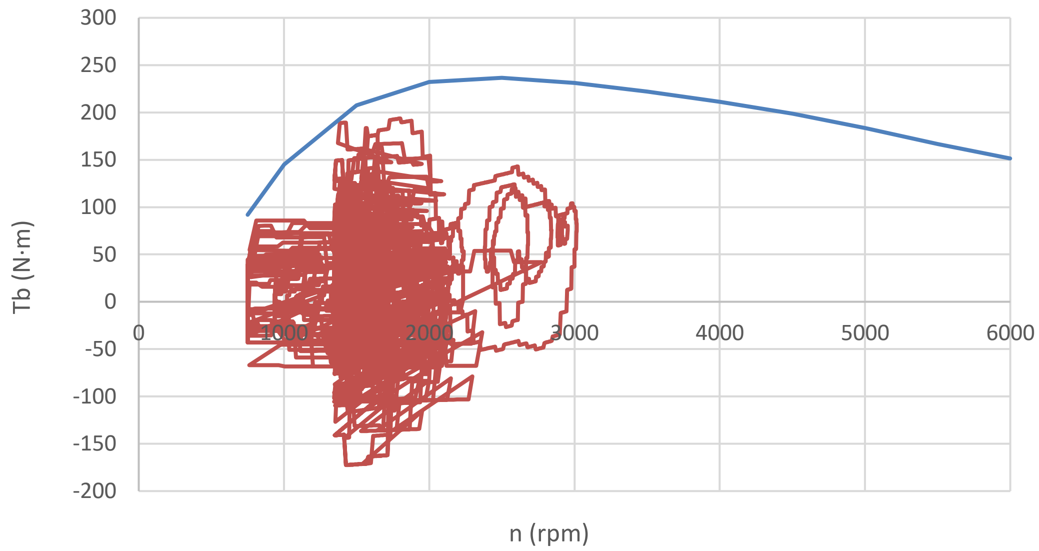

Figure 28 Engine brake torque and rotational speed during the cycle

The evolution of the engine oil temperature with distance is presented in Figure 29. Initially the oil was at room temperature and in the course of time it began to increase until stabilizing around 82 ºC. Due to the low initial temperatures of the oil, the fuel consumption was higher until reaching the final temperature. From the moment it reached its temperature peak, consumption will not be so high. Even though these values do not vary drastically, they allow us to obtain more accurate fuel consumption results.

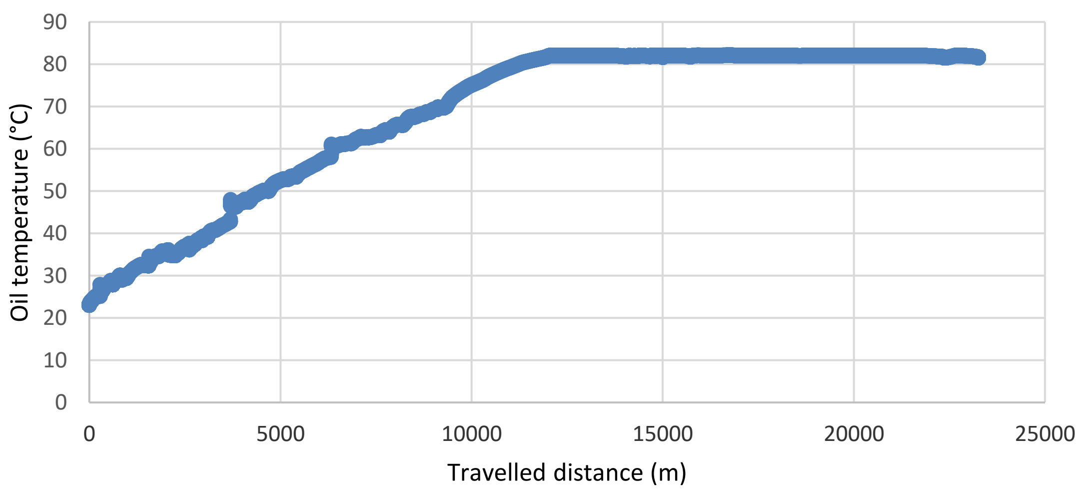

Figure 29 Evolution of engine oil temperature as a function of distance

In Table 9 are presented the results for the simulation and the experimental test for the vehicle with the original configuration. The results of the simulation show a lower fuel consumption and lower CO_2 emissions than the results of the experimental test for the whole cycle. The results of the simulation show a larger fuel consumption than the results of the experimental test for the 1^{st} phase of the cycle. The results of the simulation show a lower fuel consumption than the results of the experimental test for the 2^{nd}, 3^{rd}, and 4^{th} phases of the cycle. One possible reason for the results of the simulation give lower fuel consumptions than experimental tests is the fact that the represented mass fuel flow rate as a function of bmep as a straight line for each engine rotational speed does not give accurate results for values of bmep above 1400 kPa as the real values of fuel consumption are higher than the ones predicted by the straight line evolution.

Table 9 CO_2 emissions and fuel consumption for the original vehicle characteristics

	Simulation		Experimental test		Δ(gCO_2/km)	Δ (l/100 km)
	gCO_2/km	l/100 km	gCO_2/km	l/100 km	[%]	[%]
Total	124.53	5.24	132	5.8	-5.66	-9.66
Stopped vehicle	4.81	0.2				
Moving vehicle	119.72	5.04				
Total 1^{st} phase WLTP (Low)	173.57	7.31		7.1		2.96
Total 2^{nd} phase WLTP (Middle)	117.73	4.96		5.8		-14.48
Total 3^{rd} phase WLTP (High)	104.71	4.41		5.1		-13.53
Total 4^{th} phase WLTP (Extra-High)	127.25	5.36		6.1		-12.13

7.2. Effect of the Vehicle Mass

Since the mass of vehicle is one of the essential parameters in the dynamics of the vehicle, it is one of the variables whose effect on fuel consumption and CO_2 emissions should be studied in the simulations.

The original vehicle mass is 1178 kg and 100 kg are added to this mass as stipulated in the regulations. Of these 100 kg, 75 kg correspond to the driver's mass and the remaining 25 kg to the mass of the luggage loaded in the car [12].

As the car needs at least one person to be driven and has a capacity up to 5 persons, in this simulation we will increase the mass up to two values, m_2 and m_3, while m_1 is considered the minimum mass.

The results obtained are presented in Table 10 and the graphical representation of the results in each of the phases of the WLTP driving cycle are presented in Figure 30 and Figure 31. In Figure 32 and Figure 33 are presented the results for the total of the WLTP cycle.

Table 10 Results for the simulation of the effect of the vehicle mass

	m_1 [kg]	m_2 [kg]	m_3 [kg]
	1278	1478	1678
WLTP 1st phase (l/100 km)	7.60	7.59	7.81
WLTP 2nd phase (l/100 km)	4.96	5.18	5.60
WLTP 3rd phase (l/100 km)	4.35	4.60	4.88
WLTP 4th phase (l/100 km)	5.29	5.51	5.75
WLTP Total (l/100 km)	5.24	5.44	5.72
WLTP 1st phase (gCO_2/km)	180.56	180.22	185.54
WLTP 2nd phase (gCO_2/km)	117.74	123.01	132.91
WLTP 3rd phase (gCO_2/km)	103.33	109.30	115.81
WLTP 4th phase (gCO_2/km)	125.51	130.94	136.47
WLTP Total (gCO_2/km)	124.42	129.22	135.91

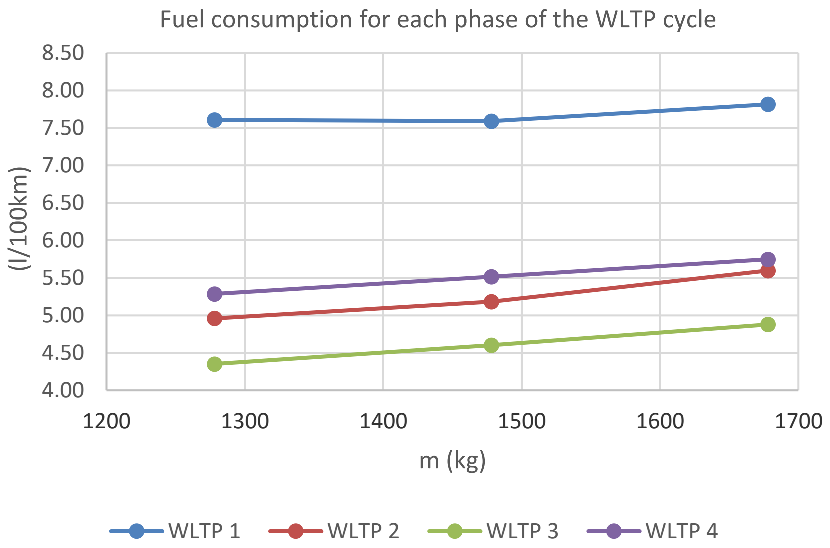

Figure 30 Evolution of the fuel consumption with the vehicle mass in each phase of the WLTP cycle

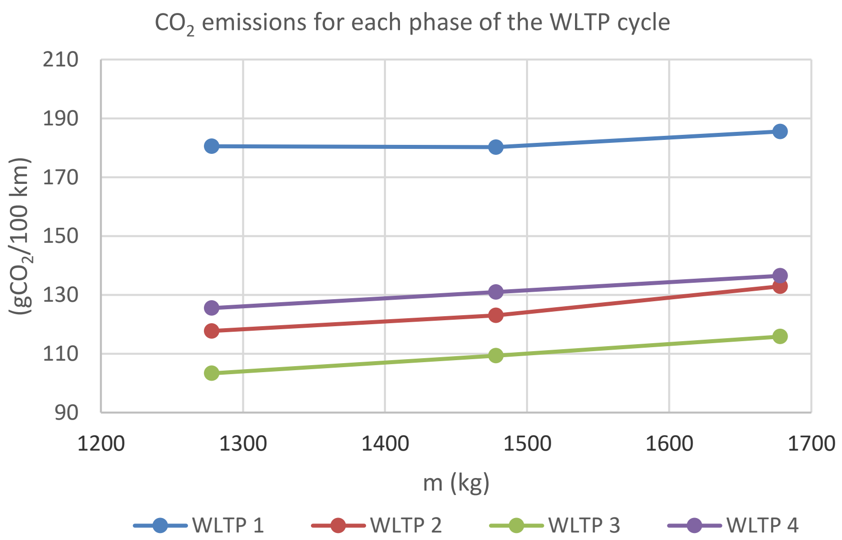

Figure 31 Evolution of the CO_2 emissions with the vehicle mass in each phase of the WLTP cycle

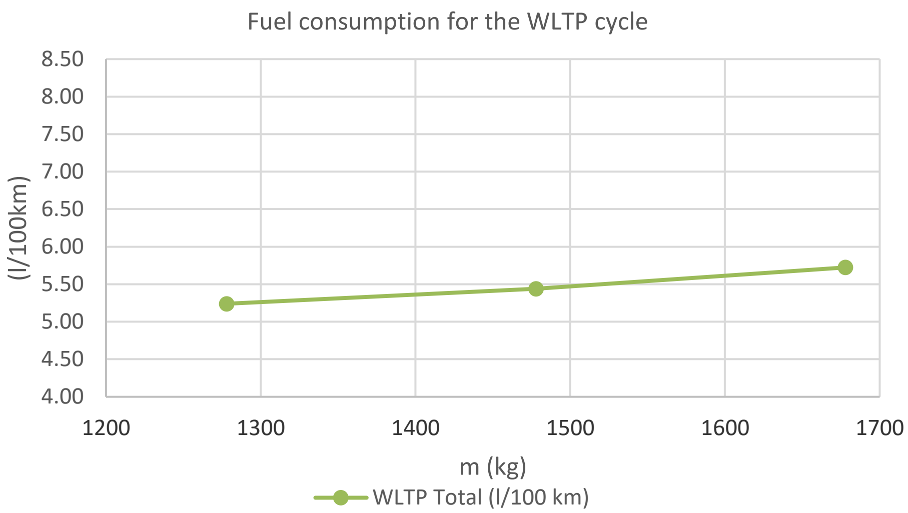

Figure 32 Evolution of the fuel consumption with the vehicle mass in the WLTP cycle

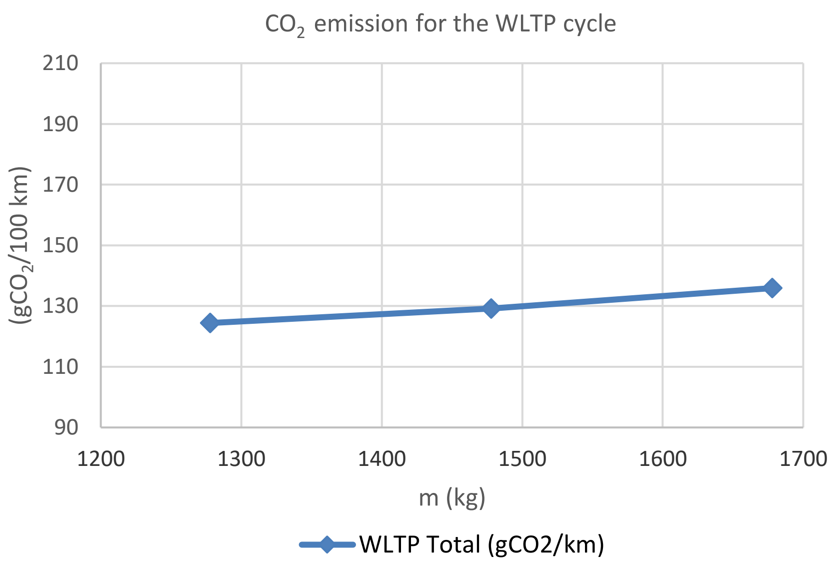

Figure 33 Evolution of the CO_2 emissions with the vehicle mass in the WLTP cycle

As it was expected, by increasing the mass of the vehicle both the fuel consumption and CO_2 emissions will increase, either in each of the cycle phases, as shown respectively in Figure 30 and Figure 31, or the total cycle, as shown respectively in Figure 32 and Figure 33. The only exception is for the 1st phase where the fuel consumption and the CO_2 emissions decrease slightly from the vehicle mass m_1 to m_2 as shown in Figure 30 and Figure 31.

7.3. Effect of the Differential Gear Ratio

In the next simulations we will vary the differential gear ratio to understand its influence. The original differential gear ratio is i_{diff2}. The chosen differential gear ratios to be studied were the one immediately below and the one immediately above, respectively i_{diff1} and i_{diff3}. These values were taken from reference [13].

The results obtained are presented in Table 11 and the graphical representation of the results in each of the phases of the WLTP driving cycle are presented in Figure 34 and Figure 35. In Figure 36 and Figure 37 are presented the results for the total of the WLTP cycle.

Table 11 Results for the simulation of the effect of the differential gear ratio

	i_{diff1}	i_{diff2}	i_{diff3}
	4.176471	4.529412	5.058824
WLTP 1st phase (l/100 km)	7.37	6.69	7.96
WLTP 2nd phase (l/100 km)	5.26	4.72	5.68
WLTP 3rd phase (l/100 km)	4.57	4.23	5.11
WLTP 4th phase (l/100 km)	5.43	5.00	6.18
WLTP Total (l/100 km)	5.39	4.93	5.99
WLTP 1st phase (gCO_2/km)	175.02	158.82	189.08
WLTP 2nd phase (gCO_2/km)	125.00	112.00	134.97
WLTP 3rd phase (gCO_2/km)	108.44	100.53	121.27
WLTP 4th phase (gCO_2/km)	128.94	118.78	146.72
WLTP Total (gCO_2/km)	127.96	117.11	142.12

These differential gear ratios did not present better results than the original differential gear ratio neither in fuel consumption nor in CO_2 emissions as shown respectively in Figure 34 and Figure 35, for the different phases of the driving cycle and as shown respectively in Figure 36 and Figure 37 for the total driving cycle. This means that the original differential gear ratio chosen for this study was in fact the best choice.

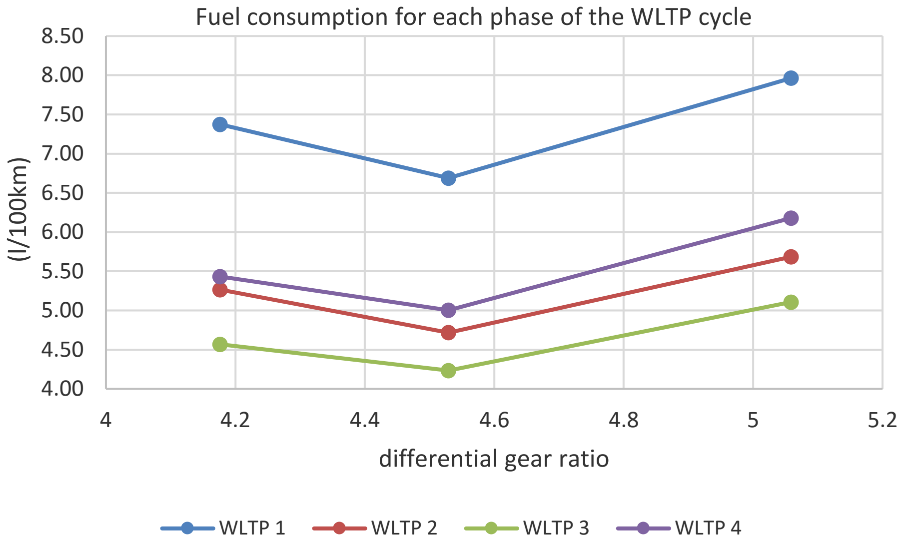

Figure 34 Evolution of the fuel consumption with the differential gear ratio in each phase of the WLTP cycle

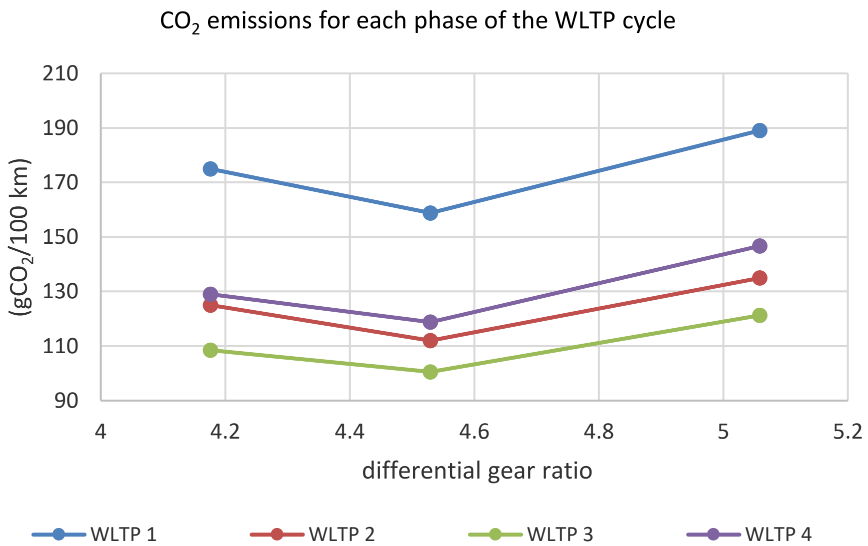

Figure 35 Evolution of the CO_2 emissions with the differential gear ratio in each phase of the WLTP cycle

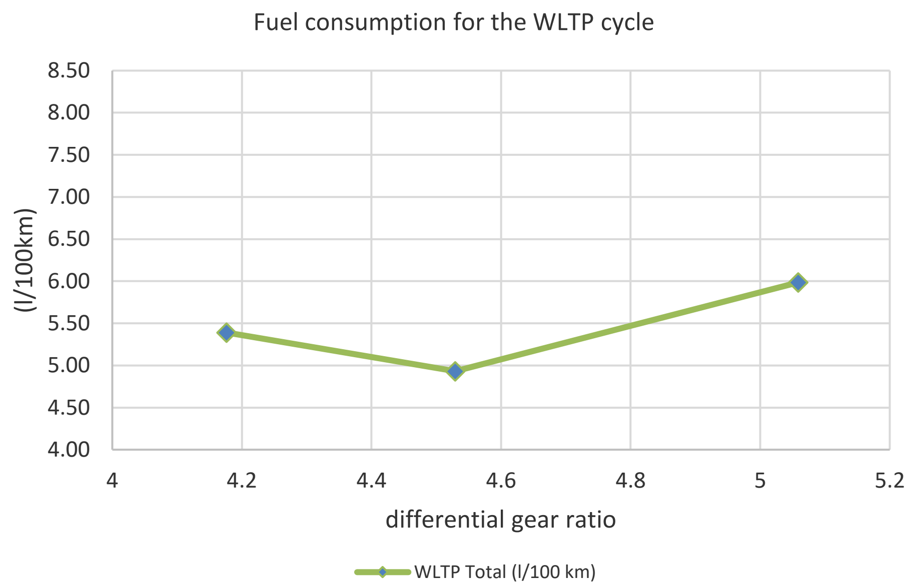

Figure 36 Evolution of the fuel consumption with the differential gear ratio in the WLTP cycle

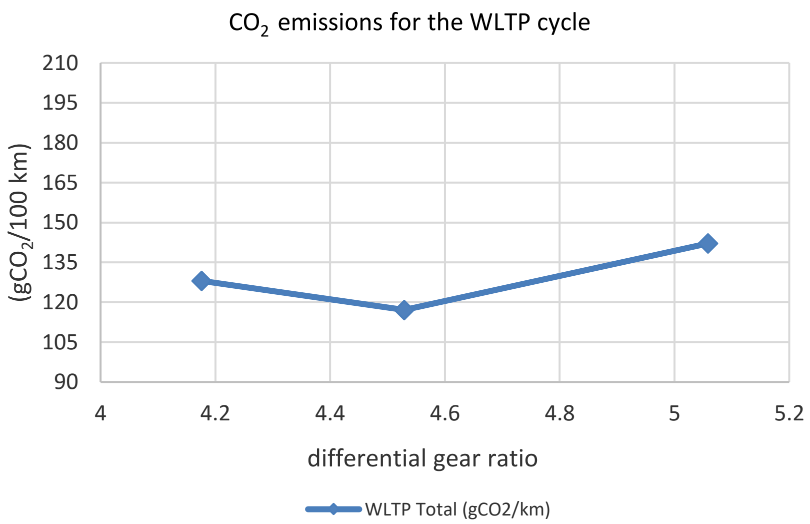

Figure 37 Evolution of the CO_2 emissions with the differential gear ratio in the WLTP cycle

7.4. Effect of the Tire Rolling Resistance Coefficient

Tire rolling resistance is the force needed to maintain the tire movement at a constant speed over a horizontal surface [14]. The tire rolling resistance is proportional to the tire rolling resistance coefficient. The lower this coefficient, the more fuel efficient is the driving.

To study its effect on the car fuel consumption and CO_2 emissions we will vary its value for a value above the original and for a value below the original.

The results obtained are presented in Table 12 and the graphical representation of the results in each of the phases of the WLTP driving cycle are presented in Figure 38 and Figure 39. In Figure 40 and Figure 41 are presented the results for the total of the WLTP cycle.

In all cycle phases the increase of the tire rolling resistance coefficient increases the fuel consumption and the CO_2 emissions as shown respectively in Figure 38 and Figure 39. The effect of the increase of the tire rolling resistance coefficient is higher in the 1st phase and it decreases from the 1st phase to the 4th phase. The explanation for this is that the importance of the tire rolling resistance compared to the vehicle drag is larger in the 1st phase and it decreases as the vehicle goes from the 1st phase to the 4th phase. The ratio of the energy expended to overcome the tire rolling resistance to the total energy expended to drive the vehicle decreases when the vehicle goes from the 1st phase to the 4th phase.

In the total cycle the increase of the tire rolling resistance coefficient increases the fuel consumption and the CO_2 emissions as shown respectively in Figure 40 and Figure 41.

Table 12 Results for the simulation of the effect of tire rolling resistance coefficient

	Crr_1	Crr_2	Crr_3
	0.0065	0.0072	0.0084
WLTP 1st phase (l/100 km)	7.04	7.60	8.01
WLTP 2nd phase (l/100 km)	4.77	4.96	5.33
WLTP 3rd phase (l/100 km)	4.26	4.35	4.51
WLTP 4th phase (l/100 km)	5.22	5.29	5.40
WLTP Total (l/100 km)	5.08	5.24	5.46
WLTP 1st phase (gCO_2/km)	167.25	180.56	190.09
WLTP 2nd phase (gCO_2/km)	113.24	117.74	126.56
WLTP 3rd phase (gCO_2/km)	101.12	103.33	107.20
WLTP 4th phase (gCO_2/km)	123.99	125.51	128.14
WLTP Total (gCO_2/km)	120.51	124.42	129.62

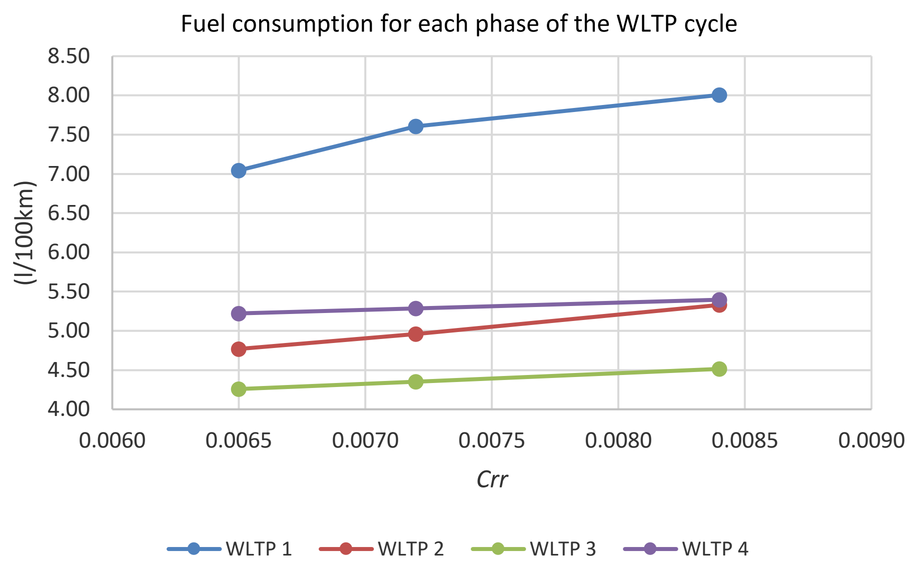

Figure 38 Evolution of the fuel consumption with the tire rolling resistance coefficient in each phase of the WLTP cycle

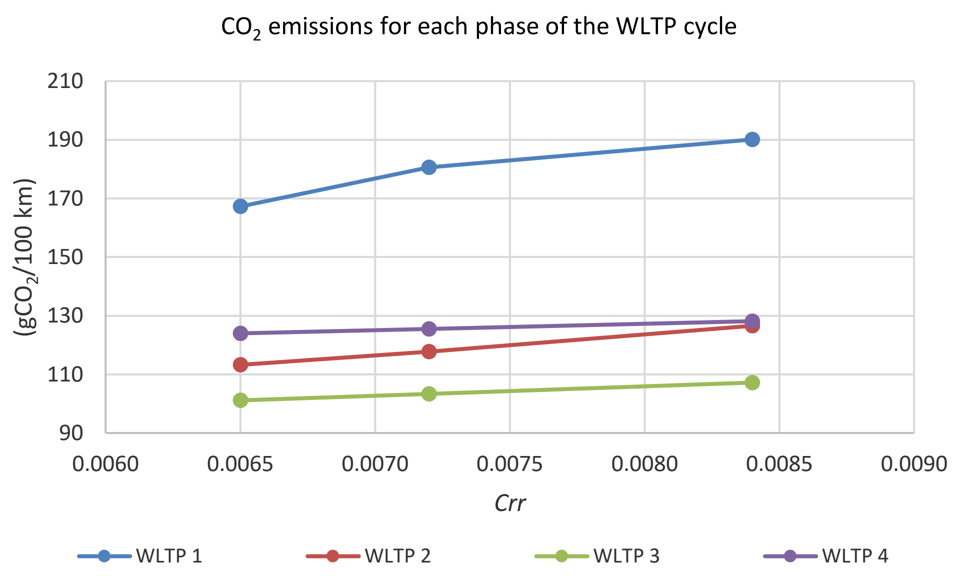

Figure 39 Evolution of CO_2 emissions with the tire rolling resistance coefficient in each phase of the WLTP cycle

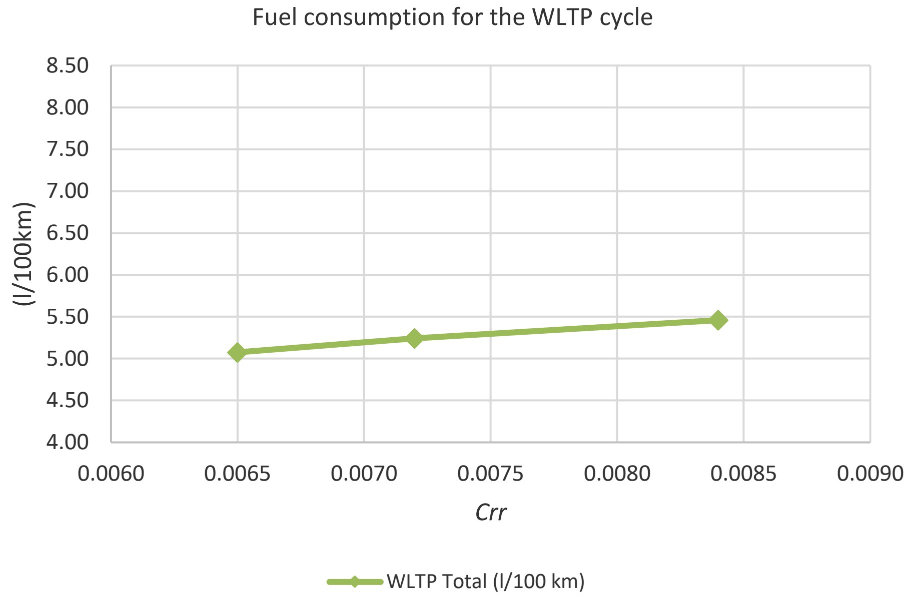

Figure 40 Evolution of the fuel consumption with the tire rolling resistance coefficient in the WLTP cycle

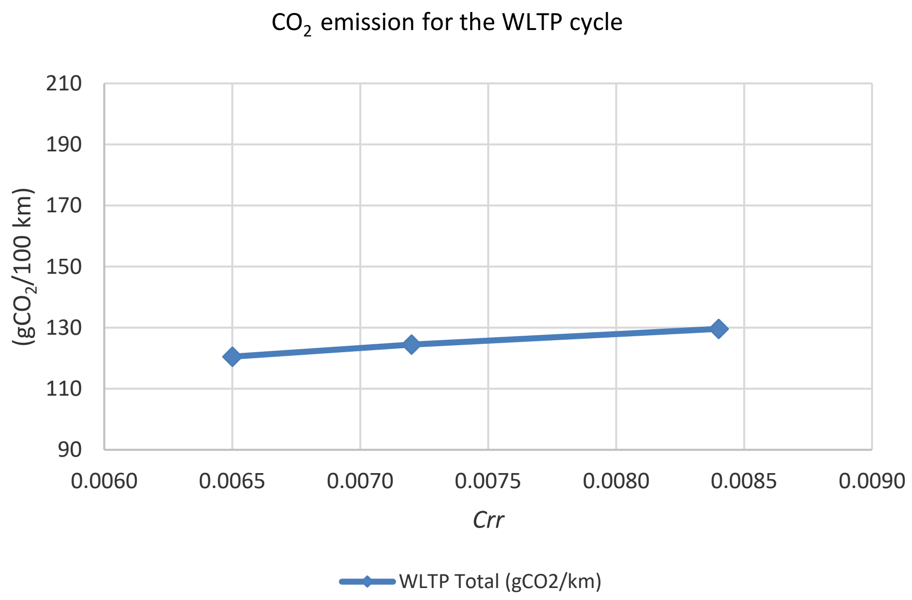

Figure 41 Evolution of CO_2 emissions with the tire rolling resistance coefficient in the WLTP cycle

7.5. Effect of the Aerodynamic Drag Coefficient

The aerodynamic drag coefficient is a measure of the effectiveness of a body shape in reducing the air resistance to the forward motion of the body [15]. For various types of automotive vehicles there is a range of drag coefficients, as presented in Figure 42.

Vehicle type drag coefficient C_D	
Saloon car	0.22–0.4
Sports car	0.28–0.4
Light van	0.35–0.5
Buses and coaches	0.4–0.8
Articulated trucks	0.55–0.8
Ridged truck and draw bar trailer	0.7–0.9

Figure 42 Typical drag coefficients for various classes of vehicles [15]

As we did for other simulations before, we will make it vary from a lower than the reference value to an upper than the reference value. The results of fuel consumption and CO_2 emissions in each of the phases of the WLTP driving cycle are presented in Table 13 as well as the visualization for these results, respectively in Figure 43 and Figure 44. The results of fuel consumption and CO_2 emissions for the total of the WLTP cycle are presented in Table 13 as well as the visualization for these results, respectively in Figure 45 and Figure 46.

Table 13 Results for the simulation for the effect of the drag coefficient

	Cx_1	Cx_2	Cx_3
	0.26	0.29	0.32
WLTP 1st phase (l/100 km)	6.84	7.59	7.70
WLTP 2nd phase (l/100 km)	4.68	5.18	5.55
WLTP 3rd phase (l/100 km)	4.17	4.60	5.01
WLTP 4th phase (l/100 km)	5.00	5.51	6.03
WLTP Total (l/100 km)	4.93	5.44	5.84
WLTP 1st phase (gCO_2/km)	162.38	180.22	182.87
WLTP 2nd phase (gCO_2/km)	111.20	123.01	131.81
WLTP 3rd phase (gCO_2/km)	98.93	109.30	118.94
WLTP 4th phase (gCO_2/km)	118.82	130.94	143.07
WLTP Total (gCO_2/km)	116.94	129.22	138.64

The fuel consumption and the CO_2 emissions increase with the drag coefficient for all the cycle phases and for the total cycle as shown in Table 13, Figure 43, Figure 44, Figure 45 and Figure 46. As shown in Figure 43 and Figure 44 there is a higher impact on the 4th phase since it is the phase with the higher speeds so it will cause a larger drag force on the vehicle, increasing in this way the fuel consumption and the CO_2 emissions. The results for the original value of the drag coefficient, C_{x2}, compared to the value of C_{x3} do not present a significant difference in fuel consumption and CO_2 emissions for the 1st phase of the cycle.

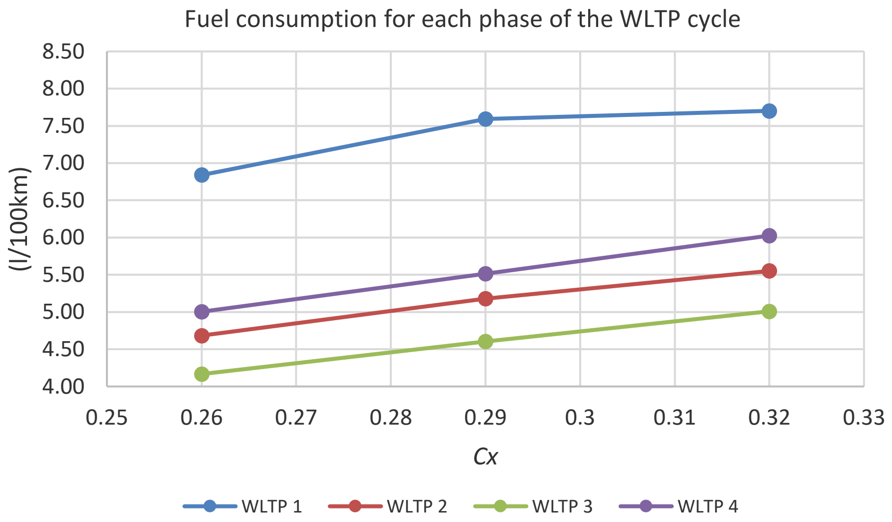

Figure 43 Evolution of the fuel consumption with the drag coefficient in each phase of the WLTP cycle

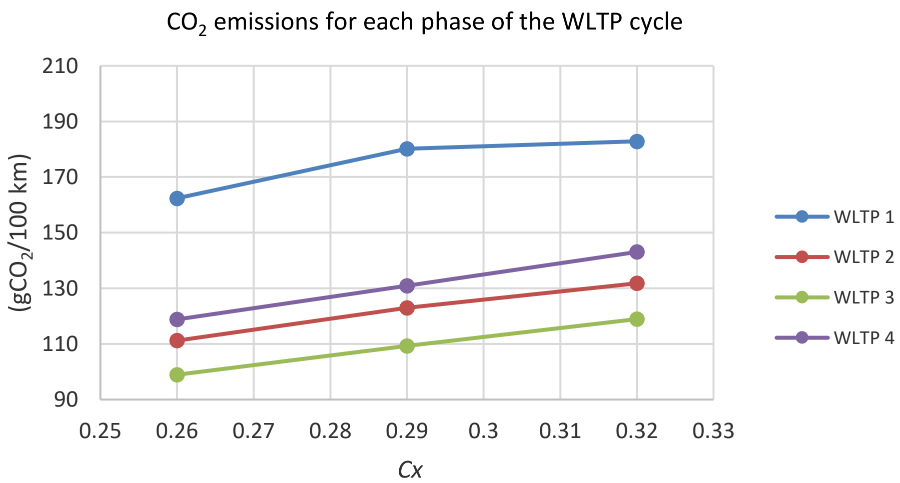

Figure 44 Evolution of the CO_2 emissions with the drag coefficient in each phase of the WLTP cycle

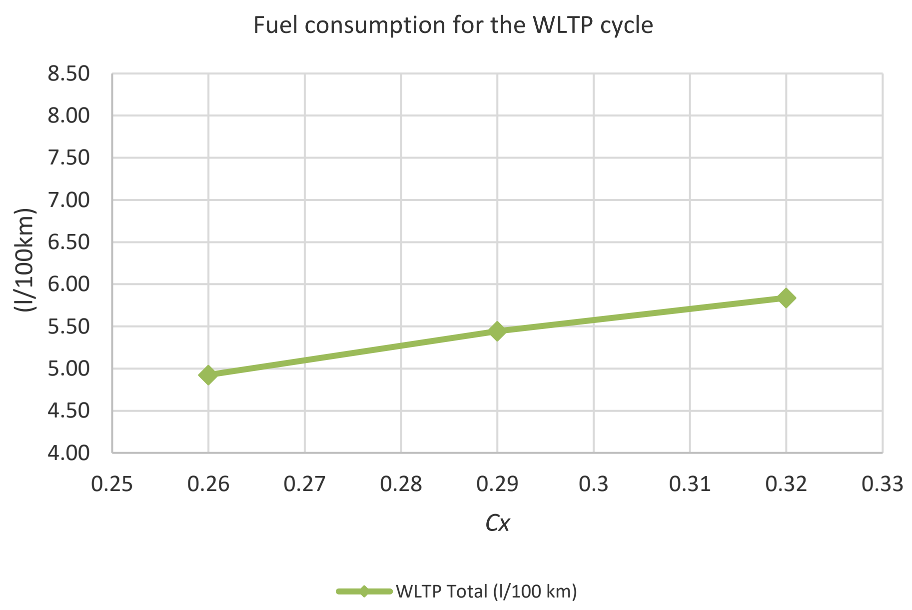

Figure 45 Evolution of the fuel consumption with the drag coefficient in the WLTP cycle

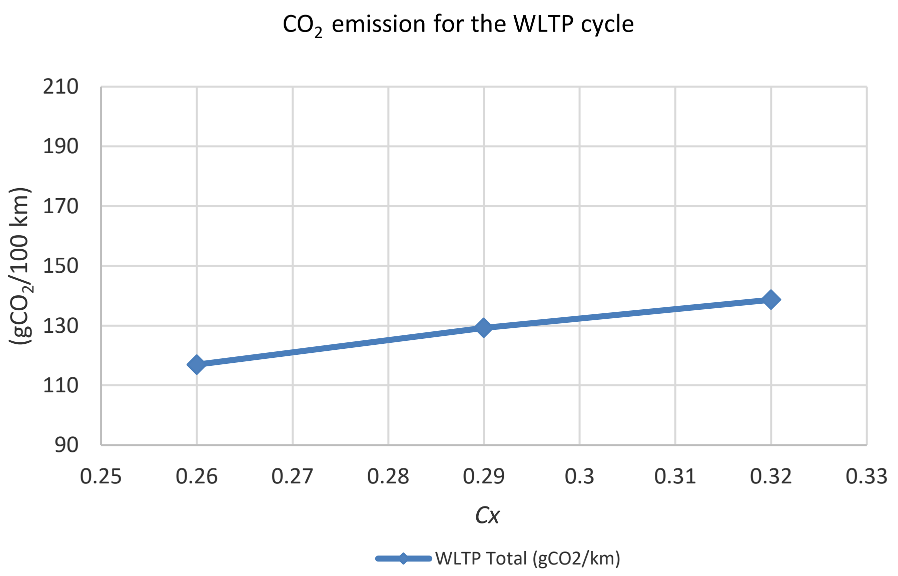

Figure 46 Evolution of the CO_2 emissions with the drag coefficient in the WLTP cycle

8. CONCLUSIONS

From this work and analysing the results for the vehicle with the original configuration it is possible to conclude that in the whole cycle and in general in all the cycle phases the results of the simulations for the fuel consumption and CO_2 emissions are lower than the results of the experimental tests.

The results of the simulations showed that the increase of the vehicle mass increases the fuel consumption as well as the CO_2 emissions.

The differential gear ratio from the original vehicle's configuration presented better results than the simulations with other differential gear ratios. For the differential gear ratios chosen for the simulations there was no difference between the gears selected during the cycle in comparison with the gear selected during the cycle with the original differential gear ratio, however this conclusion for the effect of the differential gear ratio could have been different if the differential gear ratios chosen for the simulations were more distant from the value of the differential gear ratio of the vehicle in the original configuration.

The tire rolling resistance proved to have a higher impact on the fuel consumption and CO_2 emissions in the 1st phase of the cycle and decreasing impact as the vehicle moves to the 4th phase since in the lower phases of the cycle a larger energy fraction is needed to overcome the rolling resistance when compared to the energy fraction expended to overcome the aerodynamic drag.

As it was expected the higher the value of the aerodynamic drag coefficient the higher is the fuel consumption and CO_2 emissions since the drag force increases proportionally with the drag coefficient. This impact was higher in the phases where the vehicle achieved the highest average speeds since the drag force increases proportionally with the square of the vehicle speed.

There were some approximations made during this work since not all the information required was available, for example the fmep values of the PSA EB2DT engine. The Volkswagen CBDB engine does not have the same friction as the PSA EB2DT engine and so that causes the results to lose some accuracy.

The WLTP cycle could be even more realistic, as it does not include some aggressive driving conditions that exist in real life, such as climbing hills and curving which are situations that to be performed require more power from the vehicle which results in higher fuel consumption and CO_2 emissions.

8.1. Future Research

There were some simplifications made in this work since some required information was not easy to find, like the engine friction and some dimensions related to the engine, gearbox, and radiator. This model can be improved in those aspects to increase the results accuracy.

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ANNEX 1

Annex 1

4. Additional requirements for corrections and/or modifications of gear use

The initial gear selection shall be checked and modified in order to avoid too frequent gearshifts and to ensure driveability and practicality.

Corrections and/or modifications shall be made according to the following requirements:

(a) First gear shall be selected one second before beginning an acceleration phase from standstill with the clutch disengaged. Vehicle speeds below 1 km/h imply that the vehicle is standing still;

(b) Gears shall not be skipped during acceleration phases. Gears used during accelerations and decelerations must be used for a period of at least three seconds (e.g. a gear sequence 1, 1, 2, 2, 3, 3, 3, 3, 3 shall be replaced by 1, 1, 1, 2, 2, 2, 3, 3, 3);

(c) Gears may be skipped during deceleration phases. For the last phase of a deceleration to a stop, the clutch may be either disengaged or the gear lever placed in neutral and the clutch left engaged;

(d) There shall be no gearshift during transition from an acceleration phase to a deceleration phase. E.g., if $v_j < v_{j+1} > v_{j+2}$ and the gear for the time sequence j and $j+1$ is i, gear i is also kept for the time $j+2$, even if the initial gear for $j+2$ would be $i+1$;

(e) If a gear i is used for a time sequence of 1 to 5 s and the gear before this sequence is the same as the gear after this sequence, e.g. $i-1$, the gear use for this sequence shall be corrected to $i-1$.

Example:

(i) a gear sequence $i-1, i, i-1$ is replaced by $i-1, i-1, i-1$;

(ii) a gear sequence $i-1, i, i, i-1$ is replaced by $i-1, i-1, i-1, i-1$;

(iii) a gear sequence $i-1, i, i, i, i-1$ is replaced by $i-1, i-1, i-1, i-1, i-1$;

(iv) a gear sequence $i-1, i, i, i, i, i-1$ is replaced by $i-1, i-1, i-1, i-1, i-1, i-1$;

(v) a gear sequence $i-1, i, i, i, i, i, i-1$ is replaced by $i-1, i-1, i-1, i-1, i-1, i-1, i-1$.

For all cases (i) to (v), $g_{min} \leq i$ must be fulfilled;

(f) A gear sequence $i, i-1, i$, shall be replaced by i, i, i, if the following conditions are fulfilled:

(i) Engine speed does not drop below n_{min}; and

(ii) The sequence does not occur more often than four times each for the low, medium and high speed cycle phases and not more than three times for the extra high speed phase.

Requirement (ii) is necessary as the available power will drop below the required power when the gear $i-1$, is replaced by i;

(g) If, during an acceleration phase, a lower gear is required at a higher vehicle speed for at least 2 seconds, the higher gears before shall be corrected to the lower gear.

Example: $v_j < v_{j+1} < v_{j+2} < v_{j+3} < v_{j+4} < v_{j+5} < v_{j+6}$. The originally calculated gear use is 2, 3, 3, 3, 2, 2, 3. In this case the gear use will be corrected to 2, 2, 2, 2, 2, 2, 3.

Since the above modifications may create new gear use sequences which are in conflict with these requirements, the gear sequences shall be checked twice.

APPENDIX 1

Appendix 1

n (rpm)	T_b (N.m)	$n_{f,b}$	Q_{HV} (MJ/kg)	bsfc /(g/kW.h)	T_{bmax}/T_b	bsfc/bsfc(T_{bmax})	bmep [kPa]	m_f [kg/s]
750	10.00	0.1	42.9	839.16	9.650	2.500	104.89	0.0001831
750	18.21	0.15	42.9	559.44	5.299	1.667	191.01	0.0002223
750	36.07	0.2	42.9	419.58	2.675	1.250	378.35	0.0003302
750	51.07	0.22	42.9	381.44	1.890	1.136	535.71	0.0004250
750	96.50	0.25	42.9	335.66	1.000	1.000	1012.23	0.0007067
1000	6.65	0.1	42.9	839.16	20.388	3.350	69.70	0.0001622
1000	11.61	0.15	42.9	559.44	11.667	2.233	121.81	0.0001890
1000	19.03	0.22	42.9	381.44	7.119	1.523	199.64	0.0002112
1000	24.52	0.25	42.9	335.66	5.526	1.340	257.16	0.0002394
1000	33.55	0.27	42.9	310.80	4.038	1.241	351.90	0.0003033
1000	60.65	0.3	42.9	279.72	2.234	1.117	636.13	0.0004935
1000	87.74	0.32	42.9	262.24	1.544	1.047	920.37	0.0006693
1000	122.58	0.33	42.9	254.29	1.105	1.015	1285.80	0.0009067
1000	135.48	0.335	42.9	250.50	1.000	1.000	1421.15	0.0009872
1500	6.45	0.1	42.9	839.16	33.105	3.250	67.67	0.0002362
1500	10.00	0.15	42.9	559.44	21.358	2.167	104.89	0.0002441
1500	19.68	0.22	42.9	381.44	10.854	1.477	206.41	0.0003275
1500	26.50	0.25	42.9	335.66	8.060	1.300	277.97	0.0003881
1500	32.04	0.27	42.9	310.80	6.666	1.204	336.08	0.0004345
1500	44.30	0.3	42.9	279.72	4.821	1.083	464.68	0.0005407
1500	56.20	0.32	42.9	262.24	3.800	1.016	589.51	0.0006431
1500	81.90	0.35	42.9	239.76	2.608	0.929	859.09	0.0008568
1500	96.10	0.36	42.9	233.10	2.222	0.903	1008.04	0.0009774
1500	114.70	0.37	42.9	226.80	1.862	0.878	1203.14	0.0011351
1500	142.80	0.37	42.9	226.80	1.496	0.878	1497.89	0.0014132
1500	159.40	0.36	42.9	233.10	1.340	0.903	1672.02	0.0016212
1500	172.50	0.35	42.9	239.76	1.238	0.929	1809.43	0.0018046
1500	213.58	0.325	42.9	258.20	1.000	1.000	2240.32	0.0024062
2000	6.30	0.1	42.9	839.16	36.508	3.200	66.08	0.0003076
2000	10.24	0.15	42.9	559.44	22.461	2.133	107.41	0.0003333
2000	19.69	0.22	42.9	381.44	11.681	1.455	206.54	0.0004369
2000	26.39	0.25	42.9	335.66	8.715	1.280	276.82	0.0005153

2000	32.30	0.27	42.9	310.80	7.121	1.185	338.81	0.0005840
2000	44.90	0.3	42.9	279.72	5.122	1.067	470.98	0.0007307
2000	57.50	0.32	42.9	262.24	4.000	1.000	603.14	0.0008772
2000	89.80	0.35	42.9	239.76	2.561	0.914	941.95	0.0012526
2000	108.31	0.36	42.9	233.10	2.124	0.889	1136.11	0.0014688
2000	164.24	0.36	42.9	233.10	1.400	0.889	1722.79	0.0022273
2000	180.00	0.35	42.9	239.76	1.278	0.914	1888.10	0.0025108
2000	230.00	0.32	42.9	262.24	1.000	1.000	2412.58	0.0035090
2500	6.30	0.1	42.9	839.16	36.508	3.300	66.08	0.0003845
2500	10.24	0.15	42.9	559.44	22.461	2.200	107.41	0.0004166
2500	19.69	0.22	42.9	381.44	11.681	1.500	206.54	0.0005462
2500	26.39	0.25	42.9	335.66	8.715	1.320	276.82	0.0006442
2500	31.50	0.27	42.9	310.80	7.302	1.222	330.42	0.0007120
2500	43.33	0.3	42.9	279.72	5.308	1.100	454.51	0.0008814
2500	55.54	0.32	42.9	262.24	4.141	1.031	582.58	0.0010592
2500	85.86	0.35	42.9	239.76	2.679	0.943	900.62	0.0014970
2500	102.41	0.36	42.9	233.10	2.246	0.917	1074.23	0.0017360
2500	124.86	0.37	42.9	226.80	1.842	0.892	1309.71	0.0020594
2500	173.70	0.37	42.9	226.80	1.324	0.892	1822.02	0.0028649
2500	188.67	0.36	42.9	233.10	1.219	0.917	1979.05	0.0031982
2500	230.00	0.33	42.9	254.29	1.000	1.000	2412.58	0.0042533
3000	6.43	0.1	42.9	839.16	35.770	3.200	67.45	0.0004709
3000	10.71	0.15	42.9	559.44	21.475	2.133	112.34	0.0005229
3000	20.00	0.22	42.9	381.44	11.500	1.455	209.79	0.0006657
3000	26.79	0.25	42.9	335.66	8.585	1.280	281.01	0.0007847
3000	32.86	0.27	42.9	310.80	6.999	1.185	344.68	0.0008912
3000	46.07	0.3	42.9	279.72	4.992	1.067	483.25	0.0011246
3000	59.29	0.32	42.9	262.24	3.879	1.000	621.92	0.0013568
3000	94.64	0.35	42.9	239.76	2.430	0.914	992.72	0.0019802
3000	115.00	0.36	42.9	233.10	2.000	0.889	1206.29	0.0023393
3000	153.93	0.36	42.9	233.10	1.494	0.889	1614.64	0.0031312
3000	176.79	0.35	42.9	239.76	1.301	0.914	1854.43	0.0036990
3000	230.00	0.32	42.9	262.24	1.000	1.000	2412.58	0.0052634
3500	7.50	0.1	42.9	839.16	30.267	3.000	78.67	0.0006408
3500	11.79	0.15	42.9	559.44	19.254	2.000	123.67	0.0006715
3500	22.14	0.22	42.9	381.44	10.253	1.364	232.24	0.0008598
3500	30.00	0.25	42.9	335.66	7.567	1.200	314.68	0.0010252
3500	37.50	0.27	42.9	310.80	6.053	1.111	393.35	0.0011866
3500	52.50	0.3	42.9	279.72	4.324	1.000	550.70	0.0014951
3500	67.50	0.32	42.9	262.24	3.363	0.938	708.04	0.0018022

3500	107.14	0.35	42.9	239.76	2.119	0.857	1123.84	0.0026153
3500	162.86	0.35	42.9	239.76	1.394	0.857	1708.31	0.0039754
3500	227.00	0.3	42.9	279.72	1.000	1.000	2381.11	0.0064646
4000	7.50	0.1	42.9	839.16	28.333	3.000	78.67	0.0007323
4000	12.14	0.15	42.9	559.44	17.504	2.000	127.34	0.0007902
4000	25.00	0.22	42.9	381.44	8.500	1.364	262.24	0.0011096
4000	33.21	0.25	42.9	335.66	6.399	1.200	348.35	0.0012971
4000	40.71	0.27	42.9	310.80	5.220	1.111	427.03	0.0014722
4000	58.21	0.3	42.9	279.72	3.651	1.000	610.59	0.0018946
4000	75.71	0.32	42.9	262.24	2.807	0.938	794.16	0.0023101
4000	127.50	0.35	42.9	239.76	1.667	0.857	1337.41	0.0035569
4000	177.86	0.35	42.9	239.76	1.195	0.857	1865.65	0.0049618
4000	212.50	0.3	42.9	279.72	1.000	1.000	2229.01	0.0069162
4500	7.86	0.1	42.9	839.16	25.064	2.980	82.45	0.0008634
4500	13.21	0.15	42.9	559.44	14.913	1.987	138.57	0.0009674
4500	27.14	0.22	42.9	381.44	7.259	1.355	284.68	0.0013551
4500	36.07	0.25	42.9	335.66	5.462	1.192	378.35	0.0015849
4500	45.00	0.27	42.9	310.80	4.378	1.104	472.03	0.0018308
4500	63.21	0.3	42.9	279.72	3.117	0.993	663.04	0.0023145
4500	82.50	0.32	42.9	262.24	2.388	0.931	865.38	0.0028320
4500	135.36	0.35	42.9	239.76	1.455	0.851	1419.85	0.0042482
4500	156.79	0.35	42.9	239.76	1.256	0.851	1644.64	0.0049208
4500	180.71	0.32	42.9	262.24	1.090	0.931	1895.55	0.0062032
4500	197.00	0.298	42.9	281.60	1.000	1.000	2066.42	0.0072616
5000	8.57	0.1	42.9	839.16	21.237	2.940	89.89	0.0010460
5000	13.93	0.15	42.9	559.44	13.065	1.960	146.12	0.0011334
5000	27.86	0.22	42.9	381.44	6.533	1.336	292.24	0.0015456
5000	37.14	0.25	42.9	335.66	4.900	1.176	389.58	0.0018132
5000	46.07	0.27	42.9	310.80	3.951	1.089	483.25	0.0020826
5000	64.64	0.3	42.9	279.72	2.816	0.980	678.04	0.0026298
5000	83.57	0.32	42.9	262.24	2.178	0.919	876.60	0.0031874
5000	134.64	0.35	42.9	239.76	1.352	0.840	1412.30	0.0046951
5000	155.00	0.35	42.9	239.76	1.174	0.840	1625.87	0.0054051
5000	171.43	0.32	42.9	262.24	1.062	0.919	1798.21	0.0065385
5000	179.64	0.3	42.9	279.72	1.013	0.980	1884.33	0.0073084
5000	182.00	0.294	42.9	285.43	1.000	1.000	1909.08	0.0075555
5500	8.93	0.1	42.9	839.16	18.701	2.900	93.67	0.0011989
5500	15.36	0.15	42.9	559.44	10.872	1.933	161.12	0.0013748
5500	30.00	0.22	42.9	381.44	5.567	1.318	314.68	0.0018308
5500	40.36	0.25	42.9	335.66	4.138	1.160	423.35	0.0021674
5500	49.29	0.27	42.9	310.80	3.388	1.074	517.03	0.0024509
5500	68.57	0.3	42.9	279.72	2.435	0.967	719.26	0.0030686

5500	87.86	0.32	42.9	262.24	1.901	0.906	921.60	0.0036862
5500	152.86	0.32	42.9	262.24	1.093	0.906	1603.42	0.0064132
5500	159.64	0.3	42.9	279.72	1.046	0.967	1674.54	0.0071442
5500	167.00	0.29	42.9	289.37	1.000	1.000	1751.74	0.0077313