The Compromise in Reducing Exhaust Emissions and Fuel Consumption from a Diesel CVT Powertrain Over Typical Usage Cycles

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Modern integrated powertrains allow great scope for the improvement of emissions and fuel consumption by careful optimisation of the engine speed and load selected to deliver the demanded power. A simple simulation technique is presented for the prediction of fuel economy and emissions from a passenger vehicle over typical drive cycles. The simulation may be used to investigate alternative software and generic hardware configurations at an early stage of development. Results are presented which demonstrate several steps that could improve the emissions performance of a vehicle equipped with an integrated Diesel/CVT powertrain.

Keywords / CVT, High Speed Direct Injection Diesel Engine, Simulation, Emissions

1. INTRODUCTION

Vehicle powertrains are becoming increasingly complex as the scope offered to improve vehicle performance, economy and emissions is explored. Considerable benefit may be derived from operating the engine and transmission in an integrated manner [**1**], using a single controller to interpret the driver's wishes and instruct the engine and transmission controllers in turn. Crucial to the success of such a system are the basic specification of the major components and the design of the overall powertrain control strategy.

As part of a project aimed at developing an integrated approach to CVT powertrain control [**2**] a simulation tool was required to allow the comparison of various controller strategies and hardware layouts. 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 type approval 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.

In addition to the emissions levels, predictions of engine speed and load on a second by second basis over the cycle were required to allow controller performance to be examined. Additionally the simulation was required to be readily applicable to other drive cycles and various transmission types.

Existing simulations of the system [**3**] were developed using an advanced dynamic simulation environment [**4**] using a variable time step integrator to solve the differential equations governing the system. This allows great accuracy as during periods of rapid change the time step can become very short. The major disadvantages of this technique are the long runtimes encountered and (in some cases) the large programming overheads when developing new models. To be of use during the design process the new simulation needed to be simple to calibrate and fast to run while producing results with sufficient accuracy to allow control strategies to be compared with confidence.

2. SIMULATION STRUCTURE

A simulation was developed which uses a simple fixed time step approach to reduce model development and run times by several orders of magnitude for simple simulations. The simulation is implemented within an *Excel* spreadsheet running under *Windows* on a PC.

The required drive cycle is defined as a time series of vehicle speeds. From this information and with knowledge of vehicle and transmission characteristics an engine speed and load may be calculated at each. The acceleration is simply that required to achieve the demanded speed by the end of the time step. The power requirement is determined by an empirical drag model of the vehicle expressed as a polynomial in terms of speed and the work required to accelerate the vehicle inertia. At this point inefficiencies in the powertrain may be included to arrive at a demanded engine power.

The engine speed and load are determined by the transmission ratio selected. In the work presented here a continuously variable transmission was used. The control strategies to be investigated used the concept of an ideal operating line (IOL) [**5**] to define the engine speed and torque to deliver the demanded power. This allowed engine speed to be set by a polynomial in terms of power. The engine torque required was then simple to calculate. The engine model was then used to estimate the emissions produced during each time step.

2.1 The Engine Model

For an optimum combination of speed and accuracy an empirical model of the engine was used. This approach was selected because experimental data were available for the engine under consideration. A suitable analytical model of the engine could be used as an alternative source of data. Such models tend to be large and slow running, in this case the engine model would be used to generate engine map data in a similar manner to that used on a test rig. This data would then be used in the cycle simulation.

The inputs to the engine model were engine speed and load and coolant temperature. These inputs are sufficient to represent engine operating conditon for the type of cycle under consideration. Coolant temperature is a required input if the variation in engine performance from a cold start is to be represented. Fort this work coolant temperature was taken from experimental data for a typical test. The outputs required were fuel flow and the various emissions under consideration. Experience with an earlier, transient, engine model [**6**] had suggested that it was a suitable application for a neural network.

A network of the form shown in **Figure 1** was trained using a subset of data gathered for the transient model. A sigmoidal transfer function was used for all neurons on the hidden and output layers. The engine used for the work was a 1.8 L turbocharged and intercooled direct injection Diesel engine equipped with exhaust gas recirculation. A steady state engine test rig was used to generate a basic performance map of the engine. This was was augmented by data gathered with cooler and warmer engine temperatures than normal.

The best accuracy of training was achieved with 30 neurons on a single hidden layer. Using two hidden layers introduced a tendency to instability during the training process with no increase in peak accuracy. The errors when compared to the training data for the network used are shown in **Table 1** below. Most of the predictions are subject to an RMS error of less than 5% of full scale. Fuel flow and HC predictions are particularly good. A subjective evaluation of the predictions was also used to check the general form of the surfaces represented by the network. This is an essential step in checking for generalisation and the absence of overtraining.

mass HC g/hr	**mass NOx g/hr**	**mass PM g/hr**	**fuel flow kg/hr**
2.78	4.43	5.42	2.53

Table 1 - Network Training Errors - RMS Errors on training data (% full scale)

3. MODEL VALIDATION

3.1 Experimental Data

Data were recorded during six vehicle tests performed on a chassis dynamometer over the European Drive Cycle at the chassis dynamometer facility at the Ford Engineering Centre, Dunton. The powertrain installed in the vehicle consisted of the 1.8 L Diesel engine as modelled above, a steel belt CVT modified to allow electronic control over ratio and secondary pressure, [**7**] and an oxidation catalyst. The system was controlled by a PC based algorithm using two different control structures. Both controllers used the concept of the ideal operating line as their basis, ensuring that steady state or near steady state performances were similar. Transient behaviour was governed by a 'transient shaping' technique in the first controller and using fuzzy logic in the second. The aim was to balance drivability, fuel economy and emissions considerations. Each controller was tested three times using different ideal operating lines as shown in **Figure 2**. The lines used were those for best fuel economy (BSFC), minimum oxides of nitrogen (NOx) and a mixed line which corresponded to the mimimum hydrocarbons (HC) line in **Figure 2**.

The results are presented in **Table 2** below. Test results are expressed as a mean in the form of grams per kilometre of each of the emissions since this is the form of the legislative limits. Mean fuel consumption is similarly reported as the number of litres required to drive 100km.

3.2 Simulation Data

The simulation was run using each of the IOLs for BSFC, HC and NOx. Allowances were made for transmission efficiency, the effect of ratio constraints and transmission creep as described below. The results are presented in **Table 2**

Test	Powertrain controller	Engine ideal operating line	HC g/km	NOx g/km	HC + NOx g/km	PM g/km	Fuel L/100km
1	Transient shaping	BSFC	0.1508	0.6633	0.8141	0.0823	5.99
2	Transient shaping	mixed	0.0960	0.5573	0.6533	0.0785	6.34
3	Transient shaping	NOx	0.1596	0.4349	0.5945	0.1040	6.47
4	Fuzzy logic	BSFC	0.1383	0.6917	0.8300	0.0869	6.14
5	Fuzzy logic	mixed	0.0937	0.6078	0.7015	0.0795	6.37
6	Fuzzy logic	NOx	0.1233	0.5140	0.6373	0.1100	6.60
Sim 10	Ideal	BSFC	0.2306	0.5885	0.8192	0.1791	5.756
Sim 12	Ideal	HC	0.2331	0.4977	0.7308	0.1681	5.865
Sim 11	Ideal	NOx	0.2587	0.4586	0.7173	0.2075	6.156

Table 2 - Vehicle test and simulation data for model validation

3.3 Comparison with Experimental results

Figure 3 shows a plot of the experimental data together with the simulation predictions. In **Figure 3a** the particulate matter is plotted against a sum of HC + NOx. This is a convenient format often used for the display of the results since it allows simple graphical representation of the legislative limits. The experimental results show the two control strategies tested perform in a similar manner, the main influence on emissions performance is the IOL used. The IOL for NOx gives a good HC + NOx sum but PM is poor because of the high engine speeds used. When optimising for BSFC the PM is also reduced but this is at the expense of higher NOx. This basic trade-off forms the basis for much of the optimisation work with such integrated powertrain systems. The intermediate or mixed operating line is a compromise between these two extremes as suggested by inspection of the IOL in **Figure 2**. The HC + NOx result is similar to that for the NOx line while the PM result is similar to that for the BSFC line. This IOL gives the best result when compared with the Stage 2 derogation limit for direct injection Diesel engines and is at around the level required for development targets at the time. In comparison with the simulation the most noticeable feature is the dramatically higher PM prediction. PM levels derived from the steady state rig (and used to train the neural network) are consistently around twice those measured on the chassis dynamometer due to differences in the measurement method. On the steady state rig PM is inferred from measured smoke and hydrocarbons using an empirical correlation. The coefficients of this correlation are very sensitive to engine specific variables and are difficult to determine without more data. On the chassis dynamometer PM are measured directly by observing the increase in mass of a filter paper. If this relatively constant gain error is allowed for, the behaviour of the simulation matches the experimental data well. The best IOL is still the compromise between the two extremes of NOx and BSFC. The poor PM performance of the faster NOx line is captured, as is the poor NOx performance of the BSFC line.

Figure 3b shows the fuel consumption data plotted against HC + NOx. Here the difference between the two control strategies tested is more evident. The transient shaping controller returns superior fuel consumption for each IOL due to its closer adherence to the IOL during transients. The IOL for BSFC performs best in each case as expected. The progressive deterioration in fuel economy as the two faster IOLs are used is quite marked. The simulation predicts better fuel consumption than achieved in practice. The simplistic expression used to estimate losses could be improved to close this gap but another cause of the discrepancy is that the simulation controls the powertrain in an ideal manner. No deviation from the IOL is allowed and the 'driver' always demands the correct power to allow the demanded speed to be achieved. Neither of these is true in the experimental work, causing increased fuel consumption. Indeed, the fuzzy controller is still worse than the transient shaping controller despite the plant and IOL being identical. The differences between the simulation and experiment and between the two experimental strategies are significant since fuel consumption is difficult to improve using engine changes alone. The simulation can be used to assess candidate control strategies against the ideal case.

Figure 3c shows PM plotted against fuel consumption. As in **Figure 3a** the PM predictions are around twice the measured levels although the general behaviour is replicated. Again the compromise IOL performs better than either of the extremes.

The above comparison gives a degree of confidence in the simulation. Although the absolute levels of PM are not predicted well, all the important trends and compromises are captured. This allows the simulation to be used at the concept stage to compare candidate software and hardware strategies in a simple and cost-effective way. Promising solutions will always need to be evaluated using either a full dynamic simulation or experimental work.

Sim	Ideal Line	Ratio Limits	Trans. Losses	Creep Load	HC		NOx		HC + NOx		PM		Fuel	
					g/km	% inc.	g/km	% inc.	g/km	% inc.	g/km	% inc.	L/100 km	% inc.
1	BSFC				0.2236		0.5314		0.7550		0.1565		4.790	
2	NOx				0.2446		0.3512		0.5959		0.1530		5.227	
3	HC				0.2190		0.3874		0.6064		0.1214		4.930	
4	BSFC	X			0.2310	3.3	0.4387	*-17.5*	0.6697	*-11.3*	0.1443	*-7.8*	5.032	5.1
5	NOx	X			0.2518	2.9	0.3564	1.5	0.6082	2.1	0.1634	6.8	5.319	1.8
6	HC	X			0.2326	6.2	0.3829	*-1.2*	0.6156	1.5	0.1383	13.9	5.106	3.6
7	BSFC	X	X		0.2409	4.3	0.5738	30.8	0.8148	21.7	0.1657	14.8	5.514	9.6
8	NOx	X	X		0.2690	6.8	0.4441	24.6	0.7131	17.2	0.1941	18.8	5.915	11.2
9	HC	X	X		0.2434	4.6	0.4831	26.2	0.7265	18.0	0.1547	11.9	5.623	10.1
10	BSFC	X	X	X	0.2306	*-4.3*	0.5885	2.6	0.8192	0.5	0.1791	8.1	5.756	4.4
11	NOx	X	X	X	0.2587	*-3.8*	0.4586	3.3	0.7173	0.6	0.2075	6.9	6.156	4.1
12	HC	X	X	X	0.2331	*-4.2*	0.4977	3.0	0.7308	0.6	0.1681	8.7	5.865	4.3

Table 3 - Summary of Drive Cycle Predictions including % increase when compared with previous simulation using the same IOL

4. INVESTIGATION OF TRANSMISSION EFFECTS ON FUEL ECONOMY AND EMISSIONS

In order to investigate the effect of various aspects of the transmission design several series of simulations were performed using the structure described above. Four levels of complexity have been modelled as detailed below. Table 3 below shows the results from the twelve simulations described below. The transmission characteristics compared are shown in columns 3-5 of the table.

4.1 Ideal transmission

Here the transmission is modelled as a zero power loss device with unlimited ratio capability. No catalyst is modelled. This is useful to provide a target figure for drive cycle tests using the engine calibration under investigation. Simulations of the European cycle test were performed using the BSFC line, the NOx line and the HC line. The results are shown in **Table 3** numbered 1 to 3. The worst result in terms of legislation is that obtained using the BSFC line (Simulation 1). Although the simulation predicts the best fuel consumption, the emissions predictions place the vehicle significantly outside the EEC stage 1 limit for particulates. The experimental results presented above suggest that the PM figure for such a cycle may be less by around 50% than the levels predicted. The result is also high, however, when compared with the predictions from simulations following other IOLs. This is due to the operation of the engine at high load conditions where, although BSFC is minimised, PM production is high due to the low air-fuel ratio.

The predictions for the NOx line (Simulation 2) show an improvement in NOx of 34% relative to the BSFC line but the penalty is a 9% increase in fuel consumption and a 10% increase in HC production. A small improvement in PM production is predicted. The net result is, however, not good enough to bring the result inside EEC stage 1 limits.

The HC line (Simulation 3) gives the best overall result in terms of the legislative requirements. The HC prediction is low although the BSFC line was as good here. The real benefit is that the PM prediction is the lowest of any of the three lines, indeed it is 21% better than the NOx line and 22% better than the BSFC line. This is due to the optimum line for PM being very similar to the optimum line for HC. PM production escalates rapidly either side of this optimum. The NOx prediction is 27% lower than for the BSFC line and only 10% worse than for the NOx line. These effects move the result significantly nearer to the origin and into the acceptance zone for stage 1 of the EEC directive.

The contrast between the three IOLs can be seen clearly in **Figure 4**. **Figure 4a** presents the data as a plot of PM against HC + NOx, showing the characteristic trade-off curve. This response is dominated by the trade-off between NOx and PM. The minimum PM is achieved using the HC line (similar to the PM line as in **Figure 2**). The IOLs using faster or slower engine speeds demonstrate worse PM performance. **Figure 4b** shows an equally clear trade-off between fuel consumption and HC + NOx, although here there is no optimum point evident, the fuel consumption gets worse as the NOx falls. This is due to the choice of IOLs for the simulations. If an IOL giving higher engine speeds even than those for NOx were used the NOx performance would start to deteriorate as fuel consumption rose even further. **Figure 4c** shows the trade-off between fuel consumption and particulates. Again, the IOL for HC returns the minimum PM of the three IOLs considered. The three trade-off curves may be summarised by the two main features: -

- NOx performance continues to improve as fuel consumption is allowed to rise over the range of IOLs considered here

- There is a clear optimum calibration for PM. Performance deteriorates rapidly on either side of this optimum.

Figure 5 shows the engine torques used by the three IOLs over the EUDC. The torque must rise for the IOLs using lower engine speeds. The contrast between the three IOLs is clear.

The results from the simulation using such an ideal transmission may be regarded as the best possible figure achievable using the engine under consideration. No limitation is placed on the controller in that it achieves the ideal line at all times. No losses are modelled and the ratio range is not constrained. As such these results represent a benchmark for the evaluation of practical systems. As realistic limitations on engine and transmission are progressively introduced the emissions performance will deteriorate. To investigate this three further sets of simulations were carried out.

4.2 Effect of ratio range of transmission

Table 3 and **Figure 4** show the simulation predictions when transmission ratio limitations are included in the simulation. The maximum overdrive ratio allowed is 0.4 and the lowest ratio available is 2.5, giving a variator spread of 6.25. The unconstrained case used a ratio spread of almost 12, representing a considerably different mechanical arrangement. The final drive ratio used is 5.44. This reflects the specification of the transmission used in the experimental work. A further limitation is included at this stage. The minimum engine speed while power is developed is limited to 1250 rev/min. This is an attempt to represent the behaviour of the hydro-mechanically controlled multi-plate clutch.

Table 3 also shows the percentage increases in the various emissions over the ideal case. The main effect of limiting the ratio range of the transmission is to raise the engine speeds during the higher speed parts of the test. This can be seen in **Figure 6**. The effect is most marked when following the BSFC line as here very low engine speeds are specified by ideal the line. During the low speed ECE15 cycle the minimum power-on engine speed limit of 1250 rev/min is invoked at all times. The power required can be developed at much lower engine speeds. At higher vehicle speeds the maximum overdrive ratio of the transmission is the limitation. The effect is particularly noticeable during the EUDC section of the test. The most noticeable effect on the emissions is the improvement in NOx (also shown in **Figure 7**) and PM predictions for the BSFC line. This is due to the effect of the ratio limitations requiring higher engine speeds than those used by the BSFC line to be used for part of the cycle. This means lower torques are used for the same power output, which has a negative effect on fuel consumption.

The effect of ratio limits on the NOx line results is less marked as high engine speeds are requested by the IOL, consequently the emissions are similar to the unconstrained case.

When using the IOL for HC the engine speed is increased during much of the test by the ratio limitations. This has the effect of increasing the PM emissions, as the optimum speed is unobtainable. The revised lines for HC and BSFC when ratio limitations are considered are quite similar and as expected the PM predictions are similar. The percentage changes are in opposite directions due to the different ideal case results.

In summary, the effect of ratio range limitations is to force the engine to operate at a higher speed than desirable for much of the drive cycle. The effect of this is most evident when low fuel consumption is the primary requirement. Almost doubling of the ratio range from 6.25 to 12 allows a 5% improvement in fuel consumption. In the light of the increased mechanical complexity required to achieve this and the detrimental effect on NOx production it must be questioned whether the benefit is sufficiently attractive to warrant the effort. **Figure 4** shows the same generic trade-off curves for this case as for the ideal transmission. The main difference is the general shift of the curve away from the origin of each plot. The exception as previously stated is the generally improved emissions performance of the IOL for BSFC (simulation 4) where adhering to the IOL is not possible for large sections of the cycle.

4.3 Effect of torque losses

Table 3 and **Figure 4** show the predictions when losses are included in the simulation in addition to the ratio limitations described above. Losses were included as a simple 85% torque efficiency for the transmission. This results in higher engine torques and speeds being required throughout the cycle in order to deliver the required power at the drive shafts. In a fixed ratio transmission the engine speed would remain unchanged, only the torque would increase. The action of the powertrain controller in this case is to deliver the demanded engine power at the most advantageous point according to the ideal line under consideration. This requires both the engine speed and load to change from the ideal case to maintain operation on the IOL. This model of efficiency is a simplification as the efficiency of the unit varies widely over its operating range, especially at low power throughputs. The 85% figure is suggested by experimental work to be representative over most of the working range although more sophisticated descriptions of the loss characteristics could be included.

Table 3 also shows the percentage increase in emissions when compared with the ratio-limited case with no losses. The effect of this change is detrimental to all the emissions for every ideal line. This highlights the well-known importance of transmission efficiency improvements in improving the overall efficiency of the vehicle. The effect on NOx is particularly noticeable for all the IOLs. The major increase is during the EUDC section where cumulative NOx emissions rise rapidly. The effect on fuel consumption is also significant. A change of 10% is difficult to achieve by improving the prime mover efficiency. Again, the EUDC has the greatest contribution to the total. **Figure 4** shows all three trade off curves to be further from the origin than the previous case but with the same characteristic responses.

4.4 Transmission creep

The transmission under investigation in the initial project was configured to have a creep facility at idle. This feature makes the transmission feel similar to an automatic transmission equipped with a torque converter. Here it is achieved by retaining a modest hydraulic pressure in the forward clutch at idle conditions while the transmission control lever is in the drive position. From an efficiency standpoint, the creep feature is a disadvantage. During the European test cycle there are long periods where the vehicle is held stationary using the foot brake. The legislation requires that the transmission selector be left in drive for the entire test. This means that the creep torque transmitted is a parasitic loss, causing more fuel to be burnt and producing more emissions.

Creep is modelled by setting the minimum engine torque to be 15Nm at all times. This is a fair representation of creep while the vehicle is stationary. While the vehicle is on the overrun there will be no creep loss as the vehicle will be driving the engine. This is not reflected in the current simulation although this is thought not to affect the result greatly due to the limited time spent in this condition.

The simulated emissions when creep is added to the other losses described are shown in **Table 3** and **Figure 4**. This level of simulation is the same as that used for the validation study described in section 3.2. The percentage increases in emissions when compared with the ratio-limited case with losses are also shown in **Table 3**. As expected the effect on fuel consumption is quite marked. Even more dramatic is the effect on PM production. The low speed/low load condition is particularly bad in this regard. One unexpected effect of the creep load is to improve the HC predictions by around 4%. The HC emissions of the engine considered are worst at low or zero load. Increasing the load slightly moves the engine from this poor operating point and into a cleaner region. Again, **Figure 4** shows all three trade off curves to be further from the origin than the previous case but with the same characteristic response.

5. CONCLUSION

The simple drive cycle simulation technique presented allows a useful preliminary investigation of the effects of a novel powertrain control strategies and the effect of changes to mechanical configuration when tested over an arbitary drive cycle.

Results are presented and compared with experimental data which suggest that it is possible to predict the variation in emissions performance of a vehicle over the European drive cycle resulting from varying the engine operating line in a vehicle equipped with a CVT.

Further simulations are presented demonstrating several steps that could improve the emissions performance of such integrated systems.

- A wider transmission ratio range allows more freedom to follow an optimum operating line giving useful emissions savings. For example a ratio range of 12 gives a 5% fuel economy benefit over a transmission with a range of 6.26 using the same final drive ratio. This will also have positive drivability implications in some circumstances but a detrimental effect on NOx production. The improvements are offset by the increased complexity and cost of the transmission required. These results were achieved using a direct injection Diesel engine which has very good part load efficiency. The possible improvement using a conventional gasoline engine would be greater.

- Efficiency improvements in the transmission are another effective path to improved performance whatever the control strategy. The simulation offers the scope to model the effect on fuel consumption and emissions production. Results presented here suggest that the losses in the transmission increase emissions variously from between 4 and 30% depending which pollutant and operating line is under investigation.

- The provision of a creep mode in the design of the transmission must be considered carefully due to its emissions disadvantages. PM emissions are predicted to rise by around 8% with creep while fuel consumption increases by around 4%.

- The control of the main drive clutch especially during engagement has significant effect on the emissions performance of the vehicle, especially during the low power manoeuvring typical of the ECE15 cycle. The low speed-high torque capability of the engine could not be fully utilised as this region co-incided with the clutch slip condition of the transmission used. This shortcoming is relatively simple to alleviate with more sophisticated (electronic) clutch control.

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FIGURES

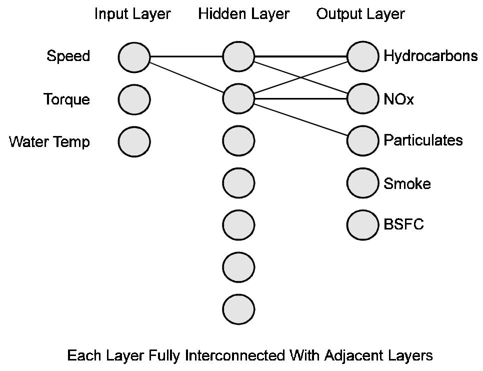

Figure 1 - Neural network used for emissions prediction

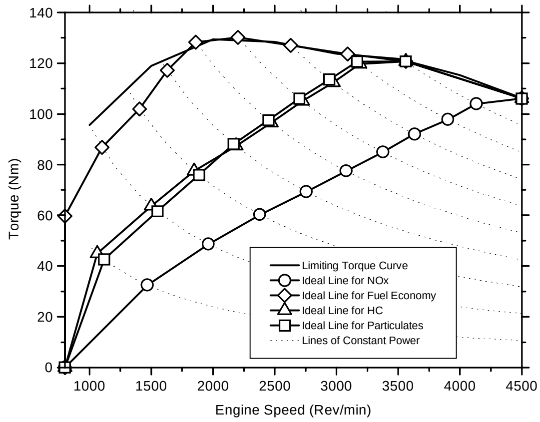

Figure 2 – NOx, HC, PM & BSFC lines

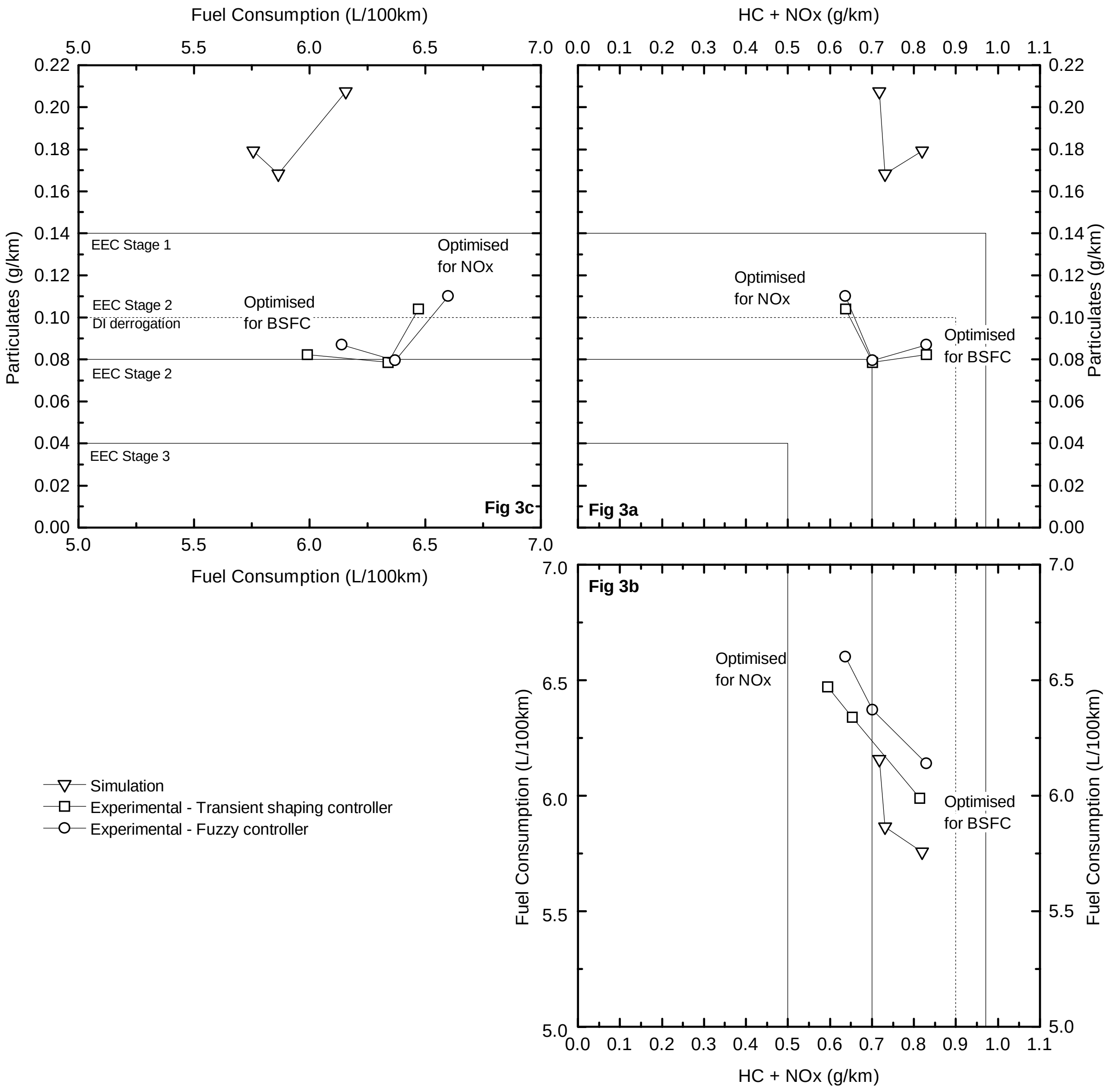

Figure 3a,b & c - Experimental results compared with simulation predictions

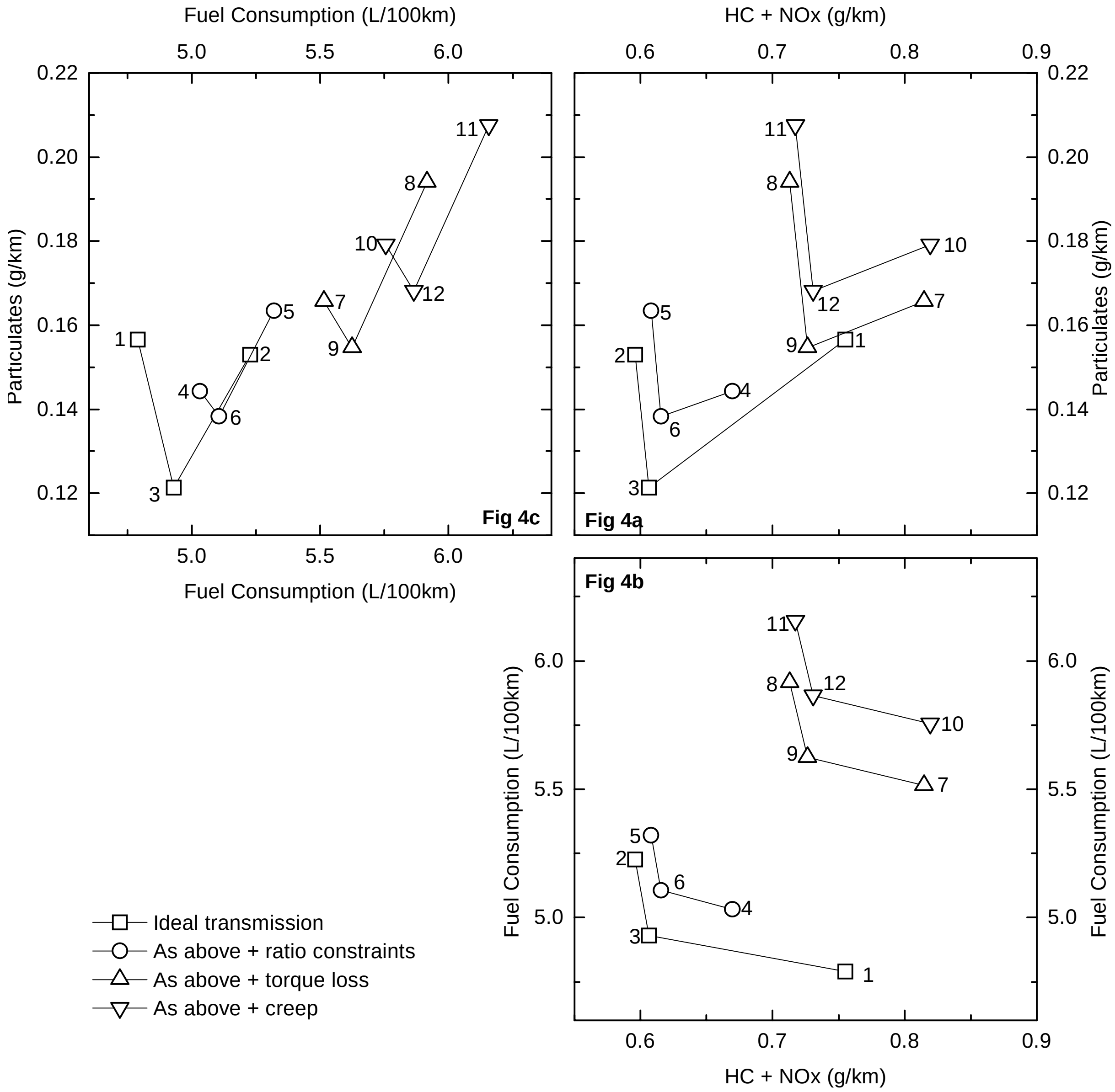

Figure 4a,b & c - Constant timestep simulation results including ratio constraints, efficiency and creep

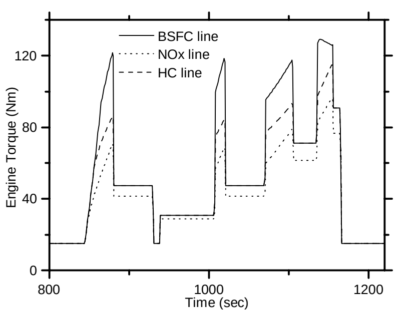

Figure 5 - Engine torque - comparison between IOLs - ideal case

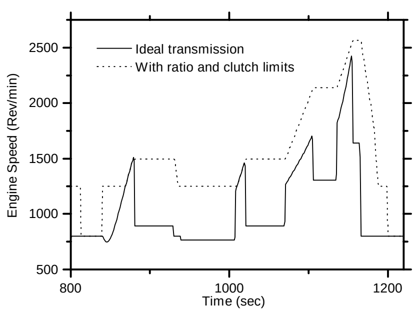

Figure 6 - Ratio limited engine speeds - BSFC line

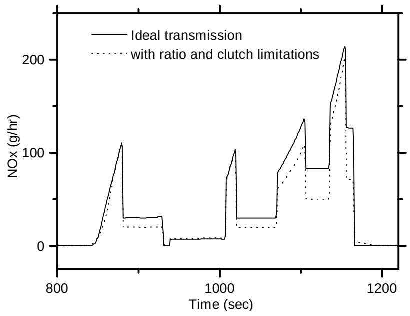

Figure 7 - NOx predictions