INTEGRATED CONTROL STRATEGIES FOR A DIRECT-INJECTION DIESEL ENGINE AND CVT

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ABSTRACT

Advanced strategies have been developed for the control of automotive CVT powertrains which allow fuel economy and emissions to be tuned easily and can include consideration of vehicle drivability.

The techniques were developed using a combination of computer simulation, rig test work and vehicle tests. The demonstrator powertrain comprised a compression belt CVT and a 1.8L direct injection turbocharged and intercooled Diesel engine, both under the drive by wire control of a supervisory microprocessor. Simulation and experimental results are presented which show the flexibility of the strategies developed and the improvements possible in performance and emissions.

INTRODUCTION

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. 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**. This 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.

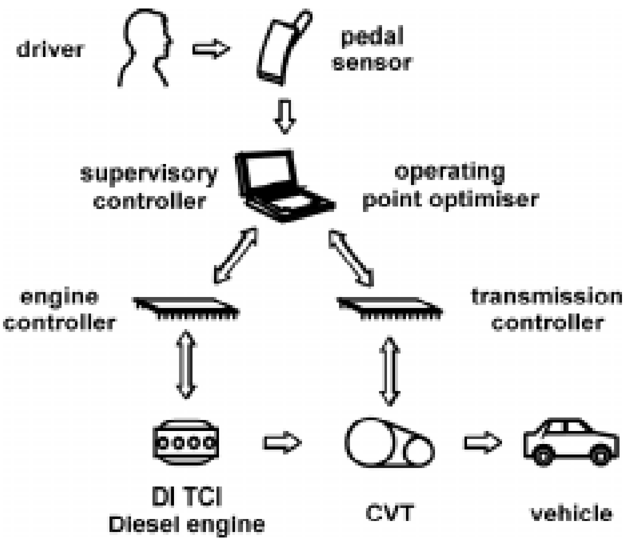

Figure 1 - Schematic of powertrain and controllers

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 and quasi steady state operation, drivability performance is not a priority, and powertrain operation can be optimised more for emissions and economy.

A project has been undertaken to develop such an integrated control strategy for a passenger vehicle powertrain. 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 confirm functional behaviour and to determine the emission and fuel economy improvements over the ECE15 + EUDC drive cycle. The same system was also installed in a vehicle for the conventional ECE15 + EUDC drive cycle test and an assessment of drivability. Results from the vehicle tests are reported in this paper. A broader range of results are also reported in reference (2).

DESCRIPTION OF POWERTRAIN

The major powertrain components were chosen to be representative of the technology level available in the five to ten years following the project. The work has been 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 (4). The specification is outlined in **Table 1.** This engine has significant fuel economy advantages over the

Engine	
Fuel system	Direct injection Diesel
Cylinders	4
Valves	2 per cylinder (SOHC)
Fuel injection system	Lucas EPIC rotary pump, Stanadyn injectors
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 130 Nm for this project due to transmission limitations)
Maximum speed	5150 rev/min
Turbocharger	Garret T2
Charge cooling	Air to air intercooler
Exhaust gas recirculation	Vacuum operated valve controlled by EPIC
Catalyst	
Type	Oxidation
Catalyst	Platinum
Monolith	Ceramic
Transmission	
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
Input Torque capacity	130Nm
Belt width	25mm

Table 1 - Specification of Demonstrator Powertrain

IDI generation of engines. It is equipped with exhaust gas recirculation (EGR) for improvement of NOx emissions.

The engine was fitted with the Lucas EPIC fuel injection system as described by Glikin (5) and Lewis (6). 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 and fuel delivery, and modified according to 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 (7). Most of these features are aimed primarily at the requirements of a manual transmission powertrain.

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 was an experimental version of the Ford CTX, modified to allow electronic control and designated CTXE. Electronic control is shown by Hendriks (8) 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 (9). It consists of a steel push belt which transmits power between primary and secondary pulleys. The specification is outlined in **Table 1.**

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

CONTROLLER STRUCTURE - DESIGN CONSIDERATIONS

The controller structure was considered in three stages.

First, the emissions considerations were defined with reference to the legislative requirements (10). The legislated emissions for vehicles with Diesel engines are HC, PM, NOx and CO. The emissions of NOx and PM 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. CO emissions are less of a problem for Diesel engines because of the excess air present for combustion and were not included here.

Secondly, fuel consumption was considered. There is no legislation currently governing the economy of individual passenger cars although future legislation is proposed (11) and economy is of interest to the customer and manufacturers alike.

Thirdly, drivability needed to be considered. This dictated that potential control strategies must take special account of transient manoeuvres and possibly reduce the emphasis on economy and emissions at such times in order to vary power delivery quickly and progressively following a change in driver demand.

The basic strategy underpinning all of the controllers developed was to optimise for fuel economy within the constraints of the emissions requirements but to subordinate these objectives during transients. The method used to achieve this was to use an *ideal operating line* (IOL) but to allow controlled transient deviations.

IDEAL OPERATING LINES

The concept of an engine economy line has been widely discussed (Stockton (12), Yang et al (13)). 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 rated speed. Driver power demand can then be placed on the economy line to determine the optimum engine speed and torque for economy.

The idea used in the classic case of the economy line can be extended to develop 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 due to their different formation mechanisms. Because of this, a compromise must be made between economy and each of the emissions so that a single 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. When legislative limits are met the fuel consumption issue may be addressed. Such a study allows a compromise operating line to be generated by applying numerical weightings to each point of each alternative line.

For the results presented here the ideal operating line (IOL) was generated by a function within the control code termed the *operating point optimiser* discussed by Brace et al (14). The optimiser took as its primary input a positive power demand produced by the supervisory controller and used this to generate an ideal engine operating point in terms of engine torque and speed. This was done by interpolation along the current ideal operating line, taking into account the weightings discussed above and current operating conditions such as coolant and/or catalyst temperatures. The IOL was generated using a neural network engine model which predicted the economy and emissions performance of the engine, given an engine speed, torque output and coolant temperature. These predictions were weighted and used by a simple optimisation routine to locate the optimum line for the conditions.

CONTROLLER ARCHITECTURE

The controller architecture was 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**, was therefore divided into two distinct areas, firstly the supervisory powertrain controller and secondly the engine and transmission controllers. The different strategies considered used the same basic structure with differences confined to the algorithms used within the supervisory powertrain controller.

The supervisory powertrain controller contained the software responsible for interpretation of driver pedal demand and the subsequent setting of engine operating point in terms of torque and speed. During transient conditions drivability considerations dictated the action of this controller. In determining the steady state operating point during less busy periods, the supervisory powertrain controller referenced the operating line optimiser.

The outputs of the supervisory controller were passed to the lower controllers. Engine torque was controlled by modulation of the EPIC fuel injection system demand signal in an open loop process using an inverse model of engine torque characteristic. Engine speed was controlled within the VDT controller by varying the transmission ratio.

SUPERVISORY POWERTRAIN CONTROL ALGORITHM

A number of supervisory powertrain 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 et al (3). Results presented here were generated using an algorithm called the *Transient Shaping* controller. The design of the algorithm was motivated by the desire to keep the engine strictly on the ideal operating line during steady operation but also to shape the trajectory followed on the engine torque speed map during a transient. To achieve this the engine torque and speed deviations from the IOL were related to driver input and plant operating point using look up tables. Larger transient demands resulted in greater deviation from the IOL. The extreme case of full pedal demand resulted in the engine torque demand saturating to ensure operation on the limiting torque curve (LTC) and a rapidly increasing engine speed to allow full power to be developed with minimum delay.

The choice of weightings to determine the IOL is clearly fundamental to the operation of the controller. Emissions and fuel consumption results will be primarily affected by the choice of IOL. Furthermore, the IOL will determine the engine operating point at the start of any transient, which will have great bearing on the dynamic behaviour of the vehicle during the transient. To assist in the calibration of these and other controller parameters a simple simulation was developed to allow the impact of calibration changes to be assessed at an early stage.

SIMULATION DESIGN

Competing control strategies were to be compared on the basis of predicted emissions (HC, NOx and PM) and fuel consumption over a typical vehicle test cycle. The new vehicle homologation procedure introduced in European Directive 91/441/EEC was selected

	CO (g/km)	NOx (g/km)	HC + NOx (g/km)	PM (g/km)	Comments
Stage 1 (DI)	3.81		1.36	0.20	in force
(IDI)	2.72		0.97	0.14	
Stage 2 (DI)	1.0		0.90	0.10	in force
(IDI)	1.0		0.70	0.08	
Stage 3	0.64	0.5	0.56	0.05	first 40 sec idle now included

Table 2 - European Diesel engine passenger car emission standards

as a representative test for the strategies. The test cycle was the urban ECE-15 cycle followed by the high speed Extra Urban Drive Cycle (EUDC), which has a maximum speed of 120 km/h. This combined cycle is known as the new European drive cycle. The limit values for exhaust emissions are shown in **Table 2**, together with draft limits for Stage 3 in 2000.

In addition to the emissions levels and fuel consumption, predictions of engine speed and load on a second by second basis over the cycle were required to allow controller performance to be examined. The simulation was required to produce results for a typical test cycle within five minutes to allow rapid comparison of a large number of candidate solutions within a realistic period. To allow this a reduced order simulation was developed which used a simple fixed time step approach to reduce model development and run times by several orders of magnitude when compared to the full dynamic simulation. The fixed time step approach dictates that only simple integration algorithms were used. A fixed time step of 1 second was chosen to achieve a suitable compromise between accuracy and volume of data. The simulation was implemented within an *Excel* spreadsheet running under *Windows* on a PC. The spreadsheet was very simple to use and allowed the user to concentrate on the engineering problem rather than on the generation of code.

MODEL STRUCTURE

The cycle is defined as a time series of vehicle speeds. From this information and with knowledge of vehicle and transmission and controller characteristics, an engine speed and load may be calculated at each step. The acceleration is simply that required to achieve the demanded speed by the end of the time step. The power requirement is determined by the work required to accelerate the vehicle inertia and an empirical drag model of the vehicle expressed as a polynomial in terms of speed. Inefficiencies in the powertrain have been included to arrive at a demanded engine power. The engine speed was set by a polynomial in terms of power representing the IOL under investigation. The engine torque required was then simple to calculate.

THE ENGINE MODEL

For an optimum combination of speed and accuracy an empirical model of the engine was used. This was only possible because sufficient experimental data were available for the engine under consideration. Analytical models of engine emission formation tend to be very large and slow running without producing results of sufficient accuracy for this application. Since the driving cycle is predominantly steady state with only modest transients a simple engine model structure was used. In addition to engine speed and load, the only other engine model input was coolant temperature, taken from experimental data for a typical test. The engine model was then used to estimate the emissions produced during each second. It was judged that other variables would reach their design points rapidly and have no noticeable effect when compared with steady state running. A neural network was used to represent the relationships between the inputs and outputs of the engine as experience with an earlier, transient, engine model (16) had suggested that it was suitable.

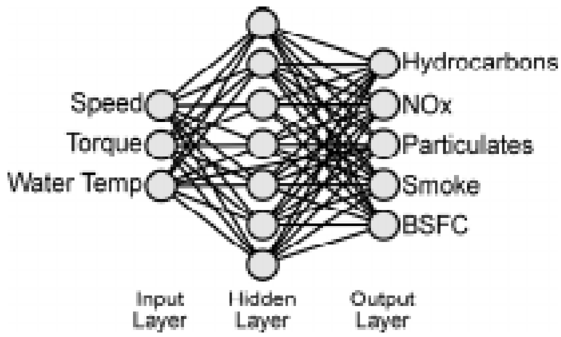

Figure 2 - Schematic of emissions prediction network

The number of network inputs and outputs were determined by the application. A network of the form shown in **Figure 2** was trained using a subset of data gathered for the transient model (16). A basic performance map of the engine at nominal operating conditions was augmented by data gathered with cooler and warmer engine temperatures than normal.

The trained network was very easy to interrogate using a PC running both the spreadsheet and neural network software within the *Windows* environment.

USE OF THE REDUCED ORDER SIMULATION

The simulation was used to study the compromises between the various emissions and fuel consumption. This was achieved by optimising for each pollutant in turn and observing the effect on others. This work was used to choose the compromise between emissions and hence IOL weightings for initial test work. Later, the effects of detail changes to transmission configuration were investigated in order to evaluate potential powertrain layouts at the concept stage.

DYNAMIC SIMULATION

A full dynamic simulation of the powertrain and vehicle was produced (15) and integrated with a detailed representation of the controller to allow testing of the complete control code in a repeatable environment. This allowed preliminary functional testing and an initial calibration of the major control loops, in particular the control of engine speed and load.

Transient behaviour was also investigated and initial values set for rate of change of engine speed and torque maps. These estimates were later refined by rig and vehicle test but the simulation was sufficiently accurate to allow some confidence that the initial settings would be quite drivable.

Some drive cycle investigations were carried out and although run times were long (over 24 hours in some cases) they did confirm the suitability of the control algorithms and the results of the reduced order simulation.

DYNAMIC POWERTRAIN TESTING

A rig was designed and built to allow dynamic testing of the powertrain under repeatable conditions in a laboratory environment with extensive instrumentation to allow a comprehensive analysis of controller behaviour. A schematic of the rig is shown in **Figure 3.**The engine and transmission were separated to allow the inclusion of an engine torque transducer. A second torque transducer was included on the output side of the locked differential. The output flange of this transducer was coupled to a large flywheel to replicate the majority of the vehicle inertia and a servo controlled hydraulic dynamometer used to simulate the vehicle drag and trim the inertia to reflect the exact test weight of the vehicle. The effect on shaft dynamics of the extra components and the absence of tyre compliance was not problematic as the bandwidth of the phenomena being studied were comparatively low.

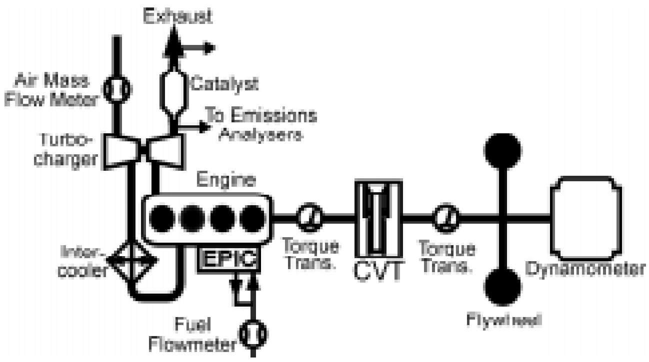

Figure 3 - Schematic, Dynamic Test Cell

Instrumentation comprised raw gas analysers, particulate measurement by TEOM, smoke assessment by exhaust opacity measurement and a series of flow, temperature and pressure transducers. Rig control and data acquisition were performed by a PC based system running a vehicle and driver emulation and closed loop control algorithms for rig fluid temperatures.

The rig was used for steady state, transient and dynamic testing of the complete powertrain running the same control algorithms used in the simulation and later, the vehicle. A series of functional tests were performed to verify that the controllers operated acceptably without serious discontinuities or other dangerous characteristics. Tests also verified that inputs and outputs were calibrated correctly, in particular that the demanded engine torque was achieved within an expected error band. Further tests were conducted to replicate the findings of the simulation and select a final set of ideal operating lines and controller calibration to be used for the vehicle work.

VEHICLE TEST WORK

DRIVABILITY STUDIES

Early in the project a series of drivability studies were conducted. Two production vehicles equipped with CVT were tested together with the demonstrator vehicle running the production specification hydromechanical transmission controller. A series of drivers were asked to assess aspects of the vehicles' drivability during track test sessions. Objective data were gathered from the same vehicles using on board data acquisition during a series of manoeuvres designed to exhibit the behaviour seen during the subjective assessment. The subjective and objective data were analysed to yield some guidelines for strategy calibration.

A second series of subjective tests were performed later in the project, this time three newly developed calibrations of the integrated powertrain controller were assessed against two production CVT vehicles. One of the most noticeable results from this work was the negative effect produced as a result of restricting the power rating of the engine. This was necessary 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. Although not a concern for low power cycles such as the ECE15 + EUDC the derated engine reduced subjective drivability ratings in comparison with earlier work using the full power available. It is evident that high engine powers are used frequently in vehicles of this class, especially during such assessments. Despite this limitation the new controllers provided borderline acceptable drivability with relatively little calibration effort. The results were sufficient to warrant an investigation of the emissions performance of the vehicle.

EMISSIONS AND ECONOMY STUDIES

A series of tests were performed on the chassis dynamometer facility at Ford Research and Engineering Centre, Dunton, UK. The cycle used was again the ECE15 + EUDC. Vehicle tests were performed using the operating lines shown in **Figure 4**. For most of the duration of the work a derated version of the full engine torque curve was used as discussed above. 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

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.

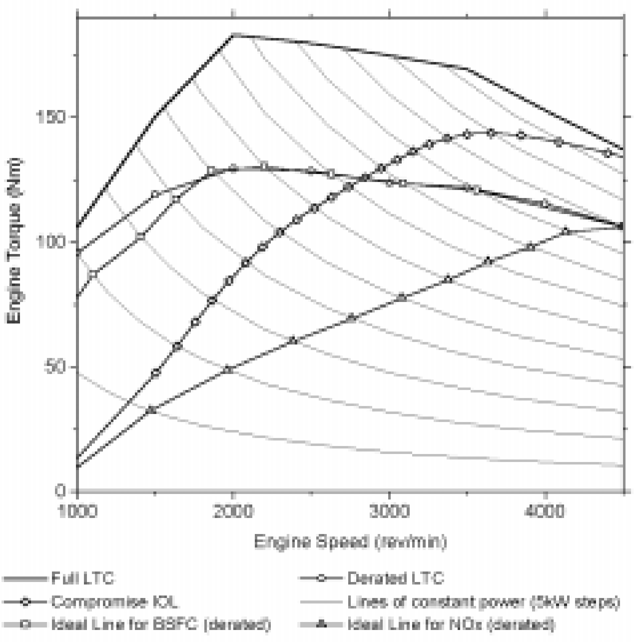

Figure 4 - Ideal operating lines lines.

Tests were performed using the recalibrated engine following the 'compromise' IOL shown in **Figure 4**. 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 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.

VEHICLE TEST RESULTS AND DISCUSSION

The results are shown in **Table 3**. **Figure 5a** 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 5b** shows the trade-off between fuel consumption and HC+NOx. **Figure 5c** shows the PM results plotted against fuel consumption. In all cases results from a manual vehicle with a comparable engine calibration are included for comparison.

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 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 6** 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 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 7**, 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 5b** 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

Test	Engine Calibration	Engine ideal operating line	HC g/km	CO g/km	NOx g/km	HC + NOx g/km	CO2 g/km	PM g/km	Fuel L/100km
1	1	IOL for BSFC	0.15	0.55	0.66	0.81	160	0.08	5.99
2	1	IOL for NOx	0.16	0.59	0.43	0.59	173	0.10	6.47
3	1	n/a (hydro mechanical)	0.15	0.67	0.59	0.75	216	0.21	8.09
4	2	mixed IOL	0.10	0.40	0.56	0.65	168	0.08	6.34
5	1	n/a (manual transmission)	0.19	-	0.48	0.67	-	0.12	6.54

Table 3 - Chassis dynamometer test results

be achieved by the Transient Shaping controller if the HC + NOx performance were set to the stage 2 limit.

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 5c 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.

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

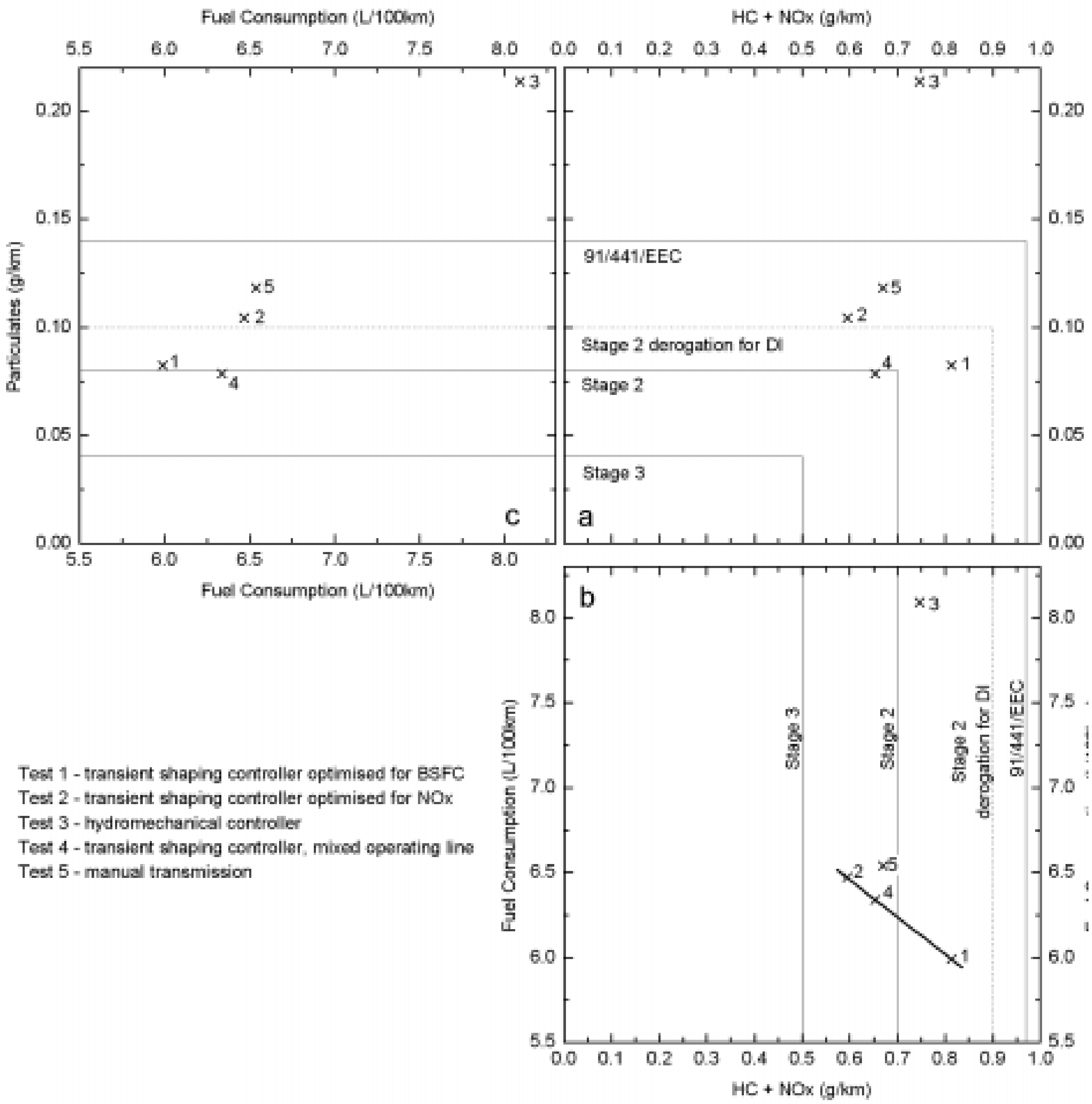

Figure 5 a, b & c - Chassis dynamometer test results above) prevents a curve being fitted with any degree of confidence, although it is estimated that the PM level at this condition would be around 0.7g/km, just on the limit of Stage 2 and comfortably below the derogation for DI allowed

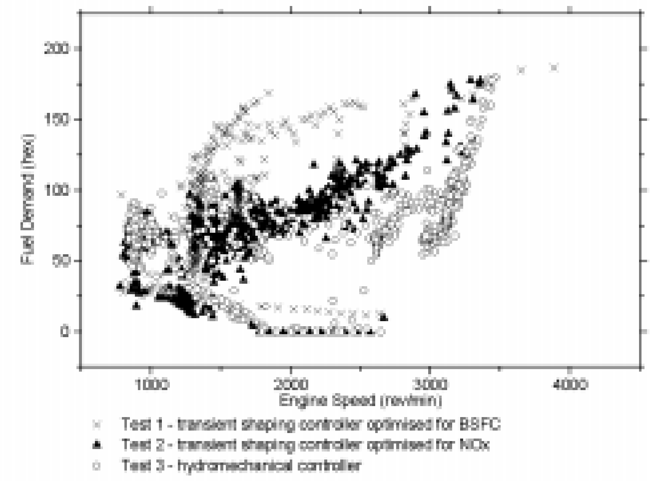

Figure 6 - Engine fuelling and speeds used during drive cycle

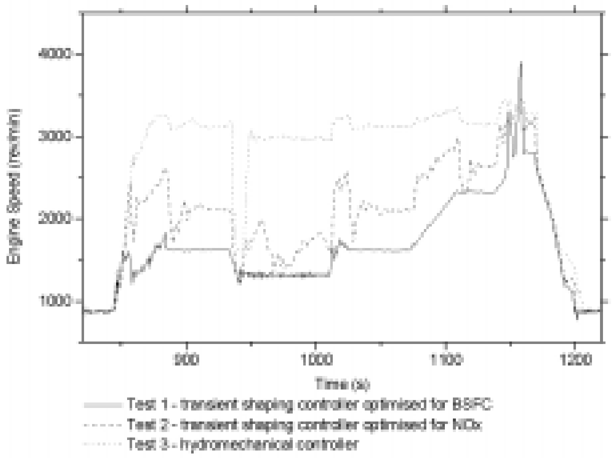

Figure 7 - Engine speeds during EUDC

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, performance and fuel consumption of the base engine have 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.

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. For the powertrain demonstrated here 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.

The integrated approach to powertrain control offers greater gains than the development of separate controllers for individual powertrain components. Future work will extend this concept to include aspects of the mechanical design of those individual components within the context of an integrated powertrain system.

A crucial element in the design and development of the controllers presented was the availability and co-ordination of a range of tools including reduced order simulation, dynamic simulation, steady state, transient and dynamic rig testing and vehicle test. Together these elements allowed the work to progress from concept to demonstration of concept in an integrated development environment, reducing potential for costly errors and incompatibilities. Such an integrated approach is vital if development times for new strategies are to be reduced to acceptable proportions.

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DEFINITIONS

BSFC: Brake specific fuel consumption
CTX: Continuously Variable transaXle
CTXE Continuously Variable transaXle (electronic control)
CO: Carbon monoxide
CVT: Continuously variable transmission
DI : Direct injection
ECE15 +
EUDC: New European drive cycle
EGR: Exhaust gas recirculation
TCi: Turbocharged and intercooled
HC: Hydrocarbons
IDI: Indirect injection
IOL: Ideal operating line
LTC: Limiting torque curve
NOx: Oxides of Nitrogen
PM: Particulate matter
TEOM: Tapered element oscillating microbalance