Optimisation of the vehicle transmission and the gear-shifting strategy for the minimum fuel consumption and the minimum nitrogen oxide emissions

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Optimisation of vehicle transmission and shifting strategy for minimum fuel consumption and NOx emissions

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Abstract

The paper outlines a computationally efficient analytical method to evaluate fuel consumption and NOx emissions during manoeuvres pertaining to the New European Drive Cycle (NEDC). An integrated optimisation procedure is also included in the analysis to minimise the Brake Specific Fuel Consumption (BSFC) and NOx emissions as objective functions. A set of optimum gear ratios are determined for 4, 5 and 6-speed transmission systems as the governing parameters in the optimisation process. The analysis highlights the determination of gear shifting strategies, based on the minimisation of either of these declared objective functions. A 7.5% reduction in the BSFC and 6.75% in NOx are attainable in the best case scenario for a 6-speed transmission and a gear shifting strategy, based on the least BSFC for the case of the chosen engine. The novel integrated analytical simulation and multi-objective optimisation has not hitherto been reported in literature. It provides the opportunity for objective intelligent-based approach to the use of gear shifting indicator technology. Results of this study also show that transmission optimization can act as an effective and inexpensive means to enhance fuel efficiency and reduce emissions.

Keywords

Optimum gear ratio, Gear shifting strategy, Fuel consumption, NOx emissions, NEDC

1. Introduction

Exhaust emissions associated with burning fossil fuels in internal combustion engines is a growing environmental concern. Many of the constituents of these
emissions contribute to greenhouse gases, which absorb heat in the atmosphere, leading to increased temperatures and global warming [1]. The increase in environmental greenhouse gases can cause result in flooding, drought, population displacement and significant damage to the eco-system [2]. Exhaust emissions also affect the quality of air with adverse health-related implications, particularly causing an increase in respiratory diseases [3].

Burning fossil fuels such as petrol and diesel not only affects the environment, but also leads to their depletion. There are significant difficulties in estimating how long reserves of fossil fuels would last [4].

For road transport alternatives to fossil fuel as a source of energy is emerging rapidly, such as hybrid or electrical propulsion systems. However, for the foreseeable future, and at least until mid-twenty first century internal combustion energy is expected to play a major role as a means of propulsion for road transport. Therefore, improved fuel efficiency and reduced emissions from IC engines and powertrain systems would remain an important research activity.

Legislations and directives regarding emission levels is becoming progressively more stringent as the automotive manufacturers strive to improve fuel efficiency with new innovative solutions or practical palliations. There have been many emergent technologies to reduce brake specific fuel consumption [5]. They include downsizing of powertrain systems, improved output power-to-weight ratio, turbo-charging, cylinder deactivation and stop-start in congested urban driving.

Fraser et al [6] carried out a drive cycle simulation with a D-class vehicle to investigate the fuel consumption benefits that could be accrued through downsizing. The original vehicle engine was a 2.0L TGDI and the ‘aggressively’ downsized selected engine was the 1.2L MAHLE engine. The authors’ simulations showed an almost 15% fuel saving.

Douglas et al [7] investigated the effect of Cylinder Deactivation (CDA) and Controlled Auto Ignition (CAI) on fuel consumption and emissions. Cylinder Deactivation (CDA) is used during low load conditions, when a number of cylinders are deactivated. This constitutes an effective engine downsizing. To maintain the engine torque with fewer cylinders, the fuel and air supplies need to be increased through application of increased throttle. Therefore, the pressure in the active
cylinders is increased, resulting in more efficient combustion. Closed valves of the
de-activated cylinders reduce the pumping losses of the engine, thus increasing its
overall efficiency. Additional fuel savings can also be accrued with a reduction in the
effective surface area of the cylinders. Therefore, less heat is lost through conduction.

Controlled Auto Ignition is a combustion strategy in which fuel and air are premixed
and ignited through air-fuel compression. Ignition occurs at multiple points, resulting
in a rapid burn rate. This controlled ignition leads to lower cylinder temperatures due
to internal EGR. The benefits of CAI are increased efficiencies and lower NOx, CO₂
and Particulate emissions. The results from drive cycle simulations on engines using
both CDA and CAI showed a fuel consumption saving of 10% and a reduction of 28%
in NOx emissions during the NEDC (New European Drive Cycle [8]).

Hybrid powertrains are an alternative approach. They also make use of energy
recovery systems to store some of the otherwise parasitic energy losses, and store
and recover the same for useful purposes, including for propulsion. Three different
hybrid systems with stored energy in a battery, or a flywheel or as high pressure fluid
in a hydraulic system were analysed by Ding et al [9]. The simulation results
showed a decrease in fuel consumption during the NEDC by approximately 31%, 33%
and 27.5% respectively [9].

Gear Shift Indicators are devices which are designed to indicate to the driver when a
gear shift should be made. The on-board computer calculates the fuel consumption,
when in any gear and suggests a shift in accordance with the lowest attainable fuel
consumption and emissions. Vagg et al [9] showed through simulations that the
vehicle fuel consumption could be reduced by 4.3% undertaking this approach. CO₂
emissions could also be reduced by 4.5% during the NEDC [9]. Norris et al [10]
showed that the gear shift indicator is able to reduce the fuel consumption by 4%
and 7% for a MINI Cooper and a Ford Transit van respectively. The Volkswagen Golf
tested in the same paper showed little improvement [10]. The fuel savings depend
highly on the vehicle and the shifting strategy. The use of shift indicators does not
require any significant modifications to the vehicle or engine. Thus, they make a
simple, inexpensive and effective way to reduce fuel consumption and emissions.

This paper investigates the gear ratios and shifting strategy in a simultaneous
manner in order to obtain an optimum design, which has not hitherto been studied in
combination. These factors can shift the engine operating point to a more efficient region, reducing fuel consumption and NOX emissions. A numerical method is developed to calculate the first gear ratio to provide adequate gradeability and a top gear ratio to reduce fuel consumption in highway driving. The intervening gear ratios are initially assumed to be equally-spaced. Subsequently, a range of new gear ratios instead of these initial intervening values are calculated. The fuel consumption, NOX emissions and 0-60 Mph acceleration times are calculated for each gear ratio combination. A multi-objective optimisation approach is used to find the optimum gearbox configuration for the specified range of gear ratios. This optimum can either provide the lowest fuel consumption, lowest NOX emissions or a trade-off between these objective functions. The 0-60 Mph acceleration times are intended to show how vehicle performance would be affected by the optimum gearbox configurations, as this is an important driveability metric. The optimum gearbox concepts do not consider any design constraints. Therefore, they are intended to be used as a target and a starting point for the transmission designers.

Simulations are carried out using the NEDC and the savings made are compared with the original gearbox currently fitted to the studied vehicle. The results show that with the addition of another gear pair, optimisation of the gear ratios and changes to the gear shifting strategy, the fuel consumption and NOx emissions can potentially be reduced by up to 7.52% and 7.6% respectively. The 0-60 Mph acceleration times remain almost unchanged, so the vehicle transient performance can be maintained with the optimum designs. Results reveal that the optimization of transmission can be considered as effective and an inexpensive alternative approach for reducing the fuel consumption and emission levels.

2. Model Description

2.1. Longitudinal dynamics

The equation of vehicle longitudinal motion is [11]:

\[ \sum F = m_w a = F_x - (F_D + F_R + F_G) \]  \hspace{1cm} (1)

where, \( F_x \) is the tractive (motive) force, \( F_D \) the aerodynamic drag, \( F_R \) the rolling resistance and \( F_G \) is any gradient loading.
The vehicle traction force [11] includes the effect of the drivetrain inertias. As the vehicle accelerates, the drivetrain components also need to accelerate which leads to a lower acceleration value:

$$F_x = \frac{T_{\text{Eng}} R_n \eta}{r_w} - \frac{a}{r_w} I_{\text{eff}}$$  \hspace{1cm} (2)

A transaxle front wheel drive (FWD) vehicle is considered (figure 2). For this configuration the effective inertia is: [11]:

$$I_{\text{eff}} = (I_E + I_{F, C} + I_{T, m})(R_n R_{F, D})^2 + I_{T, out} R_{F, D}^2 + I_D + I_S + I_W$$  \hspace{1cm} (3)
Aerodynamic drag, acting on the front projected area of the vehicle, \( A_f \) at the forward speed \( v \) is [12]:

\[
F_D = \frac{1}{2} c_D \rho A_f v^2
\]  

(4)

The rolling resistance and the coefficient of friction are calculated as [12]:

\[
F_R = \mu m_v g \cos \theta_{road}
\]  

(5)

where \( \mu = 0.01 \left(1 + \frac{2.23694v}{147}\right) \), and positive angles correspond to uphill climbing and the negative angles represent downhill manoeuvres.

The gradient force is:

\[
F_G = m_v g \sin \theta_{road}
\]  

(6)

2.2. Selection of first gear ratio

The first gear ratio is selected in order to ensure an adequate vehicle hill start capability. It is also selected to provide a low creeping speed and avoid excessive clutch use in congested traffic [13]. For these reasons the first gear ratio is fixed and no further optimisation is carried out in this respect. A hill of 1-in-3 gradient (33%) is usually used to test vehicle hill start capabilities [14], which is the approach adopted
here. In hill climbing, the applied wheel torque maintains the required acceleration to overcome the resistive forces. The acceleration of the vehicle is assumed to be constant with its value taken as the lowest starting acceleration under the NEDC conditions (0.534 m/s² [15]). The initial engine speed is assumed to be 1000 rpm (by the clutch) with the engine torque at full load. Initially no drag, rolling resistance or inertial effects are taken into account, with iterations undertaken thereafter [11]:

\[ R_{1st\ estimated} = \frac{r_w m_v}{T_{Eng}} (a_{NEDC} + g \sin \theta_{max}) \]  

(7)

The velocity of the vehicle is calculated with this gear ratio at 1000 rpm, so that the maximum resistive force at the start of the manoeuvre becomes:

\[ v = \frac{1000 \times \pi r_w}{30 \times R_{1st\ estimated}} \]  

(8)

The resistive and inertial forces are thus obtained as [11]:

\[ R_{1st} = \frac{r_w}{T_{Eng}} \left[ \alpha \left( m_v + \frac{l_{eff}}{r_w^2} \right) + F_D + F_R + F_G \right] \]  

(9)

This process is repeated iteratively until the 1st gear ratio converges to within an error tolerance of \(10^{-4}\).

2.3. Selection of top gear ratio

Traditionally, the top gear ratio for a vehicle is selected to provide the maximum speed. This is limited by the engine power and the resistive forces, predominantly the aerodynamic drag when travelling at high speeds [13]. The aim of the optimisation is to reduce fuel consumption and NOx emissions. Therefore, the selection criteria for the top gear ratio are changed to achieve these aims, noting the maximum legislated speed limit. Here, the top gear ratio is selected so that it provides maximum efficiency at the maximum motorway legal speed. For the United Kingdom, this legal limit is 70 mph (31.3 m/s). Generally, the maximum efficiency region (lowest BSFC) for an engine falls between engine speeds of 2000 and 3000 rpm. Therefore, the top gear ratio can be selected as:

\[ R_{top} = \frac{2\pi r_w N_{Eng\ max} \eta}{60 \times v_{motorway}} \]  

(10)
2.4. The intervening gear ratios

After the first and top gear ratios are determined, it is possible to estimate a set of intervening gear ratios, followed by an optimization process. One may initially set these ratios at discrete equally-spaced intervals between the first and the top gear ratios. A range of intervening ratios can then be initially calculated. For each of these ratios a range is defined as a percentage above and below those initially assumed.

2.5. Gear Shifting Strategies:

(a) Fixed engine speed: In this shifting strategy, the gear is changed once the engine has reached a predefined speed. This speed would be different, depending upon the situation. For city and highway driving, most drivers would aim to keep the engine speed relatively low (below 2500rpm) as this generally attains a better fuel consumption through early up-shifting [16]. This type of fixed speed gear change is ideal for a drive cycle simulation, where fuel consumption and emissions are the most important objectives. For situations, where increased acceleration is required, such as overtaking or joining a highway, drivers tend to allow the engine to reach higher speeds before up-shifting (>3000) as this would result in a higher wheel speeds. This type of fixed speed gear changing is ideal for an accelerative manoeuvre

(b) Minimum Fuel consumption and NOx emissions (Drive Cycle): To ensure minimum fuel consumption or NOx emissions, a gear should be selected to achieve these intended outcomes. Each potential gear should be analysed to predict the repercussions for fuel consumption and/or NOx emissions according to the instantaneous prevailing conditions. The optimum prediction should also keep the engine speed between the idle and the maximum with engine torque not exceeding the full load.

In practice, the driver would not know the required gear selection for the lowest fuel consumption or NOx emissions *apriori*. Therefore, the vehicle needs to be fitted with a gear shift indicator device or an automated shifting system. Gear shift indicator devices are already in use in some road vehicles in order to reduce fuel consumption and achieve lower emissions. The vehicle’s on-board computer is used to calculate the best gear, depending on the current speed, load and throttle position. Then, the most suitable gear is displayed by the indicator [10]. For simulation purposes it is
assumed that the driver would follow the gear shift indicator or an automatic shifting system is employed.

3. Acceleration Manoeuvre

A model for the acceleration of the vehicle is needed in order to analyse the effects of each set of gear ratio combinations on the performance of the vehicle. An acceleration manoeuvre comprises a vehicle driven at full throttle along a straight flat road until a certain criterion is met. Most manufacturers quote a 0-60 mph time. This is the criterion used in the current study.

A vehicle start model is used with the vehicle travelling at its lowest forward velocity in 1st gear with the engine speed of 1000 rpm. The first gear ratio is fixed at this speed with an adequate hill-start capability. Therefore, the same starting procedure is used in all the reported simulations.

(a)- Simulation Methodology:

1. The accelerative manoeuvre is carried out at full throttle, taking the engine torque from the full load torque curve. The time history of acceleration, velocity, displacement and the traction force are obtained through successive integrations of the equation of motion (equation (1)).

2. During the simulation, the gear shifting strategy should be monitored to ascertain whether the gear needs to be changed. For the maximum engine torque gear shifting strategy, the engine torque at the next gear is calculated. If the calculated torque exceeds the current torque, then a change of gear is required. For a fixed engine speed gear shifting strategy, a gear change is necessary, if the engine speed is greater than that pre-specified.

3. Having calculated the time histories of the engine speed and torque, the BSFC and NOx levels corresponding to the simulated conditions can be obtained from three dimensional engine maps. The mass of burnt fuel and that of produced NOx during the specified manoeuvre can be calculated as:

\[
m_{fuel} = \int \frac{BSFC \times (\frac{2\pi}{60}) N_{Eng} T_{Eng}}{3.6 \times 10^6} \, dt \tag{11}
\]

\[
m_{NOx} = \int \frac{NOX \times (\frac{2\pi}{60}) N_{Eng} T_{Eng}}{3.6 \times 10^6} \, dt \tag{12}
\]
4. Drive Cycle Analysis

Drive cycles are a set of vehicle conditions which attempt to replicate actual road driving conditions. They are used to compare fuel consumption and emissions for various road vehicles. All vehicles destined for the European market must adhere to the Euro emissions' directives and legislations. The measured emissions are taken from a NEDC drive cycle test (Figure 3). The testing is usually carried out on a chassis dynamometer, because it is difficult to achieve consistent results in a road test, although as of 2017 new testing rules require that a road test is also carried out, using a new drive cycle called the World Harmonised Light Vehicles Test Procedure (WLTP) [17].

![Figure 3: The NEDC Drive Cycle [15]](image)

During simulations, it is assumed that the vehicle would follow the drive cycle exactly and the throttle response would be instantaneous. The fuel consumption and the NOx emissions produced during the cycle are calculated. These values are used in the optimisation process in order to find the optimum gear ratios for best fuel economy or the lowest NOx emissions.
**Simulation Methodology:**

1. As the drive cycle needs to be followed precisely, the vehicle velocity is known *apriori* at each step of simulation. Therefore, the required acceleration can be found simply as: \[ a = \frac{\Delta v}{\Delta t} \].

2. The required engine torque to propel the vehicle at the required velocity and acceleration can simply be calculated through rearranging the equation of motion (equation (1)).

3. The engine speed at any prevailing gear is obtained as:
\[
N_{Eng} = \frac{60v_R}{2\pi r_w} \tag{13}
\]

4. For minimum fuel consumption and/or minimum NOx emissions, the gear shifting strategy needs to predict the upcoming conditions in all potential gear ratios at a given point in the drive cycle. It is important to select the lowest gear ratio, but one which would maintain the engine speed with least torque. This forms the basis of the approach highlighted here. However, it would be necessary in the future to ensure no sudden changes in torque surge or fade would occur as this can lead to impulsive action, inducing a plethora of drive train NVH (noise, vibration and harshness) issues such as driveline clonk or exacerbated gear rattle [18-20].

5. With the engine speed and torque evaluated, the corresponding BSFC and NOx levels can be obtained by using two-dimensional interpolation of the engine map. The burnt mass of fuel and that of produced NOx during the time history can be calculated, using equations (11) and (12).

5. **Results and Discussion**

5.1. **Vehicle and engine data:**

The vehicle considered in this study is a front wheel drive 5-speed manual transmission C-Segment 1.6L vehicle with a 4-cylinder petrol engine. The pertinent data is listed in tables 1 and 2 [7]. The appropriate maps are also presented in figures 4 and 5. The vehicle was due for release in 2005, thus subjected to Euro 4 emission legislation (for all Euro emission directives see [21]).
Table 1: Vehicle and Engine Specifications

<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass (kerb) $m_v$</td>
<td>Type</td>
</tr>
<tr>
<td>1330 kg</td>
<td>4 Stroke Petrol SI</td>
</tr>
<tr>
<td>Drag Coefficient $c_D$</td>
<td>Cylinders</td>
</tr>
<tr>
<td>0.325</td>
<td>4</td>
</tr>
<tr>
<td>Frontal Area $A_f$</td>
<td>Volume</td>
</tr>
<tr>
<td>2.01 m$^2$</td>
<td>1.6 L</td>
</tr>
<tr>
<td>Tire Radius $r_w$</td>
<td>Max Engine Speed $N_{max}$</td>
</tr>
<tr>
<td>0.2978 m</td>
<td>7000 rpm</td>
</tr>
<tr>
<td>Wheel Inertia $I_w$</td>
<td>Idle Engine Speed $N_{idle}$</td>
</tr>
<tr>
<td>0.74 kgm$^2$</td>
<td>800 rpm</td>
</tr>
<tr>
<td>Drive Type</td>
<td>Engine Inertia $I_{eng}$</td>
</tr>
<tr>
<td>FWD</td>
<td>0.1224 kgm$^2$</td>
</tr>
</tbody>
</table>

Table 2: Transmission Specifications

<table>
<thead>
<tr>
<th>Transmission Specifications [7]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
</tr>
<tr>
<td>Manual (5 Speed)</td>
</tr>
<tr>
<td>Gear Ratio</td>
</tr>
<tr>
<td>1st Gear</td>
</tr>
<tr>
<td>2nd Gear</td>
</tr>
<tr>
<td>3rd Gear</td>
</tr>
<tr>
<td>4th Gear</td>
</tr>
<tr>
<td>5th Gear</td>
</tr>
<tr>
<td>Final Drive</td>
</tr>
</tbody>
</table>

The BSFC and NOx emissions' maps can be used to calculate the fuel consumption at any given engine speed and torque. The values are given in the units g/kWh, which can be converted to the more useful form of g/s by multiplying with the prevailing instantaneous engine power [22]:

$$m_{fuel} = BSFC \times P_{Eng} = \frac{BSFC \times \left(\frac{2\pi}{60}\right) N_{Eng} T_{Eng}}{3.6 \times 10^6}$$  \hspace{1cm} (14)

$$m_{NOx} = NOX \times P_{Eng} = \frac{NOX \times \left(\frac{2\pi}{60}\right) N_{Eng} T_{Eng}}{3.6 \times 10^6}$$  \hspace{1cm} (15)
5.2. Validation Against Measurements

(a)- Validation for acceleration manoeuvre: Experimental data for a '0-60' acceleration test presented by Douglas et al [7] is used to validate the simulation model, with gear changes up to the engine speed of 6700 rpm, with each gear change duration of 0.5s. Both these criteria were used in the simulation study.
Figure 6 shows the velocity-time graph comparison between measured (experimental) and the simulated results. The 0-60 mph acceleration time for the experiment was reported as 10.89s [7] and that for the simulation study is 10.27s. The results show good correlation with a 5.7% deviation from the measured data.

(b)- Validation for NEDC (Fuel Consumption and NOx Emission): An NEDC drive cycle measurement was also presented in [7]. This is used to validate the model predictions for fuel consumption and NOx emissions. In this baseline experimental test [7] the fixed engine speed gear shifting strategy is used in the NEDC experimental test with a gear up-shift when the engine speed of 2450 rpm is reached.

The testing of a NEDC drive cycle requires a cold start. Under these conditions, the engine is not running at its optimum operating temperature. This leads to increased friction and thus increased fuel consumption. The tests used for producing the engine maps are normally carried out on a 'hot' engine operating at its optimum temperature. Therefore, a difference in results is expected in the cold start region of the NEDC. To tackle this problem, Douglas et al [7] calibrated the NEDC simulation to an experimental test using the conditions:
\begin{align}
0 &< t < 80 \ (s) \quad \dot{m}_{\text{fuel,cold}} = 4 \times \dot{m}_{\text{fuel,hot}} \\
80 &< t < 230 \ (s) \quad \dot{m}_{\text{fuel,cold}} = 1.4 \times \dot{m}_{\text{fuel,hot}}
\end{align}

One should note that this is only an approximation of the anticipated higher fuel consumption in order to compensate for the deviation of the results at cold start. A more precise calibration equation or use of specifically designed engine maps can be used in order to obtain closer values to real conditions. Also, temperature calibration was not applied to the NOx model as the results were quite similar.

The fuel flow rate and the produced NOx at idle are estimated using the experimental graphs in [7]. The average fuel flow rate at idle is 0.156 g/s and the average rate of NOx generation is 0.001264 g/s.

![Figure 7: NEDC fuel consumption (measured versus predicted)](image)

Figure 7 shows a comparison between the predicted fuel consumption and the aforementioned experimental data, both instantaneous and in accumulative form. The cumulative fuel consumed was measured at 711g in [7], whilst the predicted
value from the current analysis is 641.2g. There is a difference of 10%, which constitutes an acceptable degree accuracy. The difference is likely to be due to certain simplifying assumptions in the model, such as the quasi-static tyre model, as well as engine maps which are constructed from steady state test conditions, rather than under transient conditions of the drive cycle. These transient conditions result in lower efficiencies which cause an increase in fuel consumption [23].

Figure 8: NEDC NOx emission (measured versus predicted)

Figure 8 shows the NOx emissions (pre-catalyst) comparison between that predicted here against that measured in [7] for instantaneous and total cumulative amounts over the NEDC. The measured total NOx emission is 28.3g, whilst the predicted value is 29.3g; a difference of 3.5%. Again the correlation is acceptable and the difference is expected to be due to the use of a steady state NOx map.
5.3. Optimisation process

The first task in the optimisation process is to determine an optimum set of gear ratios which would reduce fuel consumption and NOx emissions, whilst still maintaining the vehicle acceleration performance.

In addition, the number of gear stages in the transmission system is also altered to ascertain whether any additional reductions in fuel consumption or NOx could be accrued by considering an additional gear pair. As the number of gears increases, the gearbox cost, compactness, weight and complexity also increase. A 4-speed gearbox has been tested to see if removal of one set would accrue any tangible benefit. A 6-speed gearbox has also been tested to see if sufficient fuel consumption and NOx emission reductions would occur to justify the disadvantages arising from the aforementioned issues of cost and compact-light-weight.

The 1st gear ratio and the top gear ratios are fixed, as those are based on hill-start capability and efficient motorway driving (previously noted). The intervening gear ratios are initially assumed to be equally inter-spaced. The optimisation process is applied to these intervening gear ratios. A spread of these ratios ±10% their equally spaced values are used (5 increments for a 6-speed transmission). The percentage difference in performance (BSFC and NOx) between any combination of the chosen intervening gear ratios and the initial set is used as the optimization objective function(s).

The accuracy of predictions is dependent on the time step used in the analysis. To keep the simulation run time to a minimum, whilst still maintaining the good degree of accuracy, a conservative time step of 0.5s is used. A time step sensitivity analysis is carried out, the outcome of which shows that changing the time step near the selected value had little effect on the final outcome of an optimum set of gear ratios. With these optimal configuration(s) (depending on the set objective function BSFC and/or NOx), acceleration manoeuvres were carried out in order to show how much compromise is made in terms of acceleration performance by optimizing for fuel consumption and/or NOx. All the acceleration simulations use a fixed engine speed gear shifting strategy. The gears were shifted once the engine speed reached 6700 rpm.
(a)- Fixed speed shifting Strategy

A number of drive cycle simulations, using the NEDC are carried out. A fixed speed gear shifting strategy, similar to the one presented by Douglas et al. [7] is employed in order to compare the results with the original transmission configuration. The shifting speed of 2450 rpm has been used. Simulations were carried out to find the optimum number of gear pairs. Additionally, the intervening gear ratios are allowed to alter in the pre-specified range to obtain the most optimal configuration. Two optimum sets of selected ratios corresponding to minimum fuel consumption and minimum NOx emissions are obtained.

The purpose of exercise is to ascertain whether the fuel consumption and NOx emissions can be reduced when using 4, 5 or 6-speed gearboxes.

Table 3: Optimum Combination of fixed Gear Ratios for 4, 5 and 6-speed transmissions with least fuel consumption

<table>
<thead>
<tr>
<th></th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
<th>5th</th>
<th>6th</th>
<th>Fuel</th>
<th>NOx</th>
<th>Accel. Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 Speed</td>
<td>14.8862</td>
<td>12.0128</td>
<td>6.2597</td>
<td>2.9897</td>
<td></td>
<td></td>
<td>-3.74%</td>
<td>-2.28%</td>
<td>5.04% (595.4)</td>
</tr>
<tr>
<td>5 Speed</td>
<td>14.8862</td>
<td>10.7209</td>
<td>9.8318</td>
<td>5.6657</td>
<td>2.9897</td>
<td></td>
<td>-3.55%</td>
<td>-1.96%</td>
<td>7.19% (596.6)</td>
</tr>
<tr>
<td>6 Speed</td>
<td>14.8862</td>
<td>11.2562</td>
<td>11.1404</td>
<td>6.9735</td>
<td>5.9059</td>
<td>2.9897</td>
<td>-2.89%</td>
<td>-1.17%</td>
<td>2.95% (600.7)</td>
</tr>
<tr>
<td>Original Ratios</td>
<td>14.5183</td>
<td>7.8892</td>
<td>5.4418</td>
<td>3.9548</td>
<td>3.2578</td>
<td></td>
<td>0.00%</td>
<td>0.00%</td>
<td>0.00% (618.6)</td>
</tr>
</tbody>
</table>

The results in table 3 show the optimum combination of gear ratios for 4, 5 and 6-speed transmissions, yielding least BSFC. All these transmission configurations reduce fuel consumption and NOx emissions during the NEDC compared with the original configuration of the vehicle. The 4-speed gearbox shows the greatest reduction in BSFC and NOx with reductions of 3.74% and 2.28% respectively.

The results also show that the 0-60 mph acceleration time is increased for all of the optimum gearbox configurations. Therefore, there is a trade-off between the acceleration performance and improved BSFC and NOx. However, the percentage deterioration in acceleration performance is negligible, with the worst case scenario adding a mere 0.5s to the current installed vehicle configuration.
The optimum 2nd and 3rd gear ratios for the 6-speed and 5-speed transmission are very close. This suggests that one of these gear pairs should ideally be removed to reduce the manufacturing and assembly costs.

Table 4: Optimum Combination of fixed Gear Ratios for 4, 5 and 6-speed transmissions with the lowest NOx

<table>
<thead>
<tr>
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<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
<th>5th</th>
<th>6th</th>
<th>Fuel</th>
<th>NOx</th>
<th>Accel. Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 Speed</td>
<td>14.8862</td>
<td>11.4668</td>
<td>7.6507</td>
<td>2.9897</td>
<td></td>
<td></td>
<td>-2.89% (600.7)</td>
<td>-3.03% (28.21)</td>
<td>-4.89% (9.525)</td>
</tr>
<tr>
<td>5 Speed</td>
<td>14.8862</td>
<td>13.1033</td>
<td>9.8318</td>
<td>5.3675</td>
<td>2.9897</td>
<td></td>
<td>-2.41% (603.7)</td>
<td>-2.53% (28.35)</td>
<td>9.74% (10.990)</td>
</tr>
<tr>
<td>6 Speed</td>
<td>14.8862</td>
<td>11.2562</td>
<td>11.1404</td>
<td>7.3609</td>
<td>4.8321</td>
<td>2.9897</td>
<td>-1.87% (607.0)</td>
<td>-2.18% (28.46)</td>
<td>1.20% (10.135)</td>
</tr>
<tr>
<td>Original</td>
<td>14.5183</td>
<td>7.8892</td>
<td>5.4418</td>
<td>3.9548</td>
<td>3.2578</td>
<td></td>
<td>0.00% (618.6)</td>
<td>0.00% (29.09)</td>
<td>0.00% (10.015)</td>
</tr>
</tbody>
</table>

The European emissions legislations are becoming more stringent on NOx emissions. As an approach, the transmissions may be redesigned to reduce these emissions and meet the requirements of the evolving directives. The results in table 4 show the optimum combination of gear ratios for 4, 5 and 6-speed transmission systems with lowest NOx emissions. These indicate that all of the optimum alternatives can significantly improve the current vehicle transmission and have reduced fuel consumption. The 4-speed alternative shows the greatest improvement in reducing NOx and improving BSFC by 3.03% and 2.89% respectively.

The acceleration time (0-60 mph) has increased, when compared with the current configuration, for the cases of 5 and 6-speed transmissions, but the optimum 4-speed version in fact shows improved acceleration performance.

Although the various 4-speed configurations show the least fuel consumption and NOx emissions (based on the fixed speed shifting strategy), they can have other drawbacks from drivability and comfort points of view. With widely apart gear ratios large variations in performance can ensue, which makes timely and smooth gear shifting difficult, somewhat putting the onus on the driver of a manual system.

(b)- Metric-based Shifting Strategies

The previous section dealt with alternative transmission configurations (i.e. fixed ratio gear shifting of n-speed transmissions, n=4, 5 or 6). Two different 4-speed gearboxes were found to provide optimum performance based on fixed gear shifting strategies.
These two alternatives are not necessarily the most optimum, since other shifting strategies such as one based on the metric: minimum fuel consumption or the metric: minimum NOx emissions should be investigated.

(i)- Optimizing for minimum fuel consumption:

The approach adopted is to determine a new gearbox configuration which would yield the least BSFC.

Drive cycle simulations are carried out using the minimum BSFC as the adopted performance metric for the gear shifting strategy rather than up-shifting according to a predetermined vehicle/engine speed.

Table 5: Least Fuel Consumption Gear Ratios for 4,5 and 6-Speed transmissions (minimum BSFC gear shifting strategy)

<table>
<thead>
<tr>
<th></th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
<th>5th</th>
<th>6th</th>
<th>Fuel</th>
<th>NOx</th>
<th>Accel. Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 Speed</td>
<td>14.8862</td>
<td>9.8287</td>
<td>6.2597</td>
<td>2.9897</td>
<td></td>
<td></td>
<td>-5.91% (582.0)</td>
<td>-5.18% (27.58)</td>
<td>-0.90% (9.925)</td>
</tr>
<tr>
<td>5 Speed</td>
<td>14.8862</td>
<td>10.7209</td>
<td>8.0442</td>
<td>5.3675</td>
<td>2.9897</td>
<td></td>
<td>-6.94% (575.6)</td>
<td>-6.42% (27.22)</td>
<td>1.05% (10.120)</td>
</tr>
<tr>
<td>6 Speed</td>
<td>14.8862</td>
<td>11.2562</td>
<td>9.1149</td>
<td>6.9735</td>
<td>4.8321</td>
<td>2.9897</td>
<td>-7.51% (572.1)</td>
<td>-6.72% (27.14)</td>
<td>-0.35% (9.980)</td>
</tr>
<tr>
<td>Original Ratios</td>
<td>14.5183</td>
<td>7.8892</td>
<td>5.4418</td>
<td>3.9548</td>
<td>3.2578</td>
<td></td>
<td>0.00% (618.6)</td>
<td>0.00% (29.09)</td>
<td>0.00% (10.015)</td>
</tr>
</tbody>
</table>

The results are shown in table 5, indicating that the 6-speed transmission would achieve the least BSFC during the NEDC. Therefore, the gear ratio optimisation process is carried out for the 6-speed transmission system.

Thus far, all the simulation studies have used a discrete set of gear ratios and combinations. The simulation results from this section can be used in an optimisation process for a continuous range of gear ratios. AVL CAMEO is selected as the optimisation tool [24]. This is a multi-objective optimisation tool, thus suitable for the current study. It is based on genetic algorithm [24].

The matrix of results for all of the gear ratios and combinations from the drive cycle simulations are imported into the CAMEO, which fits the data to a continuous range of variables. An accurate data-fit is essential for the quality of the optimisation process as these fitted values are used within the optimisation itself. Figure 9 shows
the fitted versus the discretely obtained simulation results. A goodness of fit is achieved.

![Figure 9: Fuel Consumption measured and predicted Data in AVL CAMEO](image)

A single objective optimisation is carried out with AVL CAMEO to find the gear ratio set with the least fuel consumption. An optimum set of gear ratios is found as presented in Table 6.

**Table 6: Optimised Gear Ratios for the Minimum BSFC**

<table>
<thead>
<tr>
<th>Gear Change Method (Drive Cycle)</th>
<th>1&lt;sup&gt;st&lt;/sup&gt;</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
<th>5th</th>
<th>6&lt;sup&gt;th&lt;/sup&gt;</th>
<th>Fuel</th>
<th>NOx</th>
<th>Accel. Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum Fuel</td>
<td>Minimum Fuel</td>
<td>14.8862</td>
<td>11.7484</td>
<td>9.1149</td>
<td>7.0314</td>
<td>4.8321</td>
<td>2.9897</td>
<td>-7.52% (572.09)</td>
<td>-6.73% (27.13)</td>
</tr>
<tr>
<td>Original Transmission</td>
<td>Fixed Engine Speed</td>
<td>14.5183</td>
<td>7.8892</td>
<td>5.4418</td>
<td>3.9548</td>
<td>3.2578</td>
<td>0.00% (618.6)</td>
<td>0.00% (29.09)</td>
<td>0.00% (10.015)</td>
</tr>
</tbody>
</table>

The results show that a further improvement in BSFC can be achieved when gear shifting strategy is based on the metric-based gear ratios rather than those based merely on vehicle speed. The fuel consumption can be further reduced by 7.52%. The NOx emissions can also be reduced by 6.73%, whilst the vehicle acceleration performance is only slightly deteriorated by 0.45% compared with the original vehicle transmission and gear shifting strategy. As a 6-speed transmission would be more expensive to design and manufacture than the 4-speed alternatives (described in the previous section), then any reductions in BSFC and reductions in NOx emissions should justify the increased manufacturing costs.
Figure 10 shows the fuel consumption (g/s) map and a comparison of the engine operating points during the NEDC for the original gearbox configuration and the optimum 6-speed gearbox with gear shifting strategy based on the least BSFC. It can be seen that the optimum gearbox configuration shifts the engine operating points to a more efficient region of the map.

![Figure 10: Fuel Consumption map and engine operating points comparison during the NEDC](image)

**(ii)- Optimizing for the minimum NOx emission:**

For this optimisation the selected metric is the minimised NOx emissions. This is a desired outcome for meeting the stringent European emission legislation. NEDC simulations are carried out using the minimum NOx emission to determine the new gear shifting strategy for various speed transmissions.

<table>
<thead>
<tr>
<th>Table 7: Least NOx emissions for 4,5 and 6-Speed transmissions</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
</tr>
<tr>
<td>-----</td>
</tr>
<tr>
<td>4 Speed</td>
</tr>
<tr>
<td>5 Speed</td>
</tr>
<tr>
<td>6 Speed</td>
</tr>
<tr>
<td>Original Ratios</td>
</tr>
</tbody>
</table>
Table 7 shows that the least NOx emissions during the NEDC can be achieved with a 6-speed transmission using the minimum NOx emission gear shifting strategy. Therefore, gear ratio optimisation is carried out for the 6-speed transmission.

The matrix of discrete results for all of the gear ratios and combinations from the drive cycle simulations are imported into the AVL CAMEO so that an optimum result can be determined. CAMEO is then used to model and fit the simulation-based data. Figure 12 shows satisfactory goodness of fit to the discrete simulation results.

Figure 12: NOx Emission Measured and Predicted Data in AVL CAMEO

A single objective optimisation based on the least NOx emissions is carried out to find an optimum set of gear ratios. The results are presented in Table 8.

Table 8: Optimum Minimum NOx Emission Gear Ratios for 4, 5 and 6-speed transmissions

<table>
<thead>
<tr>
<th>Gear Change Method (Drive Cycle)</th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
<th>5th</th>
<th>6th</th>
<th>Fuel</th>
<th>NOx</th>
<th>Accel. Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum NOx</td>
<td>14.8862</td>
<td>11.7787</td>
<td>9.1149</td>
<td>6.9735</td>
<td>4.8321</td>
<td>2.9897</td>
<td>-4.65%</td>
<td>-7.60%</td>
<td>-0.30%</td>
</tr>
<tr>
<td>Original Transmission</td>
<td>14.5183</td>
<td>7.8892</td>
<td>5.4418</td>
<td>3.9548</td>
<td>3.2578</td>
<td></td>
<td>0.00%</td>
<td>0.00%</td>
<td>0.00%</td>
</tr>
</tbody>
</table>

The results show that the NOx emissions can be reduced by 7.6% and the fuel consumption by 4.65% in this case, as well as acceleration performance by 0.3%.
Figure 12 shows the engine NOx emission rate (g/s) map and a comparison of the engine operating points during the NEDC for the original gearbox configuration and the optimum 6-speed gearbox. It can be seen that the optimum gearbox configuration shifts the engine operating point to more efficient regions of the map with lower NOx emissions (g/s).

![Figure 12: NOx emission rate map and engine operating point comparison during the NEDC](image)

**(iii)- Trade-off between Minimum Fuel Consumption and NOx Emission:** Both BSFC and NOx emission are clearly important from a commercial as well as legal perspective. Therefore, a multi-objective optimisation would be the ideal approach, in this case with both these metrics as objective functions. The results of cases (i) and (ii) clearly show that with a unitary objective function and a given transmission configuration an appropriate gear shifting indicator can be developed. In the case of the multi-objective problem clearly a degree of trade-off or priority weighting should be used between the intended outcomes. In the case studied here tables 7 and 8 yield two sets of gear ratio outcomes. However, the vehicle can only be driven with a unique gear shifting strategy. In this case the minimum BSFC objective is chosen as the primary objective because larger reductions in both fuel consumption and NOx emissions can be attained.
The previous results have shown that the lowest fuel consumption and NOx emissions during the NEDC are achieved with a 6-speed transmission for the vehicle under consideration. The matrix of all the NEDC simulation results for all the gear ratios are imported into the optimisation process. The software is then able to model and fit the fuel consumption and NOx emission data. Predictions are made for both fuel consumption and NOx emissions for any set of gear ratios with a high degree of confidence. This is shown for both set of results fitted by the optimisation routine against the discrete simulated values in Figure 13, showing high degrees of conformance.

![Figure 13: Fitted Fuel and NOx trends Versus Discretely Simulated results](image)

A multi-objective optimisation is then carried out using CAMEO to find the optimal gear ratios with the lowest fuel consumption and best attainable NOx emissions as not both criteria can be fully optimised. A 'Pareto Front' graph is shown in figure 14. All the predicted fuel consumption levels and the corresponding NOx emissions for the various determined gear ratios are shown in the figure. The dark line at the bottom highlights the 'Pareto Front' outcome.
The two optimum gear ratio sets with minimum BSFC and the least simultaneous NOx emission levels are presented in table 9.

Table 9: Minimum BSFC (primary objective) with least attainable NOx Emission (Trade-off Results)

<table>
<thead>
<tr>
<th>Gear Change Method (Drive Cycle)</th>
<th>1st Fuel</th>
<th>2nd NOx</th>
<th>3rd Time</th>
<th>4th</th>
<th>5th</th>
<th>6th</th>
<th>Fuel</th>
<th>NOx</th>
<th>Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum NOx</td>
<td>14.8862</td>
<td>9.1149</td>
<td>6.9735</td>
<td>5.0819</td>
<td>2.9897</td>
<td>-7.29%</td>
<td>573.5</td>
<td>-6.76%</td>
<td>27.12</td>
</tr>
<tr>
<td>Original Transmission</td>
<td>14.5183</td>
<td>5.4418</td>
<td>3.9548</td>
<td>3.2578</td>
<td>0.00%</td>
<td>0.00%</td>
<td>618.60</td>
<td>0.00%</td>
<td>29.09</td>
</tr>
</tbody>
</table>

The results show that for the minimum fuel consumption and least NOx emission levels on the 'Pareto Front', there is a difference of 0.25% in the fuel consumption and 0.037% in the NOx emissions. The optimum gear ratio set depends on the importance attached to these criteria in the optimisation process. However, there is only a small reduction in the NOx emissions compared with the possible reduction in fuel consumption, meaning that the gear ratio set corresponding to the minimum BSFC yields the best outcome in the case studied.
Conclusion

The paper outlines a novel computationally efficient analytical method to evaluate fuel consumption and NOx emissions during simulation of NEDC. It also provides a good test of vehicle acceleration performance. Vehicle performance during an NEDC drive cycle is assessed and verified against measured experimental tests, reported elsewhere [7].

The method sets the 1st gear ratio based on adequate vehicle hill-climb performance. The top gear ratio is selected for least BSFC under highway driving conditions. The intervening gear ratios for various transmission configurations for 4, 5 and 6-speed variety are calculated based on optimal BSFC or NOx emissions and optimised using a genetic algorithm based optimisation routine; CAMEO.

It is shown that with minimum BSFC as the primary objective function, choosing a determined set of optimum gear ratios and altering the gear shifting strategy results in reduced BSFC by 7.52% and NOx emissions by 6.73% relative to the original fixed speed gear shifts. With NOx emission level as the primary objective, optimisation of gear ratios leads to a reduction of 7.6% in NOx with a decrease in BSFC by 4.65%. In the optimised cases a 6-speed transmission shows the best outcome over the 4 and 5 speed variety, but clearly with increased manufacturing costs. With the additional manufacture and assembly costs of approximately $100 per transmission, the 6-speed BSFC-optimised 6-speed shifting strategy accrues an overall 1% improved fuel efficiency over the 4 and 5-speed optimised variety, which is regarded as quite significant.

The computationally efficient analytical simulation as well as rapid scenario-building optimisation would enable application of the methodology to gear shifting indicator technology, thus embedding a certain degree of inherent intelligent feedback to the drivers of manual transmission. For other transmissions this action can be automated.

The current study concentrated on fuel consumption and NOx emissions. In order to show the ultimate potential, maximum flexibility and less imposed constraints are considered in the model. In reality, the efficiency and emission might show contradictory behaviour to the ride comfort requirements of the vehicle. For example, the optimum gear ratios for the best fuel economy or emission level might worsen.
the drivability or shifting quality. The shifting quality can be mitigated by using technologies such as Automated Manual Transmission (AMT) in order to reduce any undue effects of performed optimization.

The presented model and conclusions are based on the overall drivetrain ratios as shown in tables 3-8. Therefore, engineers would have the flexibility to further optimize the specific ratio of transmission and final drive to achieve a desired configuration such as direct drive or over drive transmission. This is important since the direct drive configuration provides potentially a simpler and lighter design.

Considering the importance of implementing real world drive cycles such as WLTC, the same model can be used to optimize the transmission for these new drive cycles in the future.

Acknowledgements

The authors would like to express their gratitude to AVL for allowing access to the optimisation software CAMEO.

Nomenclature

\[ a \]  
Vehicle longitudinal acceleration \((m/s^2)\)

\[ A_f \]  
Frontal area of vehicle \((m^2)\)

\[ c_D \]  
Vehicle drag coefficient

\[ F_D \]  
Aerodynamic drag \((N)\)

\[ F_G \]  
Gradient force \((N)\)

\[ F_R \]  
Rolling resistance \((N)\)

\[ F_x \]  
Traction \((N)\)

\[ g \]  
Gravitational constant \((m/s^2)\)

\[ I_D \]  
Inertia of differential unit \((kgm^2)\)

\[ I_E \]  
Engine inertia \((kgm^2)\)

\[ I_{Eff} \]  
Effective inertia of powertrain \((kgm^2)\)

\[ I_{F,C} \]  
Flywheel and clutch inertia \((kgm^2)\)

\[ I_p \]  
Propshaft inertia \((kgm^2)\)

\[ I_S \]  
Driveshaft inertia \((kgm^2)\)
\( I_T \)  
Transmission inertia in the selected gear \((kgm^2)\)

\( I_W \)  
Combined inertia of the wheels and brake discs \((kgm^2)\)

\( m_{fuel} \)  
Mass of fuel burnt \((g)\)

\( m_{NOX} \)  
Mass of generated NOx \((g)\)

\( m_v \)  
Vehicle mass \((kg)\)

\( N_{Eng} \)  
Engine speed \((rpm)\)

\( N_{idle} \)  
Engine idling speed \((rpm)\)

\( N_{max} \)  
Maximum engine speed \((rpm)\)

\( NOX \)  
Rate of production of NOx \((g/kWh)\)

\( P_{Eng} \)  
Engine power \((W)\)

\( R_n \)  
Gear ratio of gear \(n\)

\( r_w \)  
Laden tyre radius \((m)\)

\( T_{Eng} \)  
Engine torque \((Nm)\)

\( \Delta t \)  
Time step of simulation \((s)\)

\( v \)  
Vehicle velocity \((m/s)\)

\( V_d \)  
Engine swept volume \((m^3)\)

Greek symbols:

\( \eta \)  
Overall powertrain efficiency

\( \theta_{road} \)  
Road angle \((degrees)\)

\( \mu \)  
Coefficient of rolling resistance

\( \rho \)  
Air density \((kg/m^3)\)

Abbreviations:

\( BMEP \)  
Brake mean effective pressure \((Pa)\)

\( BSFC \)  
Brake specific fuel consumption \((g/kWh)\)

\( ECE \ 15 \)  
Economic Commission for Europe 15

\( EUDC \)  
Extra urban drive cycle
References


[21] "What is Euro-6?", Society of Motor Manufactures and Traders, [http://www.smmt.co.uk/industry-topics/what-is-euro-6/](http://www.smmt.co.uk/industry-topics/what-is-euro-6/)

