Impact of alternative controller strategies on exhaust emissions from an integrated diesel/continuously variable transmission powertrain

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Abstract: The use of an integrated powertrain including a continuously variable transmission (CVT) offers great scope for improvements in fuel consumption and emissions, although this must be achieved without adversely affecting vehicle drivability. These conflicting aims can best be resolved by the use of novel control strategies designed from the outset with these considerations in mind. Two alternative approaches to the task are presented here. These made use of artificial intelligence and more traditional and intuitive methods to allow the maximum flexibility in operation. Both strategies incorporated a novel optimization routine described in a companion paper to locate the best operating point for the engine. The two strategies were implemented for an integrated diesel CVT powertrain and compared with an existing controller and the equivalent manual transmission powertrain. Chassis dynamometer results show the newly designed controller strategies to have significant impact on vehicle exhaust emissions, while the structure of the software allows the controller action to be highly tuneable and flexible in order to balance the vehicle drivability requirements with economy and emissions targets.

Keywords: continuously variable transmissions, emissions, fuel economy, diesel engines, neural networks, fuzzy logic, integrated powertrain control

NOTATION

b.s.f.c.	brake specific fuel consumption
CO	carbon monoxide
CO_2	carbon dioxide
CTX	continuously variable transaxle
CTXE	electronically controlled continuously variable transaxle
CVT	continuously variable transmission
DI	direct injection
EGR	exhaust gas recirculation
EPIC	electronic pumping injection control
g/km	grams per kilometre
HC	hydrocarbon
IDI	indirect injection
IOL	ideal operating line
IOP	ideal operating point
km	kilometres
kW	kilowatts
L	litres
L/100 km	litres per 100 kilometres
LTC	limiting torque curve (of a diesel engine)
N m	Newton-metres
NO_x	nitrogen oxides
NVH	noise, vibration and harshness
PM	particulate matter
VDT	Van Doorne's Transmissie b.v.

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1 INTRODUCTION

Vehicles with CVT powertrains account for a growing market share [**1**], with recent models exhibiting much improved drivability and fuel economy. Vehicles fitted with CVTs will always display unique characteristics which some drivers may find unusual or unacceptable, but there remains much room for improvement through better control. The use of a CVT allows a decoupling of engine speed from vehicle speed. Drive by wire engine control allows a similar decoupling of engine torque from driver power demand (pedal position). When combined, these two features allow great freedom in the choice of engine operating point. There has been much discussion on determining the best control strategy for this type of integrated powertrain [**2**]. A successful control strategy must consider the following points.

1.1 Economy considerations

It has long been established that the use of an integrated CVT powertrain allows an engine to operate along a line of minimum fuel consumption or an *economy line* [**2**, **3**]. The economy line is the locus of the point returning minimum fuel consumption across the operating power range of an engine. Gains in fuel economy can be impressive without any changes in engine calibration. Such a strategy may be approximated with a conventional CVT powertrain in which the driver retains direct control of the engine torque. Here the strategy will control engine speed as a function of pedal position (or engine torque). The use of an integrated powertrain allows much greater flexibility and more refined behaviour but consequently introduces more complexity into the control strategy.

1.2 Emissions considerations

The major flaw of the economy line approach is the failure to optimize exhaust emissions performance. In the current environmental and political framework, exhaust emissions must be a fundamental consideration in the development of any powertrain control strategy. Despite this there has been comparatively little published work concerning the optimization of exhaust emissions from such a system. Additionally, fuel economy must not be neglected as it will remain a crucial measure of vehicle efficiency. Central to the work reported in this paper is the extension of the economy line concept to include an evaluation of exhaust emissions. This new type of line is termed the *ideal operating line* (IOL). When optimizing for a single outcome, such as minimum fuel consumption, a true optimum line is simply generated. If this process is repeated for each of the pollutants a different line will be generated in each case owing to their differing formation mechanisms. Thus, it is not possible to arrive at a globally optimum line. To resolve this difficulty the regulated exhaust emissions are combined with fuel economy in a weighted sum, which is minimized across the operating power range of the engine. This weighted line represents a compromise between the conflicting requirements of minimizing each pollutant according to the weights selected. The factors involved in selecting an IOL, including the need for it to adapt to different environmental conditions, are discussed in a companion paper [**4**].

1.3 Drivability considerations

A further consideration which must be addressed in any practical controller is that of subjective drivability. A control strategy that ensures strict adherence of an engine to its IOL will not allow satisfactory transient vehicle performance. The maximum rate of change in engine speed is limited by the rate at which the transmission can change ratio, although a more severe limitation is the adverse effect of rapid engine speed changes on vehicle drivability. A controller designed to constrain the engine to an IOL (or equally, an economy line) can only deliver a higher power at a higher engine speed. This limitation requires the engine speed to rise very quickly to allow more power to be produced in response to any rapid increase in the driver's power demand. For the engine speed to rise quickly there must be a reduction in vehicle acceleration, as a large proportion of the engine torque is used to accelerate the engine and transmission input shaft. In extreme cases it is possible for the engine to accelerate more rapidly than under no load but only by transfer of energy from the vehicle causing it to decelerate momentarily. This is clearly an unacceptable response following a driver command to accelerate. It is conceivable that the acceleration response of the vehicle could be reduced to avoid this effect. This cannot be allowed in the converse case where there is a rapid drop in driver power demand. Vehicle response must be immediate and predictable. This necessitates rapid modulation of the engine torque to reduce the engine power even at elevated engine speeds. This is, of course, exactly the case in a conventional powertrain where the driver retains direct control of engine torque output. A successful control strategy must reconcile the contradictory aims of retaining progressive, predictable behaviour while seeking to optimize the engine torque and speed for emissions and economy.

2 CONTROLLER PHILOSOPHY

2.1 Standard control strategy

The series production, hydromechanical control strategy did not reference the IOL. Engine speed was related directly to road speed and pedal position [**4**]. Engine torque was proportional to pedal position via a conventional throttle arrangement (for spark ignition implementations). Full pedal demand forced the transmission to its lowest ratio to deliver high engine speeds in proportion to vehicle speed. A pedal demand of zero put the transmission at its highest ratio to deliver the lowest engine speed possible at the current road speed.

Intermediate pedal demands delivered engine speeds placed linearly between the two ratio limited extremes. An auxiliary shift lever position (designated *low*) raises the engine speed selected for small pedal demands to deliver a more sporting feel and greater engine braking on the overrun. This strategy has been successfully applied in a wide range of applications, delivering good subjective drivability but unremarkable fuel economy. For this work the strategy was emulated electronically within the transmission controller with proportional control of engine torque by the driver via the conventional pedal interpreter used in manual transmission vehicles.

2.2 Integrated powertrain controllers

The approach taken in this work was to use the distinction between steady and transient driving as the basis of the strategy. In particular it was seen as desirable to construct a controller that follows the IOL during more steady periods of operation but allows the engine to deviate in a controlled manner from the IOL during transients. This hybrid structure allowed quite different aims to be pursued in the steady state to those which had to be met during transient driving. Steady state operation was aimed purely at optimizing the trade-off between fuel consumption and emissions. During transient driving, this goal was subordinate to the requirement to deliver acceptable vehicle drivability. The challenge was to blend the two regimes into a seamless and progressive characteristic without unduly compromising the effectiveness of either. A further complication arose, as the IOL itself had a direct and significant effect on transient drivability. Despite the fact that the IOL was only followed during steady periods, the starting point for a transient event was, by definition, an arbitrary point on the IOL. The relative positions of the IOL and the limiting torque curve (LTC) of the engine define the amount of extra torque available for rapid manoeuvres. This is discussed in more detail in Section 3.2.1 where a specific example is presented.

In this paper, two alternative approaches to the problem are compared. The first approach used fuzzy logic to control the powertrain based on the driver's demand. The second approach was based on the use of two control functions utilizing either neural networks or more traditional grid interpolation functions to trim the power demand during transients. The new approaches were compared with an existing standard controller based on an earlier hydromechanical design incorporated in a series production transmission.

For the new control strategies it was considered that there were two principal control variables. These were the transmission ratio and the power supplied by the engine. The engine power was controlled via modulation of the engine torque, which was governed by the fuel control system. When the transmission clutch was engaged, the engine speed was controlled via the transmission ratio.

2.3 Controller architecture

The generic controller architecture was designed in a centralized hierarchical manner, shown in Fig. 1. There were two distinct functional areas: the supervisory controller and the powertrain controllers (engine and transmission). The two alternative supervisory controllers referenced an operating line optimizer function described by Brace *et al.* [**5**]. The driver power demand was passed to the optimizer function which interpolated along the IOL to return engine speed and torque coordinates describing the corresponding ideal operating point (IOP). This point formed the basis of the steady state behaviour of both controllers. The remainder of the supervisory control task attempted to determine the best compromise between the IOP returned and transient drivability requirements to arrive at a final engine speed and torque demand. This engine speed and torque vector were passed to the inputs of the powertrain controllers which were in direct control of the engine and transmission.

2.4 Fuzzy logic rule-based open-loop supervisory controller (fuzzy controller)

Fuzzy logic and fuzzy control were first introduced by Zadeh [**6**]. They have been widely implemented in domestic appliances and industrial processing plant. They have also been used in the control of conventional automatic transmissions, primarily as a means of classifying driving style or driver intent. Fuzzy logic was thought appropriate here because of its ability to represent semantic and rule-based knowledge in a well-

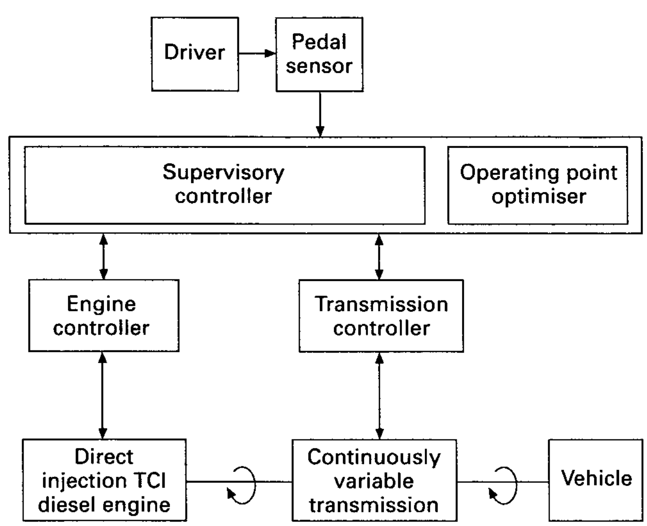

Fig. 1 Schematic of powertrain and controllers

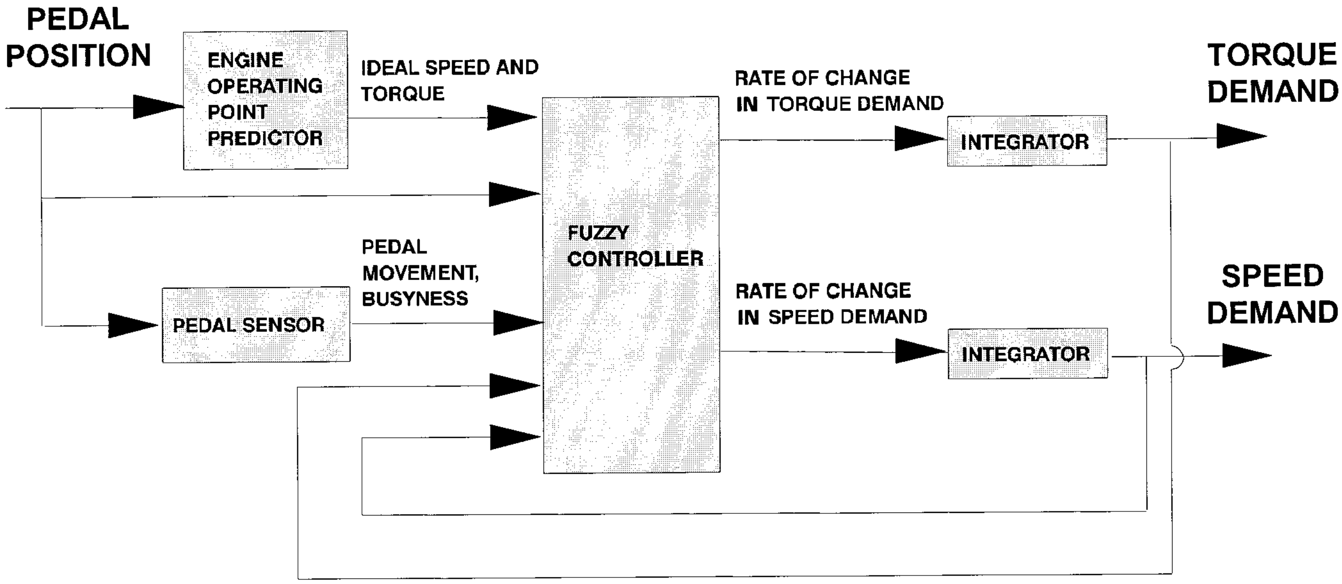

Fig. 2 Schematic of fuzzy supervisory controller

defined mathematical form and because of its ability to blend together action from different control algorithms, thus resulting in a smooth plant operation.

Figure 2 shows the fuzzy controller architecture. The fuzzy controller had five inputs, each of which was mapped to the fuzzy terms, either small or large and positive or negative. The pedal position was passed to the operating point optimizer, which returned the instantaneous IOP to the fuzzy controller. The pedal position itself, together with its rate of change and a measurement of the immediate history of its rate of change, defined as its busyness, were also inputs to the fuzzy controller. The current supervisory controller outputs of torque and speed demand were also used as inputs to the fuzzy controller. The fuzzy controller categorized the input vector and, using the expert knowledge encapsulated within the structure, generated demanded values for the rate of change in engine torque and speed. These were applied to integrators, the outputs of which were the current values of the torque and speed demands to the powertrain controllers. Steady state adherence to the IOL was achieved by ensuring that, as the difference between the instantaneous IOP and the demanded speed and torque converged to zero, the demanded rate of change of speed and torque decayed to zero. Away from the steady state the fuzzy structure allowed a smooth blend between a number of different rules, each aimed at specific driving situations. The strategy will be discussed more fully in a later paper.

2.5 Transient shaping controller

A second supervisory control structure was developed with the aim of controlling the deviation of the engine operating point from the IOL more precisely than was possible with the fuzzy logic controller. Inherent in the fuzzy logic approach used was that there was no predetermined path for the engine during a transient. For the new controller, priority during steady state operation and during slow transients was given to the low emissions/high economy objective as before. To manage the deviation from the IOL during transients the torque and speed demands were trimmed to enable movement of the powertrain along operating paths away from the IOL. This trimming or 'shaping' of the operating path during powertrain transients gave rise to the name of this approach.

Figure 3 shows the architecture of the transient shaping controller. The structure can be divided into two parts, the first concerned with the engine torque demand and the second with the engine speed demand. Links between the two parts were active during transient conditions and zero during steady state operation and are explained in the following section.

2.5.1 Engine torque demand

Measured engine speed was passed to the engine operating point optimizer (labelled as instance 1), which was used to set the corresponding ideal engine torque. This gave one component of the torque demand, shown as torque demand 1. Instance 2 of the engine operating point optimizer was used to set an instantaneous ideal torque corresponding to the measured pedal position. The error between torque demand 1 and the instantaneous ideal torque was passed, along with pedal position, to network 1. This function was used to set torque demand 2 which was the additional torque added to torque demand 1 to force the operating point to deviate

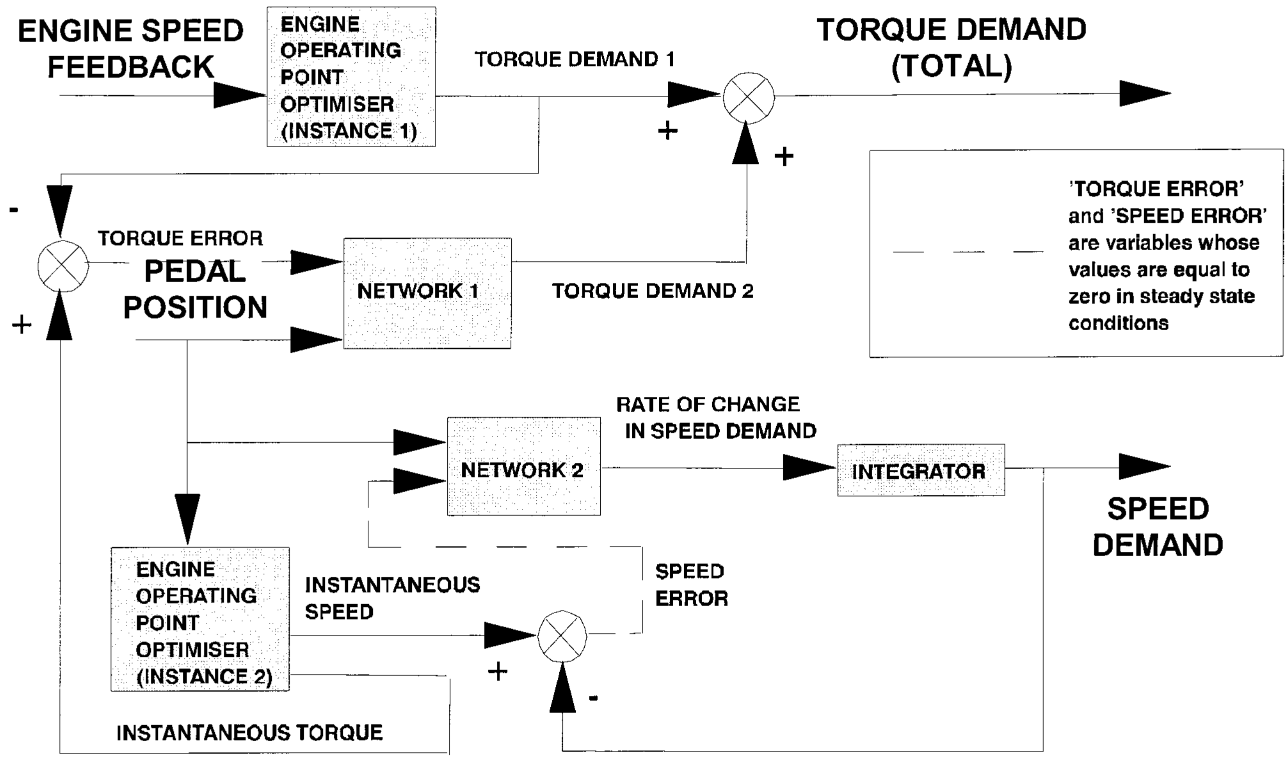

Fig. 3 Schematic of transient shaping controller

from the IOL during transient manoeuvres. When the system was in steady state the measured engine speed was equal to the ideal engine speed for the power demand as described below. This ensured that torque demand 1 was equal to the instantaneous ideal torque. The torque error was then zero, causing network 1 to set torque demand 2 to zero. In this case torque demand 1 became the (total) torque demand. This constrained the system to operate along the IOL in steady state. During transient operation the output of network 1 varied according to the surface shown in Fig. 4. For example, a large increase in power demand to the maximum will cause a large error between torque demand 1 and the instantaneous ideal torque corresponding to the demanded power. This will result in a large value for torque demand 2 which, when scaled and summed with torque demand 1, will drive the engine away from the IOL to allow enhanced transient performance. When the actual engine speed converges with the ideal engine speed as described below the torque error will decay to zero, causing torque demand 2 to fall to zero, according to Fig. 4, and the engine will once again be on the IOL.

2.5.2 Engine speed demand

Pedal position was interpreted as a power demand and passed to instance 2 of the ideal operating point optimizer which returned the corresponding instantaneous ideal engine speed. This instantaneous ideal engine speed was compared with the current speed demand. The resulting speed error was passed, with the measured pedal position, to network 2. This network set the rate of change in engine speed demand, which was the input to an integrator, which in turn determined the actual engine speed demand. In steady state the speed error approached zero, causing the output of network 2 to decay to zero, thus holding the current speed demand. This speed corresponded to the ideal engine speed returned by instance 2 of the operating point optimizer. During a transient the output of network 2 varied according to the surface shown in Fig. 5. For example, a rapid increase in pedal demand to maximum will cause a large error between the instantaneous ideal speed (the ideal speed for maximum power will be the rated speed of the engine) and the currently demanded engine speed. This in turn will cause the demanded rate of change in engine speed to be set high and positive by network 2. If the pedal is left at this position the demanded engine speed will increase to approach the ideal engine speed, reducing the error term and causing the demanded rate of change in engine speed to decay to zero as the new steady state point is reached. The transient shaping supervisory control strategy will also be discussed in more detail in the forthcoming paper mentioned earlier.

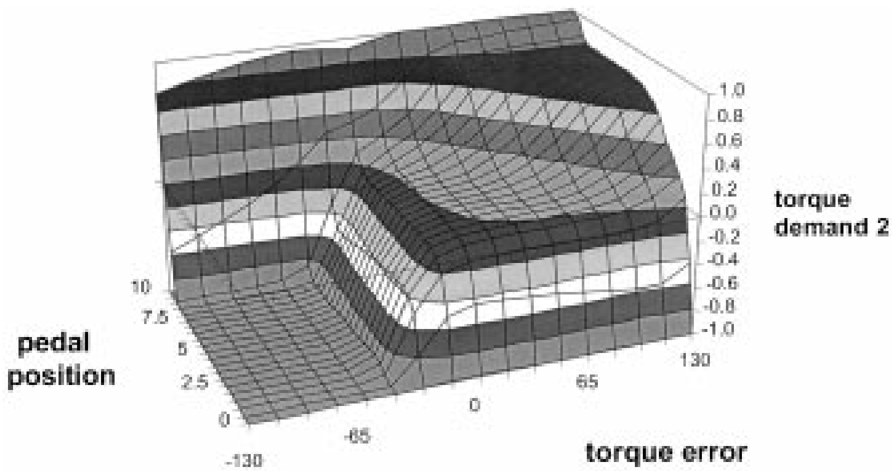

Fig. 4 Transient shaping torque trimming network

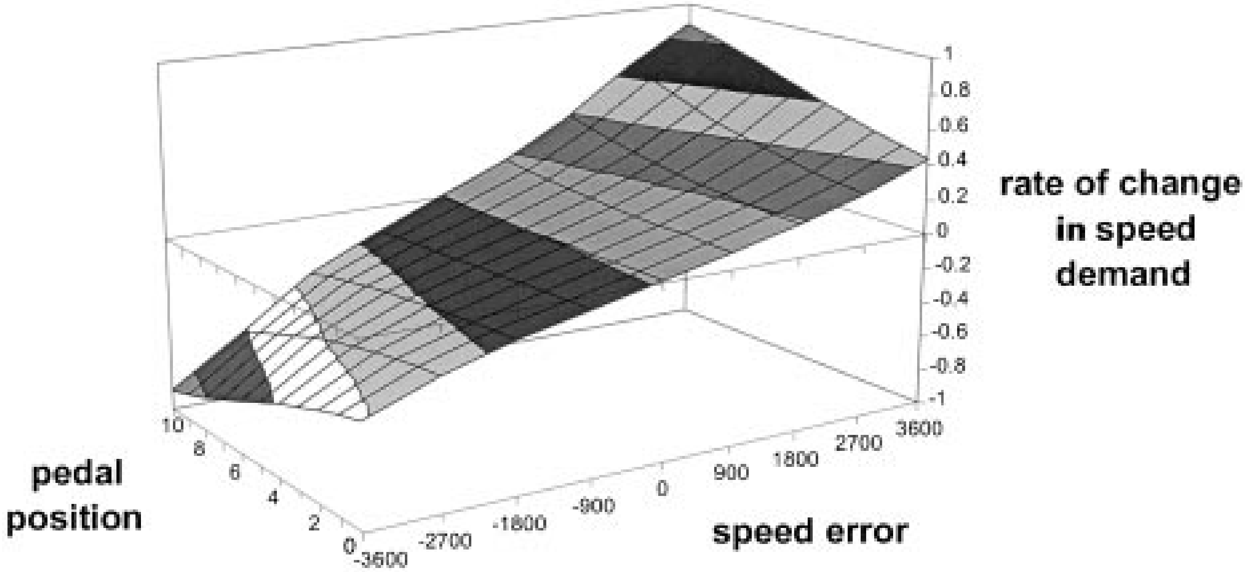

Fig. 5 Transient shaping speed rate of change network

3 APPLICATION TO DIESEL CVT POWERTRAIN

The new control strategies were developed in a generic fashion to allow their implementation for a wide range of powertrains. In order to demonstrate these strategies a simple control structure was implemented for a suitable powertrain (described below). A medium-sized passenger car (Ford Orion) was used as the experimental vehicle for the work. The system, comprising engine, transmission, vehicle and driver were modelled as described by Deacon *et al.* [**7**] using the Bath*fp* simulation software developed at the University of Bath. This enabled the early development and comparison of controller strategies. The simulation work included investigations into both dynamic system performance and vehicle emission formation during transient manoeuvres represented by the legislative test.

3.1 Transmission hardware and control

The transmission used was a modified Ford continuously variable transaxle (CTX). The version used, the largest available at the time, was rated for an input torque of 130 N m. The transmission utilizes the Van Doorne (VDT) push belt variable speed unit described by Hendriks [**6**]. The ratio is varied by altering the separation between the input (or primary) pulley sheaves which determines the radius taken up by the belt on the pulley. This must be matched by an opposite movement of the output (or secondary) pulley sheaves. Each pulley is controlled by its own hydraulic actuator. The hydromechanical control unit was upgraded to enable electrohydraulic control by the addition of two proportional solenoid valves, which were used to control the primary and secondary actuators. The transmission ratio was controlled by modulating the transmission primary pressure via the primary so-

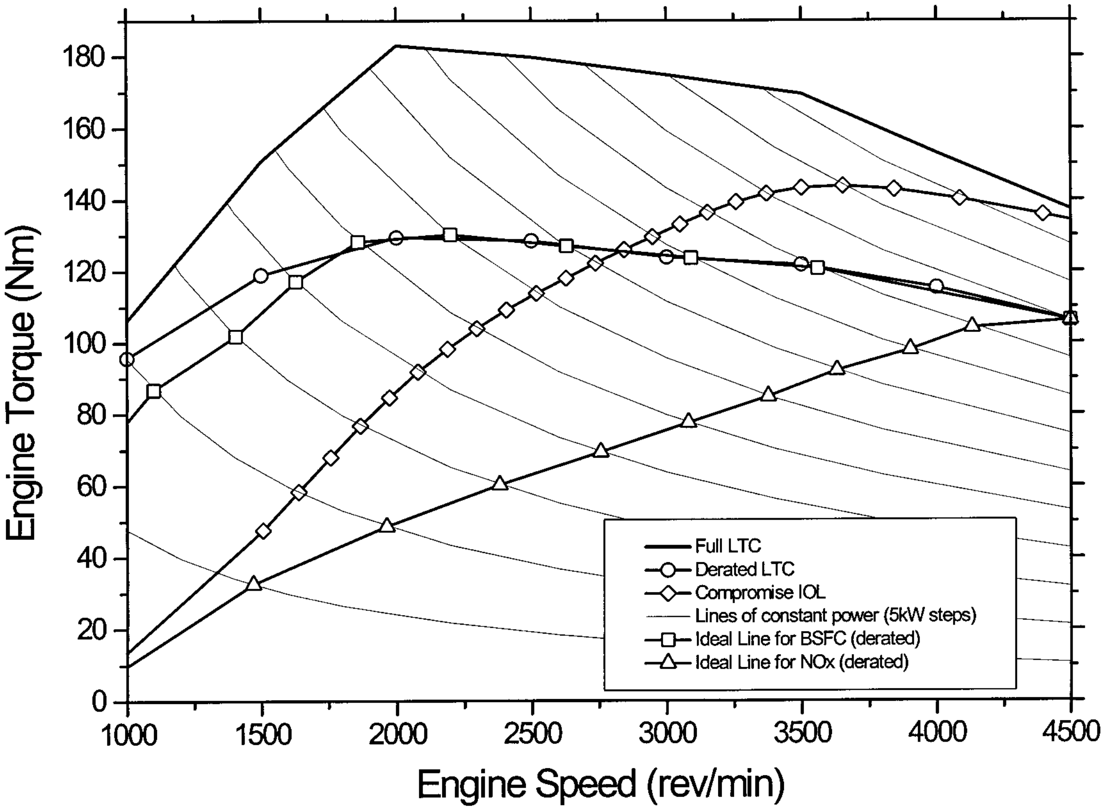

Fig. 6 Ideal operating lines

lenoid current. Rig test work using the VDT transmission controller showed that it was capable of moving the transmission ratio to the value necessary to achieve the demanded primary speed with negligible overshoot. The VDT controller was therefore retained, avoiding the requirement to develop a lower-level controller. The secondary pulley pressure was likewise controlled by the VDT controller. The secondary pressure affects the torque-transmitting capacity of the variable speed unit. Sufficient pressure must be applied to prevent the belt or pulleys being damaged by excessive slippage. Conversely, losses in the transmission are directly related to the magnitude of this pressure, requiring a delicate balance between efficiency and durability [**8, 9**]. The demand was generated by an algorithm taking as its inputs engine torque and transmission ratio. A safety factor was incorporated to enable torque transmission, in the presence of external disturbances, without gross belt slippage. The secondary pressure control loop is closed by a signal fed back from a pressure transducer. The hydraulic circuit used to control the clutch remained as in the hydromechanical version of the CTX. The modified transmission was designated CTXE.

3.2 Engine hardware and control

The engine used for this work was a prototype Ford 1.8 L direct injection turbocharged and intercooled diesel engine. The Lucas electronic pumping injection control (EPIC) system was used for both fuel injection and exhaust gas recirculation (EGR) control. The powertrain controller set the value of the EPIC fuel injection system demand signal in order to achieve the demanded engine torque. This was an open-loop process. An algorithm was developed based on a model of the EPIC demand/engine torque characteristic. Measured engine speed was an input to this algorithm, together with the demanded engine torque.

Two calibrations of engine were used. Calibration 1 was a baseline configuration used for all of the data generation, rig test and initial chassis dynamometer work. Calibration 2 incorporated some software developments derived from a separate engine development program.

3.2.1 Calibration 1

For the first phase of the work a 130 N m, derated version of the full 180 N m engine torque curve was used, as shown in Fig. 6. This was to safeguard the

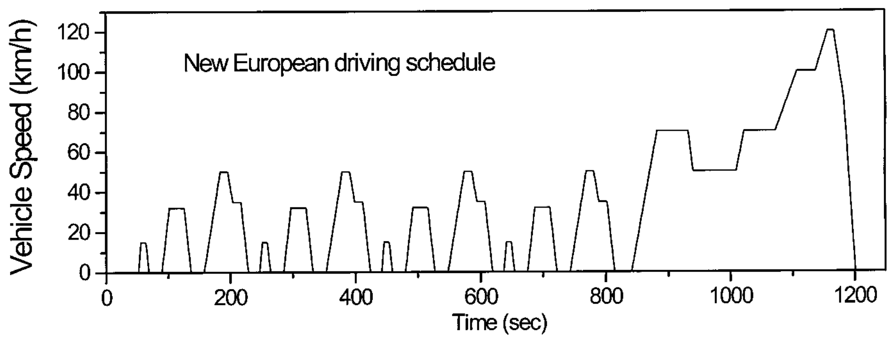

Fig. 7 New European driving schedule

transmission used and it reduced the rated engine power from 65 to 50 kW. For an initial investigation into the effect of the IOL on vehicle performance a pair of extreme IOLs was selected. These were the IOL for brake specific fuel consumption (b.s.f.c.) and the IOL for nitrogen oxides (NO_x). The b.s.f.c. line had the lowest engine speed for all powers of all the lines considered, the NO_x line the highest. Other lines for hydrocarbons (HCs), particulate matter (PM) and smoke lay between these two lines. The low speed of the IOL for b.s.f.c. has important advantages in terms of noise, vibration and harshness (NVH), although the ratio constraints of the transmission prevented it from being achieved during significant periods of steady driving. Another constraint on this line was that the control of the clutch was hydromechanical and even at the limit of the adjustment possible it opened at too high an engine speed to allow the best use of the engine to be made. Electronically controlled clutches have since overcome this problem. In this event the limiting factor would be the stability of the engine at low speed, high torque operation and the NVH characteristics at this condition. Dynamically the IOL for b.s.f.c. limits the peak acceleration achievable owing to its proximity to the LTC used. A rapid increase in torque moving from a point on the IOL directly to the LTC does not produce significant additional power. At 2000 r/min the IOL is coincident with the IOL, requiring an engine speed increase to generate more power. This is a slow operation, as discussed in Section 2. In contrast, the NO_x line is some distance beneath the LTC. At 2000 r/min almost 20 kW more power can be generated without altering engine speed. This drivability benefit is at the expense of refinement because of the higher steady state engine speeds selected.

3.2.2 Calibration 2

After completion of the main test phase the full torque curve was reinstated 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 per cent. The nozzle opening pressures of the injectors were lowered slightly to improve the idle stability. Engine dynamometer results showed the 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 Fig. 6. This represented the mean of the ideal speeds for all regulated pollutants of interest (HCs, NO_x and PM) at each engine power. The line crosses the derated torque curve at around 37 kW, although this does not affect the comparison between these and earlier tests, as this high-power region of the map is not used during the manoeuvres considered. The compromise line demonstrates an intermediate calibration between the two extremes investigated initially.

It should be emphasized that neither engine calibration considered here is representative of current builds. The emissions and fuel consumption performance of the base engine have been improved considerably since the hardware specification was fixed at the start of the work described here. It was considered 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.

4 VEHICLE TEST RESULTS AND DISCUSSION

A series of tests was performed on the chassis dynamometer facility at the Ford Engineering Centre, Dunton. The new vehicle homologation procedure introduced in European Directive 91/441/EEC was selected 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 (Fig. 7). The limit values for exhaust emissions are shown in Table 1, together with draft limits for stage 3 in 2000. The stage 3 standards will be accompanied with a change in the

Table 1 European diesel passenger car emission standards

	CO (g/km)	NO_x (g/km)	HCs + NO_x (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 s idle now included

Table 2 Chassis dynamometer test results—new European cycle

Test	Engine calibration	Engine ideal operating line	Powertrain controller	Pedal signal (for secondary pressure set point) (%)	HCs (g/km)	CO (g/km)	NO_x (g/km)	HCs + NO_x (g/km)	CO_2 (g/km)	PM (g/km)	Fuel (L/100 km)
1	1	b.s.f.c.	Transient shaping	100	0.1508	0.5535	0.6633	0.8141	159.8	0.0823	5.99
2	1	b.s.f.c.	Fuzzy logic	100	0.1383	0.4999	0.6917	0.8300	164.0	0.0869	6.14
3	1	NO_x	Transient shaping	100	0.1596	0.5889	0.4349	0.5945	172.5	0.1040	6.47
4	1	NO_x	Fuzzy logic	100	0.1233	0.5410	0.5140	0.6373	176.1	0.1100	6.60
5	1	n/a	Hydromechanical	100	0.1523	0.6745	0.5943	0.7466	215.9	0.2136	8.09
6	2	Compromise	Fuzzy logic	100	0.0937	0.3937	0.6078	0.7015	168.6	0.0795	6.37
7	2	Compromise	Transient shaping	100	0.0960	0.3996	0.5573	0.6533	167.6	0.0785	6.34
8	2	Compromise	Transient shaping	50	0.1025	0.4107	0.5203	0.6227	164.2	0.0750	6.21
9	2	Compromise	Transient shaping	30	0.0962	0.3876	0.5270	0.6232	159.7	0.0949	6.04
10	1	n/a	Manual transmission	n/a	0.1900	—	0.4800	0.6714	—	0.1180	6.54

test cycle: the current first 40 s of idle before emission sampling commences will be deleted, so that sampling will start at cranking. During each test a large number of variables were logged, including the regulated HC, CO, NO_x and PM exhaust emissions. Exhaust carbon dioxide (CO_2) was also logged, from which fuel consumption was calculated. Results for pollutants are presented in grams per kilometre travelled during the test as prescribed by the legislative limits. Fuel consumption is presented as the number of litres required to travel 100 km.

A summary of the chassis dynamometer test results is given in Table 2. Tests 1 and 2 used the IOL for b.s.f.c. Tests 3 and 4 used the IOL for NO_x. Tests 6 to 9 used the mixed or compromise IOL. These three IOLs are shown in Fig. 6. Tests 1, 3, 7, 8 and 9 used the transient shaping controller. Tests 2, 4 and 6 used the fuzzy controller. All tests used engine calibration 1 with the exception of tests 6 to 9, which used engine calibration 2. Test 5 was performed using the same powertrain running an emulation of the standard hydromechanical control strategy. Test 10 was performed using a vehicle equipped with a manual transmission for comparison.

A standard graphical format for the presentation of emission results is to plot PM against the sum of HCs and NO_x. This allows convenient representation of the legislative limits and highlights the well-known trade-off between PM and NO_x, a key variable in powertrain calibration. Figure 8a shows the results plotted in this format with the legislative limits for stages 1, 2 and 3 superimposed. Future emissions legislation will quote separate limits for HCs and NO_x, which will require a modified graphical presentation. Moreover, this presentation does not give any indication of the fuel consumption achieved by the different controllers. Fuel consumption must be viewed in the context of the emissions limits, as generally an improvement in fuel consumption will adversely affect NO_x and PM results. To this end, Fig. 8b shows the link between fuel consumption and HCs + NO_x while Fig. 8c shows the PM results plotted against fuel consumption.

4.1 Comparison between control strategies

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 (test 5). 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 focused 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 as with the IOL for NO_x described in Section 3.2.1. The high engine speeds can be observed in Fig. 9 which plots fuel demand (analogous to engine torque) against engine speed for three of the controllers considered at half second intervals over the whole drive cycle. The controller using the IOL for b.s.f.c. generally uses the highest fuel demands and lowest engine speeds, as

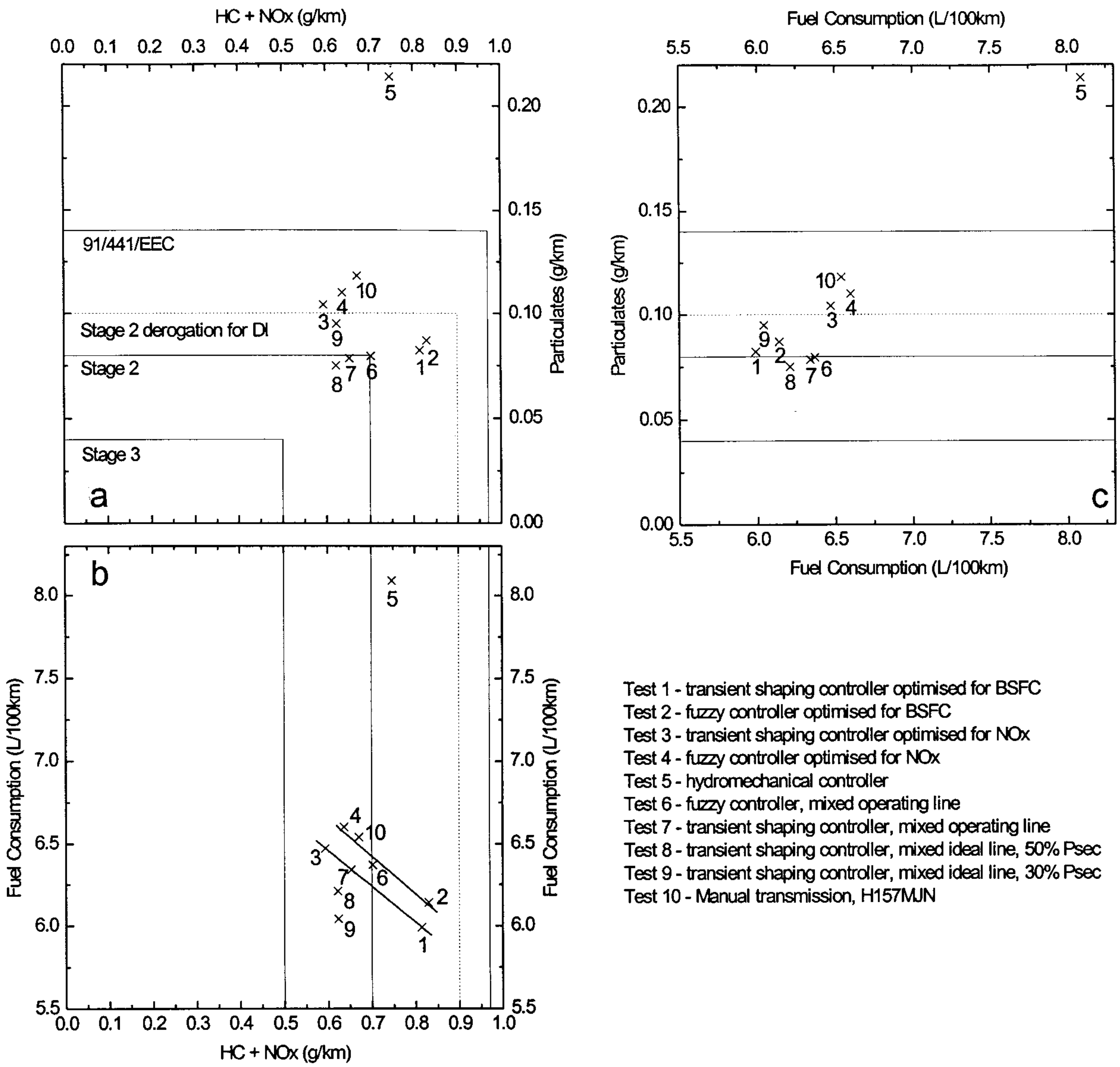

Fig. 8 Chassis dynamometer test results

expected. The controller using the IOL for NO_x 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 emphasized that the hydromechanical controller could be tuned to demand much lower engine speeds and hence give improved emissions and fuel economy. The strategy does not, however, have an IOL as its basis and is designed for implementation without drive by wire control of the engine. As such, an inferior economy and emissions performance would be expected. Further illustration of the differences between the two strategies is given in Fig. 10, which shows the engine speeds in the time domain over the extra urban portion of the test.

The differences between the two new strategies are not as marked as those discussed above, nevertheless differences are evident. In Fig. 8a the transient shaping controller (tests 1, 3, 7, 8 and 9) is consistently closer to the origin than the fuzzy controller (tests 2, 4 and 6). The differences are small but significant, a conclusion reinforced in Fig. 8b. A linear fit to the data for each controller shows the fuel consumption performance of the transient shaping controller to be around 0.2 L/100 km better than that of the fuzzy controller for an equivalent $HCs + NO_x$ performance. Although this is an improvement of only around 3 per cent, this is significant as fuel consumption improvements are difficult to achieve through changes to engine hardware. This difference is due to a combination of factors but is largely the result of a slightly different calibration. Further work has shown the calibration of each strategy to be similar in terms of drivability performance but not identical. Even allowing for the uncertainties inherent in such an assessment, the fuzzy controller had a slightly better subjective rating, corre-

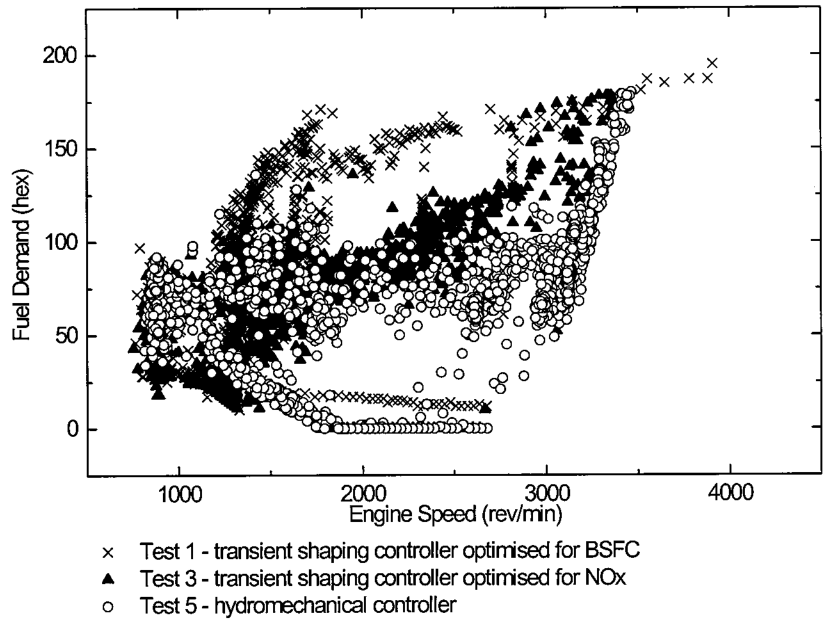

Fig. 9 Engine fuelling and speeds used during drive cycle

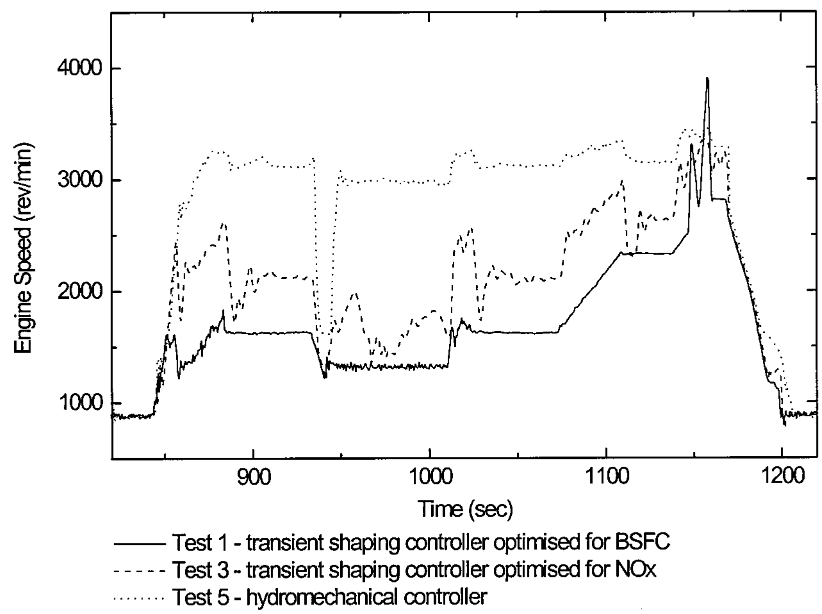

Fig. 10 Engine speeds during EUDC

sponding to the larger deviations from the IOL observed. This evidence suggests an inverse correlation between subjective drivability and exhaust emissions and economy which must be considered during the calibration process.

4.2 Comparison between IOLs

As expected, the controllers optimizing for b.s.f.c. returned the lowest fuel consumption figures (tests 1 and 2; test 9 was also very good, although improved transmission efficiency was a primary cause in this case) and those optimizing for NO_x returned the lowest NO_x (tests 3 and 4). In parallel with improved fuel consumption the IOL for b.s.f.c. gave better PM results, also on account of the low engine speeds selected (Figs 9 and 10). The mixed line fell between these extremes and looks the most impressive in relation to the stage 2 legislative requirements as shown in Fig. 8a. In general, those IOLs with higher engine speeds showed a better NO_x performance at the expense of the PM results. The relationship is not linear owing to the non-linear shapes of the emissions characteristics concerned. As extremes of IOL are approached, the incremental benefits observed are small in relation to the penalties.

The effects on emissions of improving fuel consumption must also be considered. Often it is tempting to view fuel consumption as a measure of controller performance without reference to emissions performance. Figure 8b shows the relationship between fuel economy and HCs + NO_x figures achieved by each controller to be approximately linear in the region considered. Moving the IOL from the b.s.f.c. line progressively through the mixed line to the NO_x line causes a predictable degradation in fuel consumption as engine speeds are increased for the same power. This is an important finding as it allows a judgement to be made regarding the calibration of the powertrain. A linear fit to the data suggests that an optimum fuel economy of around 6.25 L/100 km would be achieved by the transient shaping controller if the HCs + NO_x performance were set to the stage 2 limit. In practice an HCs + NO_x level somewhat below the stage 2 limit would be set to allow for production variability and long-term compliance.

One of the most challenging aspects of emissions legislation compliance today is the control of particulate matter. Limits are set very low as the effect on human health is considered disproportionately high. Fuel consumption improvements must be achieved in harmony with PM reduction wherever possible. Additionally, PM is always the most difficult pollutant to measure with high repeatability; indeed, 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 10 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 the driver, controller 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 NO_x, HCs and fuel.

From a knowledge of the PM contours across the engine operating map and an inspection of Fig. 8c it is suggested that for this powertrain there is an optimum fuel consumption of around 6.25 L/100 km, which returns the lowest PM figures. Either side of this condition the PM level starts to rise once more. The scatter on the data prevents a curve being fitted with any degree of confidence, although it is estimated that the PM level at this condition would be around 0.7 g/km, just on the limit of stage 2 and comfortably below the derogation for DI allowed until 1999. Further testing would allow this hypothesis to be more fully investigated.

4.3 Transmission efficiency effects

Test 8 had the secondary (or clamping) pressure of the transmission reduced by reducing the pedal signal input to the algorithm controlling secondary pressure by 50 per cent. This will reduce the secondary pressure by an amount less than 50 per cent, as transmission ratio has a strong influence on the set point. Nevertheless, efficiency will improve at a possible cost to long-term durability [**9**]. Figure 8b shows the effect of this change to be a marked improvement in the fuel economy versus HCs + NO_x trade-off. If the same characteristic gradient as for tests using the full secondary pressure is assumed, this would suggest a fuel consumption of around 6 L/100 km (or a 4 per cent improvement compared with the full secondary pressure). A further reduction in the pedal signal to only 30 per cent of the original level (test 9) produced a further saving which may be extrapolated using the same assumption to give around 5.8 L/100 km (or a 7 per cent improvement over the standard setting). These improvements are valuable, although the effect on the durability of the transmission must be considered. There is no 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 durability assessment program would be required before the secondary pressure could be reduced significantly.

4.4 Comparison with manual transmission

The data from the equivalent manual powertrain (test 10) shows a similar fuel consumption and HCs + NO_x performance to the fuzzy logic controller. The transient shaping controller demonstrates significantly better performance. For equivalent HCs + NO_x the transient shaping controller would offer an improvement of around 0.25 L/100 km (4 per cent) in fuel consumption. This is encouraging as it demonstrates that the CVT can fulfil its much anticipated potential with the use of advanced control strategies. Comparison with the hydromechanical controller (test 5) shows the contrasting deficit in performance typical of series production systems.

5 CONCLUSIONS

Two novel control strategies have been developed for use with integrated CVT powertrains. Both strategies have as a central concept the ideal operating line (IOL), which is an extension of the economy line concept to include exhaust emissions. Drivability is considered by allowing the engine to deviate from the ideal operating line during transient manoeuvres. The first strategy achieves this using fuzzy logic and the second through a mapped deviation of engine torque and speed from the IOL, known as transient shaping. Implementation of these control strategies for an integrated diesel CVT powertrain allowed the economy and emissions performance of the vehicle to be tuned to achieve the optimum performance from the hardware used in the study. Compared with conventional control strategies over the new European drive cycle 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 HCs + NO_x can be plotted for a given control strategy, allowing the informed calibration of the controller to return the best fuel consumption possible within the legislative constraints on emissions. It is likely that a similar trade-off between PM and fuel consumption could be developed with sufficient data.

The transient shaping controller was designed with special attention to the control of transient deviations from the IOL. This emphasis, and a slightly more conservative calibration, seems to have given the transient shaping controller the advantage in terms of emissions and economy improvements. The measured differences between them reflect both the differences in the strategies and the state of tune of the calibrations.

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