Optimisation of vehicle transmission and shifting strategy for minimum fuel consumption under EU and US legislated drive cycles

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Abstract: In recent years the importance of reducing fuel consumption and exhaust emissions from road vehicles has become paramount. This is because the exhaust gases contribute to global warming as well as adversely affect the quality of air. Emissions legislation is increasingly stringent and automobile manufacturers strive to mitigate these untoward effects and also improve fuel efficiency with new and innovative solutions.

This paper shows that gearbox configuration and shifting strategy can be optimised to arrive at an optimum design, reducing fuel consumption and NOx emissions. Such solutions are based on performance enhancement under regulated test procedures embodied in specified drive cycles, both in Europe and in United States. It is shown that a combined dynamics analysis and multi-objective optimisation can yield optimum gearbox configurations for given vehicles/engines. Furthermore, the results of the analysis can be subjected to a trade-off routine in order to find a near optimal generic solution which would meet the requirement of global design and manufacture, and simultaneously comply well with the differing requirements of various drive cycles.

Keywords: Optimum gear ratio set, Gear shifting strategy, Fuel consumption, NOx emissions

1-Introduction

1.1 - Literature Review

The major challenges currently faced by the automotive manufacturers are the need to reduce fuel consumption and exhaust gas emissions. These emissions contribute to the global warming and deteriorate the quality of air [1]. There are already stringent national and international legislations and directives with regard to limits on exhaust emissions.

There are increasing developments and technologies for improving fuel efficiency and mitigate exhaust emissions. They include powertrain downsizing within the concept of high output power-to-light weight ratio as well as Cylinder Deactivation (CDA), which make use of reducing the engine size to shift the loading to more efficient regions of the engine operational map [2]. Fraser et al [3] investigated power unit downsizing of a 2.0L engine to a 1.2L and were able to achieve a 15% reduction in fuel consumption during the NEDC [3]. Douglas et al [4] carried out simulations with CDA and another technology called Controlled Auto Ignition (CAI). Their results showed that the combination of both technologies could reduce fuel consumption and NOx emissions by 10% and 28% respectively.

Improving engine efficiency is not the only proven method. Advances in transmission design and control have also led to new methods for lowering fuel consumption and emissions. Gear Shift Indicators are able to advise the driver of the most efficient gear to select under given circumstances. Norris et al [5] showed that the factory-fitted gear shift indicators were able to reduce fuel consumption by 4% and 7% for a MINI Cooper and a Ford Transit respectively, but a Volkswagen Golf showed little improvement.

1.2 - Methodology

Previous work was carried out to develop a model which could optimise a manual transmission gear ratios and gear shifting strategy to reduce fuel consumption and emissions [6]. From the results two optimum gearbox configurations were found, one for the minimum fuel consumption and another for the minimum NOx emissions, through use of an optimisation process. These configurations were reported to lead to a reduction in fuel consumption and NOx emissions during the NEDC of 7.52% and 7.6% respectively [6].

The optimised gearbox configurations were designed specifically around the European drive cycle (NEDC). In this paper, the optimum gearbox configurations are found for the United States drive cycle (FTP-75) to see whether they conform to those found in [6]. Toa certain degree, this would infer that the optimum gear ratios for a vehicle can be independent of the selected drive cycle. There would be significant commercial advantages in this case.

2-Model Description

2.1 - Equation of Motion

Simulation are used to calculate a range of gear ratios and to ascertain the acceleration performance, fuel consumption and NOx emissions. The equation of motion are derived for simple longitudinal force balance as shown in Figure 1 [7]:

$$\Sigma F = m_v a = F_x - (F_D + F_R + F_G) \tag{1}$$

where, F_x is the wheel tractive force, F_D is the aerodynamic drag, F_R is the tyres' rolling and F_G is the gradient force due to any road inclination to the horizontal.
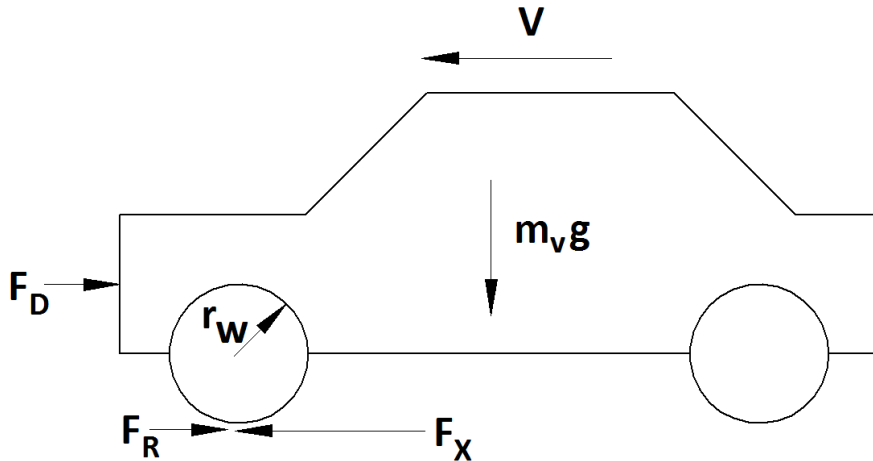

Figure 1: Forces in longitudinal vehicle dynamics on a flat road

The tractive force is [7, 8]:

$$F_x = \frac{T_{Eng} R_n \eta}{r_w} - \frac{a}{r_w^2} I_{Eff} \tag{2}$$

$$I_{Eff} = (I_E + I_{F,C} + I_{T,in})(R_n R_{F.D})^2 + I_{T,out} R_{F.D}^2 + I_D + I_S + I_W \tag{3}$$

where the effective inertia is for a front wheel drive transaxle drive train layout.

The aerodynamic drag is:

$$F_D = \frac{1}{2} c_D \rho A_f v^2 \tag{4}$$

And the rolling resistance:

$$F_R = \mu m_v g \cos \theta_{road} \tag{5}$$

Where the coefficient of rolling resistance is obtained as:

$$\mu = 0.01 \left(1 + \frac{2.23694v}{147}\right) \tag{6}$$

Any grading force is given by:

$$F_G = m_v g \sin \theta_{road} \tag{7}$$

2.2- First gear ratio selection

The first gear ratio is usually high so that it can provide adequate hill-climbing ability. Another key attribute of the first gear ratio is to provide a slow creeping speed in congested traffic. This would reduce the clutch usage by the driver [9]. For these reasons the first gear ratio will not be optimised according to the stated criteria. The equation of motion can be rearranged and modified to account for the required hill gradient, which is usually 1 in 3 [10], thus:

$$R_{1st} = \frac{r_w}{T_{Eng}\eta} \left[a\left(m_v + \frac{I_{Eff}}{r_w^2}\right) + F_D + F_R + F_G\right] \tag{8}$$

2.3 - Top gear ratio selection

The top gear ratio is selected to ensure that the vehicle can reach its maximum desired speed [9]. This speed is usually above the speed limit which means it is unlikely and indeed illegal to attain. The focus of the optimisation process is to reduce fuel consumption, so this gear ratio is selected to provide the lowest fuel consumption in highway driving speeds. Therefore, no optimisation is also required in this case.

2.4 – The intervening gear ratios

The optimisation of gear ratios is carried out for the intervening gear ratios. These gear ratios are equally spaced between first and top gears. A range of these gear ratios above and below each equally-space ratio is included for the purpose of optimisation.

2.5 - Gear shifting strategies

Two gear shifting strategies are used. The first strategy is gear shifting at fixed engine speeds. This means that up-shifting will be made once an engine speed is reached. This type of strategy is similar to the way in which most drivers change manual transmission gears. For city and motorway driving the drivers try to maintain the engine speed relatively low (2500 rpm) as early up-shifting tends to provide a better fuel consumption [11]. For situations where acceleration is the most important performance measure, drivers allow the engine to reach higher speeds (>4000 rpm) to achieve higher output power.

The other gear shifting strategy is the shifting for minimum fuel consumption. For this strategy the gear which can provide the lowest fuel consumption is selected. A gear shift indicator needs to be used to inform the driver of the most efficient gear to choose [5].

2.6 - Acceleration simulation model

The acceleration performance needs to be simulated to ensure that the driveability of the vehicle cis maintained. The industry standard is a 0-60 mph timed test. For this purpose a simulation is carried out at full throttle and the time taken to reach 60 mph is evaluated. The fixed engine speed gear shifting strategy is employed in order to ensure the quickest time.

The engine torque can be found using the full load power curve for the engine and the vehicle's velocity and acceleration can be calculated by carrying out integrations of the equation of motion (equation (1)).

2.7 - Drive cycle simulation study

Drive cycles are set to determine vehicles fuel consumption and generated emissions. Different drive cycles are used in various regions of the world due to the differences in the traffic conditions and the imposed legal speed limits. In the United States the standard is the FTP-75 and in Europe the drive cycle is the NEDC. These are shown in figures 2 and 3 [12].
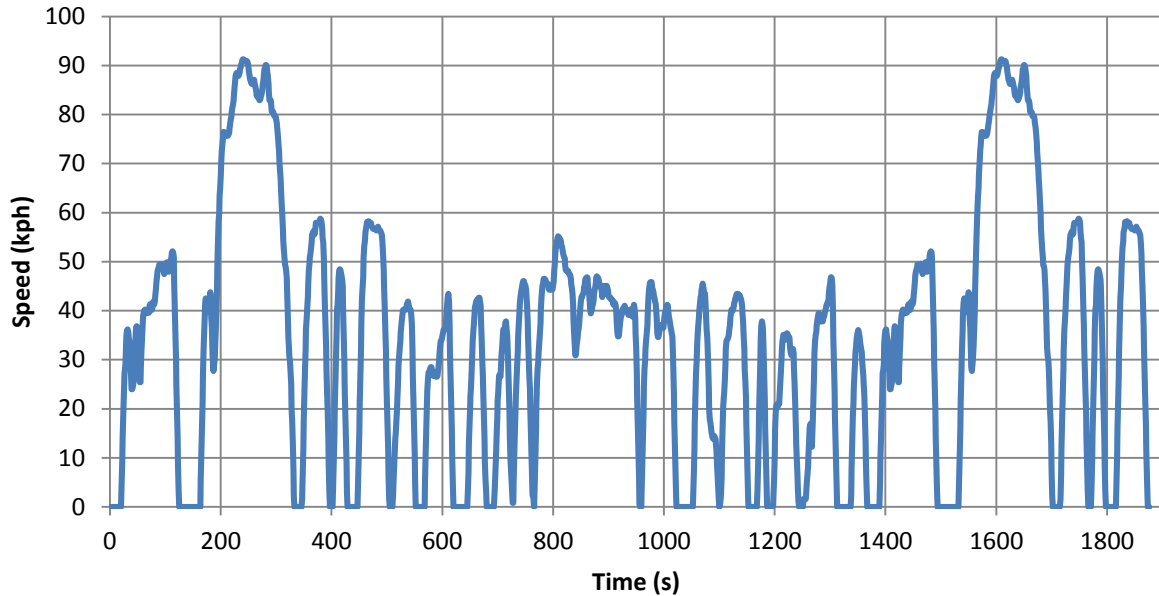

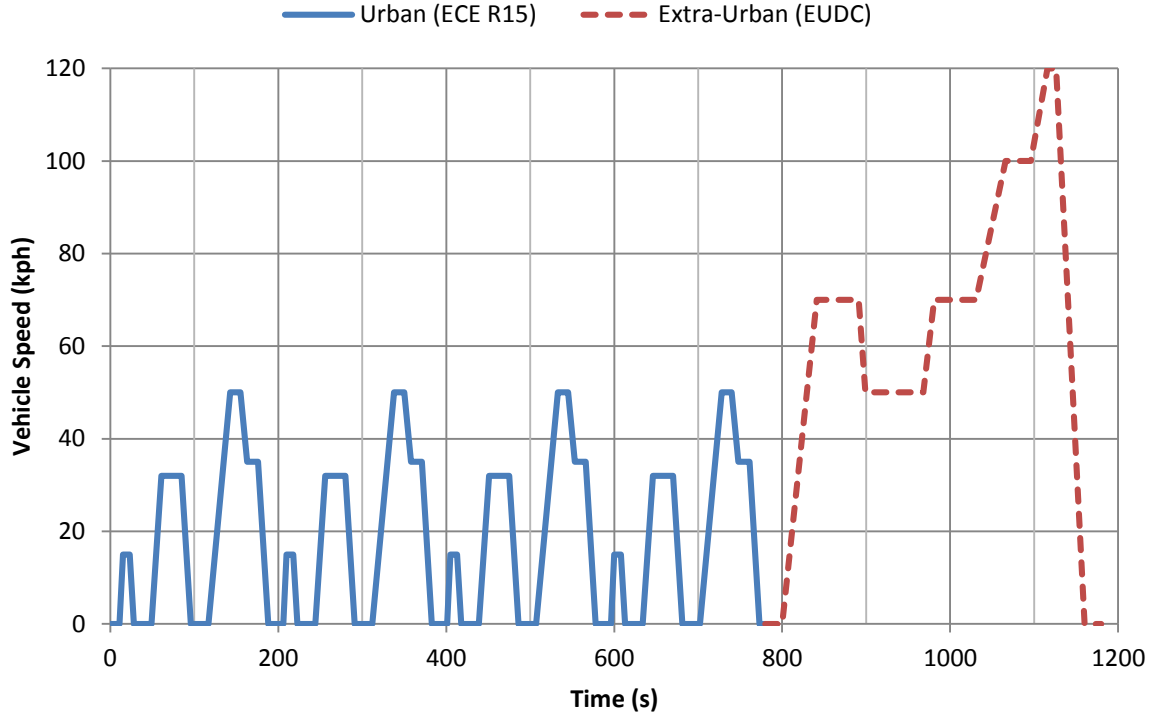Figure 2: The FTP-75 Drive Cycle [12]

Figure 3: The NEDC Drive Cycle [12]

The drive cycle needs to be followed exactly by the vehicle during the simulation study. The total fuel consumption and NOx emissions during the cycle are used in the optimisation process to determine the optimum gearbox configuration.

The vehicle's speed and acceleration at each time point is known during the drive cycle, so the equation of motion needs to be rearranged in order to find the required engine torque. The engine speed and torque can be used to find the brake specific fuel consumption (BSFC) and the NOx emission level from the appropriate engine maps. The fuel consumption and NOx emissions can be calculated by multiplying the BSFC and NOx values by the engine power:

$$m_{fuel} = \int \frac{BSFC \times \left(\frac{2\pi}{60}\right) N_{Eng} T_{Eng}}{3.6 \times 10^6} \, dt \tag{9}$$

$$m_{NOX} = \int \frac{NOX \times \left(\frac{2\pi}{60}\right) N_{Eng} T_{Eng}}{3.6 \times 10^6} \, dt \tag{10}$$

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3 - Results and Discussion

3.1 - Vehicle and Engine Specifications

The vehicle selected for optimisation is a typical European passenger car. It is powered by a 1.6L 4-cylinder engine with a five speed manual transmission [4]. The BSFC and NOx engine maps are shown in Figures 4 and 5 [4].

Table 1: Vehicle and Engine specifications [4]

Vehicle Specifications			Engine Specifications	
Mass (kerb) m_v	1330 kg		Type	4 Stroke Petrol SI
Drag Coefficient c_D	0.325		Cylinders	4
Frontal Area A_f	2.01 m^2		Volume	1.5968 L
Tire Radius r_w	0.2978 m		Max Engine Speed N_{max}	7000 rpm
Wheel Inertia I_w	0.74 kgm^2		Idle Engine Speed N_{idle}	800 rpm
Drive Type	FWD		Engine Inertia I_{eng}	0.1224 kgm^2

Table 2: Original Transmission Configuration [4]

Transmission Specifications		
Type	Manual (5 Speed)	
	Gear Ratio	Overall Ratio
1st Gear	3.583	14.5183
2nd Gear	1.947	7.8892
3rd Gear	1.343	5.4418
4th Gear	0.976	3.9548
5th Gear	0.804	3.2578
Final Drive	4.052	-

Engine BSFC (g/kWh)

Figure 4: Engine BSFC map [4]

Engine NOx pre-catalyst (g/kWh)
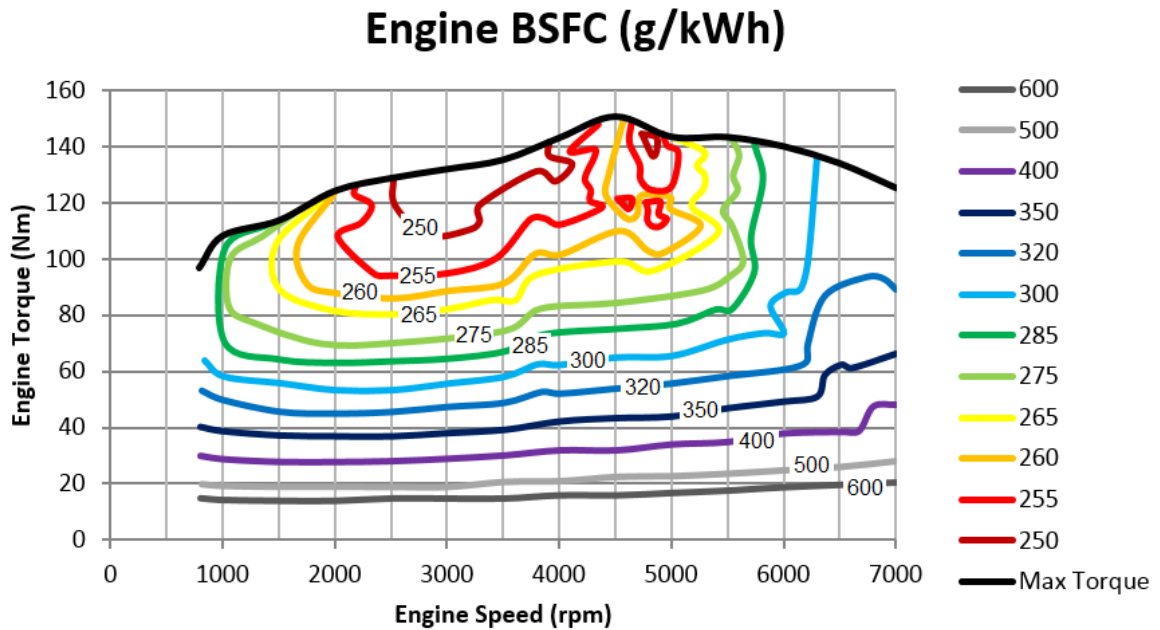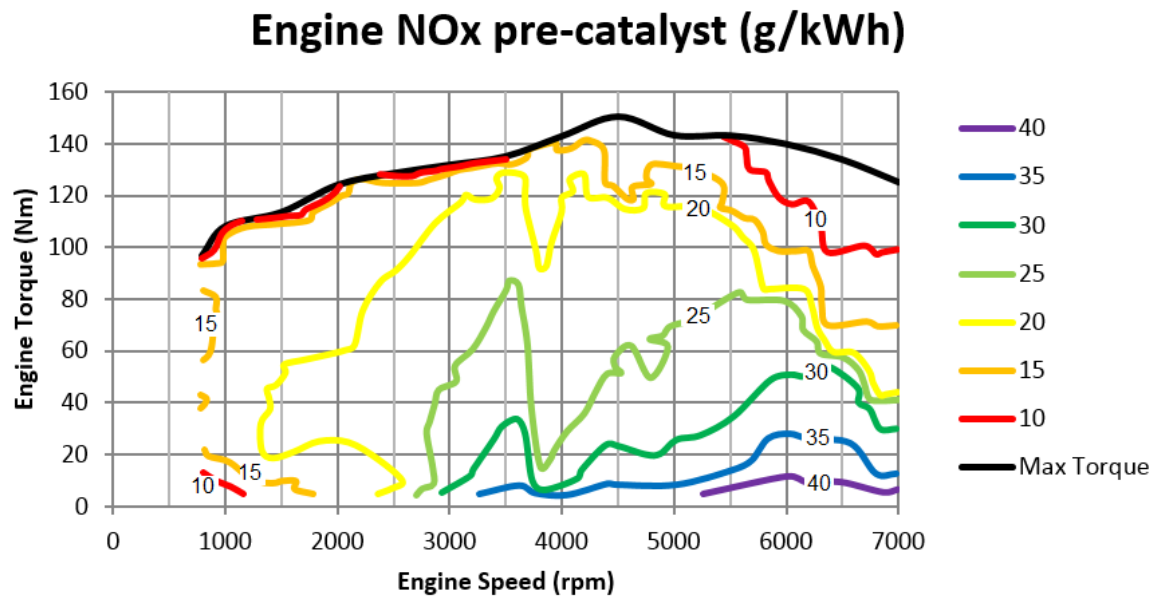

Figure 5: Engine NOx map (pre-catalyst) [4]

3.2 - Minimum Fuel Consumption Gear Ratio Optimisation

Previous reported results have shown that the lowest fuel consumption can be achieved by a 6-speed transmission with the minimum fuel consumption gear shifting strategy for the NEDC [6]. This is because the additional gear allows for earlier up-shifting which would help in reducing

fuel consumption [11]. The same transmission type is used here to find the minimum fuel consumption which could be achieved during the FTP-75 drive cycle.

The FTP-75 drive cycle requires the tests to be carried out with a cold start. This means the engine will not be running at its optimum temperature and the engine maps shown in figure 3 and 4 would not be valid during this warm up period. No temperature compensation model is available for the vehicle during a FTP-75 drive cycle so the simulations are carried out with a hot engine.

Simulations are carried out with a discrete set of gear ratios and combinations. The output from the simulation studies yields a table of gear ratio combinations and the corresponding fuel consumption and NOx emission levels. The results are then entered into an optimisation process, using the AVL CAMEO to find the minimum gear ratio set, which can exist between the discrete data points. The optimum gear ratio sets for the minimum fuel consumption during the FTP-75 and NEDC drive cycles are shown in tables 3 and 4.

Table 3: Optimum gear ratios for minimum fuel consumption during the FTP-75

	Gear Change Method (Drive Cycle)	1st	2nd	3rd	4th	5th	6th	Fuel	NOx	Accel. Time
Minimum Fuel	Minimum Fuel	14.8862	11.6727	9.1149	6.9735	4.8321	2.9897	-8.14% (766.2)	-5.01% (40.12)	-0.3% (9.985)
Original Transmission	Fixed Engine Speed	14.5183	7.8892	5.4418	3.9548	3.2578		0.00% (834.1)	0.00% (42.24)	0.00% (10.015)

Table 4: Optimum gear ratios for minimum fuel consumption during the NEDC [6]

	Gear Change Method (Drive Cycle)	1st	2nd	3rd	4th	5th	6th	Fuel	NOx	Time
Minimum Fuel	Minimum Fuel	14.8862	11.7484	9.1149	7.0314	4.8321	2.9897	-7.52% (572.09)	-6.73% (27.13)	-0.45% (9.970)
Original Gearbox	Fixed Engine Speed	14.5183	7.8892	5.4418	3.9548	3.2578		0.00% (618.6)	0.00% (29.09)	0.00% (10.015)

From the results it can be seen that there are substantial reductions in fuel consumption and NOx emissions during the FTP-75 and the NEDC drive cycles. Both optimum gearbox configurations are able to maintain the acceleration performance of the vehicle.

The optimum gearbox configurations are not identical, the second and fourth gears are slightly different for each drive cycle. This means that the optimum gearbox configuration is dependent on the drive cycle used in the analysis, but the same transmission can still provide fair performance in both cases.

3.3 - European and United States Trade-off Gearbox Configuration

As found in the previous section, the optimum gearbox configuration is dependent on the drive cycle used in the analysis. To avoid designing multiple gearboxes for any vehicle, thus increasing design and manufacturing costs, and in order to attain the highest possible performance during both the drive cycles, a trade-off for the optimum gearbox configurations should be found.

The drive cycle results for the NEDC and the FTP-75 drive cycles are entered into the AVL CAMEO and a multi-objective optimisation carried to find a set of minimum fuel consumption gear box configurations. Figure 6 shows the predicted fuel consumption values for each gear ratio combination for the NEDC and the FTP-75 drive cycles.
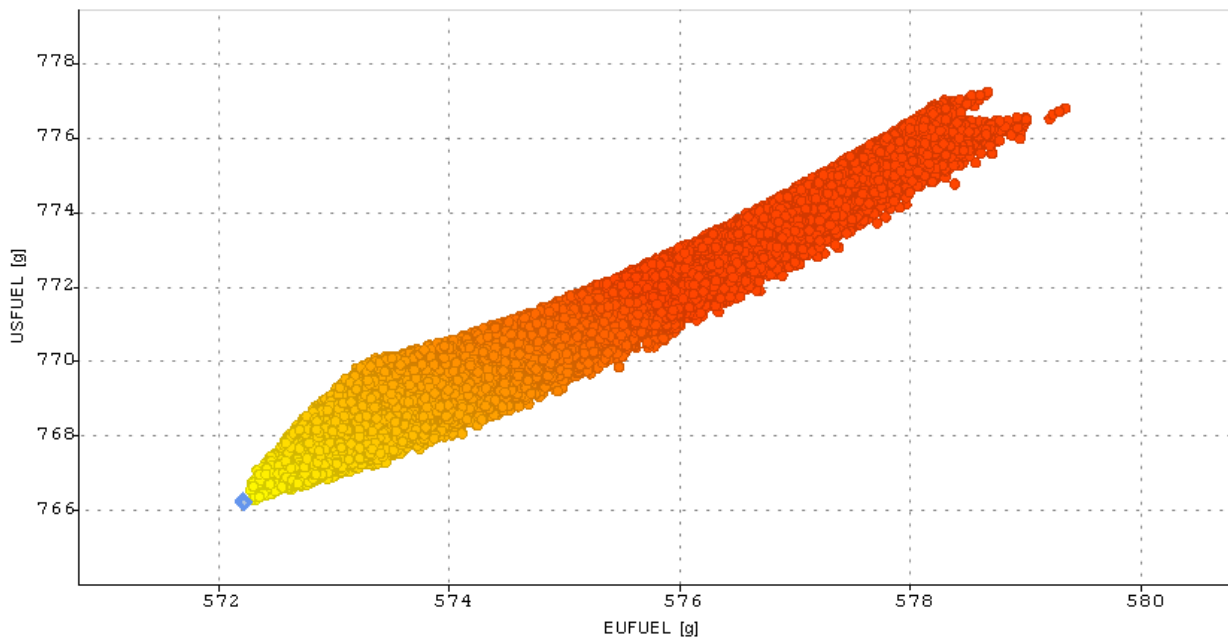

Figure 6: Predicted fuel consumption values for the NEDC and the FTP-75

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There is no unique gear box configuration which would provide the lowest fuel consumption for both the drive cycles. Therefore, a set of minimum results known as the Pareto Front are shown in Figure 7. The optimum minimum fuel consumption gearbox configurations from tables 3 and 4 are shown as the 'Min NEDC' and 'Min FTP-75' points in Figure 7.
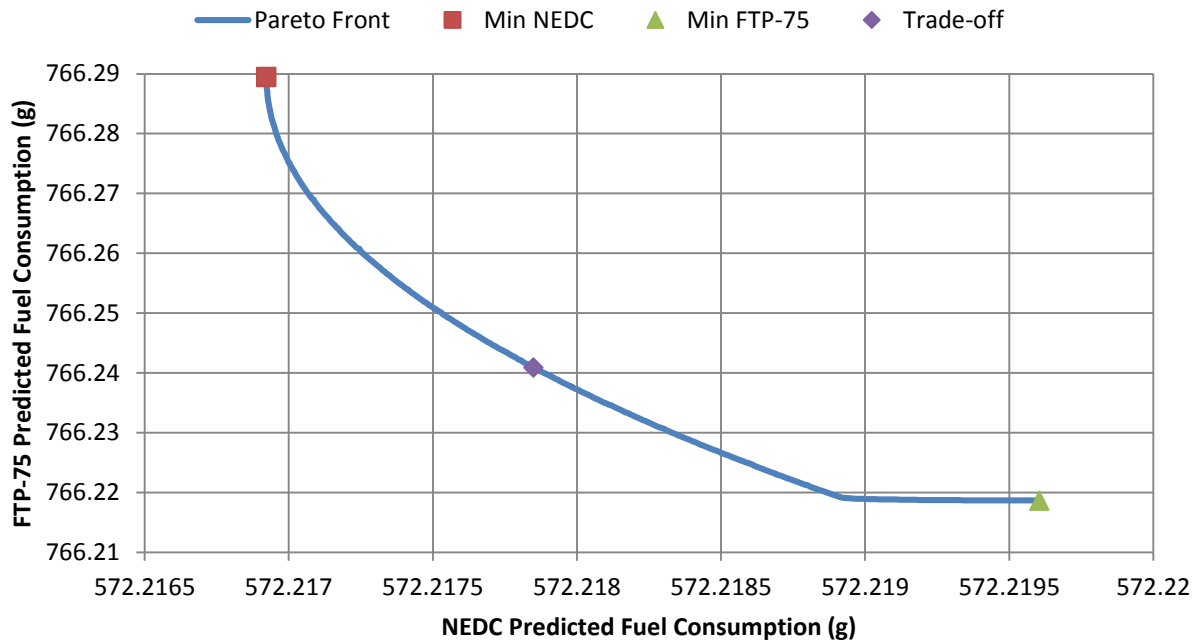

Figure 7: Minimum predicted fuel consumption points (Pareto Front)

A trade-off of the minimum results can be reached, depending on the relative importance of the fuel consumption reduction for each region. To obtain these results it a weighting ratio of 50/50 is assumed for the European and the United States drive cycles. The 'trade-off' point is shown in Figure 7. The gear ratios corresponding to this trade-off point are listed in Table 5 and the fuel consumption values recalculated in order to ensure accurate values over those predicted with AVL CAMEO.

Table 5: Trade-off optimum gearbox for minimum fuel consumption during NEDC and FTP-75

Gear Change Method (Drive Cycle)	1st	2nd	3rd	4th	5th	6th	Fuel	NOx	Time
FTP-75	14.8862	11.7455	9.1149	6.9919	4.8321	2.9897	-8.15% (766.12)	-5.02% (40.12)	-0.35% (9.98)
NEDC							-7.51% (572.16)	-6.70% (27.14)	

Overall, the trade-off gearbox can provide large reductions in fuel consumption and NOx emissions, which would help reduce the environmental impact of road vehicles in a multitude of regions.

Conclusion

The analysis approach is able to predict the BSFC for a set of gear ratios. The AVL CAMEO is able to find the optimum gear ratio set to provide the lowest fuel consumption during the FTP-75 drive cycle. The optimum gearbox configuration for the FTP-75 drive cycle is slightly different to that previously found for the NEDC for the same vehicle and engine. This means different gearboxes would be needed depending on the vehicles designated market. Designing and manufacturing different gearboxes for each region would lead to increased costs, so a trade-off to provide the best performance for both cycles is found through a trade-off process.

The trade-off gearbox configuration can potentially achieve 8.15% and 7.51% reductions in fuel consumption during the FTP-75 and the NEDC drive cycles respectively. This would also lead to 5.02% and 6.7% reductions in the NOx emissions respectively. The trade-off gearbox could also maintain the acceleration performance of the vehicle as the 0-60 time is only slightly reduced by 0.35% relative to the standard current transmission configuration.

Acknowledgments:

Nomenclature:

a	Vehicle longitudinal acceleration (m/s^2)
A_f	Frontal area of the 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_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)
NOX	Rate of production of NOx (g/kWh)
R_n	Gear ratio of the nth gear
$R_{F.D}$	Gear ratio of the final drive
r_w	Laden tyre radius (m)
T_{Eng}	Engine torque (Nm)
v	Vehicle velocity (m/s)

Greeks:

η	Overall powertrain efficiency

θ_{road} Road angles (degrees)

μ Coefficient of rolling resistance

ρ Air density (kg/m^3)

References:

[1]- Pitstick, M. E., Santini, D. J. and Chauhan, H., "Reduction in global warming due to fuel economy improvements and emissions control of criteria pollutants: new US light duty vehicles (1968-1991)", Argonne National Lab., USA, 1992.

[2]- Schernus, C., Dieterich, C., Nebbia, C., Sehr, A., Wedowski, S., and Weinowski, R., "Turbocharging of downsized gasoline DI engines with 2 and 3 cylinders", SAE Technical Paper, No. 2011-24-0138, 2011

[3]- Fraser, N., Blaxill, H., Lumsden, G. and Bassett, M.,"Challenges for increased efficiency through gasoline engine downsizing", SAE Int. J. Engines, 2(1), 2009, pp.991-1008.

[4]- Douglas, K. J., Milovanovic, N. and Blundell, D., "Fuel economy improvement using combined CAI and cylinder deactivation (CDA)- an initial study", SAE Technical Paper, No. 2005-01-0110, 2005

[5]- Norris, J., Walker, H., Stones, P., and Davies, R., "Assessing the efficacy of gear shift indicators", AEA/Millbrook, Oxford, UK, 2011.

[6]- Oglieve, C., Mohammadpour, M. and Rahnejat, H., "Optimisation of vehicle transmission and shifting strategy for minimum fuel consumption and NOx emissions", Proc. IMechE, Part D: J. Automobile Engineering, Submitted

[7]- Gillespie, T. D., "Fundamentals of vehicle dynamics", SAE, Warrendale, PA, USA, 1992.

[8]- Mohammadpour, M., Theodossiades, S., Rahnejat, H. and Kelly, P., "Transmission efficiency and noise, vibration and harshness refinement of differential hypoid gear pairs", Proc. IMechE, , Part K: J. Multi-body Dynamics, 2013, 228(1), pp.19-33.

[9]- Singh, J. and Singh, J., "Selection of gear ratio for smooth gear shifting", SAE Technical Paper, No. 2012-01-2005, 2012.

[10]- "Proving ground - test hills," HORIBA - MIRA, [Online]. Available: http://www.horiba-mira.com/our-services/test-hills. [Accessed 21 November 2015].

[11]- Vagg, C., Brace, C., Hari, D., Akehurst, S. and Ash, L., "A driver advisory tool to reduce fuel consumption", SAE Technical Paper, No. 2012-01-2087, 2013.

[12]- Barlow, T., Latham, S., McCrae, I. and Boulter, P., "A reference book of driving cycles for use in the measurement of road vehicle emissions", HMSO, Department for Transport, London, 2009.