Integrated Passenger Car Diesel CVT Powertrain Control for Economy and Low Emissions

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SYNOPSIS
A project has been undertaken to develop an integrated control strategy for a Diesel passenger vehicle powertrain. The chosen prime mover was an experimental Ford 1.8L DI TCi Diesel engine coupled to a compression belt continuously variable transmission in a Ford Orion saloon. Experimental and computer simulation studies were used to develop the necessary control algorithms. Selected control strategies were then tested experimentally to determine the emission and fuel economy improvements. The results presented suggest that integrated powertrain control is an effective route to realising the maximum potential for economy and emissions improvements from a given system.

1 INTRODUCTION

Progress in reducing harmful emissions from vehicles equipped with a CVT can be enhanced if the engine and transmission are controlled in an integrated manner. This may be achieved using a supervisory controller to co-ordinate the engine and transmission control as shown in **Figure 1**. Current production CVT systems retain a direct link between the accelerator pedal and the engine throttle (1). This effectively allows the driver to set the torque output from the engine, limiting the authority of the controller and compromising performance. Integrated control allows the software to act on a power demand from the driver to determine the optimum engine speed and torque. A constraint on such a system is that good drivability characteristics must be retained to make the vehicle commercially acceptable. This must be recognised at the concept stage.

Good vehicle drivability is characterised by the driver having ease of control of the vehicle and confidence in both predictable and desirable system responses to driver demands. It is very much dominated by the performance of the powertrain and vehicle in transient conditions. In steady state powertrain operation, drivability performance is not particularly discernible, and powertrain operation can be optimised more for emissions and economy.

A project has been undertaken at the University of Bath to develop an integrated control strategy for a passenger vehicle powertrain, described below. The project used experimental and computer simulation studies to develop the necessary control algorithms. Reduced order models were employed to investigate the relative performance of candidate control strategies by simulation. Selected control methods were then tested experimentally using a purpose built engine and transmission test rig to determine the emission and fuel economy improvements. The same system was also installed in a vehicle for the conventional ECE15 + EUDC drive cycle test and an assessment of drivability. Results for the vehicle tests are reported in this paper. A more complete description of these results will be found in reference (2).

2 DESCRIPTION OF PLANT

The major components were chosen to be representative of the technology level available in the five to ten years following the project. The work is sufficiently generic to allow its application to more advanced powertrain components as they become available.

The chosen prime mover was an experimental Ford 1.8DI TCi Diesel developed from the production 1.8IDI TCi engine as described by Lawrence (3). The specification is outlined in **Table 1.** This engine has significant fuel economy advantages over the IDI generation of engines. It is equipped with exhaust gas recirculation (EGR) for improvement of NOx emissions.

Table 1 - Specification of the Ford 1.8 L. Direct Injection Turbocharged Intercooled Diesel engine

Fuel system	Direct injection Diesel
Cylinders	4
Valves	2 per cylinder (SOHC)
Fuel pump	Lucas EPIC
Bore	82,5 mm
Stroke	82 mm
Displacement	1753 cm^3
Power Output	66kW (DIN/EEC) at 4500 rev/min
Max. Torque	180 Nm (DIN/EEC) at 2200 rev/min (limited to for this project due to transmission limitations)
Compression ratio	19:01
Idle speed	870 +10 -20 rev/min
Maximum speed	5150 +/-50 rev/min
Turbocharger	Garret T2
Charge cooling	Air to air intercooler
Exhaust gas recirculation	Via vacuum operated valve controlled by EPIC

The engine is fitted with the Lucas EPIC fuel injection system as described by Glikin (4) and Lewis (5). It consists of an electronic engine management module controlling a high pressure rotary pump and also scheduling the EGR valve opening. Steady state fuelling is determined by an algorithm relating pedal position and engine speed. Injection timing is mapped against engine speed, fuel delivery and coolant temperature. EGR valve position is mapped against engine speed and fuel delivery. There are many additional features designed to improve drivability and emissions performance, particularly during transients as reported by Martin (6).

The exhaust gases were treated using a Johnson Matthey JM DF07 oxidation catalyst. The catalyst material is platinum on a ceramic monolith. This attenuates hydrocarbon (HC), carbon monoxide (CO) and particulate matter (PM) emissions but not the oxides of nitrogen (NOx).

The transmission used is an experimental version of the Ford CTX, modified to allow electronic control and was known as the CTXE. Electronic control is shown by Hendriks (7) to be beneficial in terms of fuel economy and vehicle performance. The variator used was developed and manufactured by Van Doorne's Transmissie b.v. (VDT) and is discussed by Hendriks (8). It consists of a steel push belt which transmits power between primary and secondary pulleys. The specification is outlined in **Table 2.**

Table 2 - Specification of the Ford CTXE

Variator type	Steel compression belt
Starting device	Multi-plate wet clutch
Variator range	6.25 (2.5 $\rightarrow$ 0.4)
Final drive ratio	5.44
Torque capacity	130Nm
Belt width	25mm

The vehicle used in this work was a 1992 model year Ford Orion Saloon. The powertrain described above was installed together with comprehensive data acquisition and control systems.

3 CONTROLLER STRUCTURE - DESIGN CONSIDERATIONS

3.1 Emissions and economy considerations

Significant reductions in the emissions from today's engines are required if they are to meet future legislation (9). The legislated emissions for vehicles with Diesel engines are hydrocarbons, particulates, NOx and carbon monoxide. The emissions of NOx and particulates are usually of most concern. Most measures designed to reduce one of these pollutants have a detrimental effect on the other. To reduce both simultaneously requires a substantial improvement in the engine combustion or the use of other novel ideas. Carbon monoxide emissions are less of a problem for Diesel engines because of the excess air present for combustion. Smoke is also of interest and likely to be of increasing importance. Carbon dioxide production is directly related to the fuel economy of the vehicle. There is no legislation currently governing the economy of passenger cars although future legislation is proposed (10) and economy is of interest to the customer.

3.2 Ideal operating points and lines

The concept of an engine economy line has been widely discussed (Stockton (11), Yang et al (12)). Such a line is formed by joining together points of increasing engine power on an engine torque speed map. Each point represents the most economic torque speed combination which will produce the specified power. The line starts at zero torque and engine idling speed and finishes at the maximum power, which for automotive Diesel engines is usually at full load and rated speed.

The idea used in the classic case of the economy line can be extended to lines optimised for each of the engine emissions of concern. The ideal lines for economy and emissions, based upon steady state engine conditions, are distributed over a large part of the torque speed map as shown in **Figure 2**. Because of this, a compromise must be made between economy and each of the emissions so that an operating line can be generated. The compromise between the various ideal lines must be made by considering the level of concern associated with each of the particular pollutants and the proximity of the legislative limit to the experimental value produced when testing the particular engine vehicle combination. Such a study allows a compromise operating line to be generated by applying numerical weightings to each point of each alternative line.

3.3 Producing the ideal operating point

The ideal operating point is generated by a set of algorithms contained within the operating point optimiser, Brace et al (13). The optimiser takes, as one of its inputs, a positive power demand, produced by the supervisory controller and uses this to generate an ideal engine operating point in terms of engine torque and speed. This is done by interpolation along a pre-determined line produced by the operating line optimiser, taking into account the weightings discussed above.

4 CONTROLLER ARCHITECTURE

The controller architecture is designed in a centralised hierarchical manner. The hardware supplied by both Lucas and VDT made such a structure easier to implement than one of a distributed nature in this case. The controller architecture, shown in **Figure 1**, is therefore divided into two distinct areas, firstly the supervisory powertrain controller and secondly the engine and transmission controllers. The supervisory controller contains the software responsible for the choice of engine operating point in terms of torque and speed during transient conditions. In determining the final steady state operating point following a transient, the various alternative supervisory controllers reference the operating line optimiser. Other inputs used by the strategies included the pedal position, a measure of time, and various plant feedbacks.

The outputs of the supervisory controller are passed to the lower controllers; engine torque is controlled by modulation of the EPIC fuel injection system demand signal in an open loop process. Engine speed is controlled by varying the transmission ratio.

4.1 Supervisory controller algorithm

A number of supervisory control algorithms were developed in the course of the project. These ranged from simple rule based controllers to fuzzy logic approaches and are discussed in detail by Deacon (2). Results presented here were generated using an algorithm known as the Transient Shaping controller.

The Transient Shaping controller was motivated by the desire to keep the engine strictly on the ideal operating line during steady operation. The controller also shapes the path taken on the torque speed map by the engine during a transient. Priority during steady state operation and during slow transients is given to the low emissions/high economy objective. Clearly there is still a need to move away from the ideal line in the highly transient situation. To enable this the torque and speed demands are trimmed to enable movement of the powertrain along transient operating paths away from the ideal emissions/economy line. This trimming or ‘shaping’ of the operating path during powertrain transients gave rise to the name of this approach.

5 VEHICLE TESTS

A series of tests were performed on the chassis dynamometer facility at Ford, Dunton. The cycle used was the ECE15 + EUDC. Vehicle tests were performed using each of the three operating lines shown in **Figure 2**. For most of the duration of the work a derated version of the full engine torque curve was used as shown in **Figure 2**. This was to safeguard the transmission which was rated at 130Nm. Later versions would easily cope with the full torque but these were not available at the time. The two ideal operating lines tested at this stage of the project were the IOL for brake specific fuel consumption (BSFC) and the IOL for NOx. These were chosen as they were the two extremes. The BSFC line has the lowest engine speed for all powers of all those considered, the NOx line the highest. Other lines for HC, PM and smoke lay between these two lines.

Towards the end of the project the full torque curve was re-instated as the reliability of the transmission seemed adequate for the limited mileage required for the remaining test work. At the same time the EGR demand map was reduced by 10%. The nozzle opening pressures of the injectors were lowered slightly to improve the idle stability. Engine dynamometer results showed the emissions and fuel consumption differences between the two calibrations to be negligible in the areas used by the strategies during the drive cycle, allowing the vehicle tests to be compared directly.

Tests were performed using the recalibrated engine following the 'compromise' IOL shown in **Figure 2**. This was achieved by finding the mean of the ideal speeds for all regulated pollutants of interest (HC, NOx and PM) at each engine power. This line crosses the derated torque curve at around 37kW although this would not greatly affect the comparison between these and earlier tests as this high power region of the map is not used during the ECE test. The compromise line demonstrates an intermediate calibration, which was between the two extremes of the BSFC and NOx lines

Data were also collected from the same powertrain running an emulation of the 'conventional' hydromechanical control strategy. Here the full pedal demand forces the transmission to its lowest ratio to deliver high engine speeds. A pedal demand of zero puts the transmission at its highest ratio to deliver low engine speeds. Intermediate pedal demands deliver engine speeds placed linearly between the two ratio limited extremes.

6 VEHICLE TEST RESULTS AND DISCUSSION

The results are shown in **Table 3**. **Figure 3a** shows the PM results against the HC+NOx figures with the legislative limits indicated. Presentation in this format is useful as the trade-off between NOx and PM is a key variable in powertrain calibration. **Figure 3b** shows the trade-off between fuel consumption and HC+NOx. **Figure 3c** shows the particulates results plotted against fuel consumption. In all cases results from a manual vehicle with a comparable engine calibration are included for comparison.

Table 3 - Chassis dynamometer test results

Test	Vehicle	Engine ideal operating line	Powertrain controller	Rolls test	Pedal signal (secondary pressure) %	HC g/km	CO g/km	NOx g/km	HC + NOx g/km	CO2 g/km	PM g/km	Fuel L/100km
1	Orion J160YWC	BSFC	Transient shaping	D16103	100	0.15	0.55	0.66	0.81	160	0.08	5.99
2	Orion J160YWC	NOx	Transient shaping	D16119	100	0.16	0.59	0.43	0.59	173	0.10	6.47
3	Orion J160YWC	n/a	Hydro mechanical	D16083	100	0.15	0.67	0.59	0.75	216	0.21	8.09
4	Orion J160YWC *	new mixed	Transient shaping	D20086	100	0.10	0.40	0.56	0.65	168	0.08	6.34
5	Orion J160YWC *	new mixed	Transient shaping	D20098	50	0.10	0.41	0.52	0.62	164	0.08	6.21
6	Orion J160YWC *	new mixed	Transient shaping	D20325	30	0.10	0.39	0.53	0.62	160	0.09	6.04
7	H157MJN	n/a	Manual Transmission	-	n/a	0.19	-	0.48	0.67	-	0.12	6.54

***** new engine calibration with 10% reduction in EGR

The most striking aspect of the results is the overwhelming improvement in the figures returned by controllers following an IOL when compared with the hydromechanical type of control strategy. The same hardware was used for all the CVT tests so the improvement can be attributed entirely to control strategy. The hydromechanical strategy is largely focussed on achieving good drivability, which it does well. This is primarily due to the high engine speeds selected during power-on phases of driving. This makes large reserves of power available very quickly on demand. The high engine speeds can be observed in **Figure 4** which plots fuel demand (analogous to engine torque) against engine speed for three of the tests at half second intervals over the whole drive cycle. The controller using the IOL for BSFC generally uses the highest fuel demands and lowest engine speeds as expected. The controller using the IOL for NOx uses the highest engine speeds and lowest fuel demands of any of the IOLs proposed but even these are appreciably slower than those used by the hydromechanical strategy. It must be emphasised that the hydromechanical controller could be tuned to demand much lower engine speeds and hence improved emissions and fuel economy. The strategy does not, however, have an IOL as its basis and as such its performance would be expected to fall short of a controller that does. Further illustration of the differences between the three strategies is shown in **Figure 5**, which shows the engine speeds in the time domain over the EUDC portion of the ECE test. The high engine speeds used by the hydromechanical strategy are evident.

As expected, optimising for BSFC returned the lowest fuel consumption figures and optimising for NOx returned the lowest NOx. The mixed line fell between these extremes and as such looks the most impressive in relation to the Stage 2 requirements. **Figure 3b** shows the relationship between fuel economy and HC + NOx figures to be approximately linear in the region considered. A linear fit to the data suggests an optimum fuel economy of around 6.25 L/100km would be achieved by the Transient Shaping controller if the HC + NOx performance were set to the stage 2 limit.

During Test 5 the secondary (or clamping) pressure of the transmission was reduced by reducing the pedal signal into the controlling algorithm by 50%. **Figure 3b** shows the effect of this change to be a marked improvement in the fuel economy versus HC + NOx trade - off. If the same characteristic gradient as for the tests using full secondary pressure is assumed this would suggest a fuel consumption of around 6 L/100km (or a 4% improvement compared with the full secondary pressure) at the Stage 2 HC + NOx limit. A further reduction in the pedal signal to only 30% of the original level (Test 6) produced a further saving which may be extrapolated using the same assumption to give around 5.8 L/100km (or a 7%

improvement over the standard setting). These improvements are valuable although the effect on the durability of the transmission must be considered. There is little doubt that the original level of secondary pressure is conservative, but the transmission must not be damaged by external torque disturbances caused by pot-holes and similar occurrences. A careful development program would be required before the secondary pressure could be reduced significantly.

The data from the equivalent manual powertrain shows that the Transient Shaping controller demonstrates significantly better performance. This is encouraging as it demonstrates that the CVT can fulfil its much anticipated potential with the use of advanced control strategies.

Figure 3c shows no clear correlation between fuel consumption and particulate production. PM is always the most difficult pollutant to measure with high repeatability and the sample size here is not sufficient to draw firm conclusions. Further variability is generated by the sensitivity of PM production to driver style and controller characteristics. **Figure 5** shows a relatively large fluctuation in engine speed at the start of the final high speed portion of the EUDC during Test 1. This was caused by interactions between driver demand, controller action and powertrain dynamics leading to a large excursion in engine speed and an accompanying penalty in PM performance. Similar speed and power fluctuations of a lesser degree are apparent elsewhere and will have the effect of scattering the PM results more widely than those for NOx, HC and fuel.

From a knowledge of the PM contours across the engine operating map and an inspection of **Figure 5c** it is suggested that for this powertrain there is an optimum fuel consumption of around 6.25 L/100km, which returns the lowest PM figures. Either side of this condition the PM level starts to rise once more. The scatter on the data (discussed above) prevents a curve being fitted with any degree of confidence, although it is estimated that the PM level at this condition would be around 0.7g/km, just on the limit of Stage 2 and comfortably below the derogation for DI until allowed until 1999.

As with the other results discussed it should be emphasised that the engine calibration considered here is not representative of current builds. The emissions and fuel consumption performance of the base engine has been improved considerably since the specification was fixed at the start of the project. It was considered important that any improvements achieved should be clearly attributable to improvements in control strategy rather than variations in engine (or transmission) specification. The techniques developed here can readily be applied to an improved powertrain allowing the same judgements to be made as to the desired calibration for the powertrain controller.

7 CONCLUSIONS

The integrated control of the Diesel CVT powertrain described allows the economy and emissions performance of the vehicle to be tuned to achieve the optimum performance from the hardware considered. Compared with older control strategies the gains can be startling, bringing the CVT powertrain up to or beyond the standard achieved by the manual equivalent.

A clear trade-off between fuel economy and HC + NOx can be plotted, allowing the informed calibration of the controller to return the best fuel consumption possible within the legislative constraints on emissions. A similar trade-off between PM and fuel consumption can also be developed with sufficient data.

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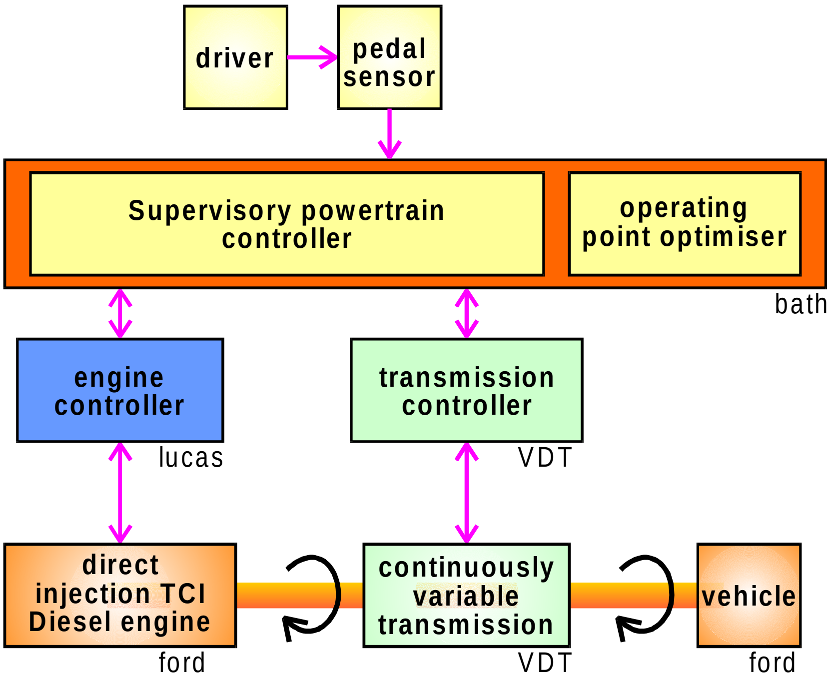


Figure 1 - Schematic of powertrain and controller

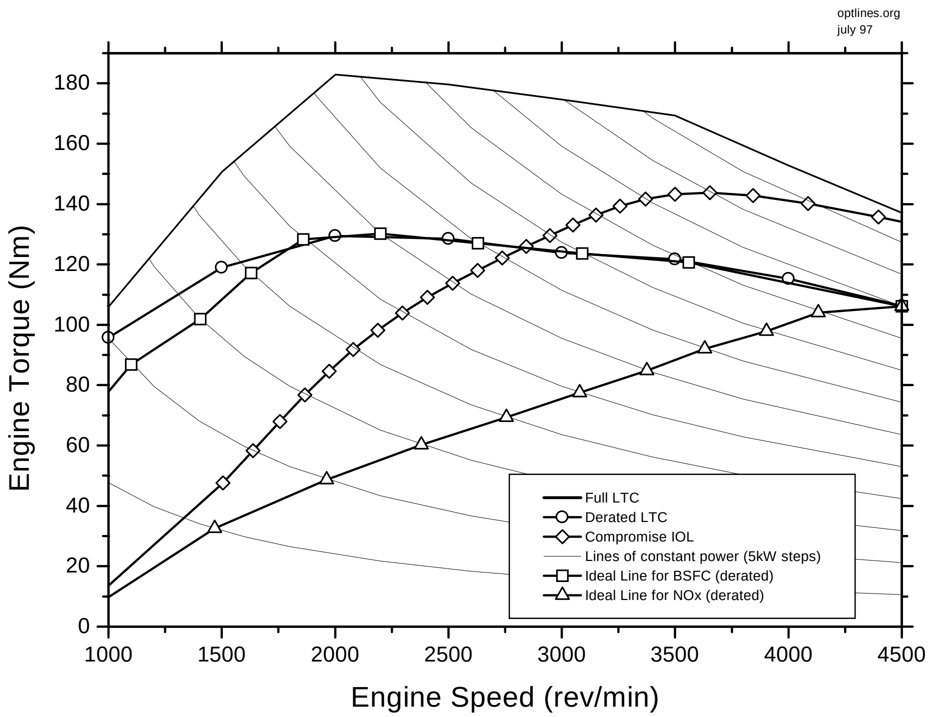
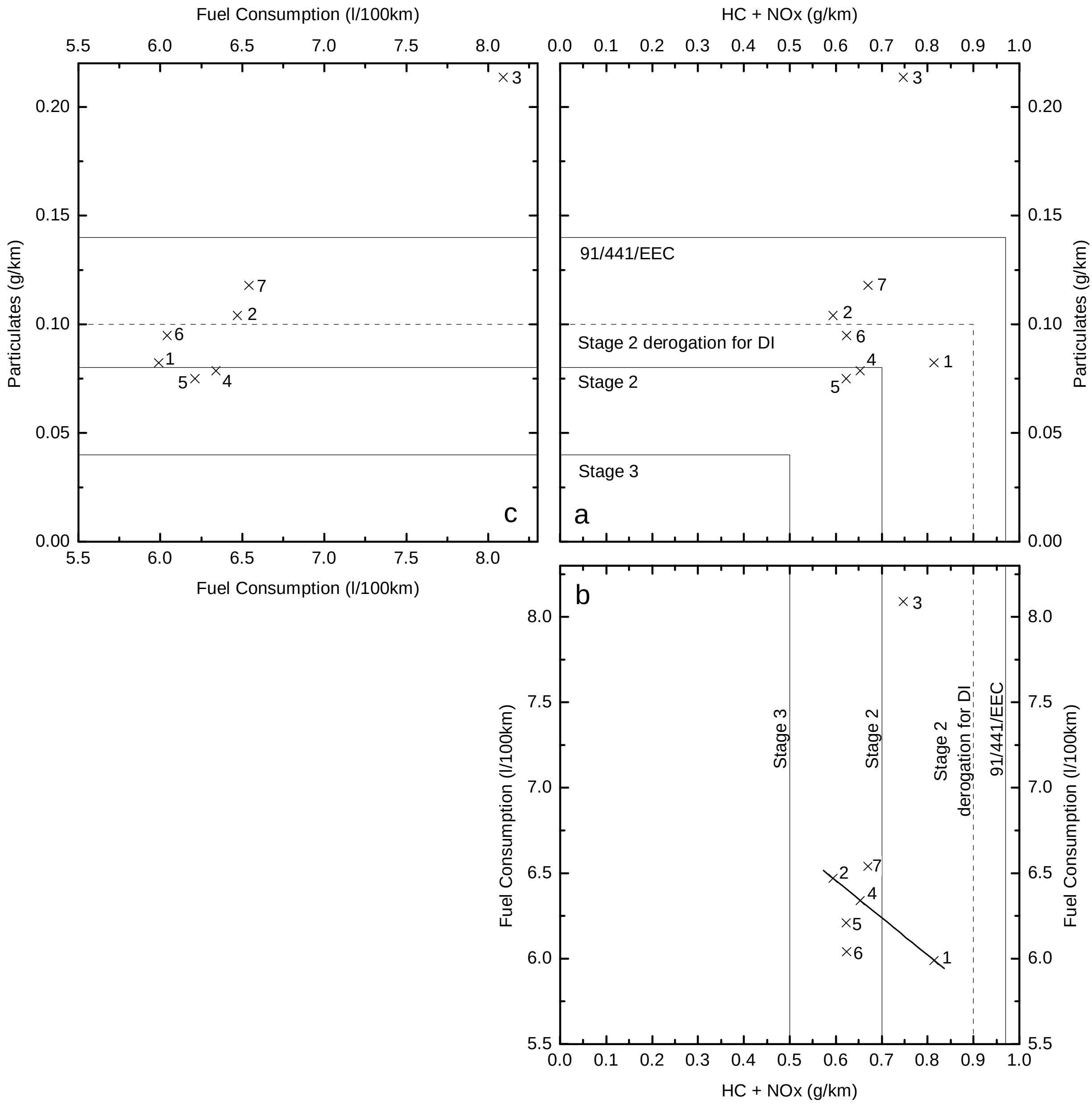


Figure 2 - Ideal operating lines



Test 1 - transient shaping controller optimised for BSFC
Test 2 - transient shaping controller optimised for NOx
Test 3 - hydromechanical controller
Test 4 - transient shaping controller, mixed operating line
Test 5 - transient shaping controller, mixed ideal line, 50% Psec
Test 6 - transient shaping controller, mixed ideal line, 30% Psec
Test 7 - Manual transmission, H157MJN

Figure 3 a, b & c - Chassis dynamometer test results

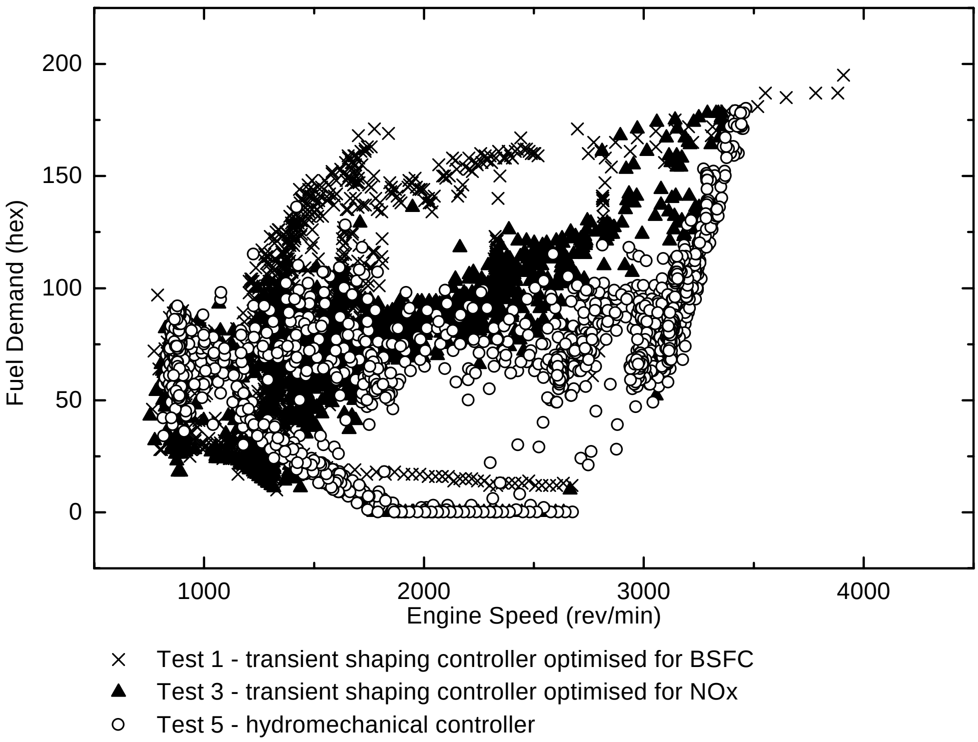


Figure 4 - Engine fuelling and speeds used during drive cycle

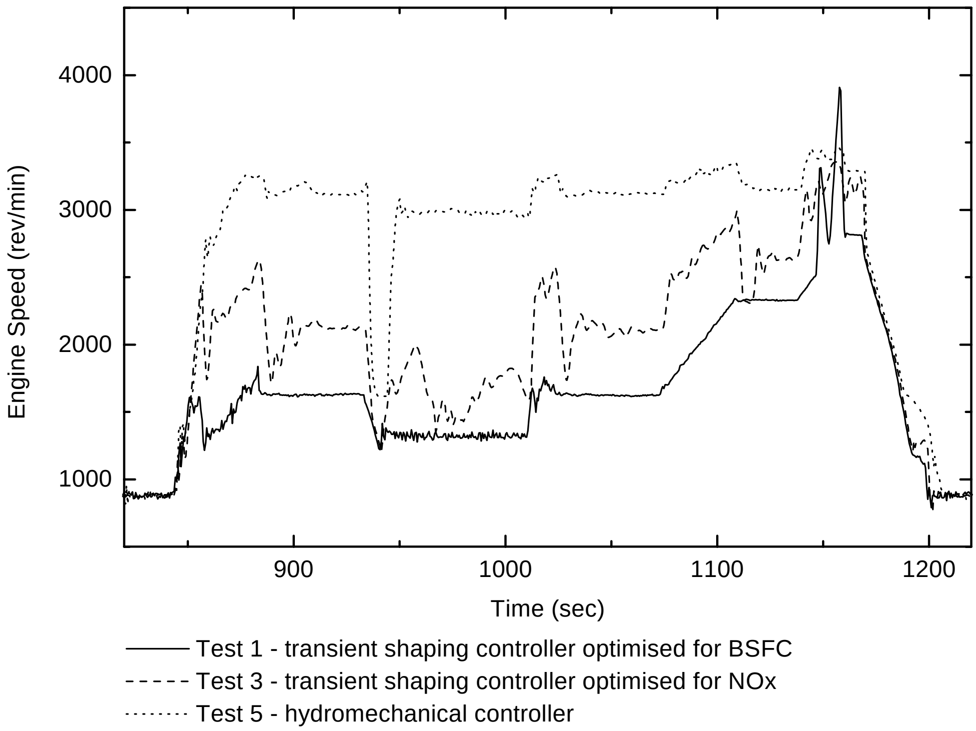


Figure 5 - Engine speeds during EUDC