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**APPROPRIATION OF GEAR RATIOS THROUGH
SIMULATION OF HOMOLOGATION TEST
PROCEDURES TO REDUCE FUEL
CONSUMPTION & CO₂ EMISSIONS.**

University of Wales Trinity Saint David

School of Engineering

Bachelor of Engineering - Motorsport Engineering

Major Project - Dissertation



Go Further

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Declaration of authenticity.

I, Danny J. Richardson, declare this dissertation is the result of my original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.

Signed: Danny John Richardson

Date: 30/04/2020

Acknowledgments.

I would like to say thank you to both my mother and father, Tina, and John Richardson, for all the support they have given me throughout my time at university.

Along with an ex-lecturer of mine, Tim Richmond for being the reason I enrolled in the first place, to which I am very grateful for the support.

Finally, I would like to say thank you to both my project supervisor Malcolm McDonald and fellow student Michael Anderson, for aiding me with academic support and helping to obtain key data that was used throughout this project.

Abstract.

This paper has been written to demonstrate the effectiveness of varying gear ratios across two of the homologating test procedures with the sole objective of reducing fuel consumption to in turn reduce the carbon dioxide emissions produced. It includes an overview of vehicle dynamics, driveline and powertrain simulation in order to equate to a torque demand, that can be manipulated alongside fuel flow data to accurately predict the total usage. Three individual vehicles were used for the investigation, each with notably different powertrain and driveline characteristics, but demonstrate many similarities within the external features, to ensure that the investigation was directed more towards the focus of the study, the effect changing gear ratios have on performance. The analysis shows that with the six-speed manual transmissions used, there is a minimum of 4.94% fuel saving and a minimum of 9.7% decrease in CO₂ production and an average decrease in BSFC of 4.46% across the NEDC. Due to the nature of the much more transient WLTP it is shown that the gear shift pattern is the most important area of study in order to potentially reduce the savings further. The results of this study have highlighted that gear ratio appropriation is an effective way to reduce the fuel consumption and emissions over the test cycles, at relatively low cost.

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1. Introduction.

Within both the passenger vehicles & transport vehicles of today, the issues surrounding combustion emissions is certainly no longer a secret. With hundreds of studies conducted from various independent bodies showing just how poor some vehicles are performing and their clear inability to control the harmful gaseous products emitted from the exhaust. Because of this unfortunate issue with internal combustion engines, there has been a magnitude of more stringent regulations developed in an attempt to control and seriously reduce the impact of some 39-million vehicles within Great Britain (Parry, 2019), along with the remaining collective of around 279-million across the rest of Europe (European Automobile Manufacturers Association, 2018).

These emissions regulations vary regarding actual vehicles type; ‘cars’, ‘vans’, ‘trucks’ etc. They’re collectively named as the “Euro emissions standards” and within the light passenger vehicles the standard is currently at Euro 6 which assesses the production of; Nitrogen oxides [NO_x], Carbon Monoxide [CO], Hydrocarbons [HC] and Particulate Matter [PM] during the New European Driving Cycle [NEDC] (RAC, 2020).

These regulations themselves are now thought to be ready for change, however there is confusion to as or when this will happen because of the change in homologation testing procedures to become the World harmonised Light vehicle Testing Procedure [WLTP]. This new cycle came into effect for any ‘brand new’ vehicles put into production from the year 2017 onwards. The introduction of these regulations and testing procedures have been the cause of a massive technological leap within recent years to improve the efficiency of ICE’s and the vehicles they are placed into.

Modifications to the powertrains have included the introduction of low friction belt/chain drives, lowered auxiliary demand losses, downsizing engine capacity, forced induction, cylinder deactivation and countless other innovations, to which Ford Motor Company have been at the forefront of. There is also the introduction and large development of alternative powertrains, in terms of ‘soft & hard’ hybrid style vehicles [HEV] and fully electric vehicles [EV]. This has become a much-preferred method of “cycle beating” since the introduction of the WLTP due to the indefinite gains from using such vehicles in comparison to solely ICE vehicles because the difference of regulations between test procedures.

Alongside this, transmission design has also been considered in depth, from efforts to eliminate mechanical losses from inertia and then the frictional losses within the components. Or by simply removing the manual control over gear choice and allowing automatic transmissions to take control. Of course, this naturally allows for the addition of

more gears and more effective gear ratios, both aiding the characteristics of the vehicle's 'driveability'.

Because of the 'fixed' nature of the manual transmission testing procedure of the NEDC and with some form of freedom within the WLTP, this paper has been written to demonstrate how the theoretical manipulation of gear ratios within a manual transmission can be used to aid in the reduction of fuel use and subsequently the combustion emissions of ICE vehicles across these homologation test cycles. Then a discussion and characterisation of the possible reasons as to perhaps why the manufactures chose not to use the ratios because of the effect on the driveability of said vehicles. Finally, look at other technologies being implemented within the internal combustion engines of today and develop a simple theoretically modelled impact, that they could have on reducing fuel consumption during the test cycles.

Objectives –

- Research and outline the fundamental regulations within the two homologation testing procedures considered within the study and demonstrate the fundamental differences between them.
- Using data provided, construct a mathematical model within Microsoft's Excel software in order to validate the methodology with respect to real measured values.
- Upon validation, develop the principle maths within MATLAB to begin the development of a gear ratio appropriation algorithm for Ford's 2.0L EcoBlue TDCi Mondeo (2019) & Volkswagen's 2.0L TFSI Passat (2006).
- Outline the potential savings that are to be had by modifying the gear ratios and identify the reasons for the changes in fuel usage.
- Develop a script in which the WLTP can be simulated in order to demonstrate an approach to be taken for potential enhancement of the gear ratios, along with a simple gear shifting strategy.

2. Timing Plan and Progression.

This project began on the 10th of October 2019 and has continued until the 4th of May 2020 because of the impact of a global pandemic. With a large project consisting of many tasks and fundamental stages that require completion, to enable progression, it was decided that a plan should be made with an estimated time for completion to ensure that the project remained on track. This timing plan was difficult to justify given the balance of available time, whilst continuing with the degree and multiple other modules, along with a lack of experience in this distinct area. However, using planning skills taken on from the group project module in the previous year, general software knowledge and vehicle simulation modelling, a rough target plan was constructed, as seen below. This plan was only to be used as a guide initially and was updated as progress was made throughout the time.

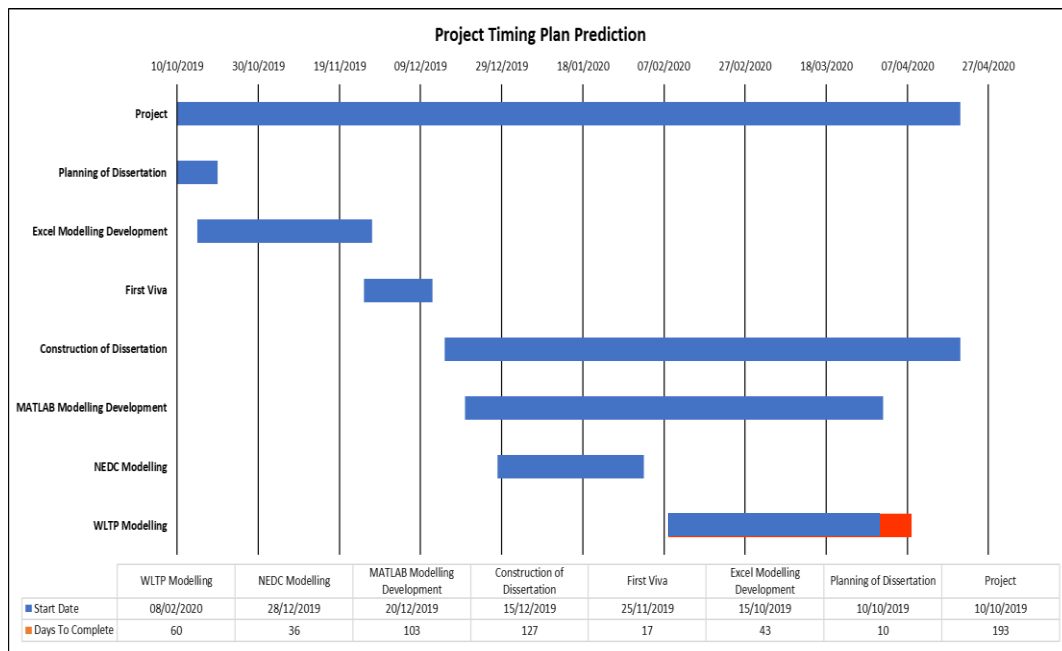


Figure 1 Overview of key points across the project.

All points within this plan are the projects milestones, meaning that they are key areas that required completion in order to continue through and simultaneously achieve the objectives laid out in the introduction.

As can be seen, the WLTP Modelling had overrun the predicted time because of the difficulties found creating various automated gear shifting scripts for the two test vehicles. The difficulty lied in the multiple series of regulatory checks and requirements that had to be met, whilst the gear ratio was being varied. However, this was eventually overcome and the task complete.

[Appendix A]

3. Drive Cycles.

3.1. New European Driving Cycle [NEDC].

The first iteration of the European homologation testing cycle was constructed as a way in which to police automotive vehicle emissions levels, specifically; Carbon Dioxide [CO₂], Oxides of Nitrogen [NO_x] and Particulate Matter [PM], both of which are a direct resultant of internal combustion of both gasoline and diesel.

Testing procedures for emissions was first brought about in 1970, but the legislative requirements were somewhat vague. Hence through the coming years, it was developed with respect to power demand, target emission levels, testing conditions, vehicle equipment requirements, vehicle power output, along with a multitude of other more stringent regulations. All focusing on reducing the pollution created from the somewhat 320-million vehicles on Europe's roads alone (European Automobile Manufacturers Association , 2018).

The cycle finally coined the title of the 'NEDC' in the year 2000 after there was a change to the start of the cycle with the removal of the "warm-up" period that previously lasted 40 seconds and the emissions were not measured until after this time (TransportPolicy.net, 2018).

This now redundant test procedure was a development upon the combination of the 'ECE' phase (Urban driving phase) and the EUDC (Extra urban phase). The ECE cycle was constructed in order to simulate the accelerations and decelerations that result from driving within a city/town environment. The phase can be characterised by the following table.

ECE Test Cycle	
Distance	1.013 [Km]
Duration	195 [s]
Average Speed	18.35 [Km/h]
Maximum Speed	50 [Km/h]
Average Acceleration	0.599 [m/s ²]
Maximum Acceleration	1.042 [m/s ²]

Figure 2 ECE Characteristics (DieselNet, 2013).

The average speed does include the three individual idling periods within the short test phase and therefore brings it down considerably. The cycle and gear position is represented in a graphically in the following figure, to demonstrate the simplicity of the phase and why it is argued that the phase is somewhat unrealistic.

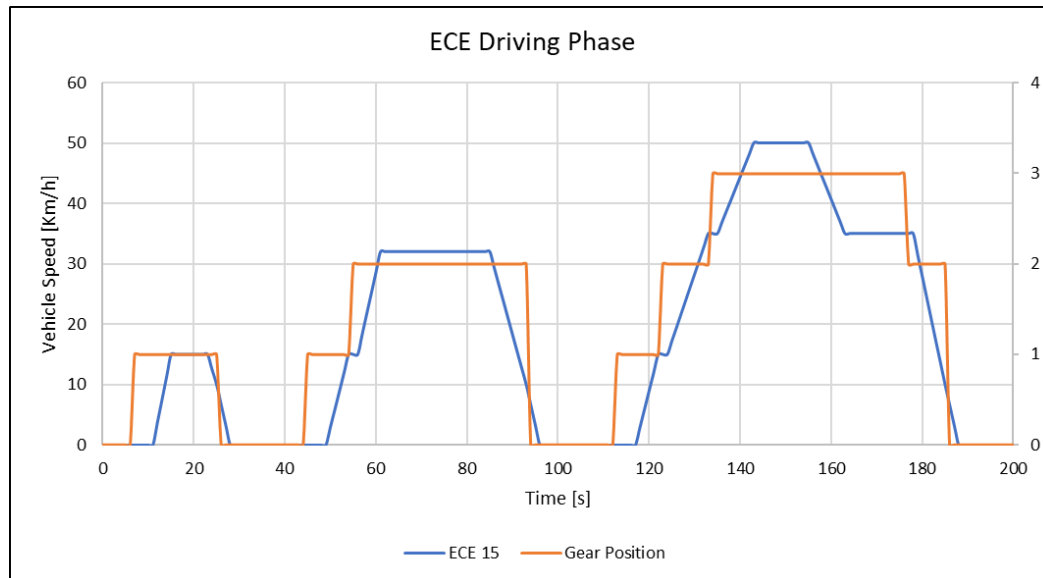


Figure 3 Urban driving cycle.

As can be seen, the cycle is created to simulate short driving periods where the first section only uses first gear, the second with first and second and then the last period of travel reaches third gear. The NEDC compiles four of the short cycles to make up the first 780 seconds of the test covering 4.05 km, referred to as ‘Urban phase’ which is part one of the cycle.

The second phase of the European driving cycle is the EUDC, constructed with the idea of simulating the vehicle when driving on higher speed ‘A’ roads, dual-carriage ways, and motorways. The extra urban phase is characterised in the following.

EUDC Characteristics	
Distance	6.95 [Km]
Duration	400 [s]
Average Speed	62.59 [Km/h]
Maximum Speed	120 [Km/h]
Average Acceleration	0.354 [m/s ²]
Maximum Acceleration	0.833 [m/s ²]

Figure 4 EUDC Characteristics (DieselNet, 2013).

Again, the average speed is including the idle periods, however they are short and do not affect the average velocity significantly. The phase is represented within the following figure and as can be seen, it is very simplistic and only represents a very idealistic driving period.

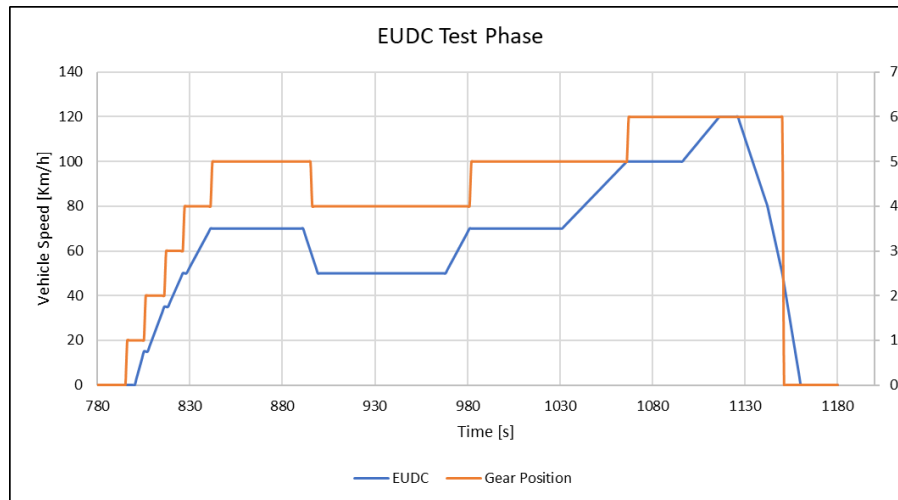


Figure 5 Extra Urban Driving Cycle.

It should be noted that this is the cycle representation for most vehicles. Although if a lower powered vehicle is to be tested, there is a slight change regarding maximum speed in which it is reduced to 90 Km/h, instead of the standard 120 Km/h.

Throughout its use across the past years from 1996 – 2017, the emissions standards for passenger vehicles were updated in conjunction with the test till it was finally decided that because “the average speed of the NEDC is unrealistically low” (verband der automobilindustrie, 2020), joint with the undeniable fact that the emissions on the real roads were vastly different to that measure in the “lab”, it should be made obsolete. However, across the years the emissions regulations for both NO_x and Particulates were reduced by 97% & 98% respectively (verband der automobilindustrie, 2020). Whilst, for petrol vehicles, CO dropped by 63% since 1993 and HC by 50% since 2001 (RAC, 2020). [Appendix B]

ECE Test Cycle	
31st December 1992	Euro 1
1st January 1997	Euro 2
1st January 2001	Euro 3
1st January 2006	Euro 4
1st January 2011	Euro 5
1st January 2015	Euro 6

Table 1 Emissions standard and their starting years (RAC, 2020).

3.2. World Harmonised Light Vehicle Testing Procedure [WLTP]

Starting from 2017 for any new generation vehicles and 2018 for newly registered vehicles, the homologation test procedure used to categorise carbon dioxide emissions [g/km] and fuel usage in miles per gallon was changed from the new European driving cycle to the world harmonised light vehicle test procedure. This change of test cycle was the first attempt at producing a test cycle to which all OEM's across the world would have to abide to and would introduce the mandatory additional test known as the Real Driving Emissions or RDE. As the name suggests, this test is conducted on real roads, with emission measuring equipment attached to the vehicle constantly recording the data across a period of 90 – 120 minutes. This cycle does include many regulations such as temperature, incline, and style of driving. Although this test sequence is still subject to conditions, because of the unpredictability of weather conditions and the accuracy of test equipment. Therefore the test is still more of a conformation of laboratory data, like NO_x production, and is not being used as a genuine representation of fuel economy (Whatcar, 2018). The RDE procedure consists of the following conditions.

- Low and High altitudes
- All year-round temperatures
- Additional vehicle payload
- Up and Downhill driving sections
- Urban, Rural and Motorway driving

[(ACEA, 2020)]

The WLTP is far more stringent in regulations than the NEDC ever was and is far more representable of real-world driving. For example, tests conducted for vehicles in Europe now use a temperature of 14°C, instead of the previous range of 20 - 30°C and this will have a knock on effect to vehicles warm up times, over fuelling periods and then change in air density for drag exhibited by the vehicle. The gear shift points for a manual transmission are no longer set and are dependent on engine speed, power, and torque demand. So, the possibility of simply changing ratios to effectively “beat” the cycle is no longer possible. The pattern has to be ran multiple times, with many regulations governing what is possible, for example the minimum engine speed gears above 2nd can go down to, the length of time held in a gear, inability to skip during acceleration, this list goes on.

The difference between the main components of the two cycles (NEDC & WLTP) can be seen in the below table. It is plain to see the sheer amount of difference just from the figures and then the style of driving and demand of the vehicle is highlighted perfectly in the graphical comparison of vehicle speeds.

	NEDC	WLTP
Test cycle	Single test cycle	Dynamic cycle more representative of real driving
Cycle time	20 minutes	30 minutes
Cycle distance	11 kilometre	23.25 kilometre
Driving phases	2 phases, 66% urban and 34% non-urban driving	4 more dynamic phases, 52% urban and 48% non-urban
Average speed	34 kilometre per hour	46.5 kilometre per hour
Maximum speed	120 kilometre per hour	131 kilometre per hour
Influence of optional equipment	Impact on CO ₂ and fuel performance not considered under NEDC	Additional features (which can differ per car) are taken into account
Gear shifts	Vehicles have fixed gear shift points	Different gear shift points for each vehicle
Test temperatures	Measurements at 20-30°C	Measurements at 23°C, CO ₂ values corrected to 14°C

Figure 6 Key area comparison of the NEDC & the WLTP (ACEA, 2020).

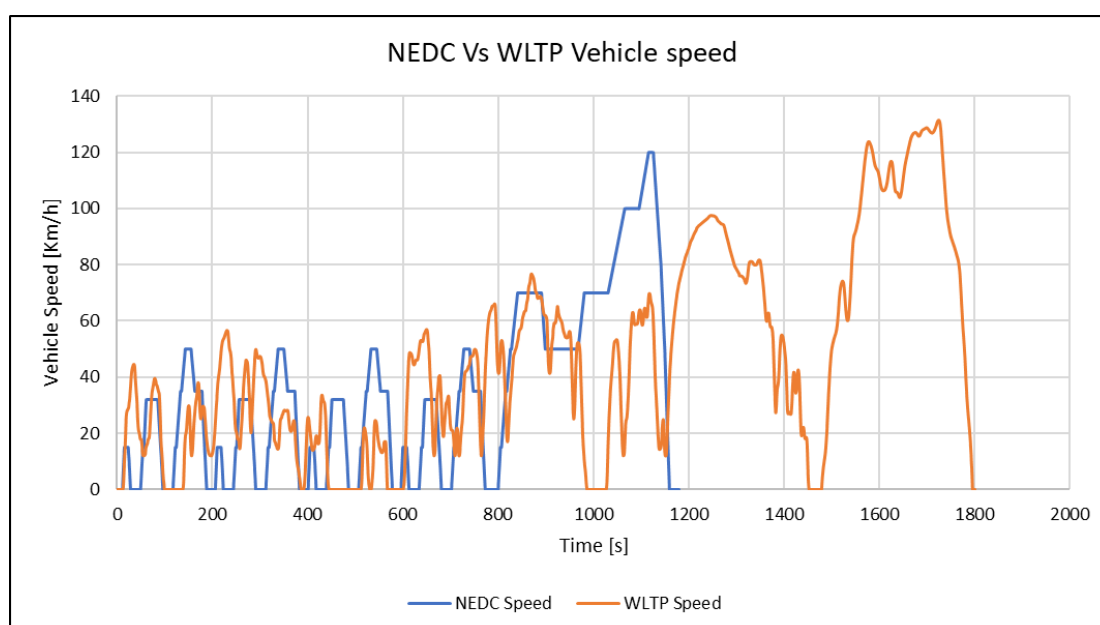


Figure 7 NEDC & WLTP Vehicle speed comparison.

Each test procedures vehicle speeds were taken from the appropriate governing documentation, Addendum 100: UN Regulation No. 101 for the NEDC and Addendum 15: Global technical regulation No. 15 for the WLTP.

4. Manual Transmissions.

4.1. Basic Operation and Characteristics.

With a manual transmission, “As the name suggests, drivers have to change the gear ratio setting rather than the transmission doing the job for them”. (Crolla, Transmissions and driveline, 2009). At this point in history vehicles are beginning to remove the real necessity of an actual driver, with autonomy becoming more reliable, automatic transmissions are now the norm for new passenger vehicles, all joint with a tremendous amount of “driver aids” equipped, it’s starting to make the manual transmissions look outdated and obsolete. Especially with the ever increasing sales of electric and hybrid vehicles that either don’t have a direct form of transmission, or they use a continuously variable transmission [CVT], it’s thought that manual transmission will be eradicated within 10 years (RAC, 2020).

However, whilst they are still “kicking about”, a manual transmission has many positives in comparison to the alternatives. For example, their average efficiencies are much higher than that of automatic transmissions, within one study conducted by Michael A. Kluger & Denis M. Long, manual gearbox efficiency was averaging at 96.2% in comparison to automatic efficiency average of 85.3%.

	Time in Gear	Representative Efficiency	Current Production Efficiency Variation
Gear 1	8%	93.5%	92-96%
Gear 2	10%	92.0%	92-97%
Gear 3	21%	94.0%	93-97%
Gear 4	20%	97.4%	93-99%
Gear 5	41%	93.8%	92-97%

Table 3 Manual transmission efficiency (Long & Kluger, 1999).

	Time in Gear *	Efficiency	Current Production Efficiency Variation
Gear 1	9%	70.3%	60-85%
Gear 2	4%	78.1%	60-90%
Gear 3	5%	86.0%	85-95%
Gear 4	32%	86.2%	85-95%
Gear 5	40%	88.7%	83-94%

* Excludes 10% for idle

Table 2 Automatic transmission efficiency (Long & Kluger, 1999).

Along with their superior efficiencies, the driveability and “feel” of a vehicle can be greatly improved if fitted with a manual transmission. This is because it requires work from the driver and brings a sense of unity and fun with the vehicle, as opposed to the simplistic and “boring” nature of an automatic or CVT transmission. However, with this ease and lack of control with an automatic, across a drive cycle alike the WLTP with its newly introduced RDE test, being able to control exactly when the vehicle changes gear is key to repeatability of low emissions and fuel usage results.

For the manual transmission, it is obvious when a vehicle has one fitted because of the required operation of a clutch pedal, or at least for common passenger vehicles without a straight cut sequential gearbox fitted. The job of the clutch is to disengage the drive between the engine and the input shaft of the box, allowing the change of gear to be much smoother and reduce potential for damage. Clutches were originally operated via the use of a cable, but they've moved onto hydraulic operation with a master cylinder on the pedal and a slave cylinder commonly moving the "clutch fork" that presses on the pressure plate and stops the transfer of drive.

The change of selected gear is helped with the use of synchronising assemblies referred to as 'synchro-meshes'. They can reduce noise, grinding & damage of gears and smoothness of changing gear. The objective is to bring the speed of the desired gear to roughly the same speed as the input shaft, allowing the gear selector to easily 'glide' over and engage the new desired gear.

Gears within a passenger vehicles manual transmission are usually paired with free spinning gears on the input shaft with those usually fixed to the output shaft. For example, Ford's 1600cc EcoBoost transmission uses the output shaft to select between 1st and 2nd that are free to spin, whilst 3rd, 4th, 5th and 6th are fixed gears on the shaft with their freely spinning pair attached to the input shaft.

The pair requires at least one independent free turning gear, because of the principle of a standard manual transmission only being able to transmit torque through one "path", seen the figure below. If more than one gear were to be engaged the transmission will simply "lock" and cause serious damage to itself and potentially the rest of the drive line. Semi-sequential gearboxes are capable of "pre-loading" gears and having multiple gears engaged because of its dual clutch pack ability to engage and disengage separate shafts of gears immediately.

[Appendix C]

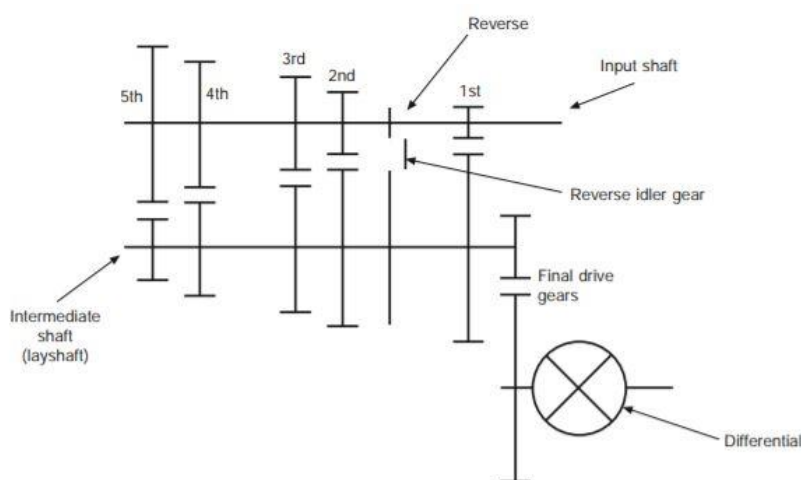


Figure 9 FWD 5spd manual transmission common construction (Crolla, Transmissions and driveline, 2009).

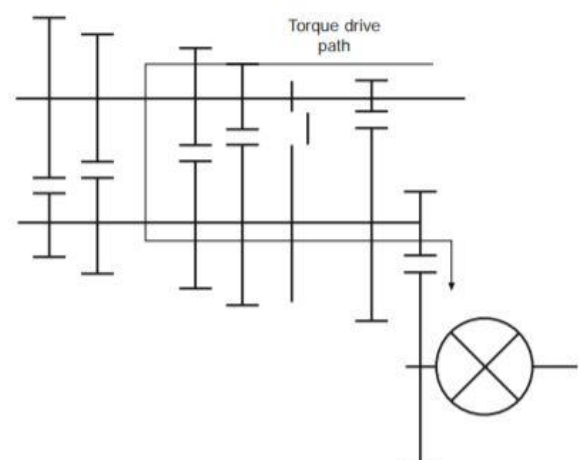


Figure 8 Depiction of a torque path through a manual transmission (Crolla, Transmissions and driveline, 2009).

5. Vehicle Modelling & Validation in Excel [NEDC].

The initial testing and development of this project was to be done using two different test vehicles, with the idea of aiding a further understanding of how to manipulate the provided data of the fuelling of said vehicle, based on the required torque demand at the associated engine speed.

The two vehicles used in this section were a Ford Mondeo 2011 [5-dr] 1600cc EcoBoost GTDI petrol Estate, a Volkswagen Passat 2006 [5-dr] 2000cc TFSI petrol Estate.

Tested Vehicles			
Ford Mondeo [2011]		VW Passat [2006]	
Fuel Type	Petroleum	Fuel Type	Petroleum
Vehicle Mass	1460Kg	Vehicle Mass	1550 Kg
Engine Cap.	1596cc	Engine Cap.	1984cc
Max Power	118 kW @ 5700 rpm	Max Power	147 kW @ 5100 rpm
Max Torque	240 Nm @ 1600 - 4000	Max Torque	280 Nm @ 1800 - 5000
No. Gears	6	No. Gears	6
Transmission Type	Manual	Transmission Type	Manual
Vehicle Type	Estate	Vehicle Type	Estate
NEDC Combined MPG	42.8	NEDC Combined MPG	34
Urban Fuel Usage	8.9 L/100Km	Urban Fuel Usage	11.1 L/100Km
Extra Urban Fuel Usage	5.3 L/100Km	Extra Urban Fuel Usage	6.6 L/100Km
CO ₂	155 g/Km	CO ₂	197 g/Km
Coe. Of Drag	0.29	Coe. Of Drag	0.29
Frontal Area	2.208 m ²	Frontal Area	2.235 m ²
Euro Emissions Band	Euro 5	Euro Emissions Band	Euro 4

Table 4 Initial Test Vehicle Comparison Table (Ford Motor Company, 2012), (ultimateSPECS, 2020), (automobile-catalog, 2011).

These vehicles have many similarities with respect to external presentation and their associated measurements, including vehicle height, length, and width, causing a similar frontal area size. This then becomes important because given that the aerodynamic drag coefficient is the same, the external drag force would therefore be of a similar nature. The vehicle test mass shows similarity also, along with both vehicles being a manual six speed transmission, meaning that the gear shifts will remain consistent because of the NEDC's fixed shift protocol. With all these similarities, it culminates to a test that has now become a derivative of essentially only the 'Power-Units' and their given 'Drivetrains'.

5.1. Vehicle & Engine Speed Assessment and Modelling.

5.1.1. Linear to Angular Wheel Speed.

The new European test cycle has a fixed vehicle speed that is given as a linear speed of which the vehicle must be at. Using these speed values, they need to be converted into meters per second [m/s] from kilometres per hour [Km/h] and this is simply done by,

$$\frac{\text{Speed in Km/h}}{3.6} = \text{Speed in m/s}$$

With this new unit of speed, the relationship between linear and angular velocity is used to calculate the speed in which the wheel is rotating.

$$V = r \cdot \omega$$

$$\omega = \frac{V}{r}$$

Where V is the linear velocity in m/s, and r is the effective rolling radius of the wheel.

5.1.2. Rolling Radius.

To obtain a ‘rolling radius’ of the driven wheel, the tyres profile is required, and the given wheel size used on the vehicles in question.

Tested Vehicles			
Ford Mondeo [2011]		VW Passat [2006]	
Front Wheel Drive		Front Wheel Drive	
Driven Wheel/s	2	Driven Wheel/s	2
Front Wheel Size	16"	Front Wheel Size	18"
Rear Wheel Size	16"	Rear Wheel Size	18"
Front Tyre Size	215/55 R16	Front Tyre Size	235/40 R18
Rear Tyre Size	215/55 R16	Rear Tyre Size	235/40 R18
Front Brake Type	Vented Disc	Front Brake Type	Vented Disc
Rear Brake Type	Solid Disc	Rear Brake Type	Solid Disc
Front Brake Size	300mm	Front Brake Size	312mm
Rear Brake Size	302mm	Rear Brake Size	282mm

Table 5 Vehicle Wheel and Brake Specifications (Ford Motor Company, 2012), (Car.info, 2020), (Eurocarparts, 2020).

Given the highly dynamic nature of tyres and the many contributing factors, alike tyre pressure, temperature, tyre construction, velocity and tyre slip (Gillespie, 1992). This creates a problem when trying to increase accuracy and because of this the rolling radius was found using a simplified equation of,

$$\text{Rolling Radius} = \left(\frac{\text{Wheel Rim Size [mm]}}{2} + \text{Tyre aspect ratio [mm]} \right) \cdot 0.97$$

For example, the VW Passat rolling radius was taken as:

$$\left(\frac{457.2}{2} + (235 \cdot 0.4) \right) \cdot 0.97 = 312.92\text{mm} - 0.31\text{m}$$

And the Ford Mondeo rolling radius was taken as:

$$\left(\frac{406.4}{2} + (215 \cdot 0.55) \right) \cdot 0.97 = 311.81\text{mm} - 0.31\text{m}$$

Using the rolling radius as a constant throughout the model is one area that would have caused a small indiscretion in the accuracy of all the model. But this would not have caused a large margin of error and therefore is believed to be adequate to achieve a high accuracy simulation of the drive cycles throughout.

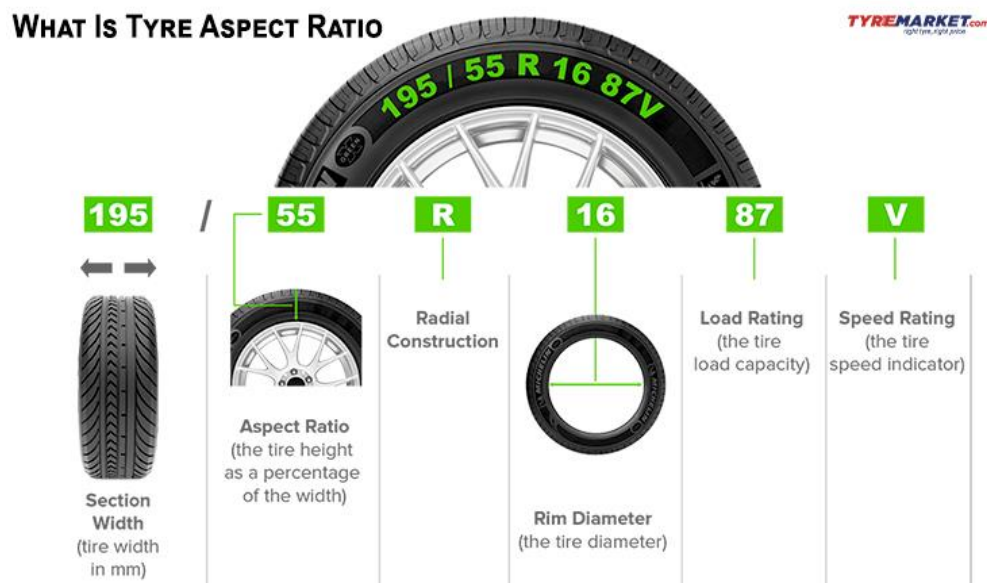


Figure 10 Depiction of the markings stamped on the sidewall of tyres (Team Tyremarket.com, 2020).

5.1.3. Final Drive and Individual Gear Ratios.

With the angular speed at the wheels, the next important factor is the final drive ratio, leading then into the individual gear ratios. The gear ratios and efficiencies for the Volkswagen Passat were provided from MAHLE Powertrain Ltd. and these efficiencies were used for both vehicles. This is because the efficiency of the Ford Mondeo transmission was unknown and uncharacterizable from the test data, due to the constant changing vehicle speed.

Tested Vehicles					
Ford Mondeo [2011]		Efficiency %	VW Passat [2006]		
Gear Ratios		94% 94% 96% 97% 97% 96% 94%	Gear Ratios		
1	3.727		1	3.36	
2	2.048		2	2.09	
3	1.357		3	1.47	
4	1.032		4	1.1	
5	0.821		5	0.87	
6	0.69		6	0.73	
Final Drive	4.067		Final Drive	3.94	

Table 6 Initial Test Vehicle Gear Ratios & Efficiencies (automobile-catalog, 2011).

Using the simple relationship of gears and their speeds, the speed after the differential and into the gearbox is simply shown as a rearrangement of:

$$\frac{\omega_{in}}{Final\ Drive\ Ratio} = \omega_{out}$$

To,

$$\omega_{out} \cdot Final\ Drive\ Ratio = \omega_{in}$$

Where, in this situation, the ω_{out} is the angular speed of the wheel and ω_{in} the speed of the output shaft within the transmission.

Naturally using the same principle for the speed at the clutch, the final drive ratio would simply be changed to the ratio of the current selected individual gear. After which, the speed found at this point is the equivalent of the current rotational speed of the crankshaft and therefore the engine.

5.2. Forces Acting on the Vehicles Driven Wheels.

The force at the wheels is the most important varying function of this testing procedure, simply because of the legislation stating that the additional inertial torque losses occurring throughout the drivetrain are neglected in the official road load determination model of the NEDC (G. Fontaras, 2016). The three main considered forces acting on the wheel come from the addition of; the vehicles inertial requirement based on the principle Newtons second law, rolling resistive forces and the aerodynamic drag forces incurred by the vehicle moving through a fluid.

Using Newtons second law:

$$Force = Mass \cdot Acceleration$$

The force required at the wheel due to having to accelerate the vehicle is found using the stated accelerations, or by using the change in velocity, multiplied by the test mass. The test mass is stated to be the vehicles kerb weight [Kg] with the addition of 100Kg mass extra, as a ballast constant for any vehicle being tested.

The rolling resistance force is a product of the traction of the tyres and their frictional, heat and sound losses. This rolling resistance force is extremely subjectable to many varying factors like; temperature, inflation pressure, vehicle speed, tyre condition/age, tyre material construction and tyre slip (Nielsen & Sandberg, 2002). Another exclusive issue of the rolling resistance is achieving a value capable of representing the coefficient. Common practice is to perform a coast down test in which both the coefficient of drag and the coefficient of rolling resistance can be approximated to a reasonable level of accuracy.

Unfortunately, neither of the test vehicles, nor a suitable test facility were available to conduct said coast-down test, therefore a rolling resistance coefficient had to be assumed based on the tyre's inflation pressure and the vehicle mass, due to these being the most notable contributing factors. Given the test vehicles are of roughly the same weight and have rolling radius that is effectively identical, using Gillespie's "common" variable values table [p. 117], within his vehicle dynamics book, a value of '0.015' was chosen to represent a coefficient for each vehicle and then used in the following equation.

$$RR\ Force = Mass \cdot 9.81 \cdot Coe_{RR}$$

Where the mass is in kilograms multiplied by the acceleration due to gravity and finally the coefficient of rolling resistance.

It could be argued that due to the difference in tyre construction and the side wall heights affecting their operation characteristics, the Ford and VW could have "vastly"

different coefficients. But again, due to the lack of testing available, these values had to be assumed and would therefore also be an area in which inaccuracy could begin to affect the results, primarily the higher speed ‘EUDC’ phase.

The aerodynamic resistive drag forces are again susceptible to various factors like, temperature, pressure, humidity, cross winds, and direction of flow. Because of these consistently varying inputs, the coefficient of drag can prove difficult to find. When conducting a coast-down test, it’s important to reduce all potential for errors and the test should be conducted multiple times in order to achieve an average coefficient until the “statistical accuracy of the average is equal to or less than 4 per cent” (UNITED NATIONS, 2005).

With the coefficient of drag found, the respective force is calculated using:

$$Drag\ Force = \frac{1}{2} \cdot \rho \cdot C_d A \cdot V^2$$

(Katz, 1995)

In which, ρ is the density of air which is regulated based on the temperature, humidity and pressure conditions of the air as stated in ‘Annex 7 – Appendix, p. 64 of UNECE R 101’. $C_d A$ is representing the coefficient of drag multiplied by the effective frontal area, usually taken as 85% of the vehicle’s width multiplied by the height (A2 Wind Tunnel, 2013). And finally, V^2 being the velocity of the vehicle, assuming there need not be corrections due to external wind influences, which is evidently the key to the equation. Because of its squared nature, the drag force grows at an exponential rate and when speed gets to around about 30m/s, it is shown to be the biggest contributing factor to resistive forces. This will seriously affect the vehicles economic performance across the EUDC ‘high speed’ phase, which is why many major manufactures have spent a considerable amount of time and resources to understand the best way to make effective use of the air, gaining in both fuel and dynamic performance. [\[Appendix D\]](#)

Tested Vehicles			
Ford Mondeo [2011]		VW Passat [2006]	
Coe. of Drag	0.29	Coe. of Drag	0.29
Frontal Area	2.208 m ²	Frontal Area	2.235 m ²
Air Density	1.2 Kg/m ³	Air Density	1.2 Kg/m ³
Air Temperature	293 [k]	Air Temperature	293 [k]
Coe. of rolling resistance	0.015	Coe. of rolling resistance	0.015
Vehicle Mass	1460Kg	Vehicle Mass	1550Kg
Test Mass	1560Kg	Test Mass	1650Kg
Test maximum positive acceleration	1.04 m/s ²	Test maximum positive acceleration	1.04 m/s ²

Table 7 Vehicle External Force Variables.

5.3. Torque Demand & Inertial Losses Through the Drive Unit.

To calculate the vehicles fuel energy requirement, the torque demand necessary to accelerate the vehicle is needed. Given the force at the wheel/s and the relationship between a torque and its equivalent force is:

$$T_{Wheel} = F_{wheel} \cdot RR$$

Where RR is the rolling radius, the torque required by the external forces acting on the vehicle is found.

Because there is multiple “rotating/moving” parts within the driveline of any vehicle, there is also additional torque required to accelerate the parts along the way, such as: the wheels, driveshafts, brake discs, wheel hubs, etc. Therefore, using the following relationships, the additional system torque is found.

$$Torque = I \cdot \alpha$$

Where I is the moment of inertia of the part and α is the angular acceleration of said part.

Angular acceleration is found using the following, or because the test is done in one second steps, one could simply do the change in angular velocity in respect to the new minus the previous value.

$$\frac{\omega_t - \omega_0}{t} = \alpha$$

Where ω_t is the angular velocity at time step t , and then ω_0 is the starting angular velocity.

Regarding the moments of inertia, the difficulty occurs because of the shape of the parts and their potential to change when in dynamic motion, i.e. tyres, universal joints, etc. To develop a higher level of accuracy, one solution would be to create a C.A.D model within a software alike SolidWorks and allow the software to calculate the concerned moment of inertia correctly. Again, unfortunately these parts were not available and therefore could not be created, hence the shapes had to be ‘simplified’ and assumed symmetric.

With the wheel and the tyre, the moments of inertia [MMI] were found separately and then added together. The wheel MMI was found using the principle that because of the shape and the lack of material removal across the “wheel disc”, a suitable equation of the following could be used.

$$MMI = \frac{1}{2} M \cdot r^2$$

The tyre's MMI was found using the following, because of its distance from the central axis of rotation and again the difficulty to model the complex shape and dimensions accurately.

$$MMI = \frac{1}{2} M \cdot (r_1^2 + r_2^2)$$

Within both equations, M is the mass of the 'object' and then r is the effective radius. For the wheel rim itself, this was used as the distance furthest from the centre. For the tyre, the radii of r_1 & r_2 is the internal and external distances, respectively. The measurements used for the Mondeo can be seen below:

Ford Mondeo [2011]	
215/55R16 6.5Jx16 ET50	
Wheel Width [mm]	165.1
Rim Radius [mm]	203.2
Tyre radius [mm]	113.74
Total Radius [mm]	316.94
Max Wheel Torque [Nm]	368.34
Wheel Rim Mass [Kg]	9
Tyre Mass [Kg]	8
Wheel inertia [Kg·m ²]	0.3716
Tyre inertia [Kg·m ²]	0.5670
Total inertia [Kg·m ²]	0.9386
Wheel circumference [mm]	1991.36

Table 8 Ford Mondeo 1600cc EcoBoost Wheel/Tyre Dimensions (Wheel-Size.com, 2020).

Brake discs, wheel hubs and driveshafts were again simplified to be of a solid disc/cylindrical shape and assumed symmetrical. Although this is not exactly realistic, the additional torque required to accelerate these parts is somewhat insignificant in comparison to that of the wheel/s or the engine itself. Therefore, the models can be simplified and seen only as an attempt to build a more realistic model. Because they were modelled as a solid disc/cylinder, the equation used is again:

$$MMI = \frac{1}{2} M \cdot r^2$$

With the radius being the point across the disc where the mass is acting.

[Appendix E]

Within the transmission, the number of moving parts that are dependent and independent of each other can be difficult to understand without viewing the internals of said gearbox. Because of the fixed nature of some gears, the sliding action of a synchro-meshes, along with the complex behaviour of variable speed differential gears, internal inertia of a transmission is extremely difficult to theoretically model on a simple equation-based approach. To achieve an adequate understanding and model of a transmission's inertia, a computer aided design software would be required to allow the ability of a motion simulation of the parts through the chosen drive-cycle. Calculating how, with all the degrees of freedom, the acceleration of the clutch, gears, shafts, and bearings would necessitate additional torque. Therefore, although it's known that gear inertia should be in the region of 0.003 – 0.025 (Greenbaum, Kluger, & Westmoreland, 1994), the internal inertia of the transmission was left out of the models to ensure there was no mis-calculations that would lead to potentially drastic inaccuracies.

Furthermore, within the gearbox there are multiple contacting surfaces usually made of cast steel/plain carbon steels, that will generate frictional losses. Bearing shells to which the gears and shafts are mounted will also add to the frictional losses inside of the transmission. Because of the interlocking nature of the gears, 'backlash' is required to help prevent the possibility of locking the gears (Shastri, Ramamurthy, & Parab, 2008). This backlash has the potential to prove advantageous because of a potentially reduced cost and reduction in noise, but miscalculation can lead to; damage, overheating and overloading of the gears and therefore a reduction in efficiency (Kumar, 2016).

Transmission efficiency has been an area in recent years that OEM's and external small engineering companies have spent a considerable amount of time and money in order to aid the reduction of fuel consumption due to wasted energy. Where as in previous years, the specifications were simply on the torque transfer capability, weight and packaging, it's now about the requirements of increasing efficiency with the other desires coming secondary to that (Kluger, Greenbaum, & Mairet, 1995). Average efficiencies within manual transmission will vary across manufacturers, but given the technology of today and a furthered understanding, it could be said that on average a mass produced manual transmission will have an overall efficiency of around 96% (Long & Kluger, 1999). For example, MAHLE found that the VW Passat's gearbox [GVT] had an overall efficiency of 95.4%, importantly with the 4th and 5th gear being of the same efficiency. Which in most cases is unusual, because it's been found that 4th gear, or the direct drive gear, can be up to around 5% more efficient than the rest of the gears (Greenbaum, Kluger, & Westmoreland, 1994).

[Appendix F]

Using the provided efficiency of each independent gear as torque is transferred, the additional torque of the transmission can be found and subsequently added onto the previously found torque up to the differential.

Using the following principle:

$$T_{axle} = T_{engine} \cdot Gear\ Ratio \cdot \mu_{gears}$$

$$T_{axle} = T_{engine} \cdot \left((n_{gear} \cdot n_{\mu}) \cdot (Final\ Drive \cdot Final_{\mu}) \right)$$

Therefore the torque transmitted to the axle/driveshaft, is simply that produced by the engine, multiplied by the current gear ratio [individual gear multiplied by the final drive ratio] including both gears signified efficiency, μ_{gear} .

The clutch in the both vehicles is a simple hydraulic single plate dry clutch system constructed and operated as shown below:

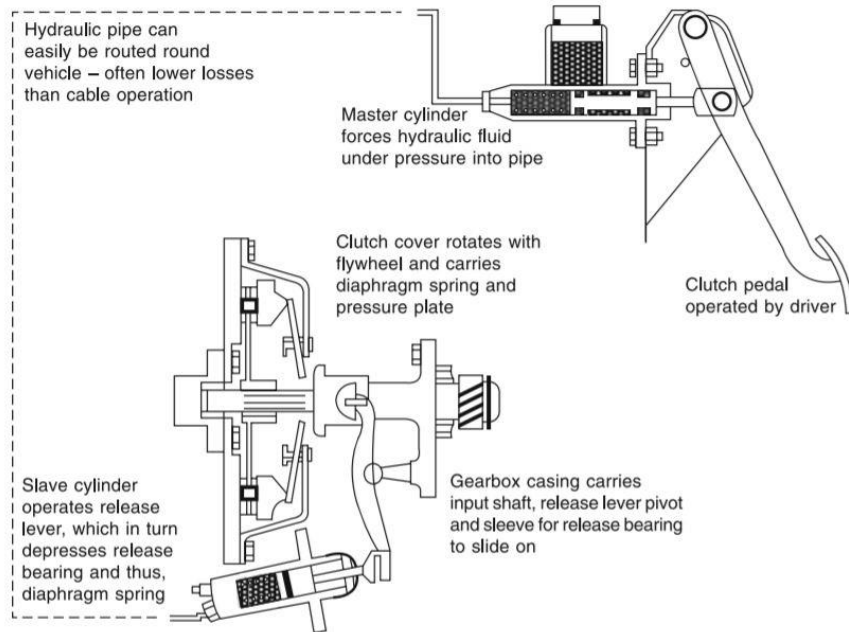


Figure 11 Diagram of clutch and actuation mechanism (Crolla, Fig. 5.1-15 Diagram of clutch and actuation mechanism., 2009).

As with the previous parts, the shape of both the clutch friction plate and the pressure plate, an actual MMI is difficult to calculate “by hand”. So, the plate was done assuming the method of a solid disc, whilst the pressure plate was disregarded. This is because, not only is the shape complex, but it is also bolted to the flywheel which will cause an offset of central axis that is nigh on impossible to model without computer aided design software.

The model could very well have been made if the parts were available, though given the somewhat trivial addition to torque, it was thought best to neglect it.

These vehicles have also been fitted with a dual mass flywheel, which changes the inertial characteristics to be much more complicated to a traditional single disc ring gear flywheel. This is because of its two sperate masses connected with a plate that can rotate a small distance independently from each other, that is then dampened with a series of internal springs that again completely change the behaviour of the rotating flywheel. Because the values of these masses, spring rates, number of springs and their dimensions are unknown, these flywheels were modelled on a single rotating solid disc principle. Therefore, the MMI was assumed to be approximately $0.15 \text{ Kg}\cdot\text{m}^2$ because it's known that a standard single mass Formula Ford flywheel has an MMI of $0.091 \text{ Kg}\cdot\text{m}^2$ (Hakewill, 2010) and then two newer dual mass flywheels have a joint inertia of around $0.12 \text{ Kg}\cdot\text{m}^2$ (Schaper, Sawodny, Mahl, & Blessing, 2009), (BOURGOIS, 2016). So, for a worst-case scenario the MMI was set higher.

Considering the last inertial force as the internals of the engine itself, given the inability to correctly calculate the MMI because of no access to the required parts the value had to be assumed. This assumption lead to a value of $0.031 \text{ Kg}\cdot\text{m}^2$ for the Ford 1600cc EcoBoost and $0.04 \text{ Kg}\cdot\text{m}^2$ for the Volkswagen 2000cc TFSI. These assumptions were made as an attempt to predict, based on two defined smaller capacity engines. One being Ford's 1000cc 3-cylinder EcoBoost having a collective internal MMI of $0.0105 \text{ Kg}\cdot\text{m}^2$ (Liu, 2014) and PSA's 1400cc 4-cylinder DV4TD 8HT turbocharged compression ignition engine having an internal MMI of $0.0307 \text{ Kg}\cdot\text{m}^2$ (MRDJA, MILJIC, POPOVIC, & KITANOVIC, 2017).

A value of $0.15 \text{ Kg}\cdot\text{m}^2$ was given within the Volkswagen model, however it is thought this value was also attempting to include the additional MMI of the flywheel attached to the engine. This is thought because when assessing the complete MMI inclusive of flywheel for the 1000cc Ford and 1400cc PSA, the complete values are 0.1452 & $0.1049 \text{ Kg}\cdot\text{m}^2$, respectively. Given the resemblance to the stated value it is probable that the VW 2000cc TFSI could have a lower internal MMI because of the reduced flywheel mass and increased engine capacity, but this would require further investigation.



Figure 13 VW Passat 2.0 TFSI Engine (JOSEPH, 2015).



Figure 12 Ford 1.6L EcoBoost Engine (Ford Motor Company, 2020)

5.4. Fuelling Across the NEDC Cycle.

5.4.1. Ford Mondeo 1600cc EcoBoost

The Ford Mondeo initial model was primarily to gain an understanding of the inertia and torque modelling because the fuel “flow” data was not present in a manner that was easily manipulated based on required torque. The fuel data was exact measured flow data for the given tested cycle, in which the full criterion that Ford used for said test was unknown. Across the model, the torque demand was consistently mismatched in magnitude compared to that of Ford’s measured data, which can be seen below. This inconsistency lead to a momentary lack of confidence in the vehicle model, however when the fuel model was developed it was found that the model holds a high level of accuracy.

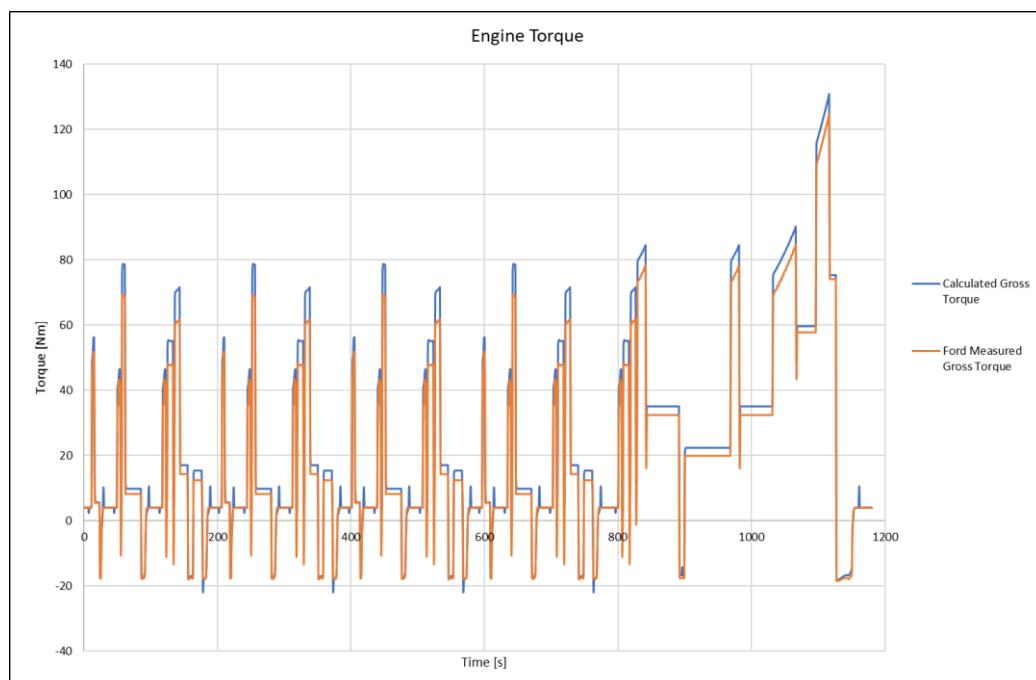


Figure 14 Comparison of calculated gross torque to Ford's measured torque.

As seen, calculated torque is consistently higher at all the acceleration points, which is why it is thought that regarding the values used, Ford perhaps were not testing this vehicle with the correct vehicle test mass (kerb + 100Kg). But instead, were perhaps running a calibration style test in which to ensure all measurements were registering and presenting their expected values.

Because the fuel flow data was equivalent of the power required by the engine across this test, of which when considering that power P , is equal to:

$$P = T \cdot \omega$$

The calculated power was obviously higher, therefore the fuel had to be found based on the assumption of linearity within Ford's "fuel map" and that it could simply be done on a ratio principle regarding the power, fuel and brake specific fuel consumption [BSFC].

With the measured fuel flow data presented in kg/h it was required to be in g/s. Done simply by:

$$\frac{g}{s} = \frac{\frac{Kg}{h}}{3.6}$$

Power was presented in Kw, therefore given that BSFC is given units of $\frac{g}{Kw \cdot h}$ and that the measured data is for every one second, the BSFC across the cycle is found using:

$$BSFC = \frac{Fuel (\frac{g}{s})}{Power (Kw)} \cdot 3600$$

To allow for ease of manipulation, the BSFC was calculated using:

$$BSFC = \frac{Fuel (\frac{g}{h})}{Power (Kw)}$$

Allowing the new fuel flow to be found by simply multiplying the measured BSFC by the calculated required power, to then be converted into g/s. This led to the following findings.

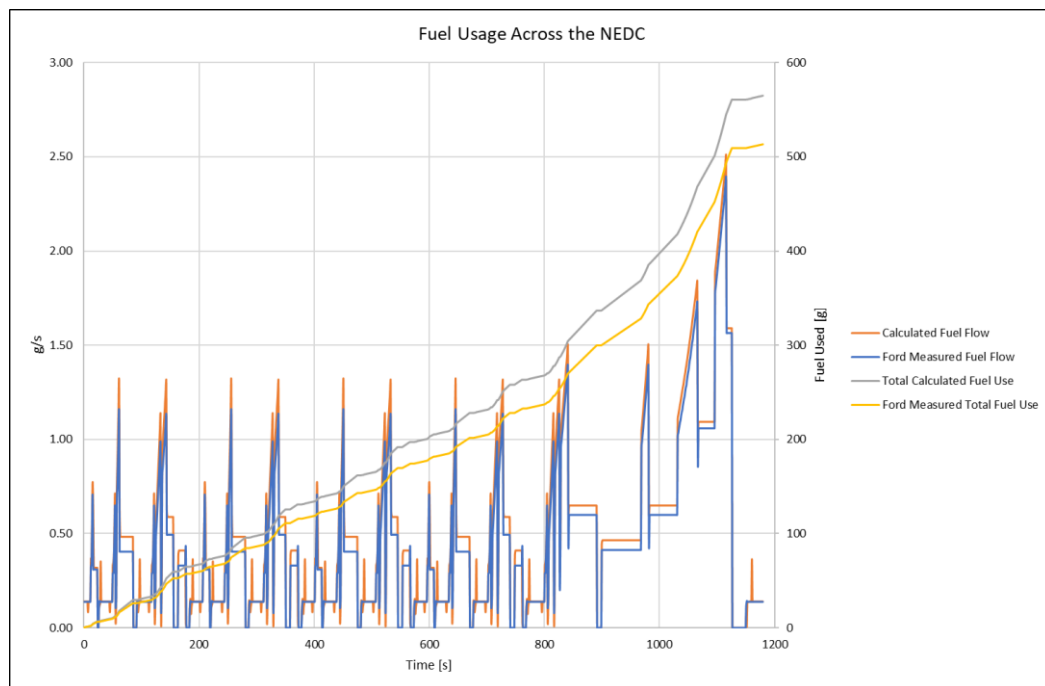


Figure 15 Calculated & Measured fuel flow and fuel used across the NEDC [Ford Mondeo 1600cc EcoBoost].

It's clear that the calculated model is seemingly grossly over predicting, but when compared to the stated values of miles per gallon and the individual phase fuel usage, the model has proved to be at a desirable level of accuracy, given the assumptions made. This will be discussed in a further section [5.6.2.].

5.4.2. Volkswagen Passat 2000cc TFSI

The data used within the Passat was measured by MAHLE Powertrain and presented in a table showing the respective fuel flow at a given engine speed and the brake mean effective pressure developed by the engine to necessitate this fuel energy.

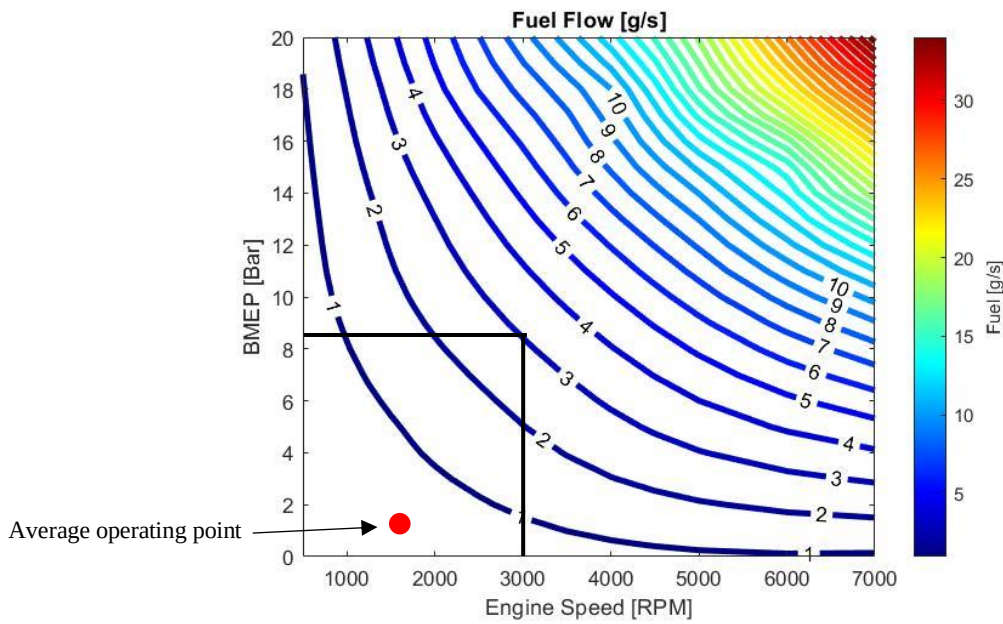


Figure 16 VW Passat BEMP Fuel flow map.

When assessing the way, that VW have generated their fuel map, it was clearly designed specifically for operating across the NEDC cycle. Because the maximum engine speed and BMEP (outlined with the two black lines in the figure above) show that the fuel flow reaches a maximum of around 3 g/s and the average operating point is in the region of 1.6 Bar BMEP and 1650RPM, keeping fuel flow to a minimum.

To calculate the fuel flow across the cycle, this required the equivalent brake mean effective pressure for the calculated torque. So, using:

$$BMEP = \left(\frac{2\pi \cdot \eta_r \cdot T_e}{V_d} \right) \cdot 10^{-5}$$

With η_r as the number of crank rotations to complete a cycle, i.e. two for a four-stroke engine. T_e is the effective torque at that point in Nm and V_d is the engine's displacement in m^3 of which is $1984\text{cc} \cdot 10^{-6} \text{m}^3$. The equation is multiplied by 10^{-5} to convert directly into units of Bar and not Pascals (Heywood, 1988).

The next fundamental equation to calculating the fuel flow is a simple interpolation relationship, shown as:

$$\frac{\text{Engine Speed} - \text{Speed Below}}{\text{Speed Above} - \text{Speed Below}} = \frac{\text{Fuel Flow} - \text{Fuel Below}}{\text{Fuel Above} - \text{Fuel Below}}$$

Rearranged into,

$$\text{Fuel Flow} = \left(\frac{\text{Engine Speed} - \text{Speed Below}}{\text{Speed Above} - \text{Speed Below}} \right) \cdot (\text{Fuel Above} - \text{Fuel Below}) + \text{Fuel Below}$$

The *Engine Speed* is the speed currently calculated, whilst *Speed Below* & *Speed Above* are the represented steps of measured engine speed: 500, 850, 1000, 1250, 1500, 1750, 2000, 2500, 3000, 3500, 4000, 4500, 5000, 6000, 7000. Represented by the figure below.

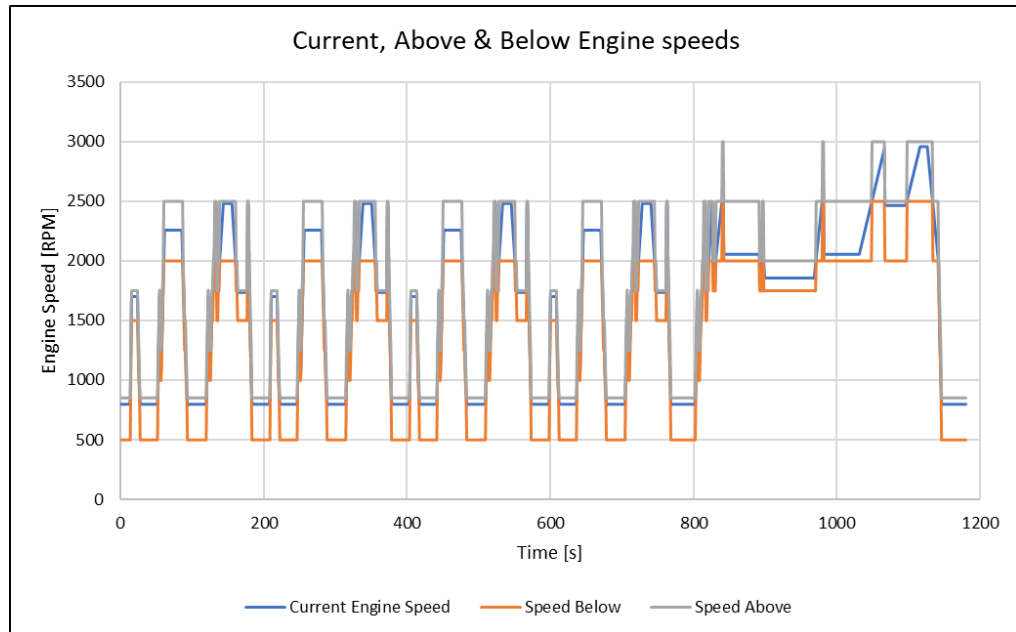


Figure 17 Current, Above and Below Engine Speeds across the NEDC Excel [VW Passat 2.0L TFSI].

With the found above and below representative engine speeds, the fuel flow for these are then required. However, with the map being dependent upon BMEP also for the desired engine speed, there was a requirement for further interpolation.

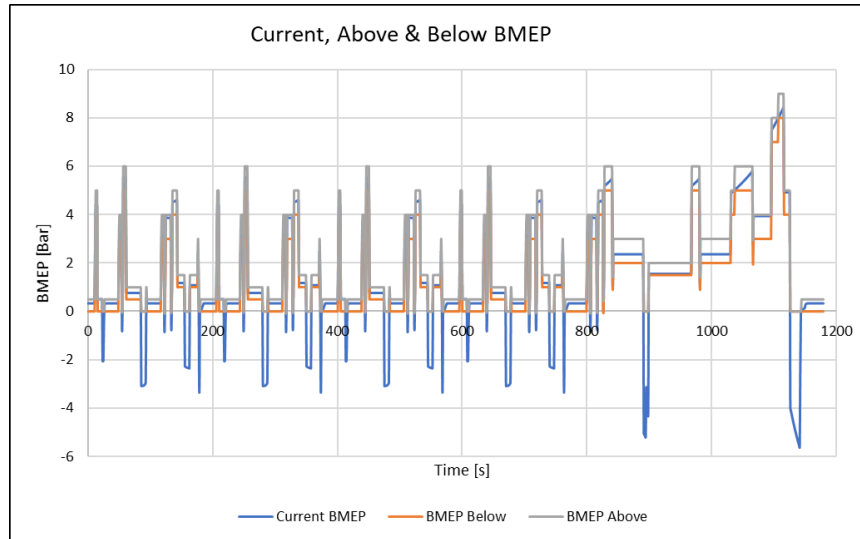


Figure 18 Current, Above and Below BMEP across the NEDC Excel [VW Passat 2.0L TFSI].

With the corrected above and below values for both the engine speed and brake mean effective pressure, the measured fuel flow for those points can be found. In order to complete this, the fuel flow values were recreated in finer steps of BMEP, i.e. from 0.5-1 down to steps of 0.001. This was done using Excel's polynomial curve fitting tool, for each of the plotted fuel flows and their associated speeds. An example of can be seen below.

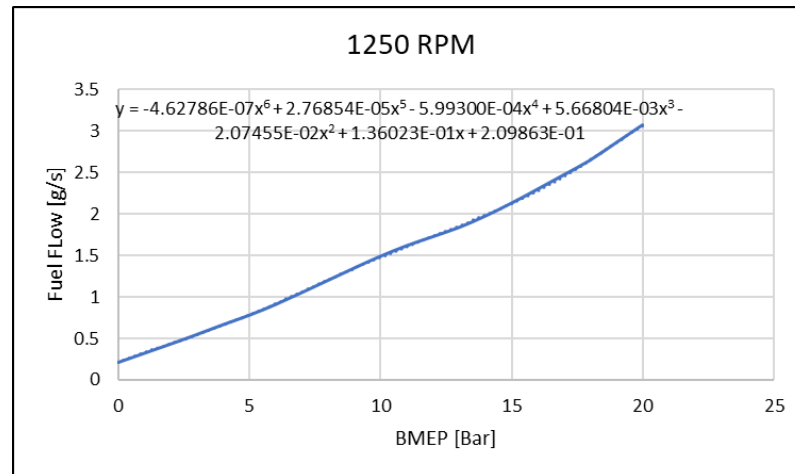


Figure 19 Polynomial curve fit for the fuel flow at 1250RPM [VW Passat 2.0L TFSI]

Excel can only take the polynomial fit to the 6th order, which can lead to difficult in accurately fitting the curve. But because of the relatively linear increase of the fuel flow, the curves were all fitted within an accuracy level of $R^2 = 9.98$ or greater, where 10 represents 100% accuracy.

Using the polynomial equation to 15 decimal places to ensure accuracy and a vector of BMEP from 0 – 20 in steps of 0.001, the fuel flow was interpolated to allow a reduction in steps and therefore increase accuracy when used in the small interpolation equation.

With all variables now available and using Excel's 'IF' & 'VLOOKUP' functions, the value of above and below BMEP was found with a statement alike, ' $=\{ \text{MIN}(\text{IF}(\$BJ\$9:\$BJ\$11008 > CS10, \$BJ\$9:\$BJ\$11008)) \}$ ' for finding the above and then ' $=\{ \text{MAX}(\text{IF}(\$BJ\$9:\$BJ\$11008 < CS10, \$BJ\$9:\$BJ\$11008)) \}$ '. The principle with these were to find the minimum value of all those greater than the actual calculated pressure in the first statement and then the maximum value of all those smaller than the calculated pressure.

With the interpolated values for each engine speed being in a column vector with BMEP on the left and fuel flow on the right, the 'VLOOKUP' function could be used to pick the associated fuel flow. The following formula was used to execute this, ' $\text{IFS}(\text{CU10}=\$BJ\$5, \text{VLOOKUP}(\text{CY10}, \$BJ\$8:\$BK\$20008, 2, \text{FALSE}))$ '. Where the above engine speed was used to characterise the vector $[\text{CU10}=\$BJ\$5]$, then using the found above BMEP $[\text{CY10}]$, this value was found in the appropriate vector $[\$BJ\$8:\$BK\$20008]$. The '2' signifies the column to choose a value from, which was the fuel flow and then 'FALSE' returns the exact value within that correct cell. The 'IFS' statement was used because of its ease to allow multiple statements within one cells "code".

Finally using the short interpolation equation the final fuel flow could be found and is represented within the below figure, along with the above and below fuel flows to show the correlation between using the finer step of BMEP in comparison to standard step of 0.5-1. The values for above and below are consistently closer than that of in figure 17, showing the step-in engine speed. This large step in speed is the reason for there still being points where the above flow can be seen to be "far" away from the calculated flow. Because the given data is at these fixed speeds, there is no way to correct this possible issue and as will become apparent within the following sections, it did not affect the model accuracy to any real extent.

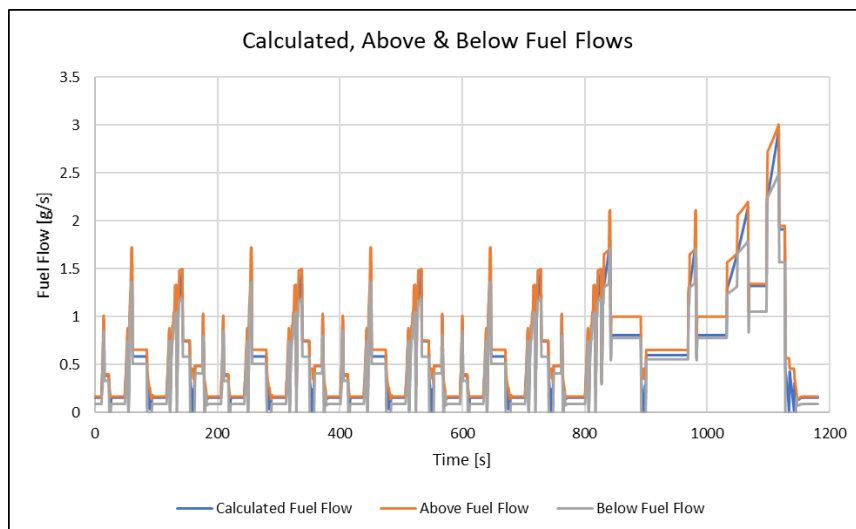


Figure 20 Calculated, Above and Below Fuel flows across the NEDC Excel [VW Passat 2.0L TFSI]

5.5. CO₂ Prediction Based on Fuel Consumption.

One of the parameters for evaluating the accuracy of the modelling is the value of carbon dioxide produced in g/Km, because it is comparable to genuine stated values from the OEM's of what their vehicles achieved during an official homologation test.

The prediction for CO₂ produced was done under the assumption of complete combustion and that the vehicle is operating at a point of Lambda 1. Lambda is a ratio of the actual air-fuel in comparison to that of its known fuel's air-fuel stoichiometric ratio, denoted with ' λ '. When at a value of 1, the actual air to fuel ratio is stoichiometric, which for gasoline is commonly known to be 14.7:1 air to fuel respectively, rich mixtures are $\lambda < 1$ & lean $\lambda > 1$ (Merker, Schwarz, Stiesch, & Otto, 2006).

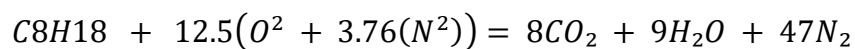
This ratio can also be referred to as the "equivalency ratio" denoted as ' ϕ ', where $\phi > 1$ is representing a "rich" mixture and $\phi < 1$ is a "lean" mixture, meaning more fuel to air mass than stoichiometric and less fuel to air mass than stoichiometric respectively. However, with the equivalency ratio, it's a "normalised quantity that provides information regarding the content of the combustion mixture" (McAllister, Chen, & Fernandez-Pello, 2011). The relationship between Lambda and equivalency ratio is:

$$\lambda = \frac{1}{\phi}$$

(McAllister, Chen, & Fernandez-Pello, 2011)

Using the principle that the ratio will remain stoichiometric and assuming complete combustion within the cylinder, the exhaust products can be equated to know the volume of water [H₂O], carbon dioxide [CO₂] and nitrogen [N] produced. Because of the variation within the chemical composition of certain fuels, primarily gasoline, joint with a lack of information of the specific chemicals used with the test fuel, the equation developed for exhaust products was done based on combustion of Iso-Octane. This fuel is the closest available chemically to common gasoline, shown as C₈H₁₈. Meaning a chain of eight carbon atoms attached with eighteen hydrogen atoms and this C_xH_x is the standard representation of a hydrocarbon fuel, with gasoline carbon said to be between C₄ – C₁₂, as opposed to diesels C₁₂ – C₂₀ (IEA-AMF, 2020).

For the combustion of 1Kmol of Iso-Octane and an AFR of 15.03:1, the balanced equation can be written as:



And given that the mass of fuel is 114.23 Kg/Kmol, mass of CO₂ produced would be 352.07 Kg/Kmol. For each Kg of fuel burnt, there will be 3.0822 Kg of CO₂ produced. With the masses being in both Kg and Kmol, the value will remain the same when

written as g/mol, so this multiplication factor can be used against the fuel flow in g/s, to present the CO₂ across the cycle. Finally, the mass of carbon dioxide per kilometre covered in order to validate against the stated values.

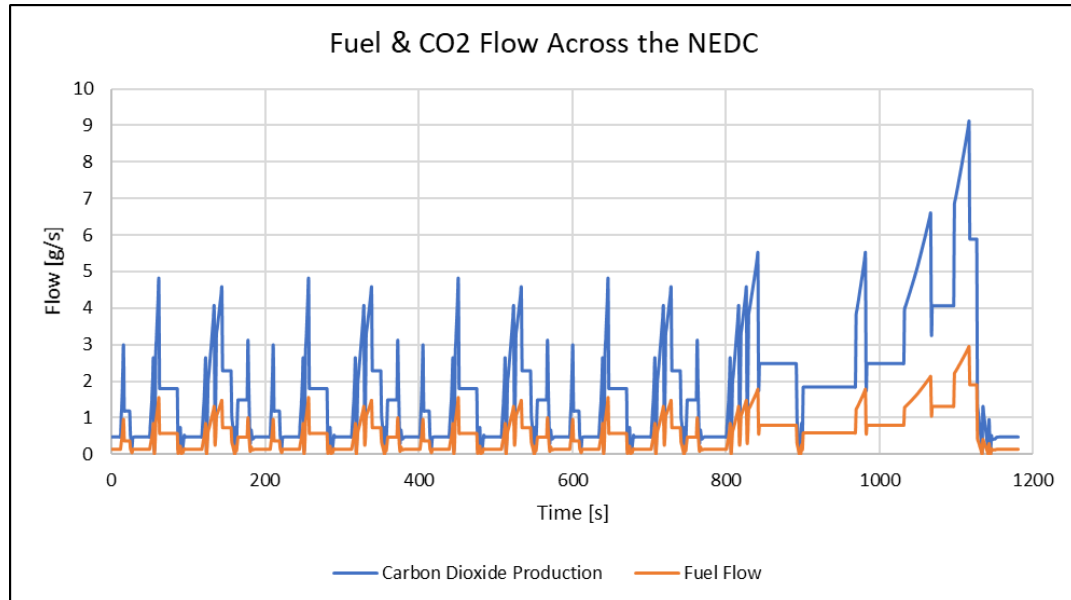


Figure 21 Fuel Flow and CO₂ Production across the NEDC [VW Passat 2.0L TFSI].

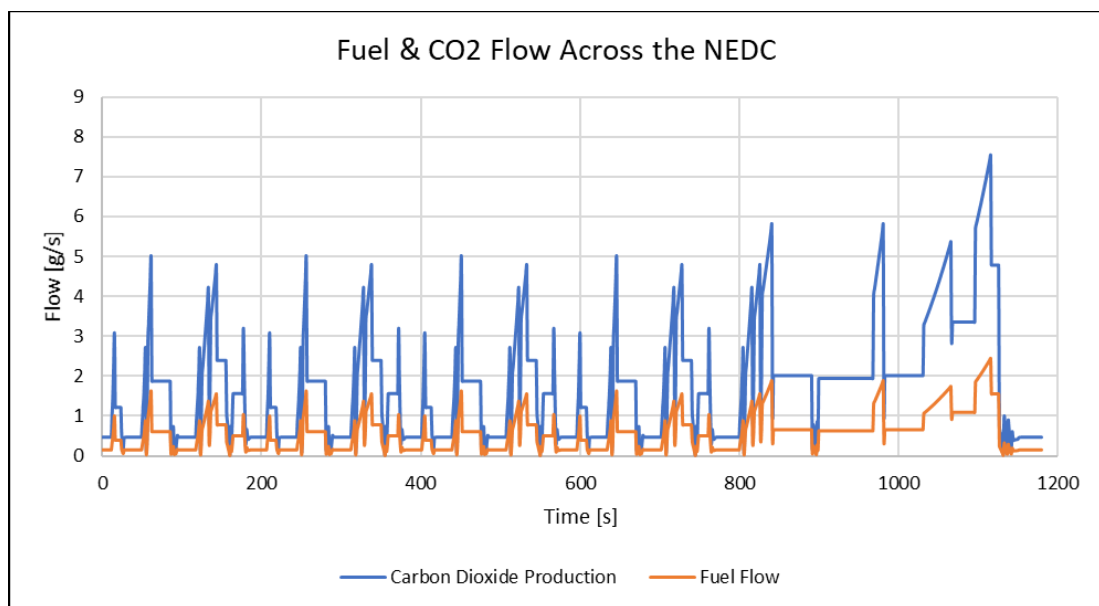


Figure 22 Fuel Flow and CO₂ Production across the NEDC [Ford Mondeo 1.6L EcoBoost].

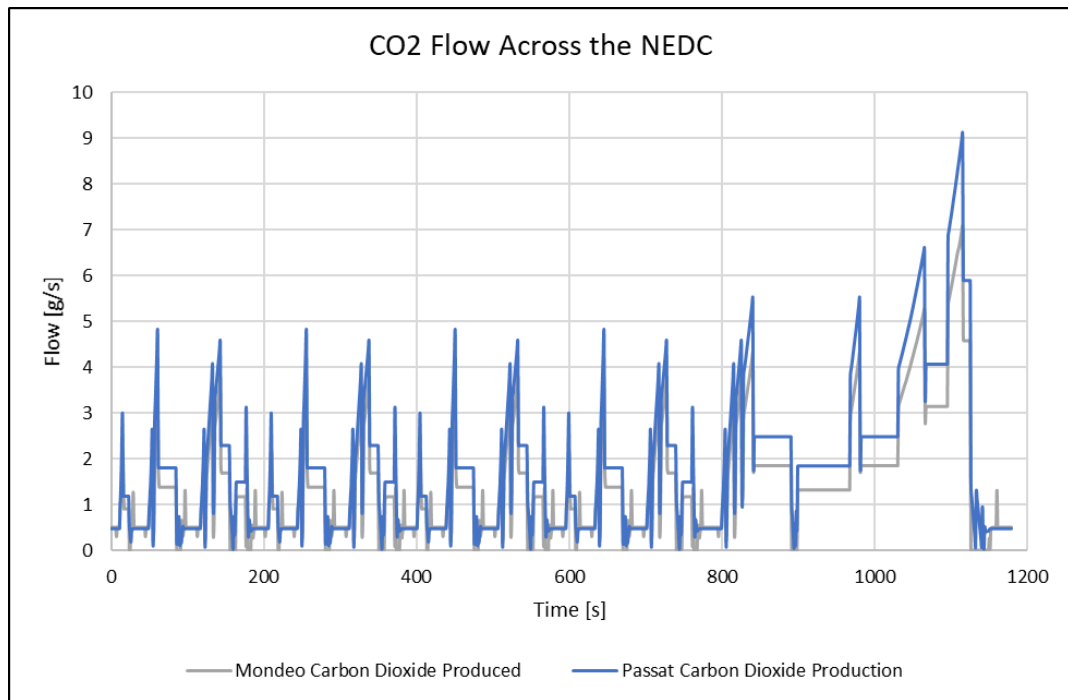


Figure 23 VW Passat & Ford Mondeo carbon dioxide production across the NEDC.

5.6. Miles Per Gallon results across the NEDC.

The final measure of accuracy for these models is matching the stated MPG achieved across the cycle. This is done using a few simple relationships regarding density, mass, and volume, shown within the following equations.

The fuel usage is stated in L/100Km for both urban and extra urban phases and then shown as a combined value across the full cycle. The urban phase concerns the first 780s and then the final 400s form the extra urban phase, where 4.07Km and 6.95Km are covered respectively for a total of 11.03Km.

Summing the total fuel flow across these phases to obtain a mass in grams, to then be divided by 1000 for the mass in Kg. Using the density of this fuel to be approximately 750Kg/m^3 (aqua-calc, 2020), although it's known that “pump” Gasoline can be within a region of $710 - 780\text{ Kg/m}^3$ (KEM, 2002) meaning there is going to be room for discrepancy within the calculation. Both vehicles are Gasoline engines therefore the maths and fuel density is the same.

The volume of the fuel used is found with:

$$Volume [m^3] = \frac{Fuel\ Mass [Kg]}{Density [\frac{Kg}{m^3}]}$$

Using the relationship between m³ and Litres:

$$Litres = m^3 \cdot 1000$$

Lastly with volume, the equivalency ratio of a Litre to a UK Gallon as:

$$1\ Gallon = 4.546\ Litres$$

Simply dividing the Litres of fuel usage by this value to equate the number of gallons used across each phase.

To obtain the miles covered across each of the phases:

$$1\ mile = 1.609\ Kilometres$$

With all variables now found the miles per gallon can be found. The values calculated for both vehicles and their official stated values can be seen within the table below, identifying a high level of accuracy on each model.

5.6.1. Volkswagen 2.0L Passat

The results in comparison to the stated values recorded and measured by MAHLE Powertrain:

	MPT Test 148	Calculated	% Difference
Urban (g)	341.5	342.02	100.2
Extra Urban (g)	374.4	388.06	103.6
	MPT Test 148		
Total Fuel Usage (g)	715.9	730.07	102.0
Total CO ₂ emissions	198	205.66	103.9
Total Accuracy %			102.4

Table 9 VW Passat Fuel and CO₂ values for predicted against MAHLE Powertrain.

And then the values against the supposed stated values from Volkswagen:

EU	Urban	
6.34	11.19	L/100Km
0.4409	0.4558	Stated Litres used
0.5174	0.4560	Calculated Litres used
7.44	11.20	L/100Km
117.34	100.05	% Difference

Table 10 Fuel usage accuracy of predicted against the stated usage.

Given the stated values are not directly from Volkswagen and that MAHLE Powertrain had measured the data themselves, this data is the preferred comparison. Meaning the accuracy of the model is with the limit of 97.6% accurate. This could be argued to be surprisingly high given the assumptions made, however the model stands to show that the mathematics and relationships used with the simulation are correct and work.

[Appendix G]

5.6.2. Ford Mondeo 1.6L EcoBoost

With the Ford, because the given test data full specifications were unknown, it is expected that the value's will be out regarding the accuracy, of which they were:

	Test	Calculated	%
Urban (g)	234.27	265.02	113.13
Extra Urban (g)	279.16	299.71	107.36
	Test		
Total Fuel Usage(g)	513.43	564.73	109.99
Total CO2 emissions (g/Km)	137.04	158.06	115.34
			111.45

Table 11 Ford measured against the calculated fuel usage and CO2 produced.

As can be seen, the calculated values are consistently over predicting in comparison to that of the measured. The reason for this is again believed to be because of the vehicle mass used for the test and not the modelling itself. This theory is backed up given the accuracy with the comparison between the modelled and officially stated values from Ford below:

	Stated	Calculated	%
Grams Urban Ford	270.87	265.02	97.84
Grams EUrban Ford	276.45	299.71	108.41
	Stated		
Total Fuel Usage(g)	547.32	564.73	103.18
Total CO2 emissions (g/Km)	155	158.06	101.97
			102.85

Table 12 Ford stated against the calculated fuel usage and CO2 produced (Ford Motor Company, 2012).

The accuracy between this model is far more than that of the comparison between the test data, again clarifying the theory of an incorrect vehicle mass used in the test. Along with the modelling mathematics being adequate to achieve an acceptable level of accuracy.

6. ‘NEDC’ Testing within MATLAB.

6.1. Recreation of the Vehicle Model Within MATLAB.

Within the MATLAB model, the Ford Mondeo was updated from the 2011 1600cc EcoBoost Petrol to a version of the 2019 Ford Mondeo 2.0L EcoBlue Diesel. The reason for this is simply because the fuel flow data was not in the correct form from the 1600cc EcoBoost. It was thought that it would also improve the investigation of powertrain and drivetrain efficiency differences and whilst allowing the detailing of differences between using a diesel & a petrol vehicle across the NEDC.

The modelling procedure conducted within Microsoft’s Excel software was carried over into MathWorks MATLAB coding software to enable the development of the gear ratio appropriation model via a series of “while loop” scripts. The principles of powertrain/driveline modelling & vehicle simulation were transferred, with the VW model compared to the excel model to ensure reproducibility. The MATLAB model returned a final total fuel use of 716.5g of fuel, while excel produced a value an initial value of 730.07g. The reasoning for this is firstly that MATLAB is a much more comprehensive software capable of more accurate interpolation/extrapolation curve fitting than excel, hence a definite discrepancy. Along with the modelling differences discussed within ‘Appendix G’ and the necessary change in ratios to accompany for the dual final drive system used within the VW Passat.

Tested Vehicles			
Ford Mondeo [2019]		VW Passat [2006]	
Fuel Type	Diesel	Fuel Type	Petroleum
Vehicle Mass	1752Kg	Vehicle Mass	1550 Kg
Engine Cap.	1996cc	Engine Cap.	1984cc
Max Power	140 kW @ 3500 rpm	Max Power	147 kW @ 5100 rpm
Max Torque	400 Nm @ 1750 - 2500 rpm	Max Torque	280 Nm @ 1800 - 5000 rpm
No. Gears	6	No. Gears	6
Transmission Type	Manual	Transmission Type	Manual
Vehicle Type	Estate	Vehicle Type	Estate
WLTP MPG Lowest	36.7 MPG	NEDC Combined MPG	34
WLTP MPG Lowest	56.6 MPG	Urban Fuel Usage	11.1 L/100Km
CO ₂ [NEDC Equivalent] 18" wheels	137 g/Km	Extra Urban Fuel Usage	6.6 L/100Km
Coe. Of Drag	0.29	CO ₂	197 g/Km
Frontal Area	2.706 m ²	Coe. Of Drag	0.29
Euro Emissions Band	Euro 6.2	Frontal Area	2.235 m ²
		Euro Emissions Band	Euro 4

Table 13 MATLAB NEDC modelling test vehicles (Ford Motor Company, 2020), (Ford Motor Company, 2020).

6.2. Gear Ratio Appropriation Modelling Script.

The gear ratio appropriation scripting process required many initial inputs that would characterise the vehicle being tested, i.e. the test mass, coefficients, speeds, etc. These standardised variables allowed for the easy manipulation and addition of the various other variables required to calculate the fuelling of the vehicle. [Appendix H, 1]

The principle used for the gear ratio appropriation testing was to run a series of loops that consisted of a full NEDC cycle and operation, where for each one done the gear ratio value would be changed in steps of 0.001. This approach was taken because of the fixed gear shift pattern of the European drive cycle, each ratio could be tested individually and would only ever identify the effect of the actual ratio on the torque requirement, and then subsequently the fuel requirement.

For the gear ratios themselves, the investigation was done on the effect of changing the ratio to 10% either side of the original, i.e. 3.024 – 3.36 – 3.696. This decision was made after research found that from two other similar studies conducted by Oglieve et al. and Eckert et al. Their relative approaches highlighted that further change in ratio will most likely reduce fuel consumption, but at cost of the performance of the vehicle. If there were to be a change of cycle or driving style i.e. the WLTP or RDE with uphill sections, the vehicle might not be capable of supplying the demanded power because of the poorly assigned gear ratio, requiring a downshift of gear that will most definitely result in unnecessary fuel usage.

6.1.1. Individual Gear ratio ‘While Loops’

The complete MATLAB script for each vehicle contains the initial assignment of vehicle characteristic variables, the original gear ratio test, ‘while loop’ variable assignments, the individually tested gear loops and finally the last run with the appropriated gear ratios.

Considering the loops for each gear, all used a simple exit criterion (i.e. ‘Post1’) and a counter (i.e. ‘Fi’) within, to allow the assignment of the following keys values: fuel usage of current test gear, average BSFC while in test gear, total CO₂ production and the gear ratio used within the test.

The initial conditions for each loop is the assignment of the gear ratio to be 10% below the original and the exit criterion set to 1. The basic construction of each gears testing loop is shown below.

```

while Post1 == 1

    Loop_Script

    FirstFG(Fi,1) = First; FirstFG(Fi,2) = FirstGearFuel1;
    FirstFG(Fi,3) = CarbonDioxideProducedFirst; FirstFG(Fi,4) = FBSFC;

    if First >= (FirstGearRatio + (FirstGearRatio*GMul))

        Post1 = 0;

    elseif First < (FirstGearRatio + (FirstGearRatio*GMul))

        First = First + 0.001;

    end

    Fi = Fi + 1;

end

```

Figure 24 Individual gear ratio testing 'while loop' principle construction.

The “Post1” exit criterion is to ensure that the test will continue through the selected difference in gear ratio, which is controlled by the internal ‘if’ statement that is continuously assessing the current ratio used for the last test. If the ratio reaches the point where it is above 10% than the original gear ratio, the script is stopped because the “Post1” variable is reassigned to 0. The variable seen titled “GMul” is simply an external variable used to potentially change the percentage of gear ratio change, if this were to be required (i.e. 10% - 15%).

6.1.1.1. Testing “Loop_Scripts”

Within the ‘while’ loops a separate script is ran, as seen in the above figure titled ‘Loop_Script’. This script contains all the additional calculations that result from a change in gear ratio, i.e. anything forward of the differential into the engine. The script calculates the transmission torque, speed and acceleration, totalled up with the additional auxiliary torque, new engine speed and eventually the fuel flow required for the new torque demand. This script contains the full test sequence and retains the original gear ratios for the other gears not being tested for this outer ‘while loop’ and for every iteration of the counter (‘Fi’) will calculate the fuel usage across the entire NEDC with the changed gear ratio. The basis of this script is used for the initial run, test runs and final run within the investigation and a full copy of the test script used for specifically first gear can be seen in Appendix H, 2.

6.1.1.2. Gear specific variables table assignment script.

Seen in the figure above, individual gear variable tables were created in order to assess the best ratio found for each gear based on the least fuel usage. These tables were assigned the names of the gear and then “FG”, i.e. ‘FirstFG’. This is where the counter was then used to allow the addition of a new row within the table for the next ratio used for testing, up until the 10% above was met.

The current ratio was assigned in column one, the fuel usage of the vehicle whilst in the currently tested gear in column two, total carbon dioxide produced within column three and finally the average brake specific fuel consumption while in the tested gear was column four.

All the individual gear specific values were found using the principle of the following line of code.

```
if TableSetup(gf,5) == 1 && TableSetup(gf,3) > 0  
    fuelFirstGear(gf,1) = FirstGearTable(gf,8);
```

Written within a counter based ‘for loop’ that counted through until the row value met the length of the test cycle, 1181. If it happens to be in that specific gear then it is assigned the value, else it’ll move onto the next gear or remain in the idle section [Appendix H, 3]. The “TableSetup” is the original table that contains the column vector of the set gear shift pattern (column five) and vehicle speed (column three).

6.1.2. Appropriated final gear ratio assignment.

On completion of the six individual gear ratio test loops and given the objective of the investigation is reduced fuel usage, the final ratio was picked based upon the least fuel used while in said gear. This is why it was so important to record the values within the independent gear tables, because then the results could be used for post processing to find an optimum set of gear ratios from the test conducted.

Using the ‘min()’ and ‘find()’ function within MATLAB, each gear’s fuel usage column was checked for the minimum fuel usage throughout all ratios tested. And with the line written as:

```
fgm = find(min(FirstFG(:,2)));  
FinalFirstGear = FirstFG(fgm,1);
```

The row of the minimum fuel usage was assigned to the variable “fgm”, to be used to assign the final ratio of the same row from the tested gears table. [Appendix H, 4]

6.3. Results of fuel usage from original to appropriated gear ratios.

Upon the final run for both vehicles, it was found that the gear ratios had shown a definite reduction of fuel usage, brake specific fuel consumption and most importantly, the mass of carbon dioxide produced.

6.3.1. Volkswagen Passat 2.0L TFSI 2006

The difference between the original gear ratio and appropriated for the VW are characterised in the following table.

	Appropriated Gear	Original Gear	Difference %
First Gear	3.0213	3.357	10
Second Gear	1.8783	2.087	10
Third Gear	1.3221	1.469	10
Fourth Gear	0.9882	1.098	10
Fifth Gear	0.783	0.87	10
Sixth Gear	0.657	0.73	10
Fuel [g]	647.06	716.49	9.6912
CO2 [g/km]	180.81	200.22	9.6912
Average BSFC [g/Kw h]	657.72	687.74	4.3647
MPG	39.975	36.101	9.6912
Mechanical Work Done [MJ]	5.8213	5.8213	0
Total Fuel Energy [MJ]	28.47	31.526	9.6912
Average Thermal Efficiency	20.447	18.465	9.6912

Table 14 VW Passat original and finalised gear ratio comparisons table.

As seen in the table, the found ratio for the least fuel used was 10% below the original gear ratio selected for the vehicle. This is because the engine speed is reduced and the required torque of the vehicle is increased, leading to a more effective positioning upon the BSFC one the engine. The fuel reduction for each individual gear can be seen in the figure below.

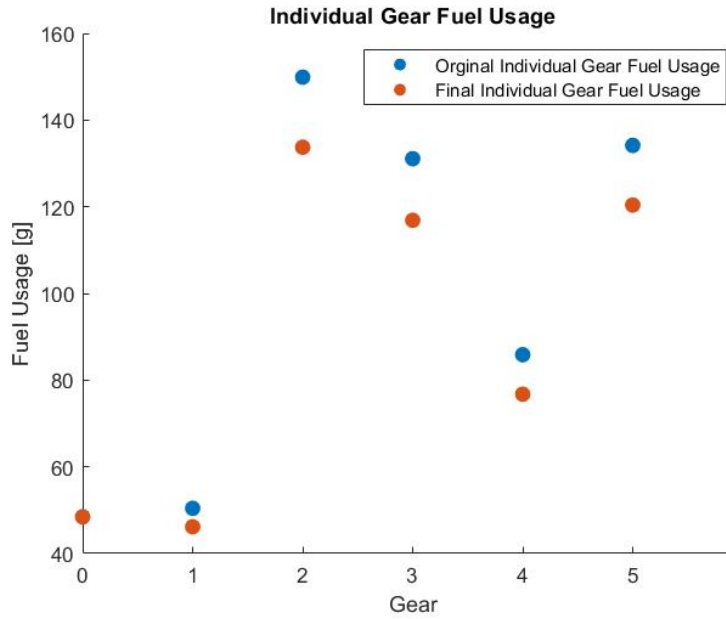


Figure 25 VW Passat individual gears fuel savings in comparison to the original gear fuel usage.

The largest saving in fuel can be seen for second gear, simply because the cycle spends the most time in this gear specifically. But it's clear that because of the set gear shift regulation of the NEDC, reducing the fuel consumption is relatively simple to do.

Regarding the change of torque effecting the positioning upon the BSFC map for this vehicle's engine, the following figures will show the torque required across the cycle, the average BSFC for each specific gear and then the brake specific fuel map itself.

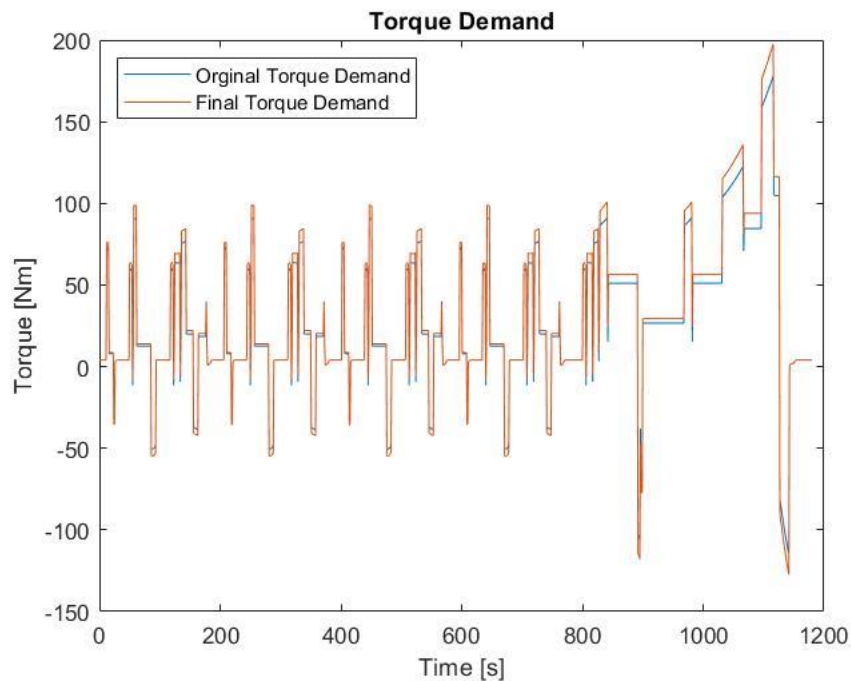


Figure 26 VW Passat torque demand across the NEDC.

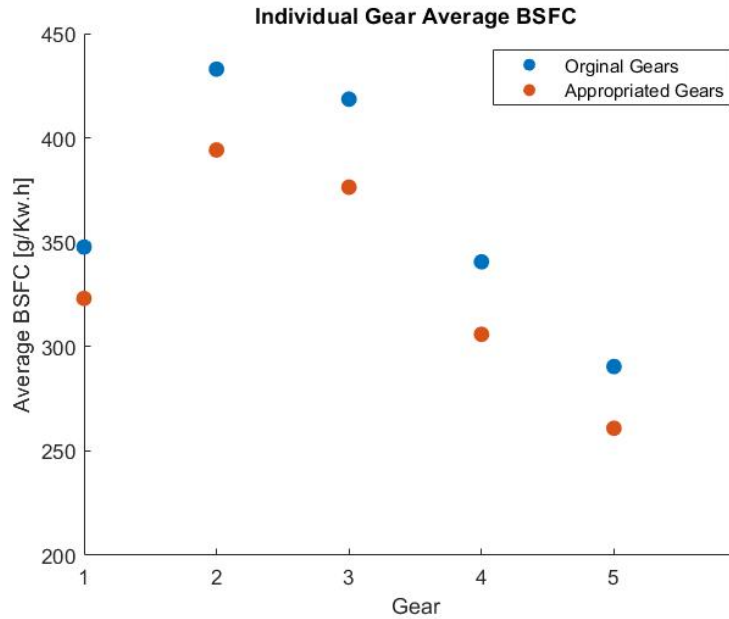


Figure 27 VW Passat individual gears average BSFC comparison.

An important note here is the difference in positioning for 2nd and 3rd compared to the higher gears. These sections are showing the higher inefficiencies for operating within the requirement of this test cycle, of which these gears are used essentially only within the urban section. This lower speed and lack of demand makes it difficult to ensure the engine is operating in a fuel-efficient position, which is one argument for when the NEDC was in use, the vehicles could've perhaps made an effective use of a hybrid system that had capability to reduce the fuel usage by a considerable amount more. Using primarily electrical energy for a large period across the urban phase, that makes up 2/3 of the entire cycle, would dramatically drop the emissions produced from the vehicle. However, this argument is again only made based on the theory of “cycle beating” and may not be the best option for ‘on-road’ vehicle performance.

To again highlight why this fuel saving and increase of operating efficiency is possible, when looking at the brake specific fuel map for this engine, with engine speed and torque as the axis. It's obvious why a higher torque will result in a more efficient use of the fuel energy for the lower engine speeds primarily. However, it should be noted that although the powertrain is operating within a more effective position, the downfall is the fact that it does require more fuel to maintain this position because of the required torque, but with the test being short and consisting of multiple idle points, the vehicle doesn't have to operate here for overly prolonged periods.

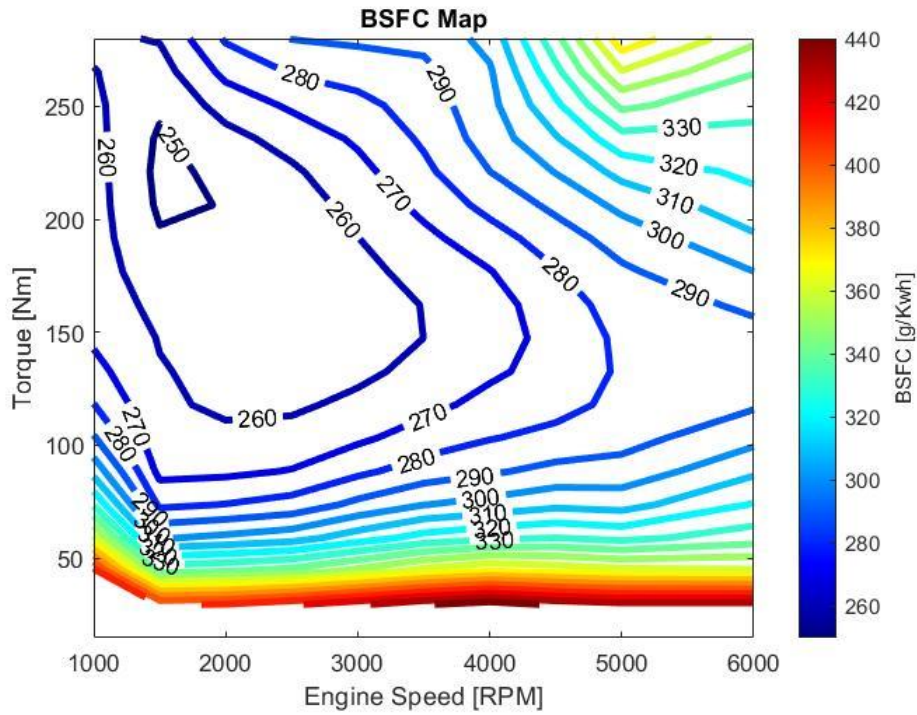


Figure 28 Contour BSFC Map of the VW 2.0L TFSI 2006 Powertrain.

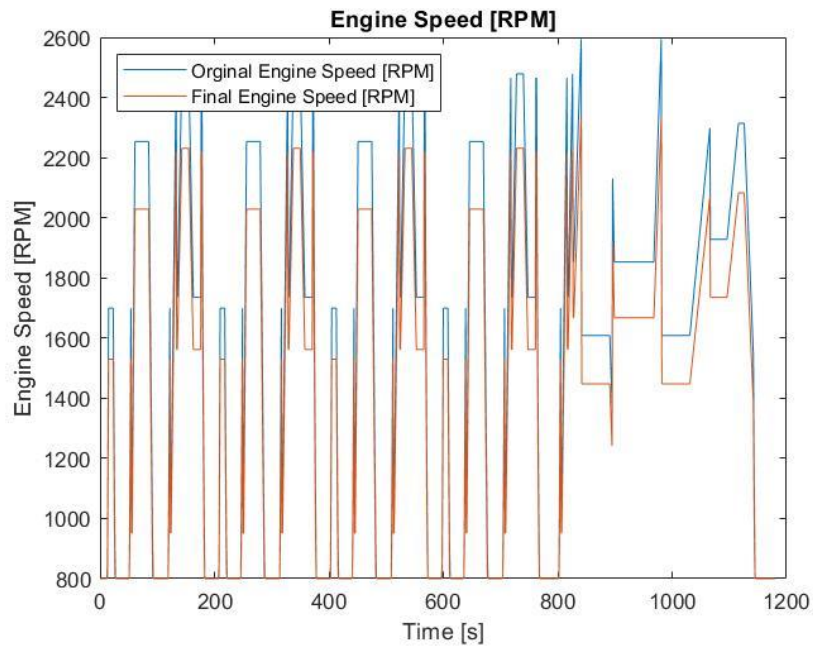
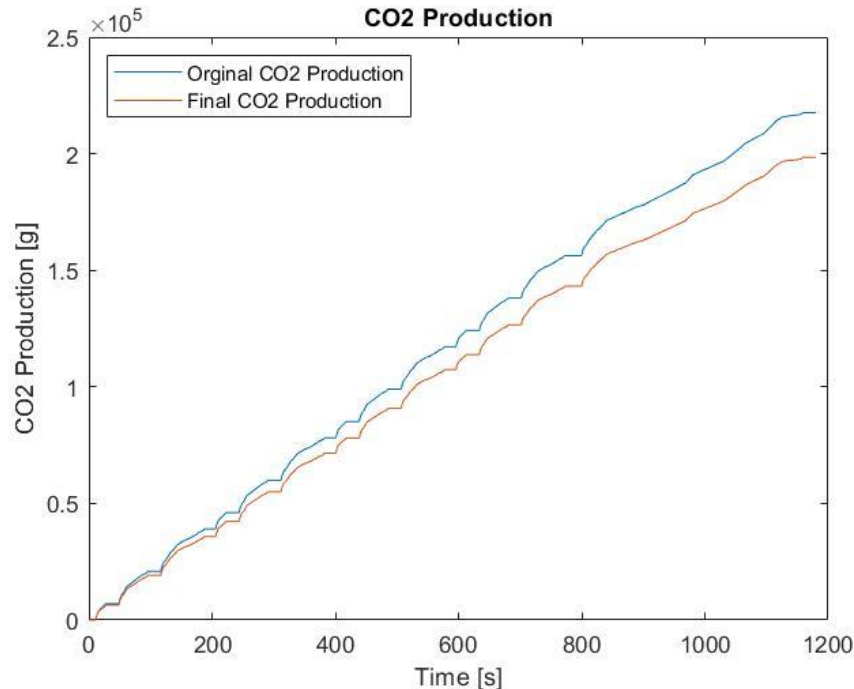


Figure 29 VW Passat difference in engine speeds due to gear ratio change across the NEDC.

As highlighted in the above, with the engine speed reduced and torque demand increased, the vehicle will remain within a much more effective positioning considering its physical parameters.

Finally and most importantly is the reduction in carbon dioxide produced across the cycle, Although this was calculated on a theoretical basis, it's been shown to be mathematically sound in the above work when used as a validation method against the measured values from MAHLE.



Throughout the cycle the appropriated gear ratio's show a clear reduction of CO₂ produced and the rate or reduction seems to remain consistent across the cycle. Therefore, it can be said that the new gear ratios would be the better option for the vehicle from a purely emissions and fuel reduction principle. It should also be noted that given the tax bracket of the time this vehicle would've been registered; the reduction of CO₂ would have led to a drop of band from J to I, reducing it from 200g/km (stated 198g/km) to 180.81g/km.

[Appendix I, 1]

6.3.2. Ford Mondeo 2.0L EcoBlue 2019

The age of the Ford Mondeo 2.0L EcoBlue Diesel vehicle, made it difficult to ensure validation because the stated values of fuel usage were against the world harmonised light vehicle testing procedure. With the best and worst found MPG and not a specific fuel volume in litres, alike all the previous stated test cycle values.

Also, because of the current legislation across Europe, Ford could state the CO₂ usage across the NEDC instead of the WLTP, this regulation is removed on the 1st of January 2021. But because Ford could state the carbon dioxide production across the NEDC it gave some idea of a validity target to match to ensure correct modelling.

The next issue within this model is that the data collected for this specific powertrain is from 2013, with the engine being released in the 2019, or at least in the Mondeo. Therefore, the engine was still in the earlier development stages with many values of the test either not correctly calibrated/corrected, but also not a full representation of the finalised stated values for the vehicle.

For example, the following were being tested or hadn't been mapped into the engines electronic control module.

- Reduced back leak injector
- Active control of lift pump
- Full VDOP
- Reduced speed water pump
- Steel Piston (as package)
- Roller Bearing Camshaft
- GTD14 Turbo vs DW10F 180PS BorgWarner

The steel pistons were being tested as a check to see if there could be any benefit for the combustion process in comparison to their other design. Also, the turbocharger was being tested and compared between a Garrett turbo and BorgWarner's to see what would be the most efficient for the engine, and it is believed that the fuel map could've been varied to accompany one turbocharger over the other and it is unknown which one was specifically used in the test. The Mondeo also incorporates "stop/start" technology that had to be introduced into model and was done so using the idea that when at standstill for at least three seconds, the engine would turn off. Of course, the only difficult is then trying to account for extra fuelling and battery charge after the vehicle is started again, causing discrepancies.

The results of the appropriation test can be seen within the following table.

	Appropriated Gear	Original Gear	Difference %
First Gear	3.2247	3.583	10
Second Gear	1.7568	1.952	10
Third Gear	1.0746	1.194	10
Fourth Gear	0.7461	0.829	10
Fifth Gear	0.8487	0.943	10
Sixth Gear	0.6804	0.756	10
Fuel [g]	620.17	652.36	4.9346
CO2 [g/km]	162.02	182.06	11.012
Average BSFC [g/Kw h]	191.72	200.86	4.5466
MPG	42.713	40.605	5.1907
Average MPG	49.644	45.617	8.8284
L/100Km	6.6147	6.9581	4.9346
Mechanical Work Done [MJ]	6.0446	6.0446	0
Total Fuel Energy [MJ]	27.287	28.704	4.9346
Average Thermal Efficiency	22.152	21.059	5.1907

Table 15 Ford Panther 2.0L EcoBlue NEDC results comparison.

Again, with the Ford, it's shown that the 10% below the original gear ratio was the most appropriate for reducing the fuel usage across the new European drive cycle. The CO₂ values were found using the flow data collected by Ford, but as stated this data clearly isn't reflective of their finalised powertrain, because the officially stated carbon dioxide production is 137g/km. There is of course discrepancies with that itself because the data provided was for the 180ps 400Nm engine, and the models used within the Mondeo and the 150ps & 190ps. The 150ps is equipped with the manual transmission and the 190ps has Ford's 8spd automatic transmission, meaning there would also be an issue with direct correlation.

When looking at the MPG values across this cycle however, the average values are roughly at the mid-point of Ford's stated values. Granted these stated are across the WLTP, but none the less it shows that the vehicle dynamic, fuelling and equivalent flow modelling is working and provides valid data, respective of the test.

The main note for this test is the lower percentage saving in comparison to the Volkswagen, coming in at 4.94% instead of 9.7%. This is thought to be primarily because the advanced technology on the Ford EcoBlue engine, but also the fact it's a diesel engine and these are known to work at a far higher thermal efficiency than that of its petrol cousin. One reason for this increase is the reduced "pumping losses" with a diesel and its ability to operate at much leaner mixtures than most gasoline engines. Although it should be noted that gasoline engines are rapidly progressing with direct injection, forced induction, variable compression and enhanced combustion processes.

Another reason for the reduced fuel saving is the vehicles already highly efficient operating point, shown with the average BSFC value being very low at 200.9g/Kw·h and it only being reduced to 191.7g/Kw·h.

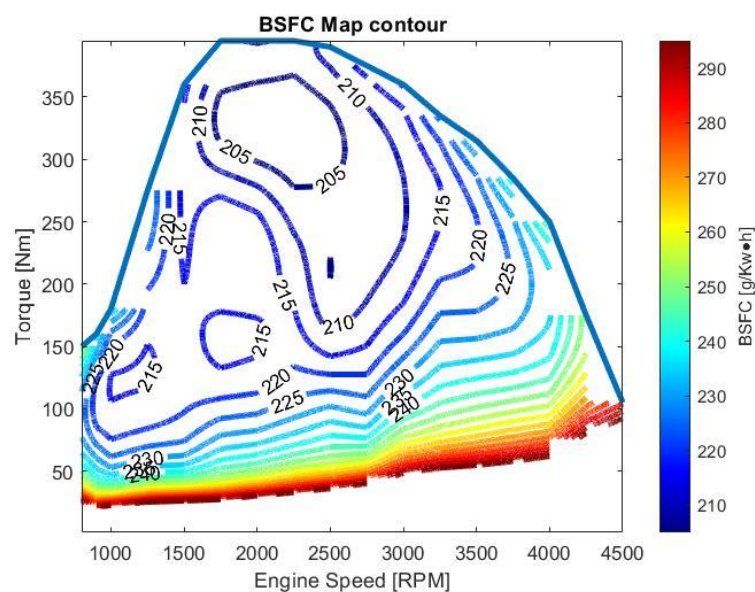


Figure 30 Ford Panther 2.0L EcoBlue Diesel engine BSFC map.

When considering the BSFC map, it's clear the engine is very well developed and designed for low speed, with high or low torque driving. With its torque band being very high from low engine speed, joint with the fact that even for the lower torque sweep the brake specific fuel consumption still doesn't reach a point that could be consider awful. It's obvious that Ford have manufactured this engine somewhat exclusively for defeating the testing procedures, instead of a purely power performance platform.

Again, the carbon dioxide production is the most important reduction that comes from reduced fuel, better combustion and further aftertreatment technology, of which the Mondeo certainly has. The vehicle is fitted with an SCR catalyst with Urea injection to reduce NO_x production, a close-coupled oxidation catalyst with a particulate filter and finally the exhaust gas recirculation channel integrated within the cylinder head (Ford Motor Company, 2016).

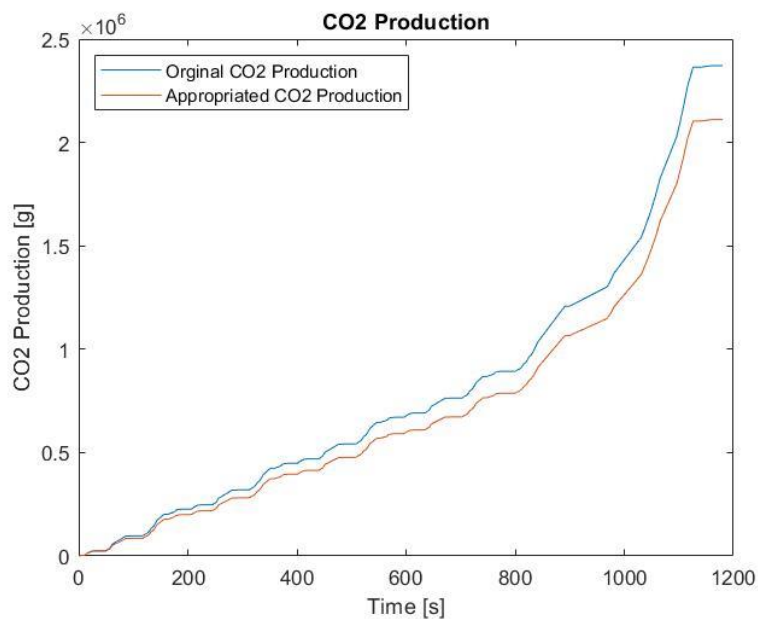


Figure 31 Ford Mondeo 2.0L EcoBlue CO2 Production across the NEDC cycle.

The carbon dioxide flow was calculated in the same manner as the fuel flow because the data provided the flow in g/s for the specific engine speed and torque. So, using the simple principle of above and below values, as previously spoke about, the CO₂ is a direct from the data Ford measured at the time. And for the appropriated gear ratios, the reduction is significant at 11%, however considering the actual stated value, this saving may not be the same if the model was completely correlated.

To highlight the reduction caused by each individual gear ratio change, the following figures show to reduced fuel use and BSFC independently. [Appendix I, 2]

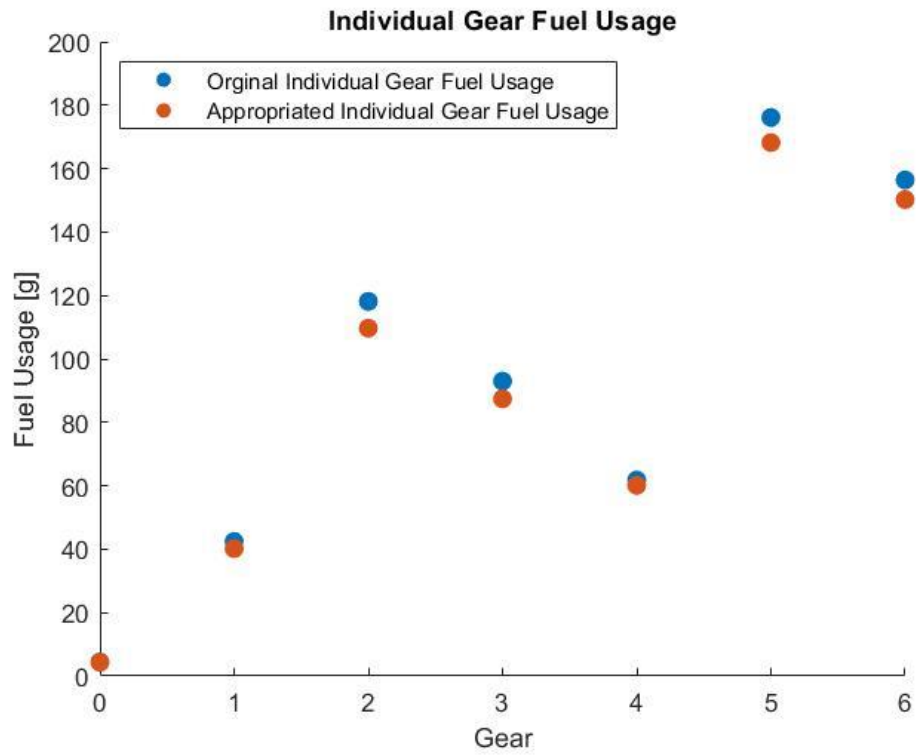


Figure 32 Ford Mondeo 2019 2.0L EcoBlue individual gear fuel usage reduction.

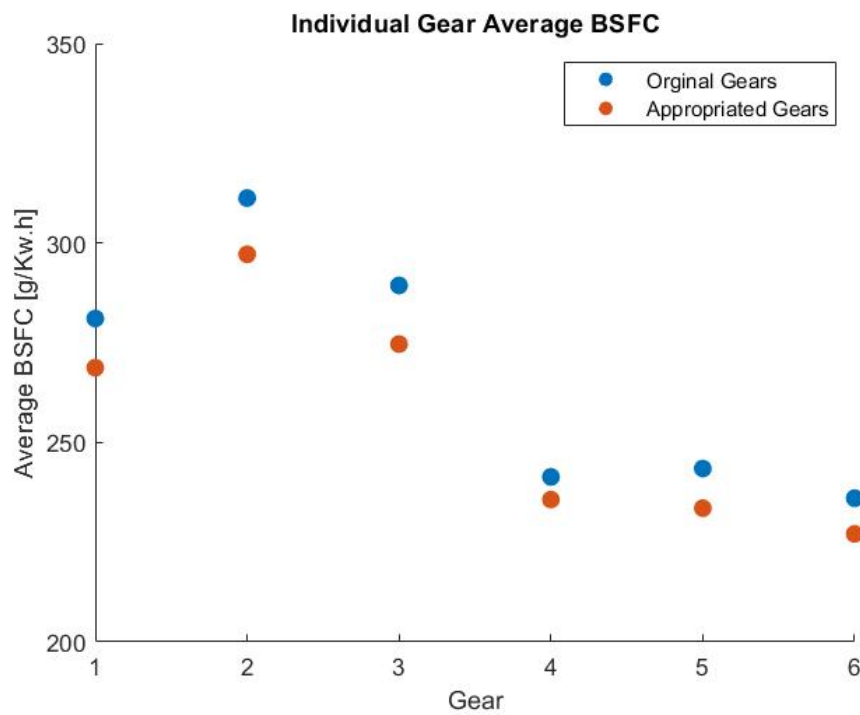


Figure 33 Ford Mondeo 2019 2.0L EcoBlue individual gear reduction in BSFC.

6.4. Change in Vehicle Acceleration Model.

Given the clear reduction possible from adjusting the gear ratios, it's difficult to understand why Volkswagen wouldn't have changed the gears from factory. For Ford however, their vehicle was developed for the WLTP, of which has a vastly different set of regulations, specifically the gear shift no longer being set. Instead it's dependent upon the maximum power, idle speed and many other regulations that halt the full change in ratios because it could possibly result in an illegal gear shifting pattern.

Therefore, to understand why perhaps VW didn't change their ratios a small investigation into the impact upon the vehicle's acceleration performance would've been affected. The following model was created within excel to try and compare the potential change in 0-62mph time and see if it was possible to characterise the effect on the vehicle's driveability. VW state their 2006 Passat 2.0L TFSI Estate it capable of reaching 62mph or 100km/h within 7.7s. This model was recreated within excel with a considerable amount of accuracy and given the difficulty to reproduce an acceleration model, because it is such a dynamic situation, this model isn't assumed to be the best representation and it's thought that a more inclusive model should be made. This model would then include; vehicle/body roll, potential traction losses, clutch slip and engine acceleration modelling.

However, the acceleration was predicted to be at 7.71s for the original gear ratios and the appropriated ratios giving a time of 7.78s. This is a very small change in time and is thought to be easily sacrificed for nearly a 10% saving of fuel, CO₂ produced and a reduced tax bracket.

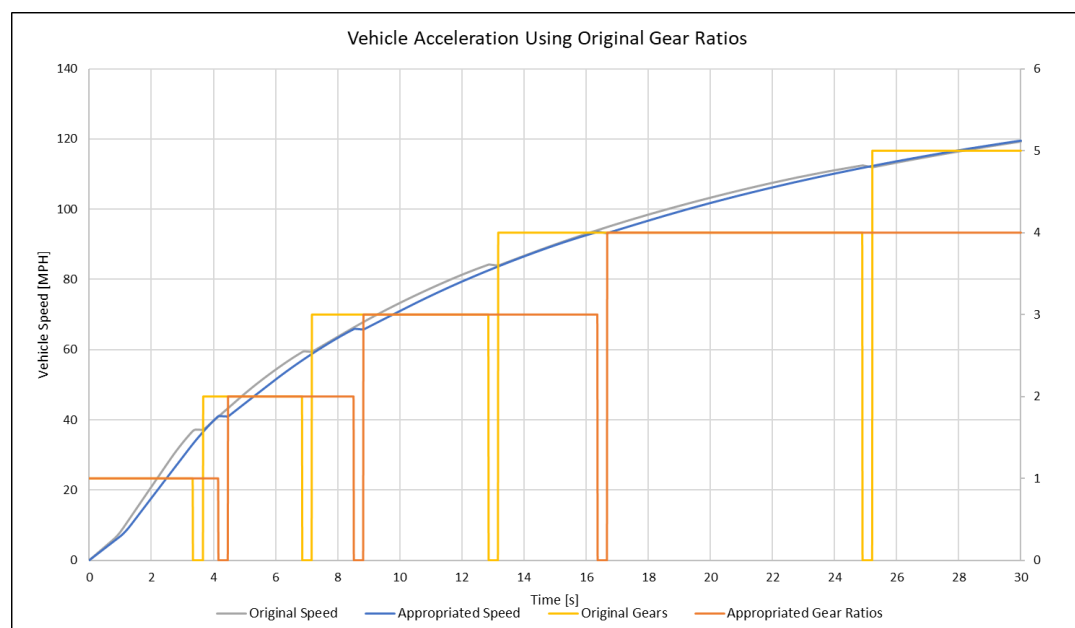


Figure 34 VW Passat vehicle acceleration model comparison with gear position.

Considering the vehicles driveability changes, it can be seen the gear is having to be held for longer than previously, which may make a dramatic difference to the higher speed vehicle acceleration feel. This is highlighted by the fact that within the same time period, with the new ratios it doesn't make it to the engine speed causing an upshift. This speed was set at 6500RPM because of the engines power curve showing this is the optimum point before the frictional losses in the engine become too high causing a rapid decrease in available torque and subsequently power.

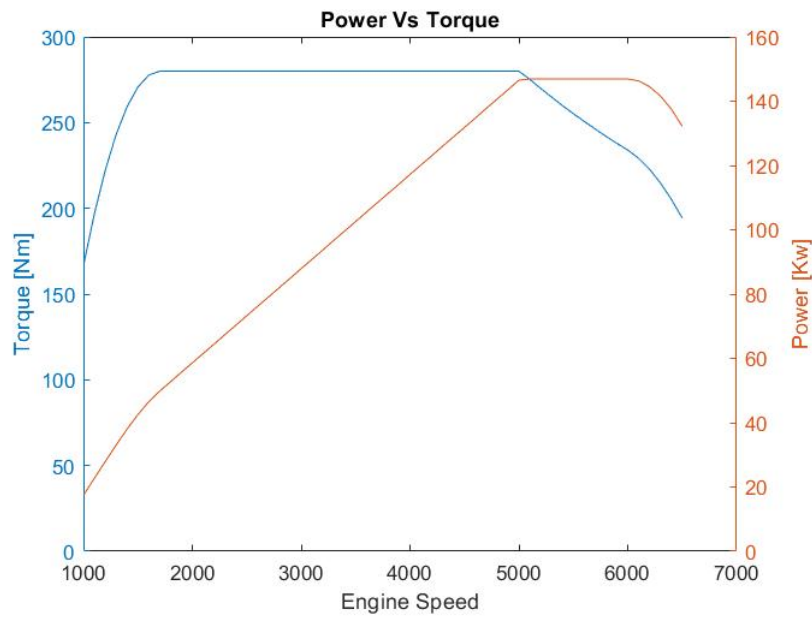


Figure 35 VW Passat 2.0L TFSI Power Vs Torque curve.

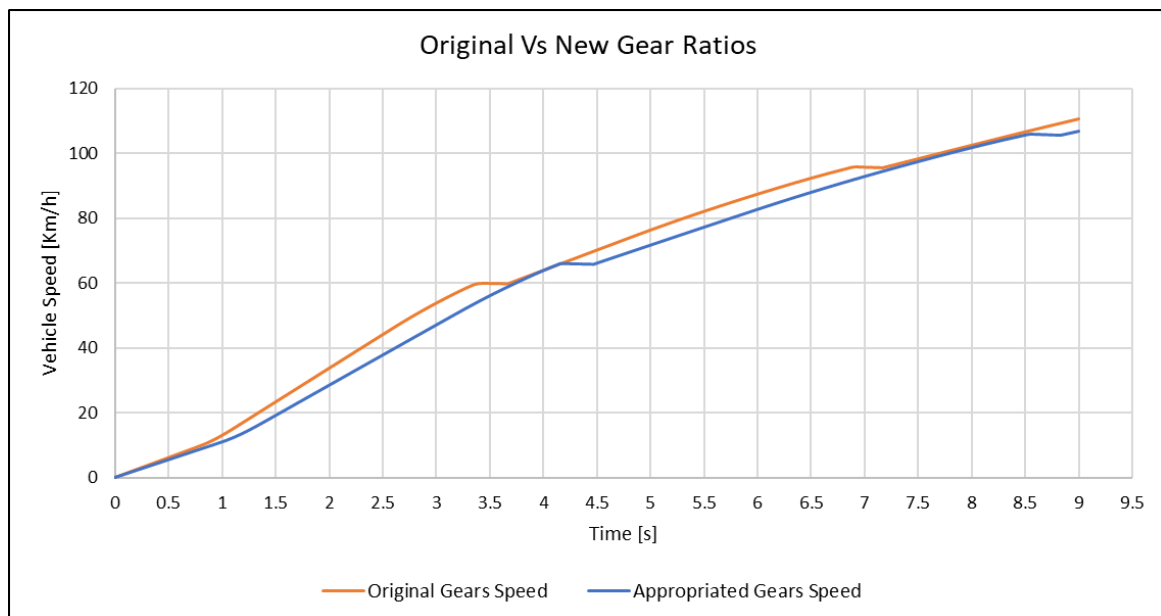


Figure 36 VW Passat 2.0L TFSI 2006 Difference in vehicle acceleration due to the change of gear ratios.

7. ‘WLTP’ Testing within MATLAB.

The testing across the WLTP was done using the same two vehicles as the NEDC model, the VW Passat 2.0L TFSI 2006 Estate and the Ford Mondeo 2.0L EcoBlue 2019 Estate. This importance between these vehicles is the manufacture years, with the VW being designed and created to defeat the NEDC and then the much newer Ford having been designed for the WLTP instead, while neither were created for both cycles. The unfortunate downside to this is the fact that the VW model is therefore unable to be validated against any measured data and can only be assumed correct. Given that this model is using the same principles as previously shown, and they were shown to be accurate, it's thought that the values found for fuel usage across this test cycle would be an accurate representation of what would be measured if it were put through a dynamometer test. As for the Ford model, the validation comes in correctly calculating the MPG achieved during the cycle and comparing it to Ford's stated best and worst achieved. This value would then be used to validate the dynamic simulation; however, the gear shift pattern is the other technical area to cause discrepancies within results. This is because although there is strict regulations for the procedure, the gear shifting pattern could show a respectable difference between those used in this study compared to Ford's testing pattern.

The WLTP complete model uses the same dynamics and calculation methods as previously stated, simply using the new modified vehicle speed provided within the regulation paper published by UNECE; Addendum 15: Global technical regulation No. 15.

[[Appendix J](#)]

7.1. Initial Gear Shift Modelling & Regulations.

The only additional modelling within this test procedure is the gear shift pattern because it is no longer set and is governed by vehicle, powertrain and drivetrain characteristics to achieve a more accurate representation of the vehicles themselves and not the test procedure, which was an issue of the NEDC.

The full technical regulations can be found within the papers following the links within ‘Appendix J’, however the most important points are:

- The minimum engine speed that will govern when the vehicle must downshift from gears above 2nd is calculated using the following:

$$n_{min_drive} = n_{idle} + 0,125 \times (n_{rated} - n_{idle})$$

Where *n_{idle}* is the engines idle speed and for the both the VW and the Ford were said to 800RPM. And then *n_{rated}* is the speed at which the maximum power from the engine is made.

- If a lower gear is required for a short period of time and the gear previous isn’t held for more than six seconds, then the previous gears shall be changed to $n_{gear} - 1$. i.e. if the pattern were 2,3,3,3,2,2,3 then it shall become 2,2,2,2,2,2,3.
- During acceleration phase, considered to be a period of 3 seconds with a vehicle speed > 1 km/h and a monotonic increase in vehicle speed, no gears shall be skipped, and they are to be held for at least 2 seconds. i.e. if the pattern were 1,1,2,3,3,3 it is to be changed to 1,1,2,2,3,3.
- During deceleration phases, considered as a period of 3 seconds with a vehicle speed > 1 km/h and a monotonic decrease in vehicle speed, only $n_{gears} \geq 2$ are to be used. Cannot downshift into 1st gear.
- During deceleration gears can be skipped if desired. During deceleration phases, if a gear is to only be held for 2 seconds, it shall be replaced by ‘0’ and then the following gears after. i.e. 4,3,3,2,2 becomes 4,0,2,2,2.
- If a gear shift pattern lasts 1 – 5 seconds where the gear prior to the pattern is lower and the gear after the pattern is the same or lower than the prior the pattern is to be adjusted. i.e. i-1,i,i,i,i,i-1 becomes i-1,i-1,i-1,i-1,i-1,i-1,i-1.

Considering these regulations, a basic pattern was constructed for the vehicles using an idealised gear shift vehicle speed that could then be adjusted to suit. For the Volkswagen, the engine speed was done on the premise of using the minimum vehicle speeds available due to the regulations on 1st gear to 2nd, 2nd minimum engine speed and then the minimum speed that the gears above 2nd could use. With this a simple ‘for loop’ was used to assign a gear value if the speeds went out of the targets initial created.

```
IGears = zeros(length(LayoutTable),1);
for Fi = 1:length(LayoutTable)
    if LayoutTable(Fi,4) == 0
        IGears(Fi,1) = 0;
    elseif LayoutTable(Fi,4) > 0 && LayoutTable(Fi,4) < Firstspeed*1.4
        IGears(Fi,1) = 1;

    elseif LayoutTable(Fi,4) > Firstspeed*1.4 && LayoutTable(Fi,4) < Thirdspeed
        IGears(Fi,1) = 2;
    elseif LayoutTable(Fi,4) > Secondspeed && LayoutTable(Fi,4) < Fourthspeed
        IGears(Fi,1) = 3;

    elseif LayoutTable(Fi,4) > Thirdspeed && LayoutTable(Fi,4) < Fifthspeed
        IGears(Fi,1) = 4;

    elseif LayoutTable(Fi,4) > Fourthspeed && LayoutTable(Fi,4) < Sixthspeed
        IGears(Fi,1) = 5;
    elseif LayoutTable(Fi,4) > Fifthspeed
        IGears(Fi,1) = 6;
    end
end
```

The “LayoutTable” variable consisted of the vehicle speeds required for the test and column four contained the linear velocity converted into angular velocity based on the vehicles rolling radius.

7.2. Corrections to Initial Gear Shift Model.

The initial model worked relatively well to develop a coherent rough gear shift pattern that would lay the basis for amendments. The VW Passat models initial pattern is shown below.

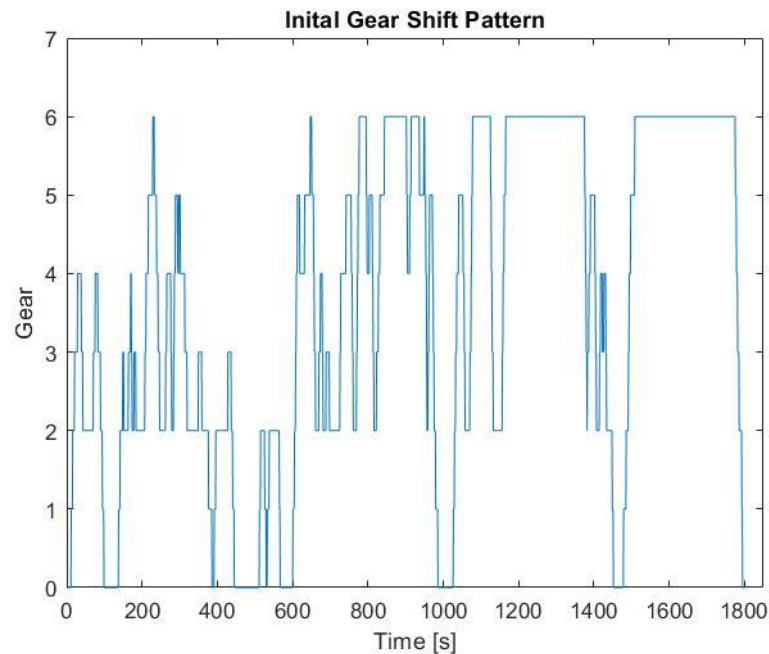


Figure 37 Initial gear shift pattern of the VW Passat across the WLTP.

It's clear that this initial modelling created a pattern with many frequent changes and sections that may need to be adjusted to suit the regulations. Because of the difficulty of the regulations and consistent changes made that will have a knock-on effect to the subsequent pattern. The script created to ensure legality of the pattern consisted of multiple iterations of tests, that would run after a few changes had been made to be sure that the engine speed hadn't dropped below the minimum at any point due to a change. Because of the amendments of gears the other regulations had to simultaneously be checked also, primarily the regulation stating the hold of 6 seconds the gear prior and after are the same. This was the most difficult to ensure because it could drastically change the model and this line of code was checked consistently throughout the iterations until a legal gear shift pattern had been found. The comparison between the initial and the finalised legal pattern for the VW is shown in the following, highlighting the amount of changes required for only one set of gear ratios.

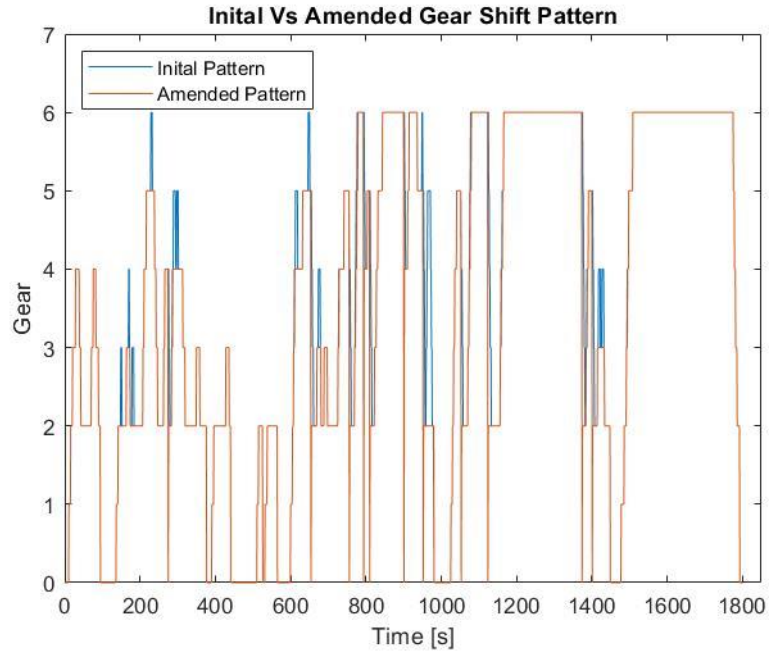


Figure 38 Initial Vs Amended Gear shift pattern for the VW across the WLTP.

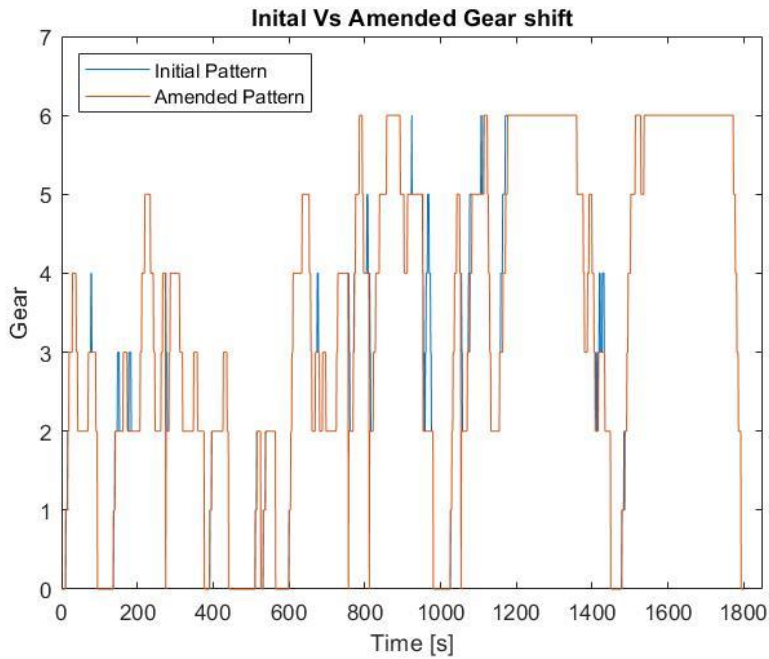


Figure 39 Initial Vs Amended Gear shift pattern for Ford across the WLTP.

This found pattern could then be used for the initial testing run of the vehicles, to use as a validity check of the modelling and gear shifting. The expected extra use for the VW is to be around 10% more CO₂ in g/km across the WLTP in comparison the NEDC. This extra use is primarily due to the change in test mass, temperature and the more transient nature of the cycle itself, highlighted in the following figure.

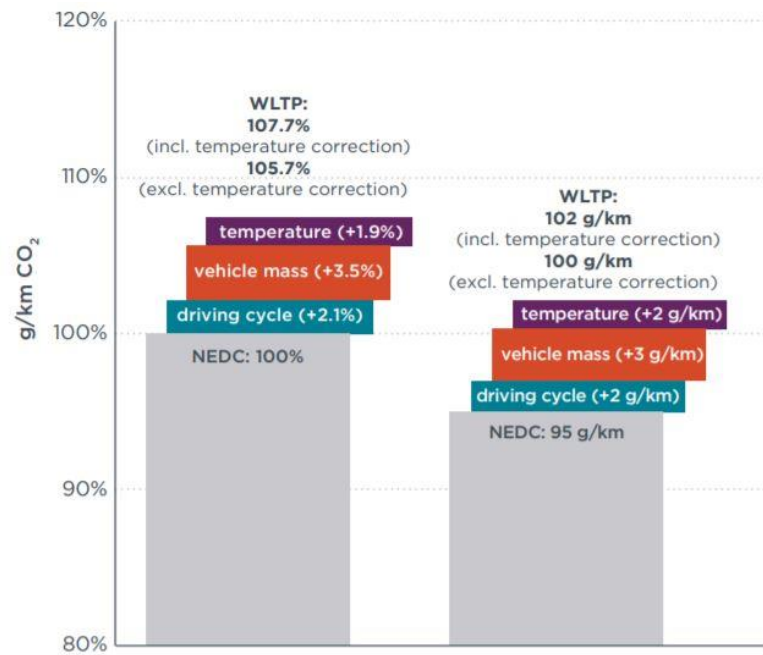


Figure 40 Expected impact of WLTP over the NEDC as a percentage of regulation change (Mock, et al., 2014).

Although the impact of the change between the NEDC and the WLTP is disputed amongst papers and their authors, it's clear than the WLTP will certainly results in additional carbon dioxide being produced because of the much more aggressive accelerations across the cycle.

Upon the initial test, the values for the VW Passat and Ford Mondeo are shown below.

	Original Gear
First Gear	3.36
Second Gear	2.09
Third Gear	1.47
Fourth Gear	1.1
Fifth Gear	0.87
Sixth Gear	0.73
Fuel [g]	1649.7
CO2 [g/km]	218.54
Average BSFC [g/Kw h]	282.5
Achieved MPG	29.886
Mechanical Work Done [MJ]	13.887
Total Fuel Energy [MJ]	72.585
Average Thermal Efficiency	19.133

Table 16 VW Passat 2.0L TFSI Original gear ratio calculated values across the WLTP.

	Original Gear
First Gear	3.583
Second Gear	1.952
Third Gear	1.194
Fourth Gear	0.829
Fifth Gear	0.943
Sixth Gear	0.756
Fuel [g]	1491.9
CO ₂ [g/km]	198.34
Average BSFC [g/Kw h]	161.42
Achieved MPG	33.046
Average MPG	56.523
Mechanical Work Done [MJ]	13.831
Total Fuel Energy [MJ]	67.137
Average Thermal Efficiency	20.602

Table 17 Ford Mondeo 2.0L EcoBlue Original gear ratio calculated values across the WLTP.

With respect to validity of these models, the VW is showing that the calculated CO₂ production is now at 218.54g/km in comparison to the stated NEDC of 198g/km, demonstrating an increase of 10.4% increase, confirming the expected increase and showing that the model is certainly within the bounds of adequate accuracy. With the Ford Mondeo, the CO₂ is again around a 9% increase from that of the NEDC, but most importantly the achieved MPG and average MPG is within the stated best and worst values from Ford at an accuracy of 95%. Therefore, these found values validate the modelling and gear shift patterns used.

7.3. Automated Gear Shift Script for Ratio Appropriation Loops.

Given the testing method used, in the multi-loop code procedure the gear ratio is consistently changed in 0.001 increments. This change will obviously affect the engine speed as the test progresses through and this has the knock-on effect to change the point where a minimum engine speed is met, meaning the gear shift pattern will have to be modified to ensure legality.

Therefore, a slightly modified version of the initial gearshift pattern script was included within the loops to reset the gear shift pattern for every test run, keeping the pattern legal. Because of the sheer number of tests, it's not effective to show all the patterns across, given that some of them wouldn't have changed. However, the following figure shows the first and last iteration of the VW's third gear test patterns.

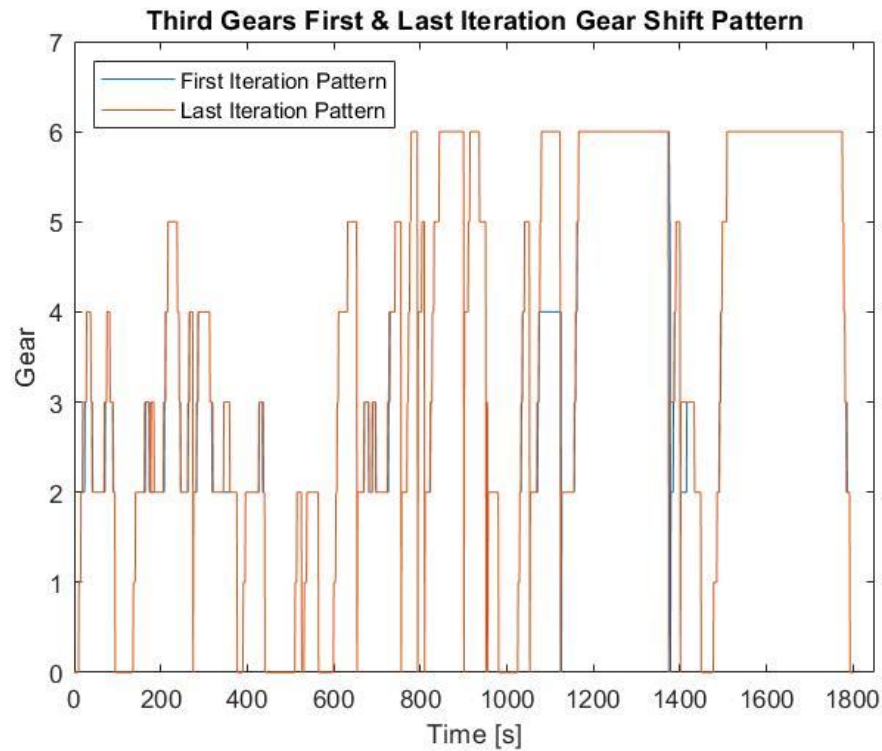


Figure 41 VW Passat 2.0L TFSI First and last iteration of loop test gear shift patterns.

This figure demonstrates the effect of the changing ratio upon the gear shift pattern and although there isn't a large change to be seen, there is a small difference in when the gear is shifted throughout the test from second to third because the engine speed set to change up for the initial layout pattern will be different. The most notable change is around 1100s and because the gear shift pattern wouldn't have had the correct amount of time to go from third to fourth and up to fifth, so the gear has had to remain in third causing a large spike in the fuel usage for this gear ratio.

[Appendix K]

7.4. Results of the appropriated gear ratios across the WLTP.

Within the modelling of the Volkswagen Passat the gear ratios would only work with a maximum percentage change of 7%. The reason for this is thought to be either a scripting issue but given the initial modelling pattern having worked and proved the 10% change in CO₂ production, the primary reason is thought to be the maximum power characteristic of the powertrain itself.

Because of the importance of the maximum power engine speed in the gear shift pattern, it's thought that a change past the 7% found, results in a pattern that doesn't conform with the power demand regulations meaning that it couldn't be used. However, as can be seen in the results below, the ratio change does portray a reduction in fuel showing the theory still remains, but the most important highlight is that the gear shift pattern is the best area to optimise for the highest reduction in fuel usage across the WLTP.

7.4.1 Volkswagen Passat 2.0L TFSI 2006 Estate WLTP Results.

The change of ratios across the cycle resulted in the following changes:

	Appropriated Gear	Original Gear	Difference %
First Gear	3.596	3.36	-7
Second Gear	1.944	2.09	7
Third Gear	1.367	1.47	7
Fourth Gear	1.023	1.1	7
Fifth Gear	0.809	0.87	7
Sixth Gear	0.679	0.73	7
Fuel [g]	1618.3	1649.7	1.9
CO ₂ [g/km]	214.39	218.54	1.9
Average BSFC [g/Kw h]	276.97	282.5	1.96
Achieved MPG	30.464	29.886	1.9
Mechanical Work Done [MJ]	13.887	13.887	0
Total Fuel Energy [MJ]	71.207	72.585	1.9
Average Thermal Efficiency	19.503	19.133	1.9

Table 18 VW Passat 2.0L TFSI gear ratio change results across the WLTP.

Given the maximum percentage change was only 7%, as stated the theory of it being the maximum reduced gear ratio remains to produce the best results for reduced fuel usage. However, as seen for first gear, this happens to be reversed and the highest possible ratio is the one the results in the least fuel usage. The only reason for this is because of the adjustable gear shift patter, and with the ratio being higher the change to second gear happens quicker and therefore less time is spent in first, reducing the overall fuel usage. There was a surprising lack of fuel saved across the WLTP for the

VW, this is thought to be because of its age, having been constructed specifically for the New European Drive Cycle instead, hence the 10% saving across the NEDC, but only a 1.9% change across the WLTP. Perhaps the fuel map for this vehicle would need adjusting to change its optimum operating point, because across the WLTP the average speed, acceleration and demand is higher than the NEDC. Along with the lack of steady state situations, where across the NEDC vehicles could thrive with a low speed, low-mid torque peak operating point ensuring low fuel usage.

On the contrary to this, considering the BSFC map of the VW Passat [figure 28] there is a fair area covered throughout the RPM range where the brake specific fuel consumption values aren't very high. So respective of this, perhaps it is only the prolonged periods of higher average engine speed that have caused the lack of fuel saving across this cycle, highlighted across the “extra-high” phase of the cycle.

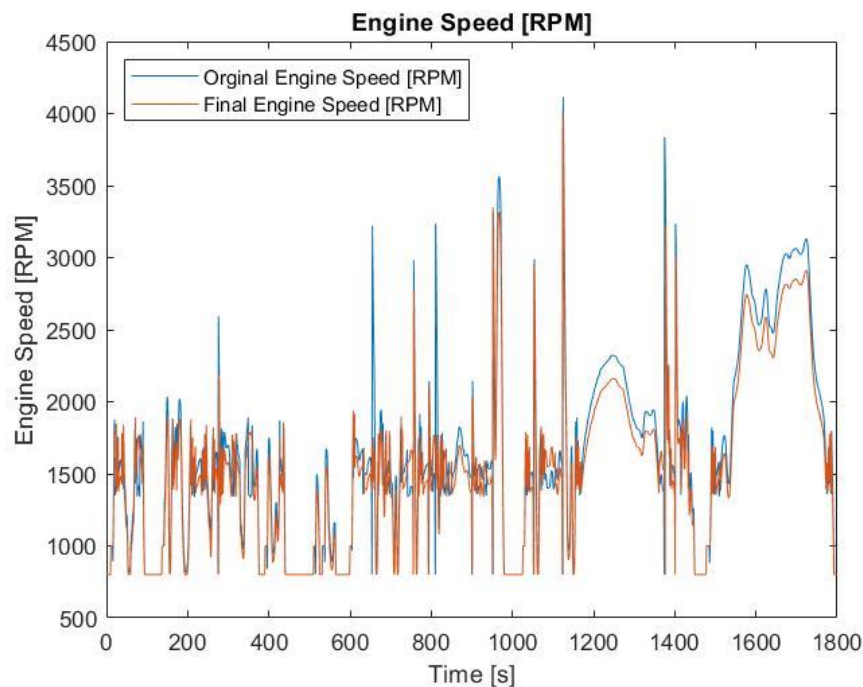


Figure 42 VW Passat 2.0L TFSI engine speeds across the WLTP.

Again, because of the changed regulation of no set gear shift pattern, the individual gear fuel usage results look very different to that of the NEDC. With the VW for gears 3, 4 and 5, the individual fuel usage is higher than that of the original gear ratio. But this is only because of the change in gear shift pattern retaining the gear for longer, but also allowing other gears to make more use of the time available, shown in the figures below.

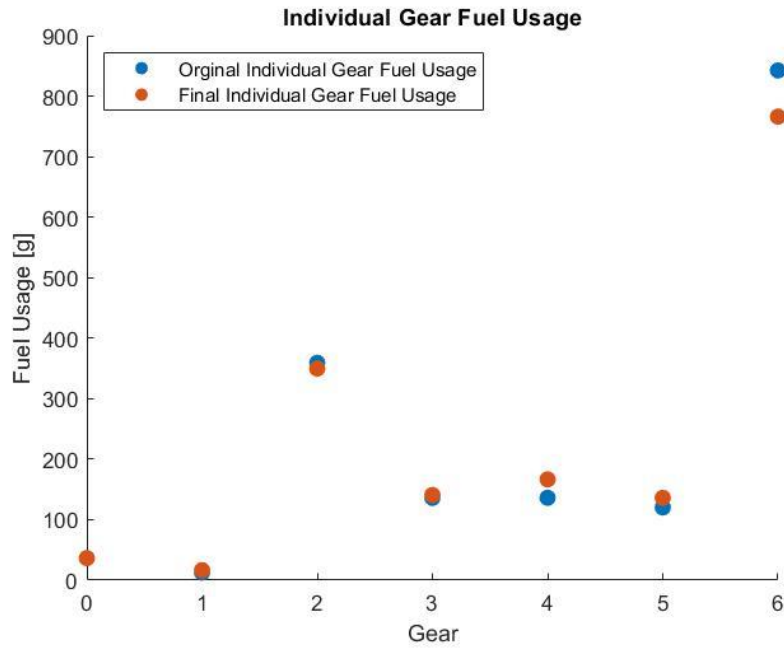


Figure 43 VW Passat 2.0L TFSI Individual gear fuel usage across the WLTP.

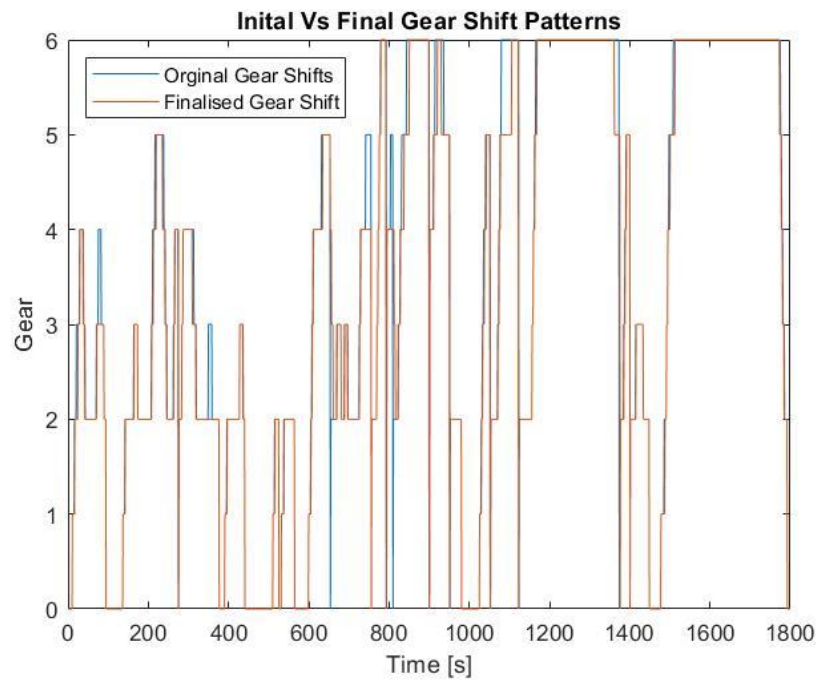


Figure 44 VW Passat 2.0L TFSI Gear shift patterns across the WLTP.

With the lack of fuel saving with appropriation of the gear ratios, it is thought that the VW Passat would benefit for a full study of the gear shift pattern and developing a further model that is capable of optimising the pattern, based on the minimisation of the brake specific fuel consumption. [Appendix L, 1]

7.4.2 Ford Mondeo 2.0L EcoBlue 2019 Estate WLTP Results.

With the Ford Mondeo having been built for the WLTP cycle the change in fuel use was expected to be low, however as can be seen in the table below, this wasn't the case.

	Appropriated Gear	Original Gear	Difference %
First Gear	3.864	3.583	-8
Second Gear	1.757	1.952	10
Third Gear	1.075	1.194	10
Fourth Gear	0.746	0.829	10
Fifth Gear	0.849	0.943	10
Sixth Gear	0.68	0.756	10
Fuel [g]	1403.9	1491.9	5.9
CO2 [g/km]	185.98	198.34	6.23
Average BSFC [g/Kw h]	160.98	161.42	0.27
Achieved MPG	35.118	33.046	5.9
Average MPG	62.717	56.523	9.88
Mechanical Work Done [MJ]	13.831	13.831	0
Total Fuel Energy [MJ]	61.772	67.137	7.99
Average Thermal Efficiency	22.391	20.602	7.99

Table 19 Ford Mondeo 2.0L EcoBlue gear ratio change results across the WLTP.

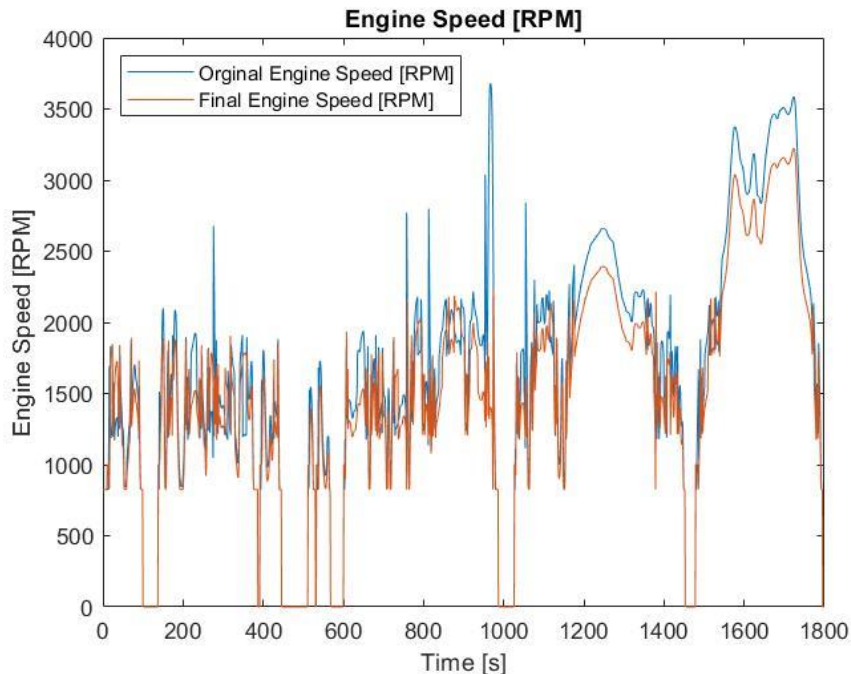


Figure 45 Ford Mondeo 2.0L EcoBlue Engine speeds across the WLTP.

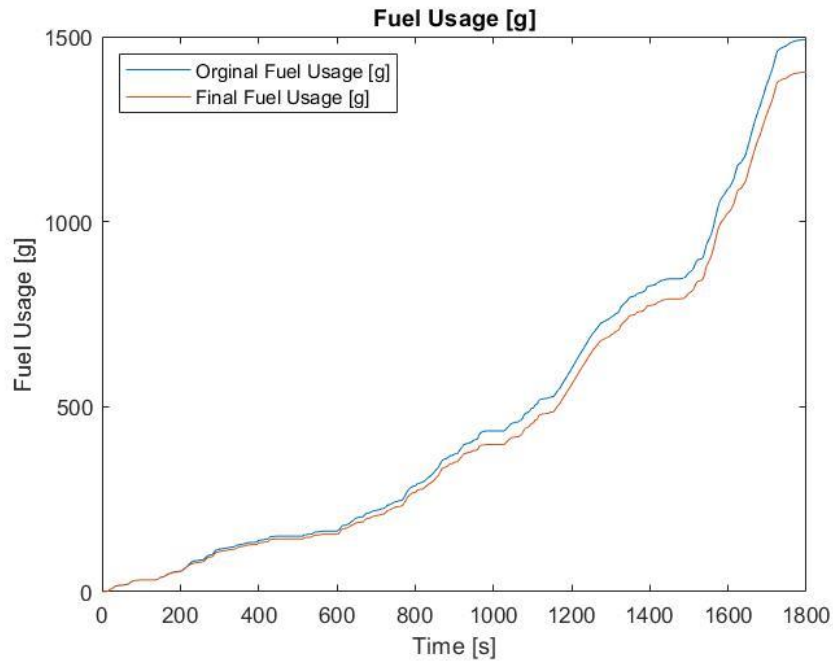


Figure 46 Ford Mondeo 2.0L EcoBlue Fuel usage across the WLTP.

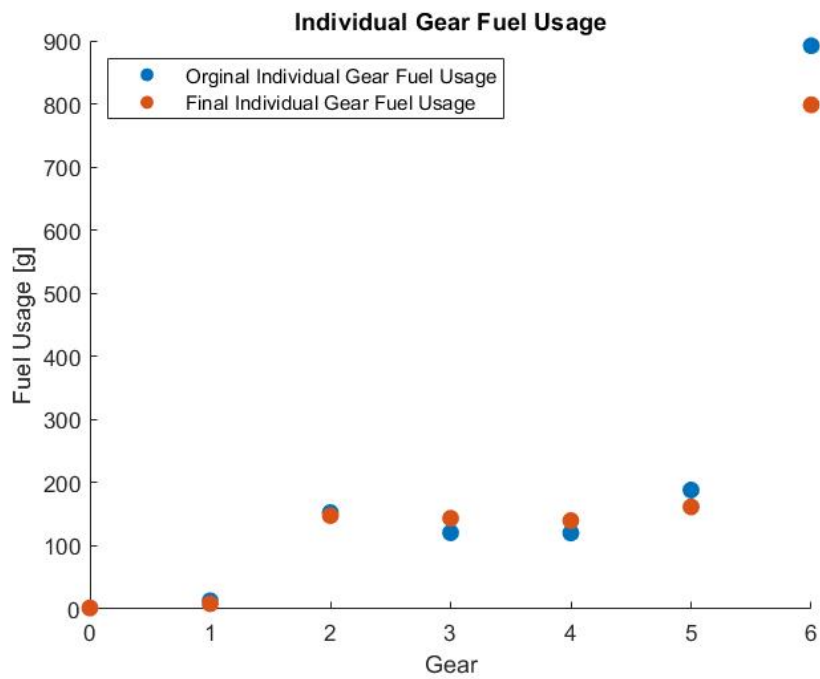


Figure 47 Ford Mondeo 2.0L EcoBlue Individual gear fuel usage across the WLTP.

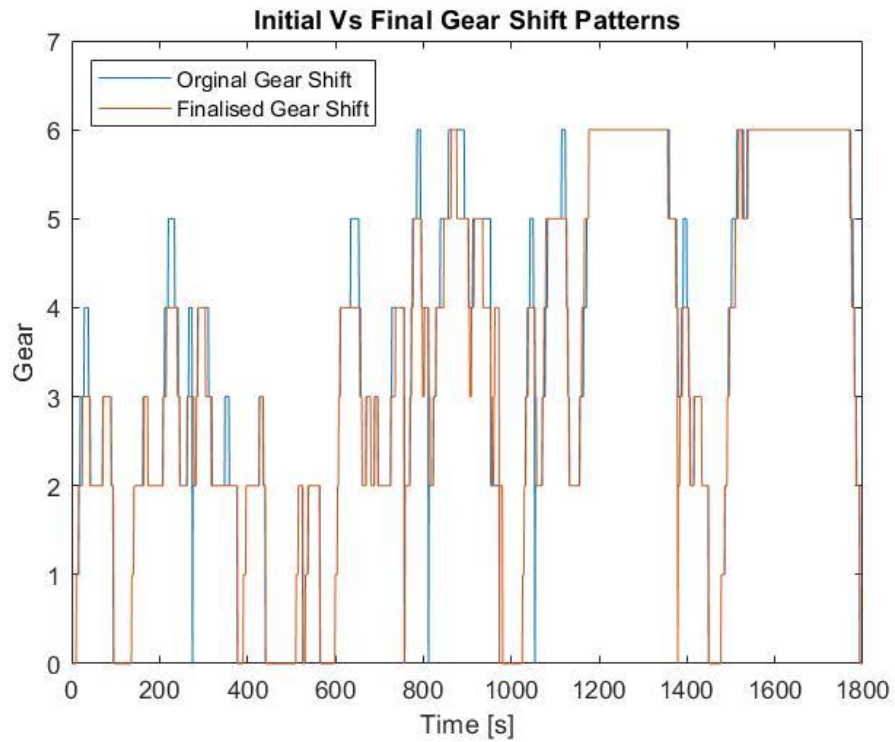


Figure 48 Ford Mondeo 2.0L EcoBlue Gear shift patterns across the WLTP.

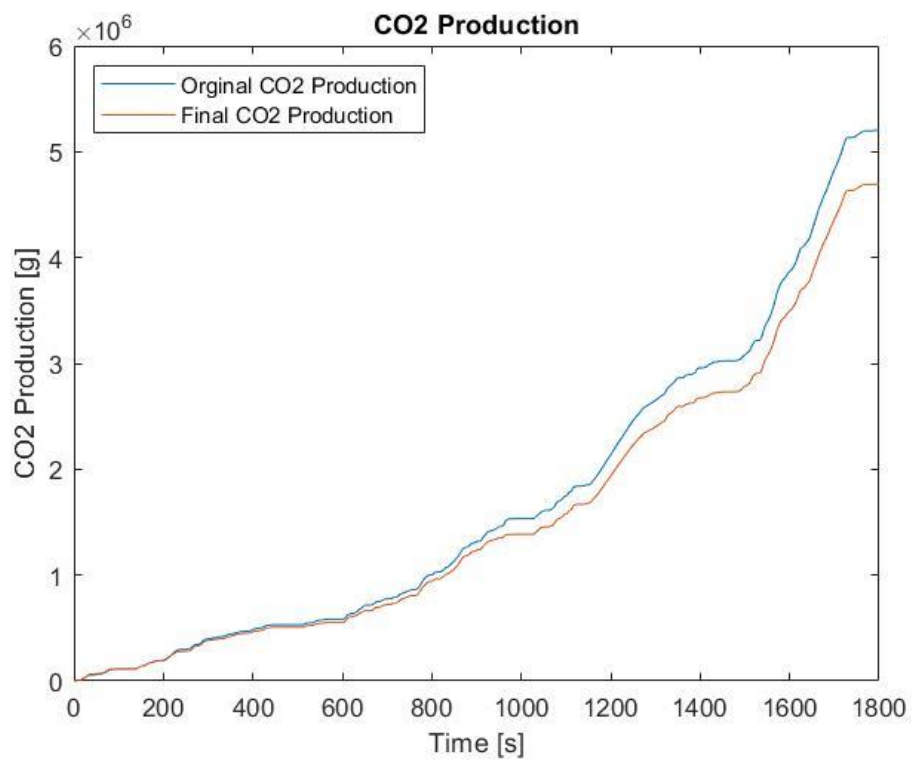


Figure 49 Ford Mondeo 2.0L EcoBlue CO2 Production across the WLTP.

The investigation resulted in a surprising reduction in fuel use for this vehicle showing a 5.9% fuel reduction and 6.23% reduction of carbon dioxide produced. Again, the maximum change below the original ratio stands to result in the least fuel used, because of the reduced engine speed, but oddly with first gear the least fuel used was found at +8% not +10%. It's thought this result occurred because the ratio allowed a change in gear shift where it spent less time in first gear, while having the engine speed lower than that of a 10% change. With the 10% change not again dropping the time spent in first gear, but instead just causing the engine to increase speed and therefore fuel.

It is shown again with the Ford, that even though singular gears like 3 and 4 are representing a higher individual fuel use, across the cycle with the effect on the gear shifting pattern, the overall fuel usage is reduced.

Although the fuel reduction is much higher than anticipated, the results demonstrate a good correlation with firstly the MPG values, with the appropriate gear ratios taking the average MPG up from Fords "best" of 56.5, to 62.7 MPG. While the achieved MPG of the cycle corresponds with Ford's found "worst" value.

And secondly, between the two cycles the vehicle that was manufactured for the cycle showed the largest margin of reduction, with the VW showing a larger reduction in fuel over the NEDC and not so much over the WLTP, while the Ford showed the opposite of this, demonstrating the impact of manufactures reason for design. The Ford was created with the intended release of 2019 meaning it would be tested over the WLTP, so this would have been the primary target for development of the vehicle's characteristics. Whereas the Volkswagen was released in 2006, where the NEDC was the only foreseeable testing procedure, at least for sales in Europe, hence the design being specifically for a good performance over this test alone.

[Appendix L, 2]

8. Conclusion.

This project consisted of three main principle aims; validation of vehicle modelling, gear ratio appropriation across the NEDC and gear ratio appropriation with a suitable gear shifting pattern across the WLTP, of which all three targets were met.

The vehicle modelling has shown, that through using simple mathematical relationships for velocity, torque and energy, a vehicle can be simulated across a homologated test procedure to a high degree of accuracy in order to calculate the fuel requirement.

The respective CO₂ production calculations for the petroleum vehicles also exhibits a high degree of accuracy, leading to the conclusion that manufactures will try to ensure the vehicle operates at, or around Lambda 1, because this is where the mathematics holds true. It is also evident from this investigation that the vehicles will be completing the combustion process to a level of nearly complete combustion within the cylinder because this is a fundamental principle of the products relationship used for the calculation of CO₂.

Considering the gear ratio appropriation across the New European Driving Cycle, it has been made clear that a reduction in gear ratio, leading to a reduction of engine speed, is a cost effective way to reduce the fuel used across the cycle and subsequently the emissions produced. This is possible because of the set gear shifting pattern and the simplicity of the NEDC itself, and it's because of this simplicity it was unanimously decided a change was needed.

Leading into the World harmonised Light vehicle Testing Procedure, from which has shown that the manufactures of the world will have to make serious changes to their vehicle's powertrain, drivetrain and external characteristics. This is due somewhat exclusively to the now regulated, vehicle dependent, manual transmission gear shifting procedure, that has been shown to be the area in which fuel could potentially be saved across this cycle instead of only modifying the gear ratios alone.

Ensuing the final conclusion that the percentage of reduction in fuel use for a vehicle across these cycles is dependent upon year of manufacture. Meaning that these passenger vehicles are being constructed specifically to complete these testing cycles with the best results and not necessarily for real world driving. This is why it is of such importance that the Real Driving Emissions [RDE] test is being introduced, to help control test cycle "cheating" and ensure a worldwide reduction in passenger vehicle pollutants.

9. Further Work.

Throughout this project there were multiple smaller avenues that could've been explored if parts and more time was available. The primary additional work would be to validate the values for weights, coefficients and moments of inertia. The weights would only be found if the vehicles were available but given the ages of them it simply had to be taken that the values stated were correct. Although there is confidence within the Ford vehicles because the owner's manuals or Ford.media documents are easily available online, but for the VW Passat the only values available are those stated on supposed specification sites. VW themselves are seemingly very reluctant to publish data out or recreate media specific documents that include the important specifications on their vehicles and powertrains.

The coefficients are a tricky value to obtain, as explained in the above report, but with access to multiple vehicles and a small OBD-2 data recording device purchased for this investigation. Coast down tests were going to be reproduced across multiple vehicles to try and gain an average value for vehicle types, i.e. hatchbacks, estates, coupes, etc. Unfortunately, at the time of the project were some of these studies could've been conducted because of "free time", weather conditions in and around Swansea are were not ideal to conduct a coast down test, with heavy rain and strong winds consistently halting the investigation.

Another area that warrants further study is the motion simulation of dynamometer testing of the transmissions used, in order to assess the internal inertia of the gears and shafts. This could then be included, and the additional torque required found, therefore increasing the realism & accuracy of the modelling.

Final section of further work would be the full gear shift pattern and gear ratio testing conducted across the WLTP. This test would be based upon the idea that for every iteration test of a step in a single gear ratio, the full tests for the other gears would be conducted and simultaneously change the gear shift pattern. The idea is to then assess the optimum gear ratios with the best gear shift pattern to try and reduce the fuel consumption further and aid in the reduction of CO₂. All with the objective of reducing the impact of emissions and show areas of importance to achieve the lowered emissions targets of 2021.

10. Covid-19 Pandemic impact.

Considering the global pandemic that forced the unforeseeable early closure of the university, the personal impact upon this project was the only the change in learning atmosphere. In the sense that it made it difficult to work from home where I have a lack of comfortable area to work, leading to a reduction in the progression rate of this report. Regarding the investigation, having access to a desktop computer that had MATLAB installed on it halted the potential development of a testing script that would've been made for the gear shifting pattern across the WLTP. The idea of the script is to allow a series of continuous runs testing every possible other gear ratio and subsequent pattern for each step of a ratio change. This script would've most likely taken at least one and a half, maybe two full days to compute and not having access to a desktop that could allow me to run a script like this stopped this area of study.

However, considering the rest of the work and the amount already conducted due to having my laptop and the time across the year without the pandemic, it is felt that there was no major impact upon this project, in comparison to other fellow students.

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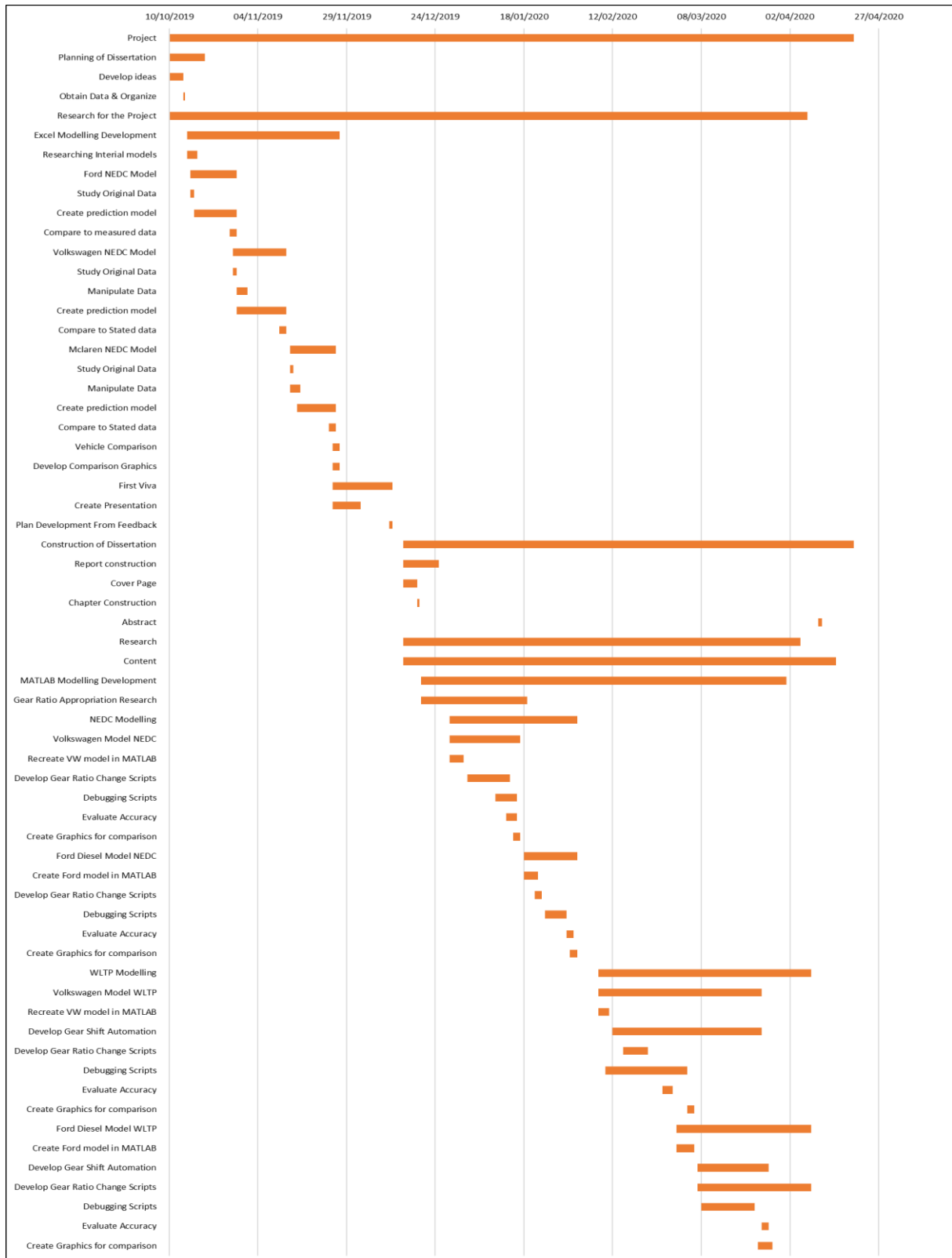
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12. Appendices.

Appendix A.

The full timing plan with all original foreseeable tasks.



Appendix B.

The following tables are the specifications for measured emissions produced by petrol passenger vehicle as the years have progressed and the regulations eventually got more stringent. The pattern of 5 years between can be seen, although with the final euro 6 emissions target starting in 2015, this was subject to a condition of vehicle release date.

The newest iteration of emissions targets are going to be put into action in the year 2021 because of the current legislation preventing it to come in earlier. These new targets are vastly more stringent than previous, joint with vastly more impacting fines and punishments for non-conformity, leading to the projected €30 Billion fines heading for the 8 major OEM's of the world.

Euro 6 Emissions	
CO	1 g/km
THC	0.1 g/km
NMHC	0.068 g/km
NOx	0.06 g/km
PM (direct injection only)	0.005 g/km
PN (direct injection only)	6.0x10⁻¹¹ Per Km

Table 20 Euro 6 Target Emissions Standards (RAC, 2020).

Euro 4 Emissions	
CO	1 g/km
THC	0.1 g/km
NOx	0.08 g/km

Table 23 Euro 4 Target Emissions Standards (RAC, 2020).

Euro 2 Emissions	
CO	2.2 g/km
HC + NOx	0.5 g/km

Table 25 Euro 2 Target Emissions Standards (RAC, 2020).

Euro 5 Emissions	
CO	1 g/km
THC	0.1 g/km
NMHC	0.068 g/km
NOx	0.06 g/km
PM (direct injection only)	0.005 g/km

Table 21 Euro 5 Target Emissions Standards (RAC, 2020).

Euro 3 Emissions	
CO	2.3 g/km
THC	0.2 g/km
NOx	0.15 g/km

Table 22 Euro 3 Target Emissions Standards (RAC, 2020).

Euro 1 Emissions	
CO	2.72 g/km
HC + NOx	0.97 g/km

Table 24 Euro 1 Target Emissions Standards (RAC, 2020).

Appendix C.

After the group project conducted in the previous year of study [2nd year], there happened to be a Ford Fiesta 1600cc EcoBoost manual transmission in the storage at the university. This transmission was thought to be a good representation of that used within the Mondeo and therefore it was decided to strip down the gearbox to both gain an understanding of the internal operation, whilst collecting measurements and photographs to re-create it within SolidWorks. The objective was to enable the possibility of calculating the moments of inertia for all internal parts, but because of the complexity of some, i.e. the differential and output-shaft, and the requirement for a full motion simulation to ensure operation. This task would have demanded far too much time in comparison to the reward.

However, the gears were still constructed along with the two shafts and the third smaller ‘spindle’ that 6th gear operates on, and below is various figures highlighting the complexity of the transmission within the CAD software, joint with photographs of the parts.

1.1. C.A.D

Input-Shaft:

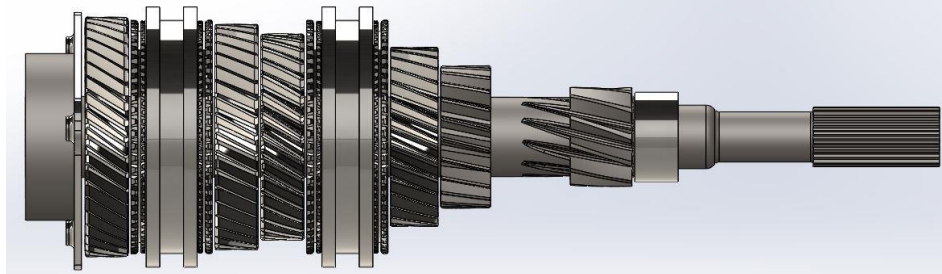


Figure 50 Side profile on the input-shaft CAD.



Figure 51 Top-side view of the input-shaft CAD.

Output-shaft:

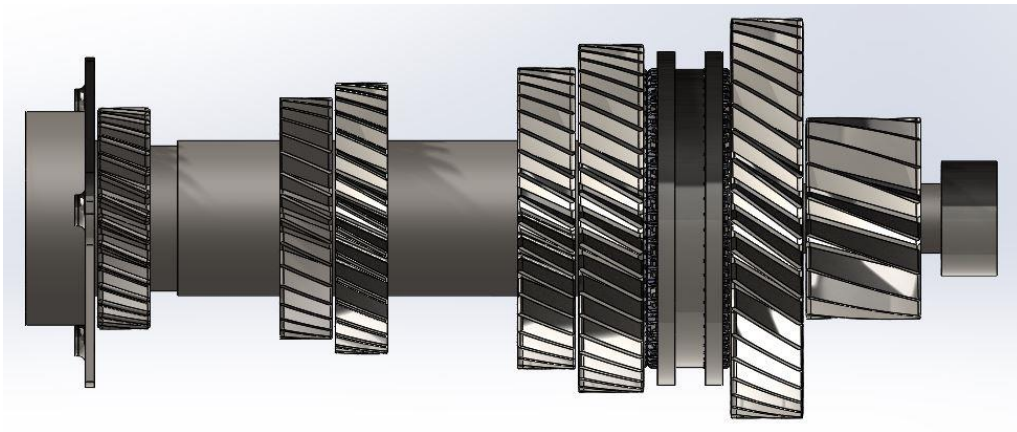


Figure 52 Side view of the output-shaft CAD.

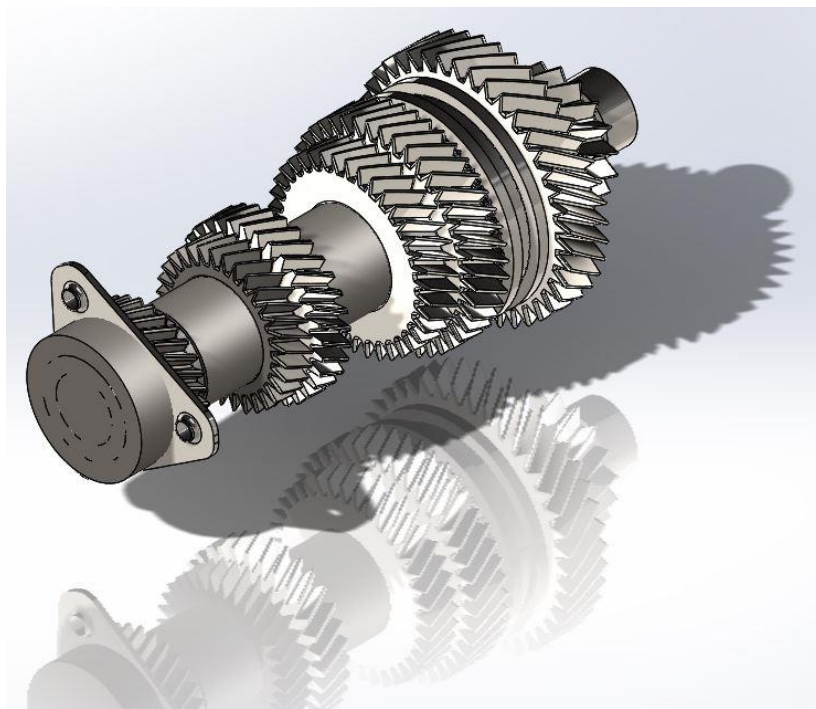


Figure 53 Top-Side profile of the output-shaft CAD.

Reverse-shaft and gear:

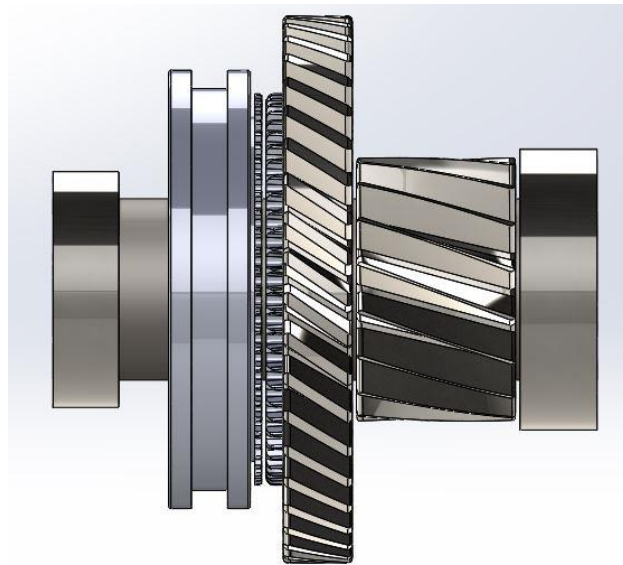


Figure 54 Side profile of the reverse gear and shaft CAD.

Full transmission excluding the differential:

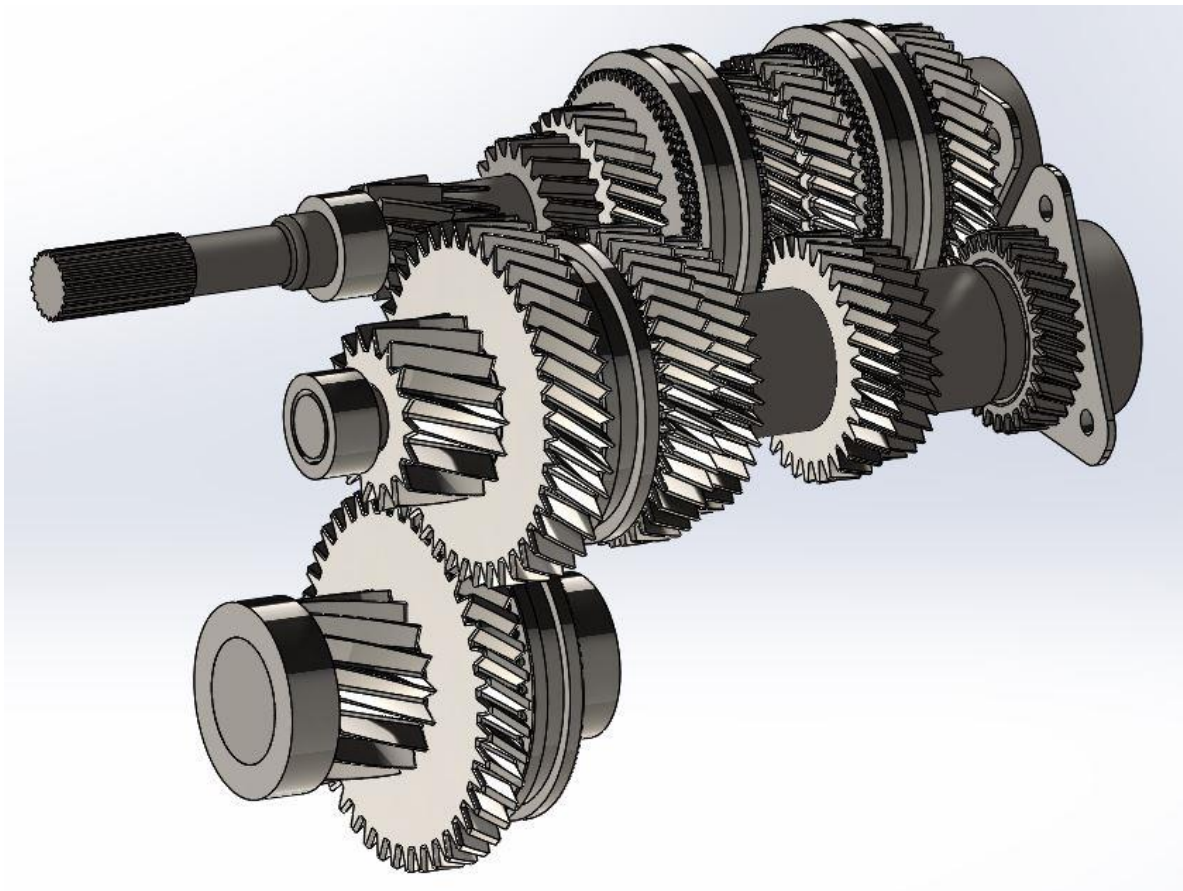


Figure 55 The three shafts in placement representative of the internal placement CAD.

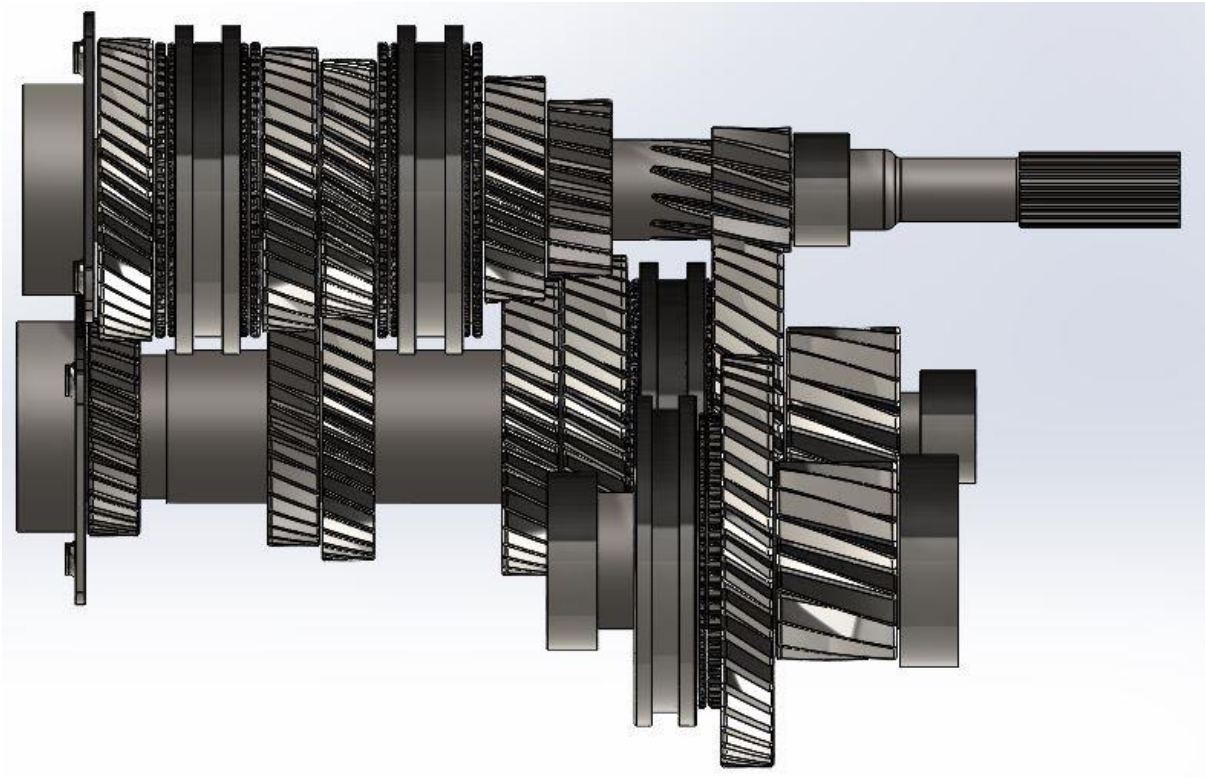


Figure 56 Full transmission side profile CAD.

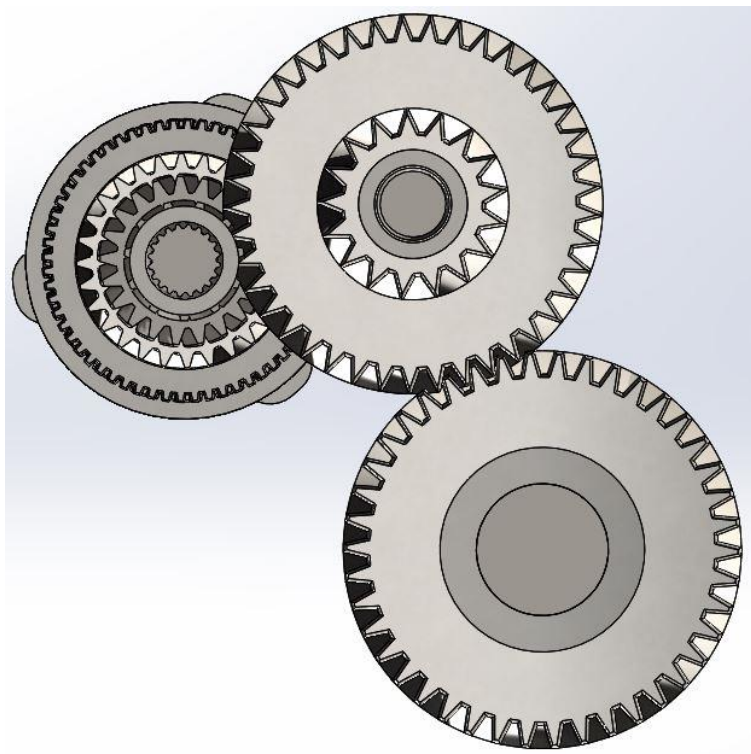


Figure 57 Positioning of the three shafts internally.

1.2. Photographs of the real transmission assembly.

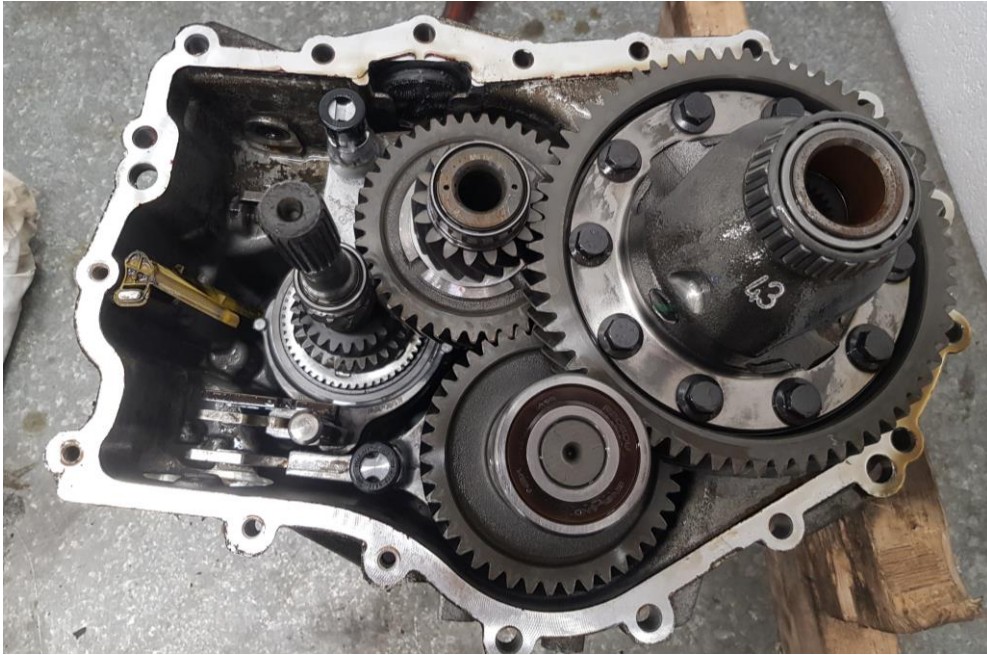


Figure 58 Top view of the real gearbox and the shaft positioning.



Figure 59 Output-shaft.



Figure 60 Input-shaft.



Figure 61 Reverse gear connection to the differential drive gear.



Figure 62 Front end of the complete transmission.



Figure 63 Transmission part number and code.

Appendix D.

Rolling resistance is a difficult value to model because it is a transient value and does not remain as constant as vehicle speed increases, due to variables such as tyre temperature which will have a knock-on effect to tyre pressure. Then if the vehicle can produce “downforce” or negative lift, there will be a large force acting on the vehicle causing a difference from original testing loads. Thomas D. Gillespie attempted to generalise characteristic behaviours of common tyres and surfaces, dependent of the variables above within the following figures:

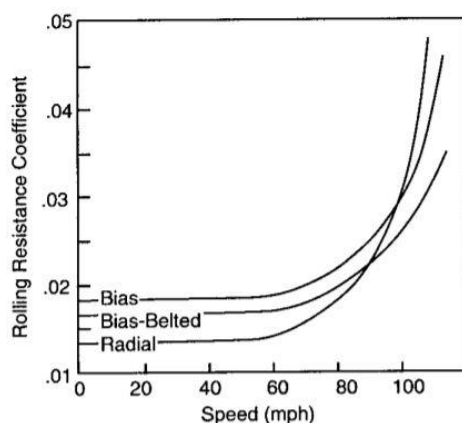


Figure 64 Tyre construction and speed influence on RR coefficient (Gillespie, 1992).

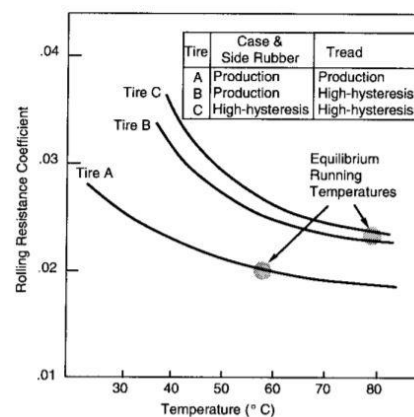


Figure 65 Tyre temperature influence on RR coefficient (Gillespie, 1992).

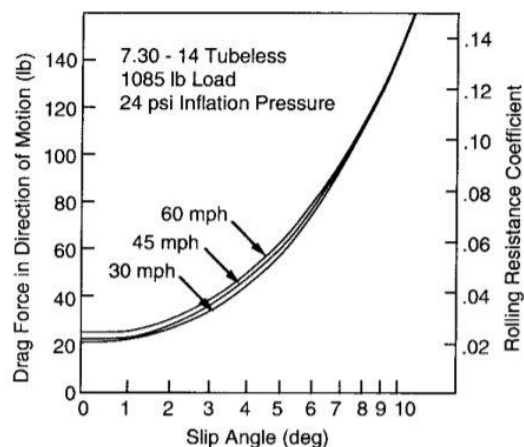


Figure 66 Slip angle and speed influence on RR coefficient (Gillespie, 1992).

Many factors contribute to a change in coefficient and it is difficult to definitively choose a value in which to be used in a model. Upon a full measurement some have tried to plot and then produce a curve fit style procedure to develop a form of equation that can attempt to predict the change in coefficient due to speed. But as shown above, that is only ever really going to be representative of a few tyres at very few points and this is again why the rolling resistance coefficient was left as constant. Another factor is that the vehicles are on a chassis dynamometer when being tested across the NEDC and WLTP, with no varying slip angles regarding turning steering wheel or modelling

the path around a corner. Speed only ever reaches 120Km/h for only six seconds and when looking at Joseph Katz's diagram of the aerodynamic force compared to that of rolling resistance, it's clear that on a straight line test, the coefficient will not make a drastic impact all the way up to the tests maximum speed.

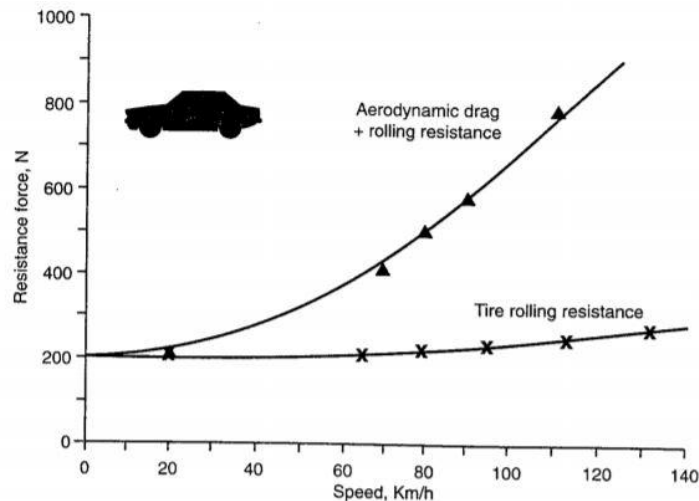


Figure 67 Aerodynamic and rolling resistance forces against speed (Katz, 1995).

In comparison to that, aerodynamic forces are a major area for potential downfall in a vehicles fuel efficiency and subsequently the emission output, not only across this test cycle, but on the roads. The aerodynamic coefficients can be found using a wind tunnel if the company wishes to pay for the time spent within one and where the value can be found more accurately because the conditions are idealised. If they were to use the 'coast down' method, cross winds could cause serious issues within their data, as well as the temperature, pressure, and humidity of the air on the test day. Lastly, the mechanical losses of the vehicle need to be known to evaluate the resistive forces acting on the vehicle, although these could be found using a test with the principle of a "sheet" over the vehicle, whilst towed by another to stop the effect of drag, shown in the following figure.

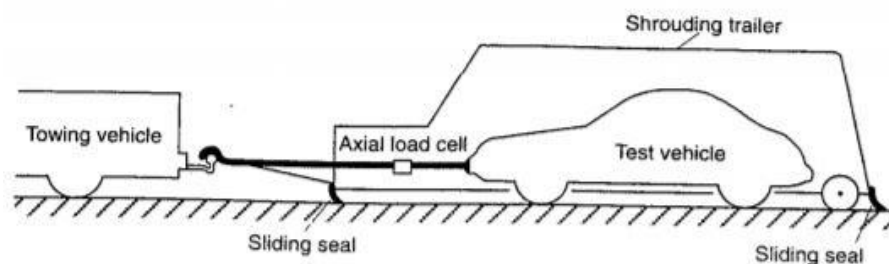


Figure 68 Type of testing procedure to calculate mechanical losses of a vehicle without drag (Katz, 1995).

For both vehicles tested the maximum force due to aerodynamic drag resistance can be seen below. These results are representative of the values used within the model, i.e. air density, frontal area and so on.

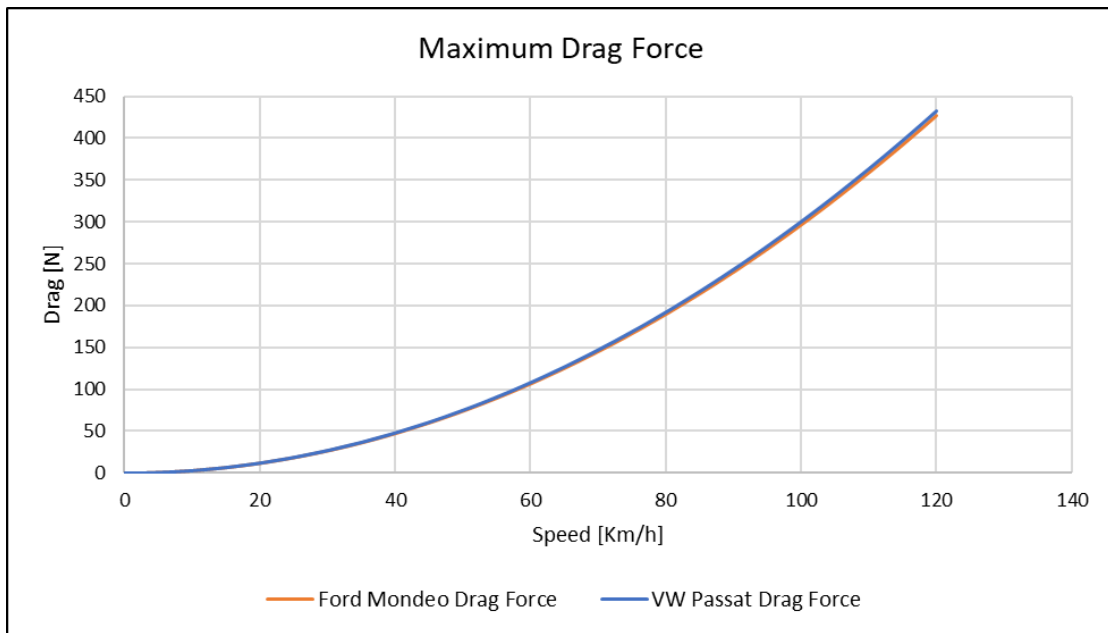


Figure 69 NEDC excel test vehicles maximum aerodynamic drag.

This is again a good reiteration of the similarities between the externals of the vehicle and that the modelling tests of these vehicles is instead a good representation of only the individual power and drivetrains.

Another point is the magnitude of the drag force on the vehicle and that in reality, these vehicles will be spending a considerable amount more time at these higher speeds when operating on real roads and this is just one reason why the NEDC was not a good model or test sequence to develop a vehicle across. It allowed multiple manufactures to “beat it” with multiple forms of electronic “trickery”, the most famous of which being Volkswagen and the ‘Diesel gate’ scandal, that demonstrated the very real difference between test bench results and real-world results.

Appendix E.

The equations used for the wheels, brake discs, hubs, driveshafts and then clutch pack with flywheel are shown below with the two simplified shapes they were based on:

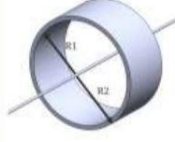
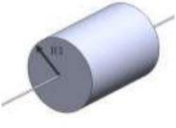
Annular cylinder about central axis	Solid Cylinder about central axis
$\frac{1}{2}M(R_1^2 + R_2^2)$	$\frac{1}{2}MR^2$
	

Figure 70 Inertia equation and the associated shape (Knight).

Most parts were modelled using the solid cylinder around a central axis because of its simple disc like profile suited to a driveshaft, clutch, flywheel, brake disc. It had to be used for the wheel disc (“rim”) and wheel hub because although they are very complex in structure, the basis remains around a cylindrical shape. The annular cylinder was used to assess the tyre, although this could have been done to a more thorough extent, the lack of actual tyre and dimensions prevented this.

The dimensions and assumed weights of the brake discs, wheel hubs, driveshaft, clutch friction plate and flywheel are shown in the table below:

Parts masses used for the three vehicles					
Volkswagen		Ford Mondeo [Petrol 1.6]		Ford Mondeo [2.0L Diesel]	
Brake discs	5.5 Kg	Brake discs	5 Kg	Brake discs	5.3 Kg
Wheel Hub	3 Kg	Wheel Hub	3 Kg	Wheel Hub	3.2 Kg
Driveshaft [short]	14 Kg	Driveshaft [short]	10 Kg	Driveshaft [short]	14 Kg
Driveshaft [long]	16 Kg	Driveshaft [long]	12.5 Kg	Driveshaft [long]	18 Kg
Friction plate	1.5 Kg	Friction plate	1 Kg	Friction plate	2 Kg
Flywheel	11 Kg	Flywheel	11 Kg	Flywheel	16.2 Kg

Table 26 Driveline parts masses for each vehicle (Eurocarparts, 2020), (Eurocarparts, 2020), (carpartssaver, 2020).

Having used the CAD of the transmission shown in Appendix C, the inertia of the shafts were found using the software’s inbuilt function. However, the accuracy of these values is not yet clear, and they were used only on the basis that they are within the region of expected values, previously stated.

Input Shaft	
Length [mm]	338
Mass [Kg]	5.6
Moment of inertia [Kg/m ²]	0.026
Output Shaft	
Length [mm]	280
Mass [Kg]	7
Moment of inertia [Kg/m ²]	0.0397
Reverse	
Length [mm]	120
Mass [Kg]	3.2
Moment of inertia [Kg/m ²]	0.0039

Figure 71 1600cc EcoBoost transmission CAD estimated values for MMI.

With all parts of the driveline, exclusive of the wheel and flywheel, it could be said that they do not have much impact, if at all, on the result of gross torque. This is because the moment of inertia is small and given the accelerations across the NEDC specifically are rather low, it is clear they would not have much effect, shown within the graph of the VW Passat's individual inertia across the cycle.

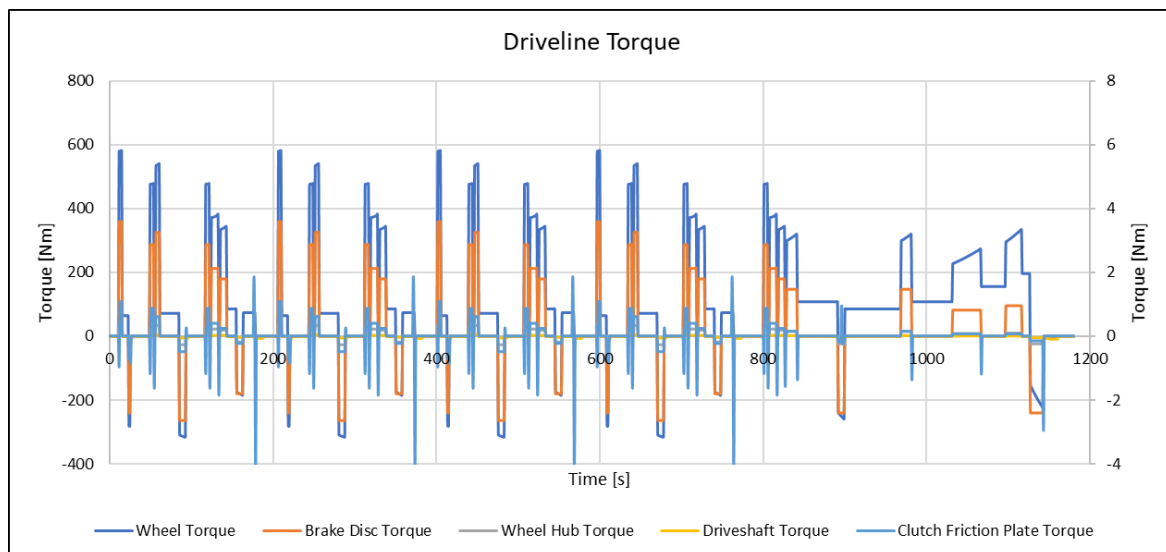


Figure 72 Driveline Torque required during the cycle [VW Passat 2.0TFSI 2006 Estate].

The wheel torque is presented with the left-hand axis and reaches nearly 600Nm, while the brake disc, wheel hub, driveshaft and clutch friction plate are on the right-hand axis and does not even make it 4Nm of torque required. Clarifying the fact that this section of modelling is not fundamental but should be included where possible.

One set of standard wheels that come on the VW Passat B6 2.0TFSI estate are shown below to highlight the complex shape and potential to drastically effect the wheel inertia.



Figure 74 VW Passat 2006 B6 2.0L TFSI Estate (Wheel-Size.com, 2020).



Figure 73 Standard VW Passat wheel rim (vrsautosalvagegateshead, 2020).

And one set of standard wheels available for the 2011 Ford Mondeo 1600cc EcoBoost Estate.



Figure 76 Ford Mondeo 1.6L EcoBoost Estate (Ford Motor Company, 2017).



Figure 75 Standard Ford Mondeo wheel rim (Ford Motor Company, 2020).

Appendix F.

A study conducted by Seaman, R. et al. that considers the “Component Inertial Effects on Transmission Design”, shows the individual moment of inertia and energy required when engine is 3000RPM. They studied a 5spd manual transmission with ratios that weren’t dissimilar to that of the VW Passat, although given the age of the paper it could be said that the transmission is out of date. However, if their results showed a worst-case scenario, the energy required to shift through the transmission is shown to only results in 2.25 KJ of energy.

FORMULA: $J_R = J (N_R/N)^2$

J = Inertia of gear being reflected.

N_R = Number of gear teeth on destination gear.

N = Number of teeth on gear being reflected.

TRANSMISSION GEAR RATIOS: 4.03/2.37/1.49/1.00/0.86

CONVENTIONAL TRANSMISSION

GEAR	INERTIA (kg-m ²)	J _R AT 1ST GEAR (kg-m ²)	J _R AT 2ND GEAR (kg-m ²)	J _R AT 3RD GEAR (kg-m ²)	J _R AT 4TH MESH (kg-m ²)	J _R AT 5TH GEAR (kg-m ²)	J _R AT REVERSE (kg-m ²)
CLUTCH	0.00791	0.12827	0.04460	0.01748	0.00791	0.00590	0.11174
INPUT	0.00041	0.00660	0.00230	0.00090	0.00041	0.00030	0.00575
C/SHAFT	0.00413	0.02156	0.00750	0.00294	0.00133	0.00099	0.01878
1 ST	0.00202	0.00202	0.00070	0.00028	0.00012	0.00009	0.00176
2 ND	0.00116	0.00332	0.00116	0.00045	0.00020	0.00015	0.00289
3 RD	0.00049	0.00357	0.00124	0.00049	0.00022	0.00016	0.00311
4 TH	0.00000	0.00000	0.00000	0.00000	0.00000	0.00000	0.00000
5 TH	0.00025	0.00552	0.00192	0.00075	0.00034	0.00025	0.00481
REVERSE	0.00112	0.00128	0.00045	0.00017	0.00008	0.00006	0.00112
IDLER	0.00016	0.00048	0.00017	0.00007	0.00003	0.00002	0.00042
SUM OF REFLECTED INERTIAS		0.17262	0.06002	0.02352	0.01064	0.00794	0.15038

Table 27 Component inertia table as presented by (Seaman, Johnson, & Hamilton, 1984).

FORMULA: $K.E. = J_n (\omega_2 - \omega_1)^2 / 2$

J_n = Effective inertia at gear being synchronized (kg-m²)

ω_1 = Synchronizer hub speed (rad/sec)

ω_2 = Speed of gear being synchronized (rad/sec)

TRANSMISSION GEAR RATIOS: 4.03/2.37/1.49/1.00/0.86

INPUT SPEED (ENGINE): 314.2 rad/sec (3000 RPM)

CONVENTIONAL TRANSMISSION

SHIFT	J _R (kg-M ²)	ω_2 (rad/sec)	ω_1 (rad/sec)	K.E. (joules)
1 - 2	0.0600	132.3	78.0	88.3
2 - 3	0.0235	211.3	132.3	73.5
3 - 4	0.0106	314.2	211.3	56.3
4 - 5	0.0079	363.8	314.2	9.8
5 - 4	0.0106	314.2	363.8	13.1
4 - 3	0.0235	211.3	314.2	124.4
3 - 2	0.0600	132.3	211.3	187.6
2 - 1	0.1726	78.0	132.3	254.1
1 - R	0.1504	-83.6	78.0	1963.1
R - 1	0.1726	78.0	-83.6	2253.5

Table 28 Energy required to shift through transmission as presented by (Seaman, Johnson, & Hamilton, 1984).

If the calorific value of the gasoline was said to be 44MJ/Kg, the transmission shifts would only require 0.05g of fuel energy to compensate for this and when considering the NEDC that contains 49 gear shifts, the fuel energy required from the accelerations of the transmission would've be 2.45g. As stated however, this is a worst-case scenario because the energy required, shown in the above table, is for shifting at 3000RPM when the average shift speed for the VW at least, is only 1200RPM.

When taking this into account it's understandable why the modelling within this report would never be 100% accurate because there is always going to assumptions made and parts that are difficult to model, but none the less with the value being only 0.34% of the total use it's understandable why this section of modelling is disregarded for the NEDC.

Another point for consideration within transmissions is the oil used to lubricate the gears within and the effect temperature and oil viscosity could have. A study conducted by Höhn B. et al. titled "Optimisation of Gearbox Efficiency" demonstrates a comprehensive study on contributing factors, but one important part is the effect of fluid temperature on efficiency, their study found:

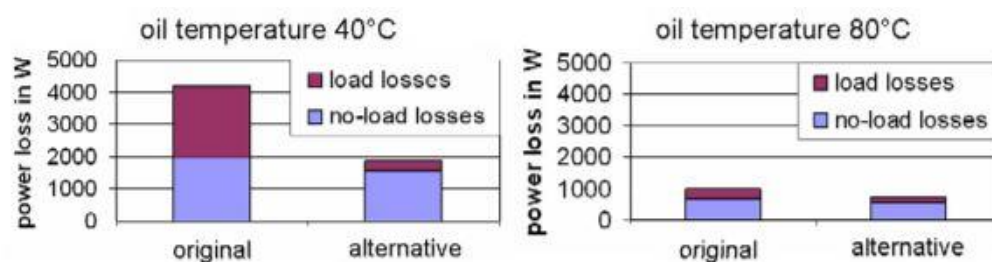


Figure 77 Oil temperature affect upon losses within a transmission as presented by (Höhn, Michaelis, & Hinterstoißer, 2009).

Their alternative design study also poses interesting because the only change was the style of bearing used for the transmission shaft, they had changed from a taper roller type bearing to ball bearing type. And although the no-load situations didn't show much reduction, the important loaded losses are a considerable amount less than the taper rolling, exclusively at the lower temperature.

This idea could show very positive results for the earlier stages of testing procedures, where all temperatures are low and additional fuel will be added to try reach correct operating temperature as soon as possible. If the losses during this section could be made as minimal as possible it should stand to show even more saving of fuel and reduction in emissions across the test procedures.

Appendix G.

Unfortunately, much later within this project it was found that the VW Passat GVT 6spd manual transmission contains a dual final drive system, of which 5th and 6th gear operate with a ratio 3.087 instead of the first thought 3.944. This 3.944 ratio is only operable when 1st – 4th gear are selected. Because of this ratio change, it enable the author to correct the initial NEDC model to be more accurately representable of the official stated fuel use and emissions values, instead of the supposed MAHLE measured values. MAHLE found there to be 374g of fuel used across the extra urban phase and this was thought to be high, however having used the single final drive ratio, this value was matched with high accuracy. It's therefore thought that perhaps MAHLE were conducting a research and development test on behalf of Volkswagen, to test the changes of what a single and dual final drive system would have on the vehicle, for the specific reason of reducing fuel consumption. Having implemented the secondary final drive ratio, it did reduce the Excel modelled fuel usage by around 20g of fuel across the extra urban phase.

The secondary final drive ratio was stated on a workshop manual print, used by VW.

Manual gearbox		6-speed 02Q, front-wheel drive		
Identification code		GVT	HDV	HVS
Manufactured	from to	11.04 04.06	11.04 04.06	11.05 04.06
Allocation	Engine	2.0 l - 147 kW	2.0 l - 100 kW turbo diesel 2.0 l - 103 kW turbo diesel	2.0 l - 125 kW turbo diesel
Ratio: $Z_2 : Z_1$	Final drive I	71 : 18 = 3.944	69 : 20 = 3.450	70 : 19 = 3.684
	Final drive II	71 : 23 = 3.087	69 : 25 = 2.760	70 : 24 = 2.917
Capacity of manual gearbox		2.3 litre	2.3 litre	2.3 litre
Drive shaft flange Ø		107 mm	107 mm	107 mm

Table 29 VW Transmission codes, allocations, final drive ratios and oil capacities (workshop-manuals.com, 2020).

The details of the ratio change can be seen within the comparison tables below showing the new fuel usage against MAHLE's measured and the stated values for the 2006 VW Passat 2.0L TFSI estate.

	MPT Test 148	Calculated	% Difference
Urban (g)	341.5	341.81	100.1
Extra Urban (g)	374.4	356.96	95.3
	MPT Test 148		
Total Fuel Usage (g)	715.9	698.77	97.6
Total CO₂ emissions	200.05	201.22	100.6
Total Accuracy %			98.4

Table 30 Dual final drive ratio change to fuel usage against MAHLE test data.

As can be seen, the change to the extra urban phase fuel usage is a fair amount, considering the calculated fuel usage previously was over predicting at 388g of fuel. So given the extra urban phase is the higher speed section where aerodynamic drag becomes a serious contributing factor, perhaps the drag coefficient, frontal area or the air density has an issue. But having said that, the MATLAB model has raised the fuel usage again into an area of higher accuracy. It is slightly overpredicting, but with the measured fuel use of 715.9g, the MATLAB model has calculated 716.5g total fuel usage. Therefore, the excel model could simply be lacking with the interpolation ability for fuelling, given that MATLAB is capable of many other forms of curve fitting, as well as taking a polynomial curve fit to the 9th order where it matches exactly.

EU	Urban	
6.3	11.2	L/100Km
0.4382	0.4562	Stated Litres used
0.4759	0.4557	Calculated Litres used
6.84	11.19	L/100Km
108.62	99.90	% Difference

Table 31 VW Passat dual final drive results against the stated fuel usage values.

Back with the excel modelling, when compared to the supposed “official” figures, the model is coming out far over on the extra urban phase with 0.54 L/100km of extra fuel being used. The main difficulty here is the inaccessibility to definitive values from VW themselves, without a VIN code to able to access the official handbook for the vehicle. All values seen or found from research are all slightly different and in turn changes the accuracy of the model from under back to over predicting for fuel use and CO₂ produced. However, given that the minimum accuracy is within 95% for all stated values found, it’s believed that this model has no issues regarding inertial models and/or interpolation of fuelling and the technique is correct.

Appendix H.

1. General characteristics and external forces script layout

The basic script construction used for all tests that included the creation of the main table used to assign the time stamp, vehicle speed, gear, clutch operation, “mode” and the variables required for fuelling manipulation. This script also contained the calculated external values for torque and speed up to the differential because these remain consistent, regardless of the gear ratio in the transmission.

The following text is the basic setup script used for the VW Passat.

NEDC VW Passat

```
clc  
clear
```

General

```
format short  
VehicleMass = 1559;  
TestMass = VehicleMass + 100;  
IdleRPM = 800;  
EngineDis = 1984;  
Gravity = 9.81;  
FlywheelMI = 0.3;  
EngineMI = 0.04;  
TotalDistance = 11.03;  
load("GearTimeVelocityMode.mat");  
TableSetup = zeros(1181,35);  
TableSetup(:,1) = GearTimeVelocityMode.Times;  
TableSetup(:,2) = GearTimeVelocityMode.Velocitykmh;  
TableSetup(:,3) = GearTimeVelocityMode.Velocitykmh./3.6;  
TableSetup(:,4) = GearTimeVelocityMode.Acceleration;  
TableSetup(:,5) = GearTimeVelocityMode.Gear;  
TableSetup(:,6) = GearTimeVelocityMode.Mode;
```

```

TableSetup(:,7) = GearTimeVelocityMode.Clutch;

EngineSpeedsb = [500 850 1000 1250 1500 1750 2000 2500 3000 3500 4000
4500 5000]';

EngineSpeedsa = [850 1000 1250 1500 1750 2000 2500 3000 3500 4000 4500
5000]';

% Tyre size
% Front wheels are 215/55 ZR16
% Rear wheels are 215/55 ZR16

```

Aero

```

ADensity = 1.18;
FrontArea = 2.235;
DragCo = 0.3;
CdA = FrontArea*DragCo;

```

Brakes & Hub

```

% Front Discs

FrontOd = 312;
FrontId = 262;
FrontDMass = 5;
FrontDiscMI = 0.5*FrontDMass*(((FrontOd/1000)+(FrontId/1000))/2)^2;

% Rear Discs

RearOd = 310;
RearId = 260;
RearDMass = 5;
RearDiscMI = 0.5*RearDMass*(((RearOd/1000)+(RearId/1000))/2)^2;

% Hub MI

HubR = 137;
HubMass = 3;

```

```
HubMI = 0.5*HubMass*(HubR/1000)^2;
```

Wheels

```
RollingRadius = 0.31;
```

```
RollResis = 0.015;
```

```
WheelMI = 1.05;
```

Driveshafts

```
% Left Driveshaft
```

```
LengthLD = 0.49;
```

```
RadLD = 0.025;
```

```
LDMass = 14;
```

```
LDMI = 0.0088;
```

```
% Right Driveshaft
```

```
LengthRD = 0.79;
```

```
RadRD = 0.025;
```

```
RDMass = 16;
```

```
RDMI = 0.01;
```

Gears

```
% Efficiencies
```

```
FirstE = 0.94;
```

```
SecondE = 0.94;
```

```
ThirdE = 0.95;
```

```
FourthE = 0.97;
```

```
FifthE = 0.97;
```

```
SixthE = 0.96;
```

```
FinalDriveE1 = 0.94;
```

```
FinalDriveE2 = 0.94;
```

```
% Gear Ratios
```

```

FirstGearRatio = 3.357;
SecondGearRatio = 2.087;
ThirdGearRatio = 1.469;
FourthGearRatio = 1.098;
FifthGearRatio = 0.87;
SixthGearRatio = 0.73;
FinalDriveGearRatio14 = 3.944;
FinalDriveGearRatio56 = 3.087;

```

Fuel variables Row Vectors

```

load("FuelFlowPassat.mat")

% 500 RPM

FiveHundredBMEP = zeros(23,1);
FiveHundredFuel = zeros(23,1);
FiveHundredBMEP(:,1) = FuelFlow.BMEP(1:23);
FiveHundredFuel(:,1) = FuelFlow.FuelFlow(1:23);

% 800 RPM

EightHundredfiftyBMEP = zeros(23,1);
EightHundredfiftyFuel = zeros(23,1);
EightHundredfiftyBMEP(:,1) = FuelFlow.BMEP(24:46);
EightHundredfiftyFuel(:,1) = FuelFlow.FuelFlow(24:46);

% 1000 RPM

OneThouBMEP = zeros(23,1);
OneThouFuel = zeros(23,1);
OneThouBMEP(:,1) = FuelFlow.BMEP(47:69);
OneThouFuel(:,1) = FuelFlow.FuelFlow(47:69);

```

```
% 1250 RPM
```

```
OneThouTwoFiftyBMEP = zeros(23,1);  
OneThouTwoFiftyFuel = zeros(23,1);  
OneThouTwoFiftyBMEP(:,1) = FuelFlow.BMEP(70:92);  
OneThouTwoFiftyFuel(:,1) = FuelFlow.FuelFlow(70:92);
```

```
% 1500 RPM
```

```
OneThouFiveBMEP = zeros(23,1);  
OneThouFiveFuel = zeros(23,1);  
OneThouFiveBMEP(:,1) = FuelFlow.BMEP(93:115);  
OneThouFiveFuel(:,1) = FuelFlow.FuelFlow(93:115);
```

```
% 1750 RPM
```

```
OneThouSevFiftyBMEP = zeros(23,1);  
OneThouSevFiftyFuel = zeros(23,1);  
OneThouSevFiftyBMEP(:,1) = FuelFlow.BMEP(116:138);  
OneThouSevFiftyFuel(:,1) = FuelFlow.FuelFlow(116:138);
```

```
% 2000 RPM
```

```
TwoThouBMEP = zeros(23,1);  
TwoThouFuel = zeros(23,1);  
TwoThouBMEP(:,1) = FuelFlow.BMEP(139:161);  
TwoThouFuel(:,1) = FuelFlow.FuelFlow(139:161);
```

```
% 2500 RPM
```

```
TwoThouFiveBMEP = zeros(23,1);  
TwoThouFiveFuel = zeros(23,1);  
TwoThouFiveBMEP(:,1) = FuelFlow.BMEP(162:184);
```

```

TwoThouFiveFuel(:,1) = FuelFlow.FuelFlow(162:184);

% 3000 RPM

ThreeThouBMEP = zeros(23,1);
ThreeThouFuel = zeros(23,1);
ThreeThouBMEP(:,1) = FuelFlow.BMEP(185:207);
ThreeThouFuel(:,1) = FuelFlow.FuelFlow(185:207);

% 3500 RPM

ThreeThouFiveBMEP = zeros(23,1);
ThreeThouFiveFuel = zeros(23,1);
ThreeThouFiveBMEP(:,1) = FuelFlow.BMEP(208:230);
ThreeThouFiveFuel(:,1) = FuelFlow.FuelFlow(208:230);

% 4000 RPM

FourThouBMEP = zeros(23,1);
FourThouFuel = zeros(23,1);
FourThouBMEP(:,1) = FuelFlow.BMEP(231:253);
FourThouFuel(:,1) = FuelFlow.FuelFlow(231:253);

% 4500 RPM

FourThouFiveBMEP = zeros(23,1);
FourThouFiveFuel = zeros(23,1);
FourThouFiveBMEP(:,1) = FuelFlow.BMEP(254:276);
FourThouFiveFuel(:,1) = FuelFlow.FuelFlow(254:276);

% 5000 RPM

FiveThouBMEP = zeros(23,1);

```

```

FiveThouFuel = zeros(23,1);
FiveThouBMEP(:,1) = FuelFlow.BMEP(277:299);
FiveThouFuel(:,1) = FuelFlow.FuelFlow(277:299);

% 6000 RPM

SixThouBMEP = zeros(23,1);
SixThouFuel = zeros(23,1);
SixThouBMEP(:,1) = FuelFlow.BMEP(300:322);
SixThouFuel(:,1) = FuelFlow.FuelFlow(300:322);

% 7000 RPM

SevenThouBMEP = zeros(23,1);
SevenThouFuel = zeros(23,1);
SevenThouBMEP(:,1) = FuelFlow.BMEP(323:345);
SevenThouFuel(:,1) = FuelFlow.FuelFlow(323:345);

```

Fuel Variables Row Vectors

```

% 500 RPM

FiveHundred = zeros(23,2);
FiveHundred(:,1) = FiveHundredBMEP;
FiveHundred(:,2) = FiveHundredFuel;

% 800 RPM

EightHundredfifty = zeros(23,2);
EightHundredfifty(:,1) = EightHundredfiftyBMEP;
EightHundredfifty(:,2) = EightHundredfiftyFuel;

% 1000 RPM

```

```
OneThou = zeros(23,2);  
OneThou(:,1) = OneThouBMEP;  
OneThou(:,2) = OneThouFuel;
```

```
% 1250 RPM
```

```
OneThouTwoFifty = zeros(23,2);  
OneThouTwoFifty(:,1) = OneThouTwoFiftyBMEP;  
OneThouTwoFifty(:,2) = OneThouTwoFiftyFuel;
```

```
% 1500 RPM
```

```
OneThouFive = zeros(23,2);  
OneThouFive(:,1) = OneThouFiveBMEP;  
OneThouFive(:,2) = OneThouFiveFuel;
```

```
% 1750 RPM
```

```
OneThouSevFifty = zeros(23,2);  
OneThouSevFifty(:,1) = OneThouSevFiftyBMEP;  
OneThouSevFifty(:,2) = OneThouSevFiftyFuel;
```

```
% 2000 RPM
```

```
TwoThou = zeros(23,2);  
TwoThou(:,1) = TwoThouBMEP;  
TwoThou(:,2) = TwoThouFuel;
```

```
% 2500 RPM
```

```
TwoThouFive = zeros(23,2);  
TwoThouFive(:,1) = TwoThouFiveBMEP;  
TwoThouFive(:,2) = TwoThouFiveFuel;
```

```
% 3000 RPM
```

```
ThreeThou = zeros(23,2);  
ThreeThou(:,1) = ThreeThouBMEP;  
ThreeThou(:,2) = ThreeThouFuel;
```

```
% 3500 RPM
```

```
ThreeThouFive = zeros(23,2);  
ThreeThouFive(:,1) = ThreeThouFiveBMEP;  
ThreeThouFive(:,2) = ThreeThouFiveFuel;
```

```
% 4000 RPM
```

```
FourThou = zeros(23,2);  
FourThou(:,1) = FourThouBMEP;  
FourThou(:,2) = FourThouFuel;
```

```
% 4500 RPM
```

```
FourThouFive = zeros(23,2);  
FourThouFive(:,1) = FourThouFiveBMEP;  
FourThouFive(:,2) = FourThouFiveFuel;
```

```
% 5000 RPM
```

```
FiveThou = zeros(23,2);  
FiveThou(:,1) = FiveThouBMEP;  
FiveThou(:,2) = FiveThouFuel;
```

```
% 6000 RPM
```

```

SixThou = zeros(23,2);
SixThou(:,1) = SixThouBMEP;
SixThou(:,2) = SixThouFuel;

% 7000 RPM

SevenThou = zeros(23,2);
SevenThou(:,1) = SevenThouBMEP;
SevenThou(:,2) = SevenThouFuel;

%% Curve Fitting

load("FuelCurveFits.mat")

```

Wheel Force

```

% Rolling Resistance

RollingR = zeros(1181,1);
RollingR(:,1) = (TestMass*Gravity)*RollResis;

% Mass Force

LinearForce = zeros(1181,1);
LinearForce(:,1) = TestMass.*TableSetup(:,4);

% Aero Force

AeroForce = zeros(1181,1);
AeroForce(:,1) = (0.5*ADensity*CdA).*(TableSetup(:,3).^2);

% Wheel Force

WheelForce = zeros(1181,1);
WheelForce(:,1) = RollingR + LinearForce + AeroForce;

```

```

% With Stationary Force

for wf = 1:length(TableSetup(:,1))

    if TableSetup(wf,3) == 0 && TableSetup(wf,4) == 0

        WheelForce(wf,1) = 0;

    end

end

TableSetup(:,8) = WheelForce;

```

Wheel Angular Velocity and Acceleration

```

% Velocity

WheelRads = TableSetup(:,3)./RollingRadius;
TableSetup(:,9) = WheelRads;

% Acceleration

WheelRadss = TableSetup(:,4)./RollingRadius;
TableSetup(:,10) = WheelRadss;

```

Wheel Torque

```

WheelTorque = zeros(1181,1);

for wt = 1:length(TableSetup(:,1))

    if TableSetup(wt,6) == 7 && TableSetup(wt,7) == 0

        WheelTorque(wt,1) = 0;
    end
end

```

```

elseif TableSetup(wt,3) > 0

WheelTorque(wt,1) = WheelForce(wt,1).*RollingRadius;

end

end

TableSetup(:,11) = WheelTorque;

```

Brake & Hub Assembly Torque

```

% Brake discs [4 of]
% T = Ia

DiscTF = (FrontDiscMI*2).*TableSetup(:,10);
DiscTR = (RearDiscMI*2).*TableSetup(:,10);
FrontBT = DiscTF + DiscTR;

% Hub Torque

HubT = (HubMI*4).*TableSetup(:,10);

% Total Torque

BHT = FrontBT + HubT;

TableSetup(:,12) = BHT;

```

Outer Driveshaft Torque

```

OuterDsT = zeros(length(TableSetup),1);

for gb = 1:length(TableSetup(:,1))

```

```
if TableSetup(gb,6) == 7 && TableSetup(gb,7) == 0
```

```
OuterDsT(gb,1) = 0;
```

```
else
```

```
OuterDsT(gb,1) = BHT(gb,1) + WheelTorque(gb,1) +  
(WheelMI*4).*TableSetup(gb,10);
```

```
end
```

```
end
```

```
TableSetup(:,13) = OuterDsT;
```

Inner Driveshaft Torque

```
InnerDsT = OuterDsT + (LDMI.*TableSetup(:,10)) +  
(RDMI.*TableSetup(:,10));
```

```
TableSetup(:,14) = InnerDsT;
```

Differential Output Angular speeds

```
DiffOutSpeed = TableSetup(:,9);
```

```
DiffOutAccel = TableSetup(:,10);
```

```
TableSetup(:,16) = DiffOutSpeed;
```

```
TableSetup(:,17) = DiffOutAccel;
```

Wheel force without braking effect

```
DiffOutTorque = zeros(length(TableSetup),1);
```

```
for tit = 1:length(TableSetup(:,1))
```

```
if TableSetup(tit,6) == 7 && TableSetup(tit,7) == 0
```

```

DiffOutTorque(tit,1) = 0;

else

DiffOutTorque(tit,1) = InnerDsT(tit,1);

end

end

TableSetup(:,15) = DiffOutTorque;

```

2. The complete testing loop script fundamental layout.

```

%% Efficiency of Gears

format shortG

FinalE1f = FinalDriveGearRatio14*FinalDriveE1;
FinalE2f = FinalDriveGearRatio56*FinalDriveE2;
FEf = (First*FirstE)*FinalE1f;
SEf = (SecondGearRatio*SecondE)*FinalE1f;
TEf = (ThirdGearRatio*ThirdE)*FinalE1f;
FoEf = (FourthGearRatio*FourthE)*FinalE1f;
FiEf = (FifthGearRatio*FifthE)*FinalE2f;
SiEf = (SixthGearRatio*SixthE)*FinalE2f;

```

Transmission in Torque

```

TransInTFirst = zeros(1181,1);

for itf = 1:length(TableSetup(:,1))

    if TableSetup(itf,5) == 0

        TransInTFirst(itf,1) = 0;

    elseif TableSetup(itf,15) == 0

        TransInTFirst(itf,1) = 0;
    end
end

```

```

elseif TableSetup(itf,5) == 1
    TransInTFirst(itf,1) = DiffOutTorque(itf,1)./FEf;
elseif TableSetup(itf,5) == 2
    TransInTFirst(itf,1) = DiffOutTorque(itf,1)./SEf;
elseif TableSetup(itf,5) == 3
    TransInTFirst(itf,1) = DiffOutTorque(itf,1)./TEf;
elseif TableSetup(itf,5) == 4
    TransInTFirst(itf,1) = DiffOutTorque(itf,1)./FoEf;
elseif TableSetup(itf,5) == 5
    TransInTFirst(itf,1) = DiffOutTorque(itf,1)./FiEf;
elseif TableSetup(itf,5) == 6
    TransInTFirst(itf,1) = DiffOutTorque(itf,1)./SiEf;
end
end

FirstGearTable = TransInTFirst;

```

Transmission in Speed & Acceleration

```

% Diff in speed

DiffInSpeedFirst = zeros(length(TableSetup),1);
DiffInAccelFirst = zeros(length(TableSetup),1);

for Dif = 1:length(TableSetup)

    if TableSetup(Dif,5) == 1 || TableSetup(Dif,5) == 2 ||
    TableSetup(Dif,5) == 3 || TableSetup(Dif,5) == 4

        DiffInSpeedFirst(Dif,1) =
        TableSetup(Dif,9).*FinalDriveGearRatio14;

        DiffInAccelFirst(Dif,1) =
        TableSetup(Dif,10).*FinalDriveGearRatio14;

    elseif TableSetup(Dif,5) == 5 || TableSetup(Dif,5) == 6

```

```

        DiffInSpeedFirst(Dif,1) =
TableSetup(Dif,9).*FinalDriveGearRatio56;

        DiffInAccelFirst(Dif,1) =
TableSetup(Dif,10).*FinalDriveGearRatio56;

    end

end

% Speed

TransInSFirst = zeros(1181,1);

for isf = 1:length(TableSetup(:,1))

    if TableSetup(isf,5) == 0 && TableSetup(isf,3) == 0

        TransInSFirst(isf,1) = 0;

    elseif TableSetup(isf,15) == 0 && TableSetup(isf,7) == 1

        TransInSFirst(isf,1) = IdleRPM*((2*pi)/60);

    elseif TableSetup(isf,5) == 1

        TransInSFirst(isf,1) = DiffInSpeedFirst(isf,1).*(First);

    elseif TableSetup(isf,5) == 2

        TransInSFirst(isf,1) =
DiffInSpeedFirst(isf,1).*(SecondGearRatio);

    elseif TableSetup(isf,5) == 3

        TransInSFirst(isf,1) =
DiffInSpeedFirst(isf,1).*(ThirdGearRatio);

    elseif TableSetup(isf,5) == 4

        TransInSFirst(isf,1) =
DiffInSpeedFirst(isf,1).*(FourthGearRatio);

    elseif TableSetup(isf,5) == 5

        TransInSFirst(isf,1) =
DiffInSpeedFirst(isf,1).*(FifthGearRatio);

    elseif TableSetup(isf,5) == 6

```

```

        TransInSFirst(isf,1) =
DiffInSpeedFirst(isf,1).*(SixthGearRatio);

        end

end

FirstGearTable(:,2) = TransInSFirst;

for trf = 1:length(TableSetup(:,1))

    if TableSetup(trf,7) == 0 && TableSetup(trf,3) > 0

        TransInSFirst(trf,1) = TransInSFirst(trf-1,1)*0.8;

    end

end

% Acceleration

TransInAFirst = zeros(length(TableSetup(:,1)),1);

iaf = 2;

TransInAFirst(1,1) = 0;

while iaf <= length(TableSetup(:,1))

    TransInAFirst(iaf,1) = (TransInSFirst(iaf,1) - TransInSFirst(iaf-
1,1));

    iaf = iaf+1;

end

for eF = 1:length(TableSetup(:,1))

    if TableSetup(eF,7) == 0

        TransInAFirst(eF,1) = 0;

    end

end

```

```
FirstGearTable(:,3) = TransInAFirst;
```

Flywheel

```
FlywheelTmFirst = FlywheelMI.*TransInAFirst;
```

```
FlywheelTFirst = FlywheelTmFirst + TransInTFirst;
```

```
FirstGearTable(:,4) = FlywheelTFirst;
```

Engine Speed & Acceleration

```
EngineSpeedFirst = zeros(1181,1);
```

```
for esf = 1:length(TableSetup(:,1))  
    if TransInSFirst(esf,1) < IdleRPM*((2*pi)/60)  
        EngineSpeedFirst(esf,1) = IdleRPM;  
    elseif TransInSFirst(esf,1) > IdleRPM*((2*pi)/60)  
        EngineSpeedFirst(esf,1) = TransInSFirst(esf,1)./((2*pi)/60);  
    end  
end
```

```
for fbf = 1:length(TableSetup(:,1))  
    if TableSetup(fbf,5) == 1  
        Enf(fbf,1) = EngineSpeedFirst(fbf,1);  
    else  
        Enf(fbf,1) = 0;  
    end  
end
```

```
EnSf = Enf(:,1);  
EnSf(EnSf == 0) = [];  
NewEngineSpeedsFirst(:,Fi) = EnSf;
```

```
% Acceleration
```

```

EngineAccelFirst = zeros(1181,1);

for eaf = 1:length(TableSetup(:,1))
    if FirstGearTable(eaf,2) < IdleRPM*((2*pi)/60)
        EngineAccelFirst(eaf,1) = 0;
    elseif TableSetup(eaf,16) > 0 && TableSetup(eaf,7) > 0
        EngineAccelFirst(eaf,1) = TransInAFirst(eaf,1);
    elseif TableSetup(eaf,6) == 7 && TableSetup(eaf,7) == 0
        EngineAccelFirst(eaf,1) = TransInAFirst(eaf-1,1)*0.75;
    end
end
end

```

Engine and Auxiliary Torque

```

EngineTFirst = EngineMI.*EngineAccelFirst;

FirstGearTable(:,4) = EngineTFirst;

AuxiliaryT = GearTimeVelocityMode.AuxTorque;

FirstGearTable(:,5) = AuxiliaryT;

```

Gross Engine Torque

```

GrossTFirst = EngineTFirst + AuxiliaryT + FlywheelTFirst;

for fbf = 1:length(TableSetup(:,1))
    if TableSetup(fbf,5) == 1
        GTof(fbf,1) = GrossTFirst(fbf,1);
    else
        GTof(fbf,1) = 0;
    end
end
end

```

```

GTf = GTof(:,1);
GTf(GTf == 0) = [];
NewEngineSpeedsFirst(:,(Fi + 674)) = GTf;
FirstGearTable(:,6) = GrossTFirst;

```

BMEP

```

BMEPFirst = (((4*pi.*GrossTFirst)./(EngineDis*10^-6)).*10^-5);

```

```

FirstGearTable(:,7) = BMEPFirst;

```

Engine Speeds above & below

```

AboveRPMFirst = zeros(length(EngineSpeedFirst),1);
BelowRPMFirst = zeros(length(EngineSpeedFirst),1);

for eaf = 1:length(EngineSpeedsa(:,1))

    for atf = 1:length(TableSetup(:,1))

        if EngineSpeedsa(eaf,1) > EngineSpeedFirst (atf,1)

            baf = find(EngineSpeedsa(:,1) > EngineSpeedFirst(atf,1));
            baf = baf(1,1);
            AboveRPMFirst(atf,1) = EngineSpeedsa(baf,1);

        end

        if EngineSpeedsb(eaf,1) < EngineSpeedFirst(atf,1)

            daf = find(EngineSpeedsb(:,1) < EngineSpeedFirst(atf,1));
            daf = daf(end,1);
            BelowRPMFirst(atf,1) = EngineSpeedsb(daf,1);

        end

    end

end
end

```

Fuelling

```

PickAboveFirst = zeros(length(EngineSpeedFirst),1);
PickBelowFirst = zeros(length(EngineSpeedFirst),1);

for Kic = 1:length(EngineSpeedFirst)

    if AboveRPMFirst(Kic,1) == 850

        Pi850ABMEPFirst =
find(PickAbove850First(:,2)>BMEPFirst(Kic,1));

```

```

    Pi850AFuelFirst = PickAbove850First(Pi850ABMEPFirst,1);
    PickAboveFirst(Kic,1) = min(Pi850AFuelFirst);

elseif AboveRPMFirst(Kic,1) == 1000

    Pi1000ABMEPFirst =
    find(PickAbove1000First(:,2)>BMEPFirst(Kic,1));
    Pi1000AFuelFirst = PickAbove1000First(Pi1000ABMEPFirst,1);
    PickAboveFirst(Kic,1) = min(Pi1000AFuelFirst);

elseif AboveRPMFirst(Kic,1) == 1250

    Pi1250ABMEPFirst =
    find(PickAbove1250First(:,2)>BMEPFirst(Kic,1));
    Pi1250AFuelFirst = PickAbove1250First(Pi1250ABMEPFirst,1);
    PickAboveFirst(Kic,1) = min(Pi1250AFuelFirst);

elseif AboveRPMFirst(Kic,1) == 1500

    Pi1500ABMEPFirst =
    find(PickAbove1500First(:,2)>BMEPFirst(Kic,1));
    Pi1500AFuelFirst = PickAbove1500First(Pi1500ABMEPFirst,1);
    PickAboveFirst(Kic,1) = min(Pi1500AFuelFirst);

elseif AboveRPMFirst(Kic,1) == 1750

    Pi1750ABMEPFirst =
    find(PickAbove1750First(:,2)>BMEPFirst(Kic,1));
    Pi1750AFuelFirst = PickAbove1750First(Pi1750ABMEPFirst,1);
    PickAboveFirst(Kic,1) = min(Pi1750AFuelFirst);

elseif AboveRPMFirst(Kic,1) == 2000

    Pi2000ABMEPFirst =
    find(PickAbove2000First(:,2)>BMEPFirst(Kic,1));
    Pi2000AFuelFirst = PickAbove2000First(Pi2000ABMEPFirst,1);
    PickAboveFirst(Kic,1) = min(Pi2000AFuelFirst);

elseif AboveRPMFirst(Kic,1) == 2500

    Pi2500ABMEPFirst =
    find(PickAbove2500First(:,2)>BMEPFirst(Kic,1));
    Pi2500AFuelFirst = PickAbove2500First(Pi2500ABMEPFirst,1);
    PickAboveFirst(Kic,1) = min(Pi2500AFuelFirst);

elseif AboveRPMFirst(Kic,1) == 3000

    Pi3000ABMEPFirst =
    find(PickAbove3000First(:,2)>BMEPFirst(Kic,1));
    Pi3000AFuelFirst = PickAbove3000First(Pi3000ABMEPFirst,1);
    PickAboveFirst(Kic,1) = min(Pi3000AFuelFirst);

elseif AboveRPMFirst(Kic,1) == 3500

```

```

        Pi3500ABMEPFirst =
find(PickAbove3500First(:,2)>BMEPFirst(Kic,1));
        Pi3500AFuelFirst = PickAbove3500First(Pi3500ABMEPFirst,1);
        PickAboveFirst(Kic,1) = min(Pi3500AFuelFirst);

    elseif AboveRPMFirst(Kic,1) == 4000

        Pi4000ABEMPFirst =
find(PickAbove4000First(:,2)>BMEPFirst(Kic,1));
        Pi4000AFuelFirst = PickAbove4000First(Pi4000ABEMPFirst,1);
        PickAboveFirst(Kic,1) = min(Pi4000AFuelFirst);

    end
end

```

```

for Pic = 1:length(EngineSpeedFirst)

    if BelowRPMFirst(Pic,1) == 500

        Pi500BBMEPFirst =
find(PickBelow500First(:,2)>BMEPFirst(Pic,1));
        Pi500BFuelFirst = PickBelow500First(Pi500BBMEPFirst,1);
        PickBelowFirst(Pic,1) = min(Pi500BFuelFirst);

    elseif BelowRPMFirst(Pic,1) == 850

        Pi850BBMEPFirst =
find(PickBelow850First(:,2)>BMEPFirst(Pic,1));
        Pi850BFuelFirst = PickBelow850First(Pi850BBMEPFirst,1);
        PickBelowFirst(Pic,1) = min(Pi850BFuelFirst);

    elseif BelowRPMFirst(Pic,1) == 1000

        Pi1000BBMEPFirst =
find(PickBelow1000First(:,2)>BMEPFirst(Pic,1));
        Pi1000BFuelFirst = PickBelow1000First(Pi1000BBMEPFirst,1);
        PickBelowFirst(Pic,1) = min(Pi1000BFuelFirst);

    elseif BelowRPMFirst(Pic,1) == 1250

        Pi1250BBMEPFirst =
find(PickBelow1250First(:,2)>BMEPFirst(Pic,1));
        Pi1250BFuelFirst = PickBelow1250First(Pi1250BBMEPFirst,1);
        PickBelowFirst(Pic,1) = min(Pi1250BFuelFirst);

    elseif BelowRPMFirst(Pic,1) == 1500

        Pi1500BBMEPFirst =
find(PickBelow1500First(:,2)>BMEPFirst(Pic,1));
        Pi1500BFuelFirst = PickBelow1500First(Pi1500BBMEPFirst,1);
        PickBelowFirst(Pic,1) = min(Pi1500BFuelFirst);
    end
end

```

```

elseif BelowRPMFirst(Pic,1) == 1750

    Pi1750BBMEPFirst =
find(PickBelow1750First(:,2)>BMEPFirst(Pic,1));
    Pi1750BFuelFirst = PickBelow1750First(Pi1750BBMEPFirst,1);
    PickBelowFirst(Pic,1) = min(Pi1750BFuelFirst);

elseif BelowRPMFirst(Pic,1) == 2000

    Pi2000BBMEPFirst =
find(PickBelow2000First(:,2)>BMEPFirst(Pic,1));
    Pi2000BFuelFirst = PickBelow2000First(Pi2000BBMEPFirst,1);
    PickBelowFirst(Pic,1) = min(Pi2000BFuelFirst);

elseif BelowRPMFirst(Pic,1) == 2500

    Pi2500BBMEPFirst =
find(PickBelow2500First(:,2)>BMEPFirst(Pic,1));
    Pi2500BFuelFirst = PickBelow2500First(Pi2500BBMEPFirst,1);
    PickBelowFirst(Pic,1) = min(Pi2500BFuelFirst);

elseif BelowRPMFirst(Pic,1) == 3000

    Pi3000BBMEPFirst =
find(PickBelow3000First(:,2)>BMEPFirst(Pic,1));
    Pi3000BFuelFirst = PickBelow3000First(Pi3000BBMEPFirst,1);
    PickBelowFirst(Pic,1) = min(Pi3000BFuelFirst);

elseif BelowRPMFirst(Pic,1) == 3500

    Pi3500BBMEPFirst =
find(PickBelow3500First(:,2)>BMEPFirst(Pic,1));
    Pi3500BFuelFirst = PickBelow3500First(Pi3500BBMEPFirst,1);
    PickBelowFirst(Pic,1) = min(Pi3500BFuelFirst);

elseif BelowRPMFirst(Pic,1) == 4000

    Pi4000BBMEPFirst =
find(PickBelow4000First(:,2)>BMEPFirst(Pic,1));
    Pi4000BFuelFirst = PickBelow4000First(Pi4000BBMEPFirst,1);
    PickBelowFirst(Pic,1) = min(Pi4000BFuelFirst);

end

end

```

Fuel flow Calculated

```

FuelFlowCalcFirst = zeros(length(EngineSpeedFirst),1);

for fcf = 1:length(EngineSpeedFirst)

```

```

    FuelFlowCalcFirst(fcf,1) = (((EngineSpeedFirst(fcf,1)-
BelowRPMFirst(fcf,1))/(AboveRPMFirst(fcf,1)-
BelowRPMFirst(fcf,1)))*(PickAboveFirst(fcf,1)-PickBelowFirst(fcf,1))) +
PickBelowFirst(fcf,1);

end

```

Fuel usage for each

```

FuelUsageUrbanFirst = sum(FuelFlowCalcFirst(1:780,1));

FuelUsageEUFirst = sum(FuelFlowCalcFirst(781:end,1));

TotalFuelUseFirst = sum(FuelFlowCalcFirst);

FirstGearTable(:,8) = FuelFlowCalcFirst;

```

Carbon Dioxide Production

```

CarbonDMulti = 3.082227; % Stoichiometric fuel/air mass
multiplier
CarbonDioxideProducedFirst =
(CarbonDMulti*TotalFuelUseFirst)/TotalDistance;

```

Gear Specific fuel usage

```

Idlefuel = zeros(length(TableSetup(:,1)),1);
Coastfuel = zeros(length(TableSetup(:,1)),1);
fuelFirstGear = zeros(length(TableSetup(:,1)),1);
fuelSecGear = zeros(length(TableSetup(:,1)),1);
fuelThirdGear = zeros(length(TableSetup(:,1)),1);
fuelFourthGear = zeros(length(TableSetup(:,1)),1);
fuelFifthGear = zeros(length(TableSetup(:,1)),1);
fuelSixthGear = zeros(length(TableSetup(:,1)),1);

for gf = 1:length(TableSetup(:,1))

    if TableSetup(gf,5) == 1 && TableSetup(gf,3) > 0

        fuelFirstGear(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 2

        fuelSecGear(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 3

```

```

        fuelThirdGear(gf,1) = FirstGearTable(gf,8);

elseif TableSetup(gf,5) == 4

        fuelFourthGear(gf,1) = FirstGearTable(gf,8);

elseif TableSetup(gf,5) == 5

        fuelFifthGear(gf,1) = FirstGearTable(gf,8);

elseif TableSetup(gf,5) == 6

        fuelSixthGear(gf,1) = FirstGearTable(gf,8);

elseif TableSetup(gf,5) == 0 || TableSetup(gf,5) == 1 &&
TableSetup(gf,3) == 0

        Idlefuel(gf,1) = FirstGearTable(gf,8);

elseif TableSetup(gf,5) == 0 && TableSetup(gf,3) > 0

        Coastfuel(gf,1) = FirstGearTable(gf,8);

end
end

IdleFuel1 = sum(Idlefuel);
CoastFuel1 = sum(Coastfuel);
FirstGearFuel1 = sum(fuelFirstGear);
SecondGearFuel1 = sum(fuelSecGear);
ThirdGearFuel1 = sum(fuelThirdGear);
FourthGearFuel1 = sum(fuelFourthGear);
FifthGearFuel1 = sum(fuelFifthGear);
SixthGearFuel1 = sum(fuelSixthGear);
TotalF1 = (FirstGearFuel1 + SecondGearFuel1 + ThirdGearFuel1 +
FourthGearFuel1 + FifthGearFuel1 + SixthGearFuel1 + IdleFuel1 +
CoastFuel1);

```

Power

```

angularSFirst = TableSetup(:,22).*((2*pi)/60);
FirstGearTable(:,9) = angularSFirst;

powerFirst = (angularSFirst.*TableSetup(:,26))./1000;
FirstGearTable(:,10) = powerFirst;

```

BSFC

```

bsfcFirst = zeros(length(TableSetup(:,1)),1);
bsfcFirst(:,1) = (FirstGearTable(:,8)./powerFirst).*3600;

```

```

FirstGearTable(:,11) = bsfcFirst;

FirstBSFC = zeros(length(TableSetup(:,1)),1);

% First BSFC

for fbf = 1:length(TableSetup(:,1))
    if TableSetup(fbf,5) == 1
        FirstBSFC(fbf,1) = bsfcFirst(fbf,1);
    else
        FirstBSFC(fbf,1) = 0;
    end
end

FB = FirstBSFC(:,1);
FB(FB == 0) = [];
FBSFC = mean(FB, 'all');
NewEngineSpeedsFirst(:,(Fi + 1348)) = FB;

```

3. Gear specific variable assignment ‘for loop’ principle script used for fuel usage across all testing.

```
Idlefuel = zeros(length(TableSetup(:,1)),1);
Coastfuel = zeros(length(TableSetup(:,1)),1);
fuelFirstGear = zeros(length(TableSetup(:,1)),1);
fuelSecGear = zeros(length(TableSetup(:,1)),1);
fuelThirdGear = zeros(length(TableSetup(:,1)),1);
fuelFourthGear = zeros(length(TableSetup(:,1)),1);
fuelFifthGear = zeros(length(TableSetup(:,1)),1);
fuelSixthGear = zeros(length(TableSetup(:,1)),1);
for gf = 1:length(TableSetup(:,1))

    if TableSetup(gf,5) == 1 && TableSetup(gf,3) > 0

        fuelFirstGear(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 2

        fuelSecGear(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 3

        fuelThirdGear(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 4

        fuelFourthGear(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 5

        fuelFifthGear(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 6

        fuelSixthGear(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 0 || TableSetup(gf,5) == 1 && TableSetup(gf,3) == 0

        Idlefuel(gf,1) = FirstGearTable(gf,8);

    elseif TableSetup(gf,5) == 0 && TableSetup(gf,3) > 0

        Coastfuel(gf,1) = FirstGearTable(gf,8);

    end
end
IdleFuel1 = sum(Idlefuel);
CoastFuel1 = sum(Coastfuel);
FirstGearFuel1 = sum(fuelFirstGear);
SecondGearFuel1 = sum(fuelSecGear);
ThirdGearFuel1 = sum(fuelThirdGear);
FourthGearFuel1 = sum(fuelFourthGear);
FifthGearFuel1 = sum(fuelFifthGear);
SixthGearFuel1 = sum(fuelSixthGear);
TotalF1 = (FirstGearFuel1 + SecondGearFuel1 + ThirdGearFuel1 + FourthGearFuel1 +
FifthGearFuel1 + SixthGearFuel1 + IdleFuel1 + CoastFuel1);
```

4. Final gear ratio and comparison table assignment.

%% Variables assignment

```
FV = round((((FirstGearRatio*(GMul*2))*1000)+2)/2);
SV = round((((SecondGearRatio*(GMul*2))*1000)+2)/2);
TV = round((((ThirdGearRatio*(GMul*2))*1000)+2)/2);
FiV = round((((FourthGearRatio*(GMul*2))*1000)+2)/2);
FiiV = round((((FifthGearRatio*(GMul*2))*1000)+2)/2);
SiV = round((((SixthGearRatio*(GMul*2))*1000)+2)/2);
```

```
FirstVariables(1:4,1) = FirstFG(1,:);
FirstVariables(1:4,2) = FirstFG(FV,:);
FirstVariables(1:4,3) = FirstFG(end,:);
```

```
SecondVariables(1:4,1) = SecondFG(1,:);
SecondVariables(1:4,2) = SecondFG(SV,:);
SecondVariables(1:4,3) = SecondFG(end,:);
```

```
ThirdVariables(1:4,1) = ThirdFG(1,:);
ThirdVariables(1:4,2) = ThirdFG(TV,:);
ThirdVariables(1:4,3) = ThirdFG(end,:);
```

```
FourthVariables(1:4,1) = FourthFG(1,:);
FourthVariables(1:4,2) = FourthFG(FiV,:);
FourthVariables(1:4,3) = FourthFG(end,:);
```

```
FifthVariables(1:4,1) = FifthFG(1,:);
FifthVariables(1:4,2) = FifthFG(FiiV,:);
FifthVariables(1:4,3) = FifthFG(end,:);
```

```
SixthVariables(1:4,1) = SixthFG(1,:);
SixthVariables(1:4,2) = SixthFG(SiV,:);
SixthVariables(1:4,3) = SixthFG(end,:);
```

```
fgm = find(min(FirstFG(:,2)));
FinalFirstGear = FirstFG(fgm,1);
```

```
sgm = find(min(SecondFG(:,2)));
FinalSecondGear = SecondFG(sgm,1);
```

```
tgm = find(min(ThirdFG(:,2)));
FinalThirdGear = ThirdFG(tgm,1);
```

```
fogm = find(min(FourthFG(:,2)));
FinalFourthGear = FourthFG(fogm,1);
```

```
figm = find(min(FifthFG(:,2)));
FinalFifthGear = FifthFG(figm,1);
```

```
sigm = find(min(SixthFG(:,2)));
FinalSixthGear = SixthFG(sigm,1);
```

```
FinalFirstFuel = FirstFG(fgm,1);
```

```

FinalSecondFuel = SecondFG(sgm,1);
FinalThirdFuel = ThirdFG(tgm,1);
FinalFourthFuel = FourthFG(fogm,1);
FinalFifthFuel = FifthFG(figm,1);
FinalSixthFuel = SixthFG(sigm,1);

```

```

FinalFirstBSFC = FirstFG(fgm,1);
FinalSecondBSFC = SecondFG(sgm,1);
FinalThirdBSFC = ThirdFG(tgm,1);
FinalFourthBSFC = FourthFG(fogm,1);
FinalFifthBSFC = FifthFG(figm,1);
FinalSixthBSFC = SixthFG(sigm,1);

```

Appendix I.

1. VW Passat additional graphs.

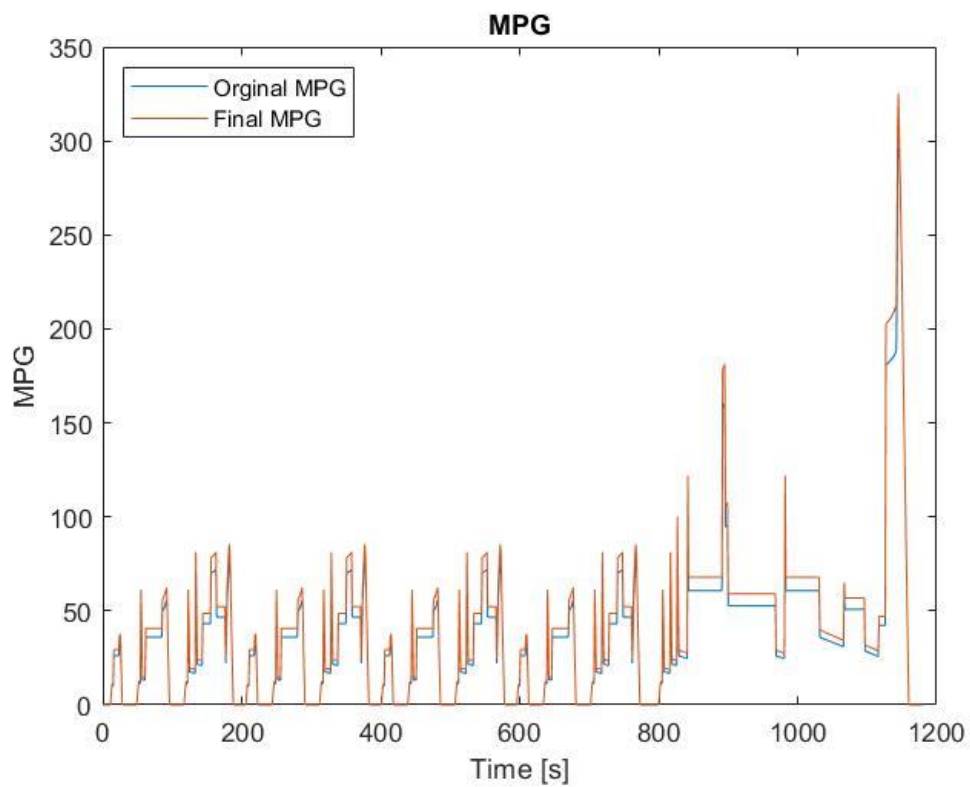


Figure 78 Miles per gallon of the VW Passat across the NEDC cycle.

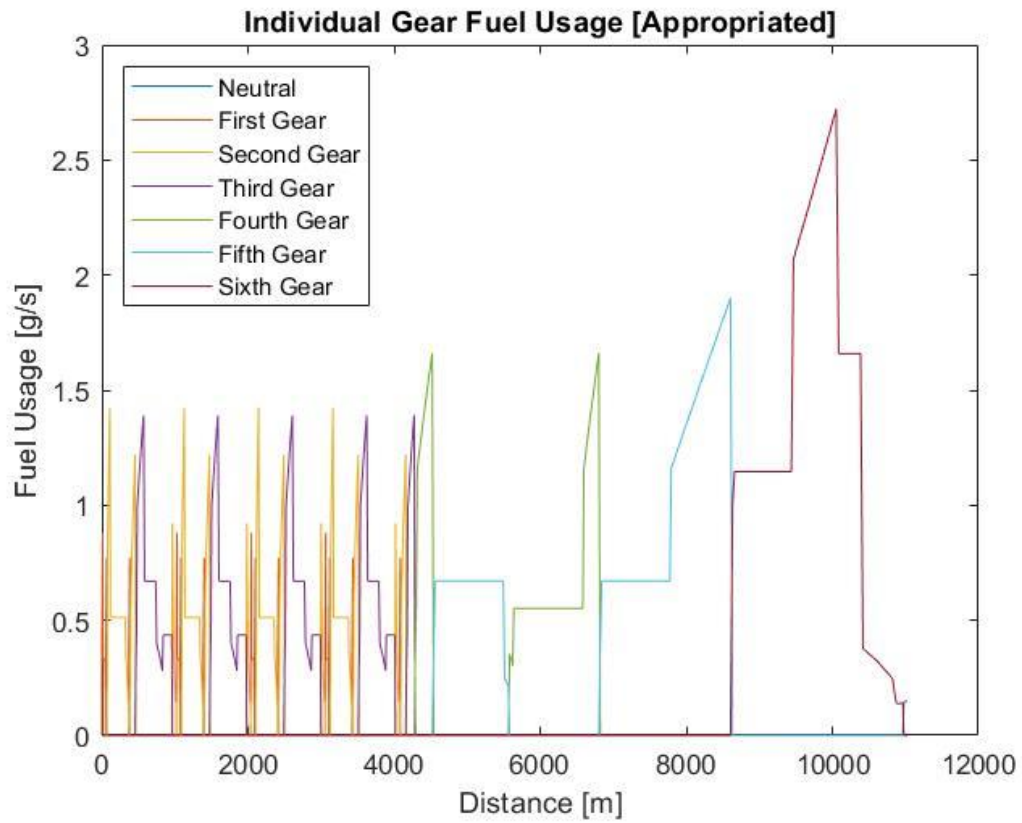


Figure 79 VW Passat appropriate individual gear fuel flow across the cycle as gear position changes.

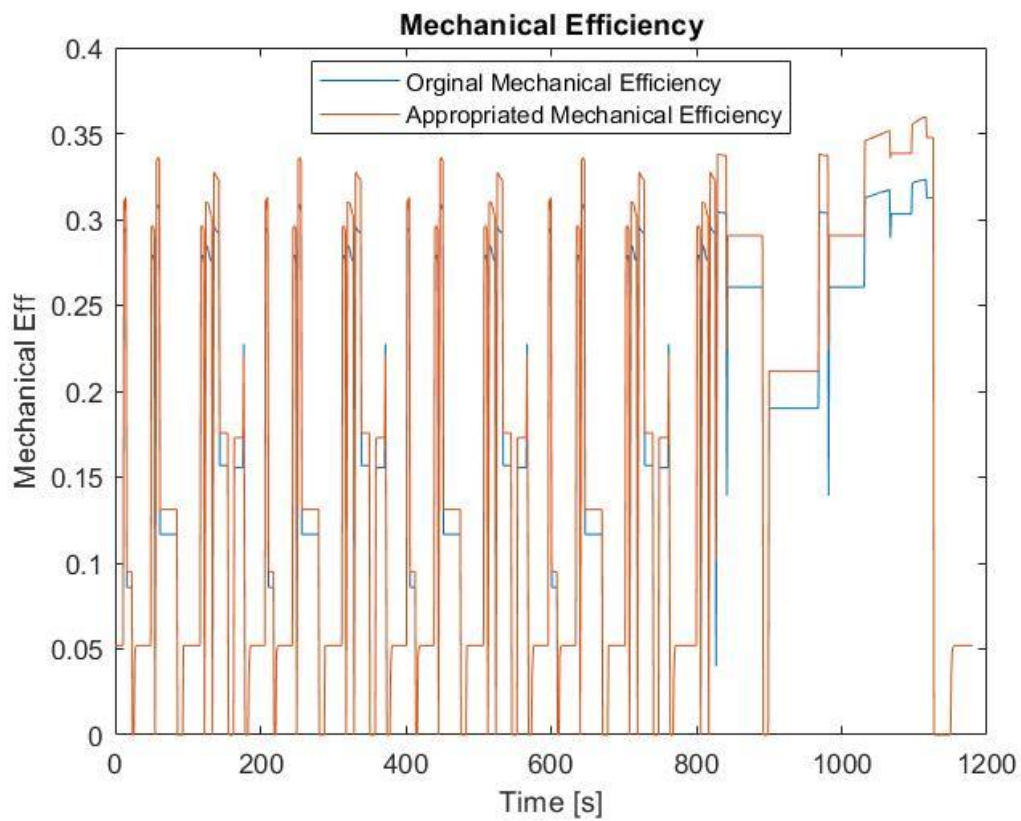


Figure 80 Mechanical efficiency percentage of the VW Passat with the original & appropriated gear ratios across the NEDC.

Minimum Speed New [MPH]	Minimum Speed Original [MPH]	Variation %
4.8764	4.3887	10
7.8438	7.0594	10
11.144	10.029	10
14.909	13.418	10
24.04	21.636	10
28.65	25.785	10

Table 32 Theorised change in minimum vehicle speed due to the changed gear ratio, assuming minimum engine of 800RPM.

Petrol car (TC48) and diesel car (TC49)

Band	CO2 emission (g/km)	Single 12 month payment	Single 12 month payment by Direct Debit	Total of 12 monthly instalments by Direct Debit	Single 6 month payment	Single 6 month payment by Direct Debit
A	Up to 100	£0	£0	N/A	N/A	N/A
B	101-110	£20	£20	£21	N/A	N/A
C	111-120	£30	£30	£31.50	N/A	N/A
D	121-130	£125	£125	£131.25	£68.75	£65.63
E	131-140	£150	£150	£152.25	£79.75	£76.13
F	141-150	£165	£165	£168	£88	£84
G	151-165	£205	£205	£210	£110	£105
H	166-175	£240	£240	£246.75	£129.25	£123.38
I	176-185	£265	£265	£273	£143	£136.50
J	186-200	£305	£305	£315	£165	£157.50
K*	201-225	£330	£330	£341.25	£178.75	£170.63
L	226-255	£565	£565	£582.75	£305.25	£291.38
M	Over 255	£580	£580	£598.50	£313.50	£299.25

Table 33 Tax brackets for petrol and diesel vehicles registered 01/03/01 & 31/03/17 (UK Government , 2020).

2. Ford Mondeo additional graphs.

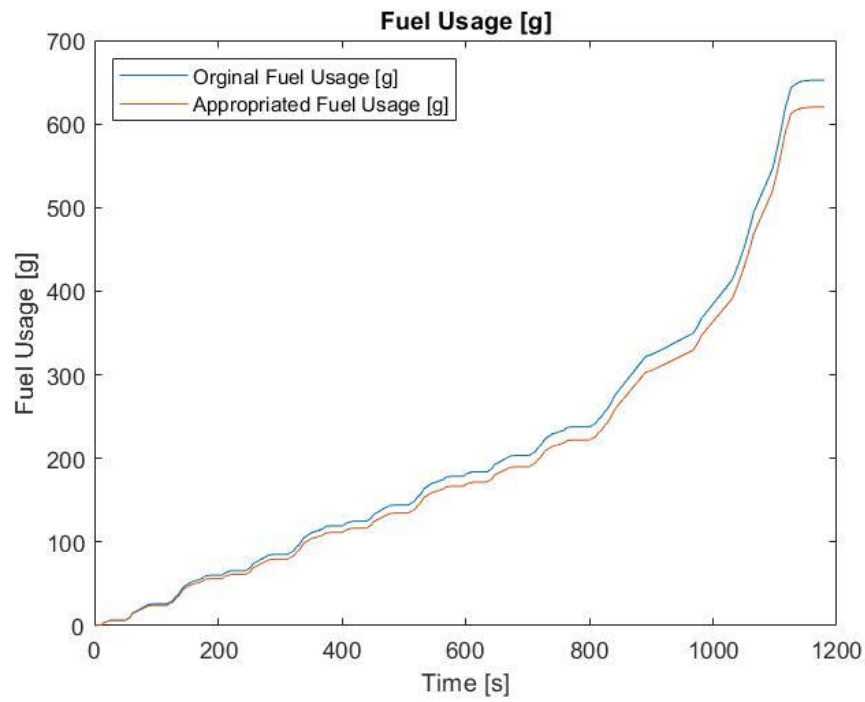


Figure 81 Ford Mondeo 2.0L EcoBlue fuel usage over time across the NEDC.

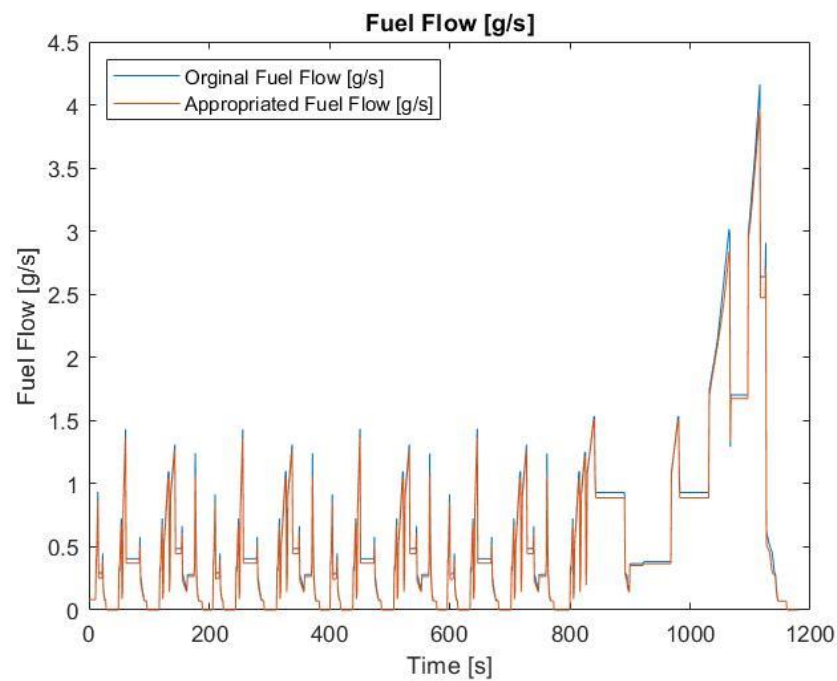


Figure 82 Ford Mondeo 2.0L EcoBlue fuel flow over the NEDC.

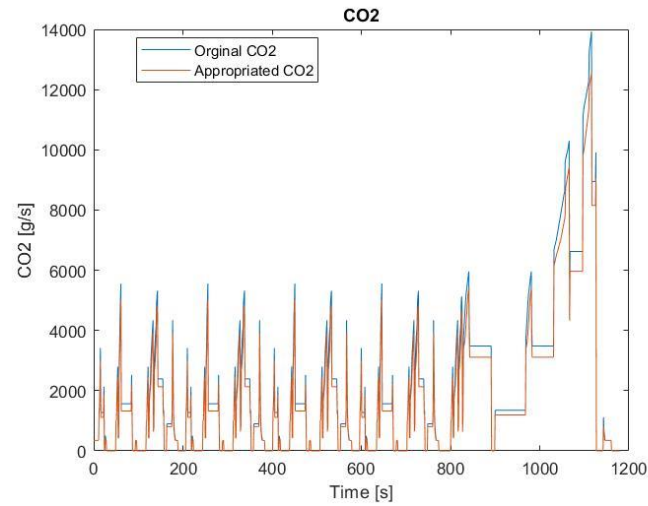


Figure 83 Ford Mondeo 2.0L EcoBlue CO2 flow across the NEDC.

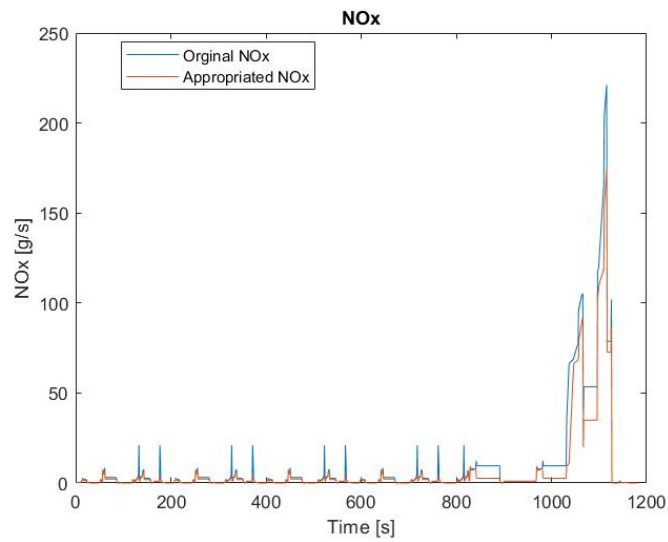


Figure 84 Ford Mondeo 2.0L EcoBlue NOx production across the NEDC.

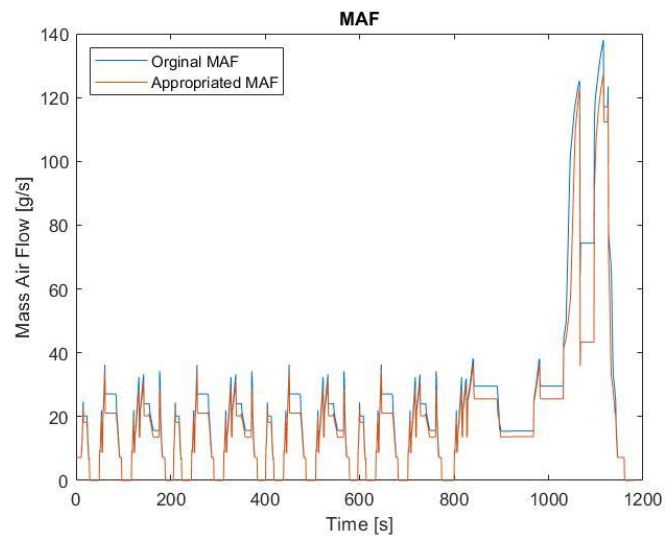


Figure 85 Ford Mondeo 2.0L EcoBlue MAF across the NEDC.

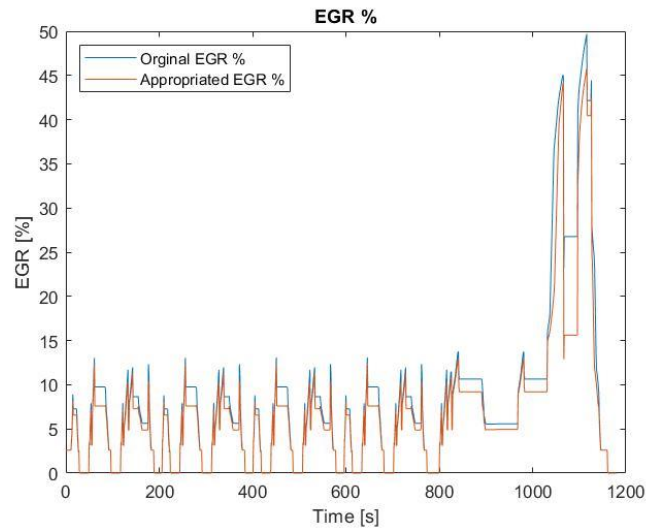


Figure 86 Ford Mondeo 2.0L EcoBlue EGR percentage across the NEDC.

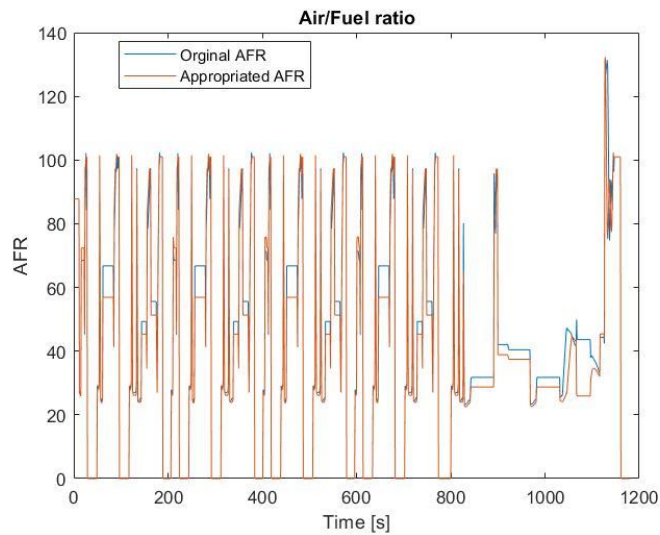


Figure 87 Ford Mondeo 2.0L EcoBlue AFR across the NEDC.

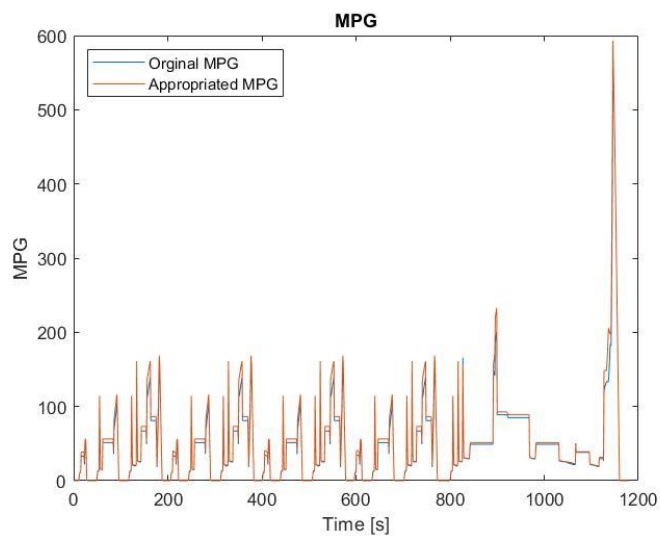


Figure 88 Ford Mondeo 2.0L EcoBlue MPG across the NEDC.

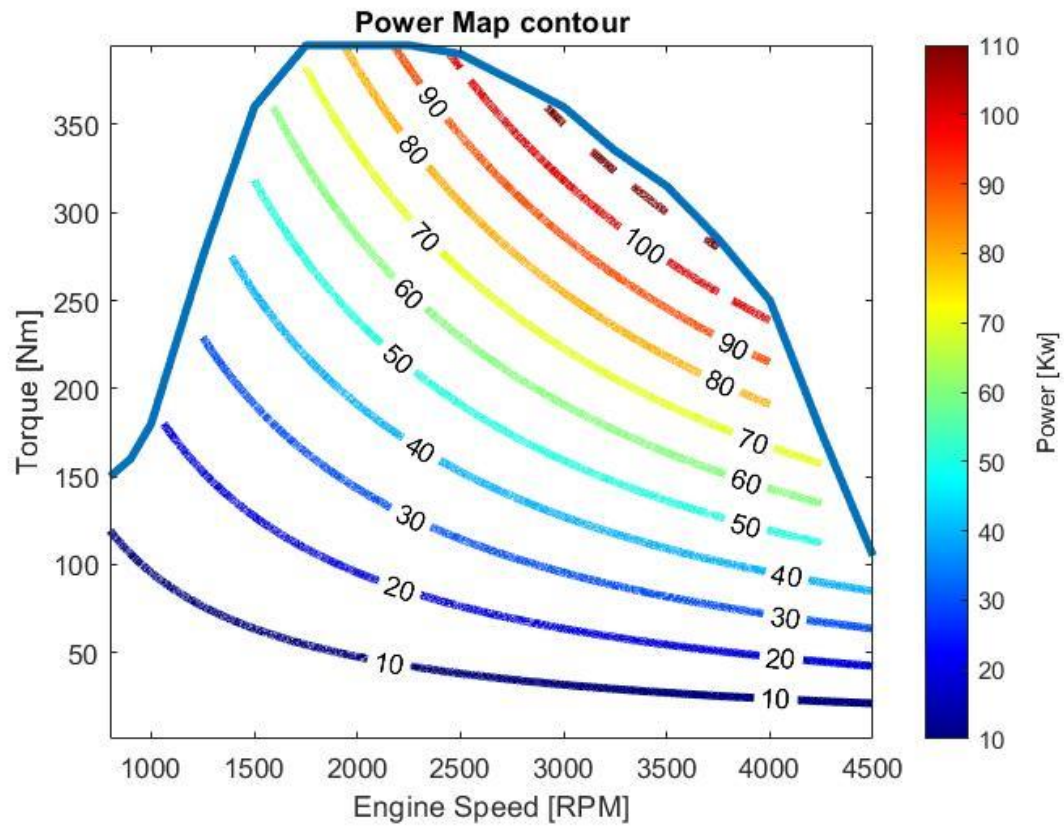


Figure 89 Ford Mondeo 2.0L EcoBlue Power contour against Torque and Engine speed.

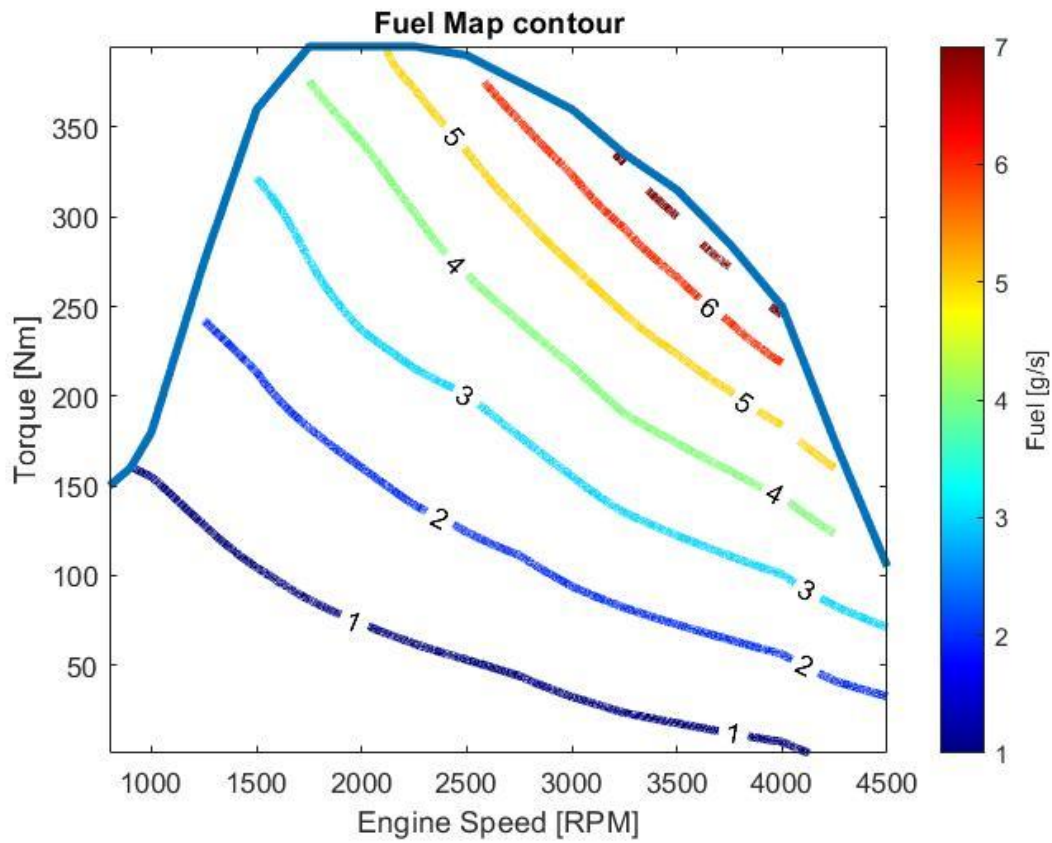


Figure 90 Ford Mondeo 2.0L EcoBlue Fuel flow map.

Minimum Speed New [MPH]	Minimum Speed Original [MPH]	Variation %
4.3714	3.9342	10
8.0238	7.2215	10
13.118	11.806	10
18.893	17.004	10
24.911	22.42	10
31.073	27.966	10

Figure 91 Ford Mondeo 2.0L EcoBlue theorised minimum vehicle speeds due to gear ratio changes.

Appendix J.

The following link is to the full specifications explained and regulated by the United Nations Economic Commission for Europe [UNECE]. This is the regulatory paper including the governing equations for the legal gear shift pattern, minimum engine speeds, power demands and full vehicle specifications that are to be adhered by any manufacture producing a newly registered vehicle within Europe.

Pages 398 – 418 include the specified vehicle speed at the time period for class 3 vehicles, of which both the tested vehicles are. They also are required to conduct the “extra-high” phase of the test.

Pages 428 – 435 are those concerning the gear shift pattern regulations and those governing minimum engine speeds and power requirements.

Two links will be shown in the hope it remains accessible.

<https://op.europa.eu/en/publication-detail/-/publication/7d1c640d-62d8-11e7-b2f2-01aa75ed71a1/language-en>

The pages in the following link may be slightly different to that in the above, but the content and regulations remain the same.

https://www.unece.org/fileadmin/DAM/trans/doc/2018/wp29other/EU_2017_1151.pdf

Appendix K.

The following script is that used for correcting the gear shift pattern, showing multiple iterations within it to ensure that a previous change hadn't caused another issue further on. Within the script there is called subscripts with the standard title of "EngineSpeedScript..." and these were used continuously to assess an issues with maintaining the engine speed above minimum because of a gear change.

It is shown at the top of the script the basic overview of how the initial pattern is constructed and as can be seen, the only change within this is the gear ratio that is being tested. This script was for the VW's third gear testing loop, hence within the "Speeds" section, only third gear has been changed from "...RatioO".

Third Gear Shift Checking Script

Testing

First iteration

```
Firstspeed = FSChange/(FirstRatio0*FinalRatio0);
Secondspeed = NminD3456/(SecondRatio0*FinalRatio0);
SecondSpeedDecel = ESMin2/(SecondRatio0*FinalRatio0);
Thirdspeed = NminD3456/(Third*FinalRatio0);
Fourthspeed = NminD3456/(FourthRatio0*FinalRatio0);
Fifthspeed = NminD3456/(FifthRatio0*FinalRatio0);
Sixthspeed = NminD3456/(SixthRatio0*FinalRatio0);

Gears = zeros(length(LayoutTable),1);

for Pi = 1:length(LayoutTable)

    if LayoutTable(Pi,4) == 0

        Gears(Pi,1) = 0;

    elseif LayoutTable(Pi,4) > 0 && LayoutTable(Pi,4) < Firstspeed*1.4

        Gears(Pi,1) = 1;

    elseif LayoutTable(Pi,4) > Firstspeed*1.4 && LayoutTable(Pi,4) <
Thirdspeed

        Gears(Pi,1) = 2;
```

```

elseif LayoutTable(Pi,4) > Secondspeed && LayoutTable(Pi,4) <
Fourthspeed

    Gears(Pi,1) = 3;

elseif LayoutTable(Pi,4) > Thirdspeed && LayoutTable(Pi,4) <
Fifthspeed

    Gears(Pi,1) = 4;

elseif LayoutTable(Pi,4) > Fourthspeed && LayoutTable(Pi,4) <
Sixthspeed

    Gears(Pi,1) = 5;

elseif LayoutTable(Pi,4) > Fifthspeed

    Gears(Pi,1) = 6;

end

end

for Sit = 1:length(LayoutTable)

    if TableSetup(Sit,6) < 0 && Gears(Sit,1) == 2 && LayoutTable(Sit,4)
> SecondSpeedDecel

        Gears(Sit,1) = 2;

    elseif TableSetup(Sit,6) <= 0 && LayoutTable(Sit,4) <
SecondSpeedDecel

        Gears(Sit,1) = 0;

    elseif Gears(Sit-1,1) == 2 && LayoutTable(Sit,4) < SecondSpeedDecel

        Gears(Sit,1) = 0;

    else

        Gears(Sit,1) = Gears(Sit,1);

    end

end
end

```

```

format short G

FirstG = zeros(1,1);

for FE = 2:length(TableSetup)
    if TableSetup(FE-1,3) == 0 && TableSetup(FE,3) > 0
        FirstG(FE-1,1) = 1;
    else
        FirstG(FE,1) = 0;
    end
end

fgs = find(FirstG == 1);

Gears(fgs,1) = 1;

TableSetup(:,7) = Gears;

% Engine Speed Test 1

EngineSpeedScriptFirstIt_TG

% Launch in First Gear Corrections

for KFG = 1:length(TableSetup)
    if TableSetup(KFG,7) == 0 && TableSetup(KFG,3) == 0
        EngineSpeedT(KFG,1) = EngineSpeedT(KFG,1);
    elseif TableSetup(KFG-1,7) == 1 &&
((TableSetup(KFG,4)*(FinalRatio0*FirstRatio0))*(60/(2*pi))) < 1800
        TableSetup(KFG,7) = 1;
    else
        TableSetup(KFG,7) = TableSetup(KFG,7);
    end
end

```

```

        end

    end

    % First Gear Changes for  $V > 0$ 

    EngineSpeedScriptSecondIt_VW_TG

    Check = EngineSpeedT;
    Check(:,2) = TableSetup(:,7);

    GearT = Check(:,2);

    % Any Below Minimum Engine Speed

    for te = 3:length(TableSetup)

        if EngineSpeedT(te,1) < MinRPM && GearT(te,1) >= 3

            GearT(te,1) = GearT(te,1) - 1;

        else

            GearT(te,1) = GearT(te,1);

        end

    end

    GearT(:,2) = TableSetup(:,8);

    EngineSpeedScriptThirdIt_VW_TG

    GearT(:,3) = EngineSpeedT;

    NGearT = Gears(:,1);

    % Held For Long enough

    FO = 7:length(TableSetup(:,1))-1;

    for ep = FO

```

```

    if GearT(ep,1) == 0

        GearT(ep,1) = GearT(ep,1);

    elseif GearT(ep-7,1) == GearT(ep,1) && GearT(ep-3,1) ~= GearT(ep,1)

        NGearT(ep-1,1) = NGearT(ep-7,1);
        NGearT(ep-2,1) = NGearT(ep-7,1);
        NGearT(ep-3,1) = NGearT(ep-7,1);
        NGearT(ep-4,1) = NGearT(ep-7,1);
        NGearT(ep-5,1) = NGearT(ep-7,1);
        NGearT(ep-6,1) = NGearT(ep-7,1);

    end

end

NGearT(:,2) = TableSetup(:,8);

for gv = 4:length(TableSetup)-1

    if Gears(gv,1) == 0

        NGearT(gv,1) = NGearT(gv,1);

        elseif ((NGearT(gv-2,1) + NGearT(gv-1,1) + NGearT(gv,1))/3) ==
NGearT(gv,1) && (NGearT(gv-3,1) == NGearT(gv,1) - 1 || NGearT(gv-3,1)
== NGearT(gv,1) + 1 || NGearT(gv-3,1) == NGearT(gv,1)) &&
(NGearT(gv+1,1) == NGearT(gv,1) - 1 || NGearT(gv+1,1) == NGearT(gv,1) +
1 || NGearT(gv+1,1) == NGearT(gv,1))

            NGearT(gv,1) = NGearT(gv,1);

        elseif ((NGearT(gv-3,1) + NGearT(gv-2,1) + NGearT(gv-1,1))/3) ==
(NGearT(gv,1) || (NGearT(gv,1) - 1) || (NGearT(gv,1) + 1))

            NGearT(gv,1) = NGearT(gv,1);

        else

            NGearT(gv+1,1) = NGearT(gv,1);

        end

    end

end

for qw = 1:length(TableSetup)

```

```

    if NGearT(qw,1) == 2 && NGearT(qw+1,1) == 1
        NGearT(qw+1,1) = 2;
    else
        NGearT(qw,1) = NGearT(qw,1);
    end
end

EngineSpeedScriptFourthIt_VW_TG

format shortG

NGearT(:,3) = EngineSpeedT;

for bv = 1:length(TableSetup)
    if NGearT(bv,1) == 1 && NGearT(bv,2) == 2 && NGearT(bv,3) > IdleRPM
        NGearT(bv,3) = IdleRPM;
    elseif NGearT(bv,1) == 2 && NGearT(bv,3) < (ESMin2*(60/(2*pi)))
        NGearT(bv,3) = IdleRPM;
        NGearT(bv,1) = 0;
    end
end

NGearT(:,4) = TableSetup(:,3);

EngineSpeedScriptFifthIt_VW_TG

for lo = 1:length(TableSetup)
    if NGearT(lo,3) < MinRPM && NGearT(lo,1) >= 3
        NGearT(lo,1) = NGearT(lo,1) - 1;
    elseif NGearT(lo,3) < (ESMin2*(60/(2*pi))) && NGearT(lo,1) == 2
        NGearT(lo,1) = 0;
    end
end

```

```

else
    NGearT(lo,1) = NGearT(lo,1);
end
end

NGearT(:,3) = EngineSpeedT;

for gy = 4:length(TableSetup)-1
    if Gears(gy,1) == 0
        NGearT(gy,1) = NGearT(gy,1);

        elseif ((NGearT(gy-2,1) + NGearT(gy-1,1) + NGearT(gy,1))/3) ==
NGearT(gy,1) && (NGearT(gy-3,1) == NGearT(gy,1) - 1 || NGearT(gy-3,1)
== NGearT(gy,1) + 1 || NGearT(gy-3,1) == NGearT(gy,1)) &&
(NGearT(gy+1,1) == NGearT(gy,1) - 1 || NGearT(gy+1,1) == NGearT(gy,1) +
1 || NGearT(gy+1) == NGearT(gy,1))

            NGearT(gy,1) = NGearT(gy,1);

            elseif ((NGearT(gy-3,1) + NGearT(gy-2,1) + NGearT(gy-1,1))/3) ==
(NGearT(gy,1) || (NGearT(gy,1) - 1) || (NGearT(gy,1) + 1))

                NGearT(gy,1) = NGearT(gy,1);

            else

                NGearT(gy+1,1) = NGearT(gy,1);

            end
        end
    end

EngineSpeedScriptSixthIt_VW_TG

NGearT(:,3) = EngineSpeedT;

BrakeOddsRemove_VW_TG

NGearsT(:,2) = NGearT(:,2);

EngineSpeedScriptSeventhIt_VW_TG

```

```

NGearST(:,3) = EngineSpeedT;
NGearST(:,4) = TableSetup(:,3);

for vc = 2:length(TableSetup)-1

    if NGearST(vc,1) >= 3 && NGearST(vc,3) < MinRPM

        NGearST(vc,1) = NGearST(vc,1) - 1;

    end

end

BrakeOddsRemove_VW_Second_TG

for mj = 2:length(TableSetup)-1

    if NGeT(mj-1,1) == 1 && NGeT(mj+1,1) == 2 && NGeT(mj,1) == 0

        NGeT(mj,1) = 1;

    else

        NGeT(mj,1) = NGeT(mj,1);

    end

end

EngineSpeedScriptEighthIt_VW_TG

NGeT(:,2) = TableSetup(:,8);
NGeT(:,3) = EngineSpeedT;
NGeT(:,4) = TableSetup(:,3);

for xc = 2:length(TableSetup)-2

    if NGeT(xc-1,1) > NGeT(xc,1) && NGeT(xc+1,1) == 0

        NGeT(xc,1) = 0; NGeT(xc+1,1) = NGeT(xc+2,1);

    end

end

for df = 11:length(TableSetup)

```

```

    if NGeT(df,1) < NGeT(df-3,1) && NGeT(df-10,1) == 0 && NGeT(df-5,1)
> NGeT(df,1)

        NGeT(df-9:df-1,1) = NGeT(df,1);

    else

        NGeT(df,1) = NGeT(df,1);

    end

end

Belows = find(NGeT(:,1) == 7);

NGeT(Belows,1) = 2;

for er = 3:length(TableSetup)-2

    if NGeT(er,1) > NGeT(er-2,1) && NGeT(er,1) == NGeT(er-1,1) &&
NGeT(er,1) > NGeT(er+1,1)

        NGeT(er,1) = NGeT(er+1,1); NGeT(er-1,1) = 0;

    else

        NGeT(er,1) = NGeT(er,1);

    end

end

for to = 2:length(TableSetup)-1

    if NGeT(to,1) < NGeT(to-1,1) && NGeT(to,1) < NGeT(to+1,1) &&
NGeT(to-1,1) == NGeT(to+1,1)

        NGeT(to,1) = NGeT(to-1,1);

    else

        NGeT(to,1) = NGeT(to,1);

    end

end

FinalisedGearsThird = NGeT(:,1);

```

Appendix L.

1. Volkswagen Passat 2.0L TFSI Estate 2006 WLTP additional graphs.

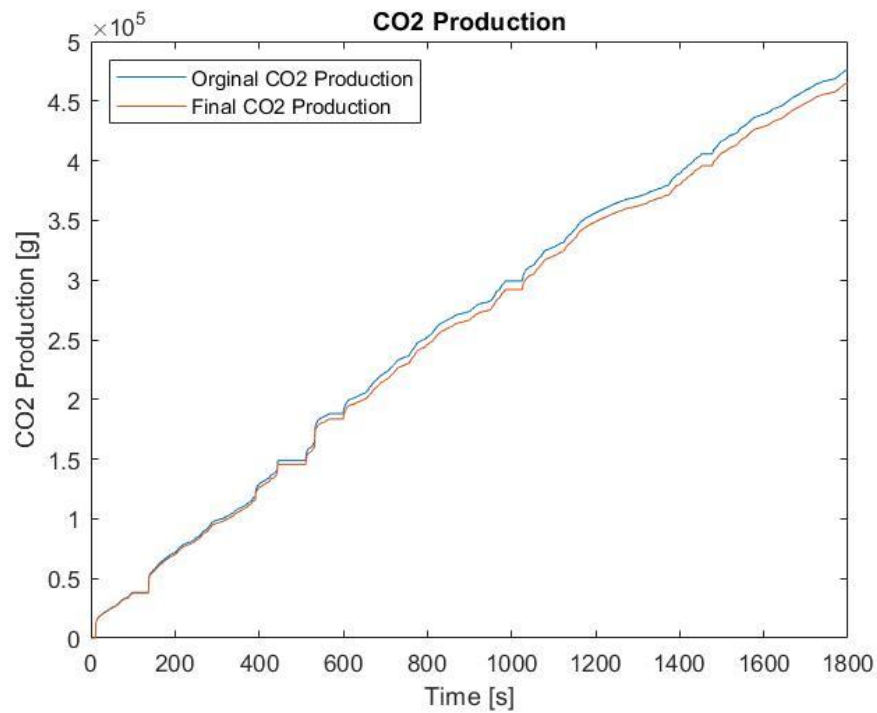


Figure 92 Volkswagen Passat 2.0L TFSI CO2 production across the WLTP.

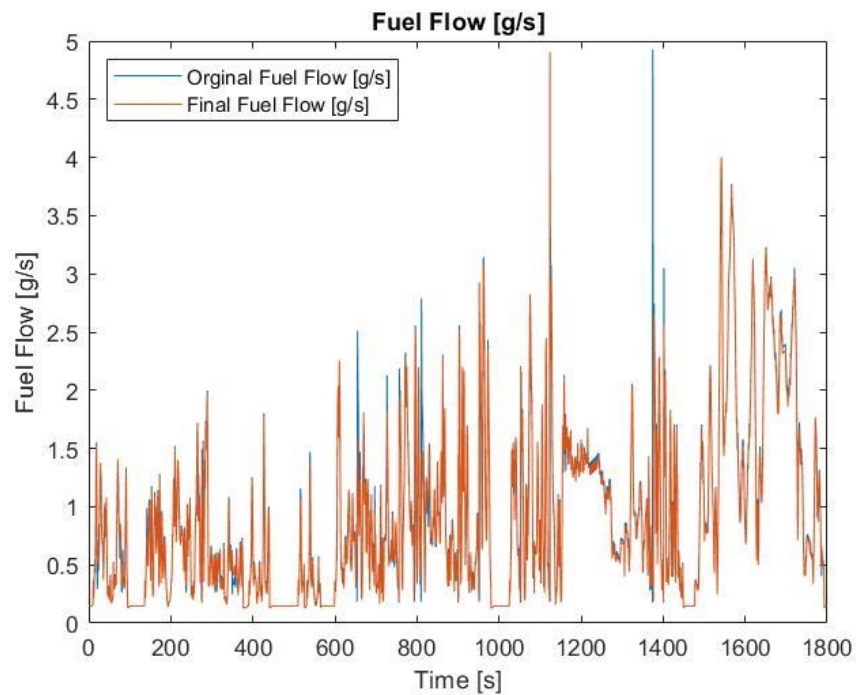


Figure 93 Volkswagen Passat 2.0L TFSI Fuel flow across the WLTP.

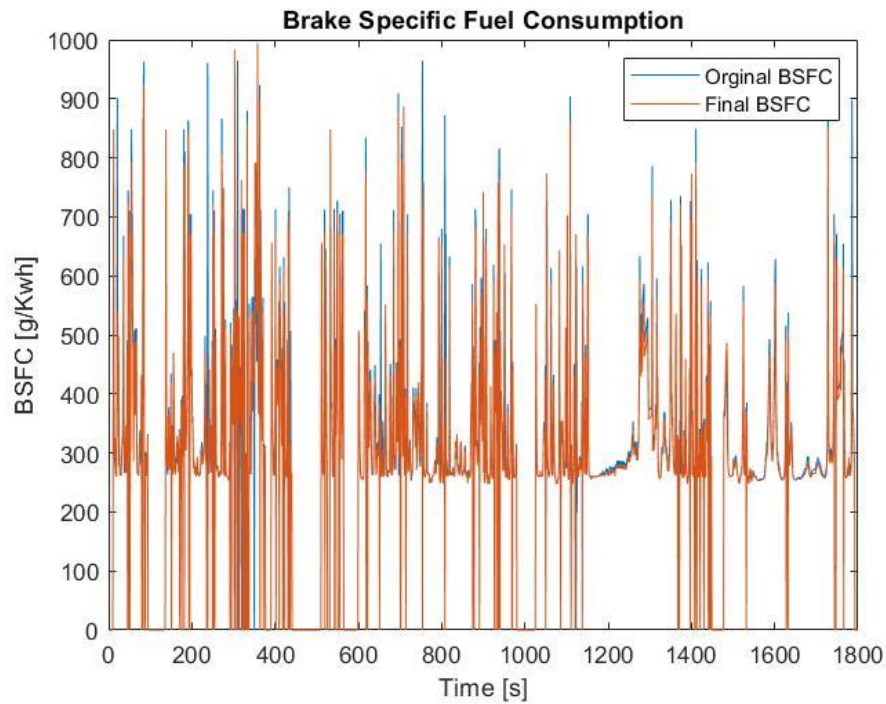


Figure 94 Volkswagen Passat 2.0L TFSI BSFC across the WLTP.

2. Ford Mondeo 2.0L EcoBlue Estate 2019 WLPT additional graphs.

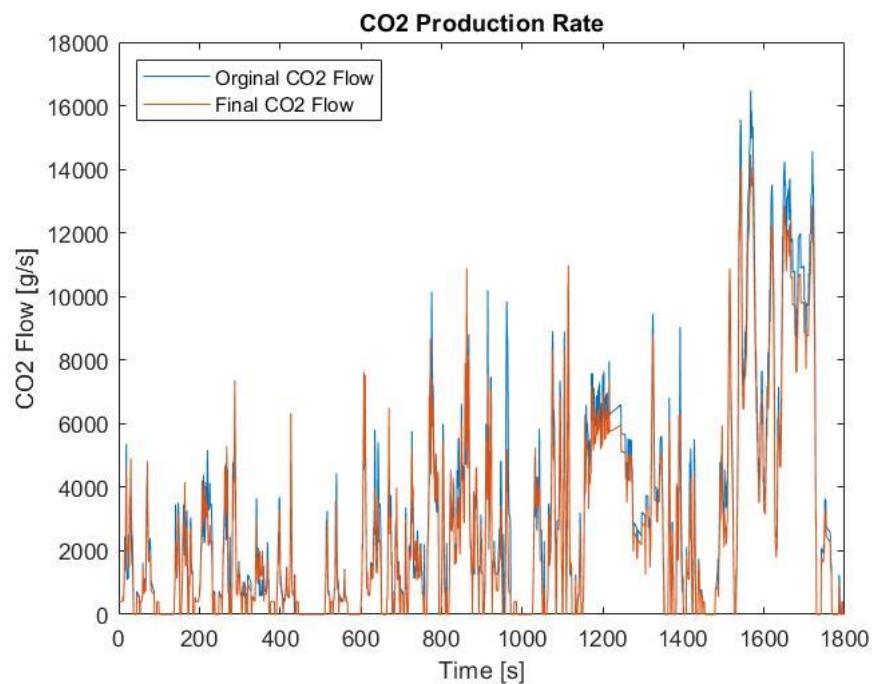


Figure 95 Ford Mondeo 2.0L EcoBlue Flow rate of CO2 Production across the WLTP.

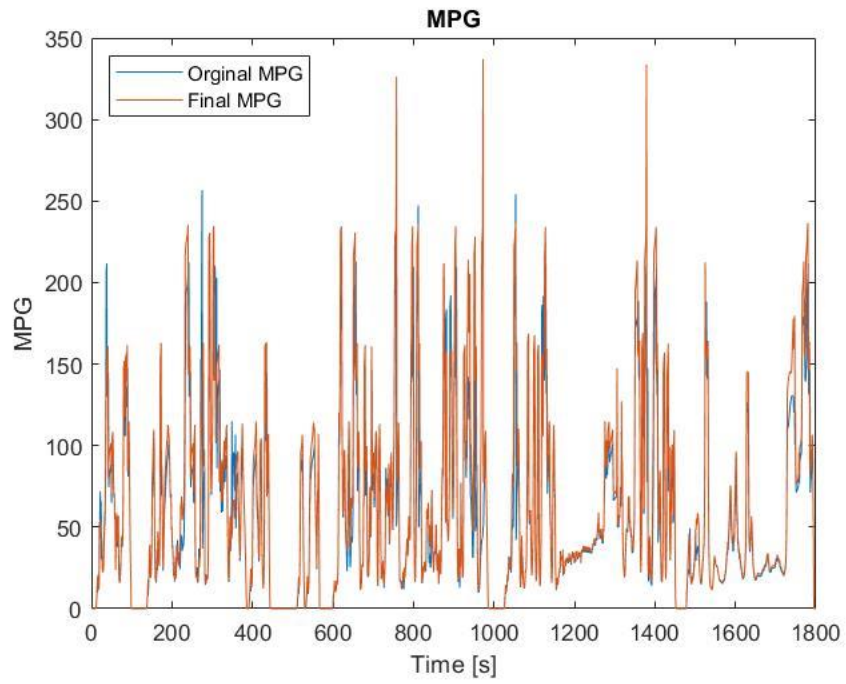


Figure 96 Ford Mondeo 2.0L EcoBlue MPG across the WLTP.