Chapter 4 - Operating point optimiser model

4.1 Use of models in control

Many control strategies incorporate models of part or all of the plant. One of the primary aims is to estimate the value of variables which are not practical to measure but which may advantageously be used as inputs to the control algorithm.

Engine emissions and engine brake torque are good examples of such variables. While it is possible to measure such variables in a laboratory the process is expensive, time consuming and requires costly specialised equipment. Some applications of the internal combustion engine such as marine propulsion and power generation are able to support the development of continuous monitoring systems for torque and/or emissions due to the large capital cost of the plant and the expert nature of the maintenance and operation.

The requirements of mass produced motor vehicles preclude the fitment of expensive components or systems which require frequent maintenance or adjustment. This must be reconciled with ever increasing demands for conformity of emissions, refinement, and enhanced performance. Such qualities are today achieved with the aid of sophisticated microprocessor controllers (although good controllers are only able to enhance basically capable plant, they should not be viewed as a panacea for basic performance deficiencies). Modern engine control systems require data from devices such as hot film air flowmeters, exhaust gas oxygen sensors, injector needle velocity transducers and knock sensors. These instruments are mass produced developments of laboratory equipment, generally exhibiting good linearity and robustness. Due to the large numbers of vehicles produced it has been possible to develop and produce such items at a realistic cost for routine fitment.

4.1.1 Emissions monitoring

Currently this is not the case with the measurement of the concentrations of exhaust gas pollutants. The only such instrument is the exhaust gas oxygen (EGO) sensor. These are used in fuel injected spark ignition engine controllers to ensure unity equivalence ratio is maintained. At this condition there will be no excess oxygen in the exhaust gas. This is important for the successful operation of a three way catalyst. Diesel engines, by their nature, operate at very lean conditions when compared to an SI engine. This leads to a large oxygen content in the exhaust gas. The lower temperature of the exhaust gas from a Diesel engine also provides difficulties as the output voltage from EGOs is strongly dependent on sensor temperature. Heated EGOs (or HEGOs) have been developed to allow their use in lean burn engines. If the gas stream is hot as in a stoichiometric engine the sensor does not require heating and is sometimes termed a UEGO sensor. Amstutz (1992) used a sensor with regulated heating to further extend the useful range of the device down to a Lambda of 3.5. This enabled a system to be developed which controlled EGR rate on a turbocharged DI Diesel engine by a closed loop with feedback from the HEGO sensor. The response time of the sensor was significant and had to be allowed for in the control system.

4.1.2 The modelling alternative

It is possible that similar devices may be developed in the future to allow on-line monitoring of other gases in the Diesel engine exhaust. These would allow control strategies to react to the data and optimise engine operation in a sophisticated manner. Until such devices become available this type of control may only be attempted by using an estimator or engine model to predict the emissions at any instant. Such an engine model would need to be relatively compact, have low computational overheads and a rapid update time to be of use in an on-board controller. If these criteria were met the model could conceivably provide a rapid estimate of the variable in question, not affected by instrument delay and preferably free of noise.

Advanced control structures

A second, and important, benefit of the modelling approach is that it facilitates more advanced control structures. Instrumentation is useful in determining the current or historical condition of the plant. As such it may be used as part of a feedback, or reactive, controller. Enhanced performance, particularly during transients may be achieved by adopting a feed*forward*, or *pro*active, structure. Such a strategy requires knowledge of the plant characteristics to enable sensible control inputs to be generated in the open loop or forward path. A traditional structure such as three term control may then be used to close the loop or *trim* the control input to achieve the set point. Such a strategy can offer good dynamic performance with minimal steady state errors. Crucially the open loop term must be estimated with sufficient accuracy to reduce the task of the feedback structure. A good model of the plant enables this to be achieved.

Engine torque control

A similar argument may be put forward for the case of engine torque control. Reliable torque transducers of modest cost are being developed and may soon be fitted to vehicle driveshafts in some applications. It is more difficult to measure the brake torque of the engine at the transmission input shaft due to the compact arrangement of modern powertrains. This limitation may be overcome, but in the short to medium term a reliable prediction of engine torque is extremely useful to the powertrain control engineer, both to estimate current torque output and to calculate fuelling levels to achieve future desired torques.

Willans lines
For a Diesel engine it is possible to predict brake torque with reasonable accuracy using the Willans line method. Willans lines for the 1.8DI TCi are shown in **figure 4.1**. It can be seen that for a given engine speed engine, brake torque increases linearly with fuel flow rate. The gradient and intercept of these straight lines varies with engine speed as a function of friction in the engine. Speed dependent friction losses for the engine may be estimated by extrapolating the Willans lines to the zero fuel axis and reading off the friction load. These are shown in **figure 4.2**.

Non linearities
Losses which vary non-linearly with load will force the torque to deviate from this straight line relationship with fuelling. A good example of this is the throttle loss in a spark ignition engine. In the Diesel engine under consideration here there is no throttle plate. This is one of the major contributing factors to the superior fuel economy of the Diesel engine. It has the coincidental benefit to the control engineer of resulting in a simpler relationship between fuel flow and torque production. This is not, however, the whole story. There will be non-linear losses within the engine which, though small, will combine to give a significant error in this simplistic approach. This effect is more apparent at conditions away from the nominal. The major perturbing factors are engine temperature, air charge temperature, boost pressure, EGR flow, exhaust back pressure and to a lesser extent, fuel temperature. Engine temperature will affect friction losses and thermodynamic losses due to heat transfer. The other factors are primarily noticeable due to their effect on air fuel ratio in the cylinder and the degree of pumping work performed by the engine. Combustion efficiency varies with air fuel ratio and in the extreme there may be insufficient oxygen present to fully burn the fuel. This should not happen in a properly regulated modern Diesel engine, even during transients.

Errors
The combined effect of the various errors may not be significant for many purposes, indeed a Willans line type relationship was used successfully in the Integrated Powertrain Control project to schedule the fuel pump demand to achieve the desired engine torque. One effect which may cause difficulty is a systematic variation in the error across the engine speed/load map. It is quite possible for the torque prediction to be too high in some areas of the map and too low in others. In this case undesirable effects may be observed during transient manoeuvres as the operating point moves from one region to the other. The engine power and resulting vehicle acceleration may be perceptibly non-linear with respect to the accelerator pedal position. To counter this effect the algorithm used in later controllers was modified to counter the effect of systematic non-linearities in the engine system such as the EGR map and exhaust back pressure. The resulting expression can become quite complex and eventually it may be better to adopt a more detailed model.
Engine torque prediction
Various researchers have proposed models of internal combustion engines aimed at predicting torque produced. Rizzoni (1989) describes a method for predicting engine torque from the measured crankshaft speed fluctuations. The instrumentation required is simply the standard engine speed magnetic sensor detecting the passing of the flywheel ring gear teeth.

Engine torque may also be related to peak cylinder pressure, although here an estimate must be made of engine friction to separate the brake torque from the indicated torque. This is not a trivial task, especially under varying operating conditions.

Neural network torque estimation
The factors discussed earlier which influence engine torque were measured as part of the neural network study. The resulting network can predict torque to a fair degree of accuracy as seen in some of the results in **Chapter 6**.

4.2 Operating point prediction

4.2.1 Definition

One of the fundamental concepts in the Integrated Powertrain Control project was that of an Ideal Operating Point (IOP). This is defined as the engine speed and load which delivers the desired power whilst producing the least undesirable emissions. A locus of IOPs may be drawn across the engine speed/load map and referred to as an Ideal Operating Line (IOL). Clearly the undesirable emissions are more wide ranging than simply the traditional concern relating to CO_2 (directly analogous to fuel consumption). Environmental pressure, in the form of type approval regulations, demands ever lower levels of the harmful products of combustion as discussed in Chapter 1. In the Diesel engine these are CO, uHC, NOx and PMs. If the Diesel engine is functioning correctly there should be insignificant production of CO compared to a SI petrol fuelled engine. As such CO is not considered in the optimisation process.

4.2.2 Difficulties in defining a global optimum

The units used for this exercise are non dimensionalised by relating their mass flow to brake specific work (grams per kilowatt hour). Values presented in these units are usually prefixed *Brake Specific (BS)*. This enables easy comparison between different power levels. Due to the nature of the various pollutants and their different origins (as discussed earlier) it is difficult to build an engine where the various emissions are at a minimum simultaneously. If lines of constant power are considered on an engine torque/speed map the desired power can be achieved at either low load, high speed or alternatively low speed and high load. All intermediate points on the constant power line must also be considered. The optimum point for each power curve is the speed/load combination which delivers the least brake specific mass of the pollutant in question.

BSNOx will generally be lower at high engine speeds for a given power output due to the lower temperatures and pressures developed in the cylinder at such a condition. BSFC will generally be minimised at higher engine loads and lower speeds to achieve the desired power. This is due largely to the lower friction losses at the lower engine speed. BSuHC and BSPM are generally lowest at some engine speed between the high level for best BSNOx and the low speed for best BSFC. At higher engine speeds the mixing imperfections lead to less optimised combustion and at lower speeds the lower air fuel ratio required leads to more uHC and soot formation.

By careful manipulation of the engine calibration it may be possible to move these optimum speeds closer together. One way to achieve this is to manipulate the EGR schedule. EGR is used to reduce NOx emissions by slowing the reaction rates but has the unwanted effect of increasing the particulate levels. This trade off may be used to set an EGR level such that the minimum BSPM production for a given power output lies at the same speed/load point as the minimum BSNOx point. Experience with the 1.8DI TCi (**figure 4.3**) indicates that this point will also be near the optimum for BSuHC. Brake Specific Fuel consumption is unlikely to be a minimum at this condition. This synchronisation of emissions is not generally a consideration for engine calibration engineers as they may well be more concerned with lowering the absolute levels rather than subtle positioning of the minima. In any case a calibration which achieved this objective at one set of operating conditions may well be entirely unsuitable at a different set of conditions.

4.2.3 Variations in operating conditions

This variation of emissions performance with conditions should be accounted for in the IOL generation. Operating conditions may be characterised in a number of ways. The most obvious is perhaps engine temperature. This is conveniently represented by engine coolant temperature since this information is available from the fuel pump control system. Better results may be achieved by measuring the cylinder head temperature near to the flame face. This was not possible on the engines used due to their experimental nature. Oil temperature is also important although the engine used employed a heat exchanger to keep the oil temperature close to the water temperature in most circumstances. The oil temperature will have most effect directly following a cold start when viscosity is at a maximum. IC engines generally produce more uHC when cold due to longer ignition delay and quenching effects of cold surfaces. By the same token they would be expected to produce less NOx due to lower peak pressures and temperatures, although Belardini (1991) observes that the increased friction work at low engine temperatures leads to increased NOx for an equivalent brake power. Friction work will be related more to oil temperature than water temperature, suggesting its possible inclusion as an independent variable in any emissions prediction routine.

4.2.4 Other transients

Ambient air temperature will have an influence on the emissions due to its effect on air fuel ratio although its relationship with inlet air temperature is less direct in a turbocharged and intercooled engine with EGR.

Boost pressure, injection timing, EGR fraction, exhaust manifold pressure and other similar variables vary too rapidly to be considered in an IOL generator. Such variables may be assumed to have a sufficiently fast response to stabilise within a few fractions of a second (several seconds at most) at any given operating point. The plant will require a finite time to move to a new operating point, engine speed slew being the limiting variable. As such the boost pressure and similar variables will track the operating point with sufficient speed to prevent them from appearing as independent variables during the time frame of a powertrain transient. This makes their inclusion in the IOL generator superfluous. Clearly there are some manoeuvres where this statement is not true. The obvious case being the rapid increase in torque demand. This will occur after a large step increase in accelerator pedal position. The only acceptable control action to deliver the required vehicle response is to move the engine onto the Limiting Torque Curve (LTC) by increasing the fuelling rapidly, followed by increasing the engine speed to raise the power output. Here the IOL is of secondary importance and drivability is the dominant factor.

In general, changes in power demand which occur rapidly result in a deviation from the ideal line in order to deliver acceptable drivability. It is only during such manoeuvres that secondary engine parameters such as boost pressure will deviate grossly from their steady state values for the engine operating point. The IOP will be achieved by the controller only during more settled periods of operation. As such the IOP should represent the ideal point *at steady conditions*. In addition the recommended operating point for a constant power demand should not change too rapidly as this will lead to oscillatory and non-linear performance which is undesirable. In general the fastest transient which should be considered in generating an IOL is the engine coolant temperature. This has a time constant of several minutes when moving from one power condition to another and during a warm up following a cold start. The effect of the catalyst may be important to the formulation of an IOL. This is discussed below.

4.2.5 Weighted sum

Given the fact that in general the various emissions are lowest at different operating points there cannot be a global optimum operating point. A compromise solution must be achieved taking into account the relative importance of each pollutant. These rankings may be used to arrive at a weighted sum of the emissions which allows an IOL to be set. These weightings are difficult to decide upon. Here some evaluation using a drive cycle simulation tool can be extremely instructive. If the primary goal is to achieve good performance over one of the legislative cycles then the limits set down for the various pollutants will be influential to the outcome. For example, the 1996 EEC limits for passenger vehicles equipped with a DI Diesel engine stipulate a NOx + HC figure of 0.90 g/km over the ECE15 + EUDC test. The limit for PM is just 0.10 g/km. On this basis the PM production would seem to be the limiting factor in gaining type approval. This is not quite the case, a modern Diesel engine is designed to produce low levels of PM compared to the other pollutants. Indeed, the legislation is drawn up with the input of experts to reflect the expected state of the art at the time of enforcement. As such the various species will be regulated at different levels relating to the proportions expected from a typical engine. There must also be consideration of the relative toxicity of the pollutants, although this is the subject of much debate currently, and in any case, if unrealistic targets are set they will lose their intended impact. The legislation should direct the activity of researchers, not prevent the sale of motor vehicles.

4.2.6 Catalyst effects

Another important consideration is that for the purposes of type approval the emissions are measured at the tailpipe. The oxidation catalyst will become operational shortly after the start of the test. The catalyst will henceforth remove some of the particulate matter and most of the uHC. Under these conditions NOx may be the dominant factor in deciding the IOL (and also base engine calibration) as it is not currently possible to reduce NOx emissions significantly in an oxygen rich exhaust stream such as that produced by a Diesel engine.

Willcock (1993) reveals that emissions, especially of uHC are influenced by exhaust manifold conditions due to the reactions which take place there. One may speculate that NO and NO_2 are also affected by the exhaust manifold conditions, although the excess oxygen present should inhibit the reduction process. These effects are not considered explicitly here as all of the emissions measured during the work were sampled after the exhaust manifold. Any reactions taking place in the manifold will be lumped with those occurring in the cylinder. Transient effects may lead to certain deficiencies in this approach.

Catalyst light-off

The effects of the catalyst should be considered when formulating an IOL. Consideration of uHC production is less important if the catalyst can be relied upon to oxidise the major proportion. This will not occur if the catalyst is below its *light-off* temperature (200-300^oC). Above this temperature the exothermic reaction within the monolith will further heat the assembly, sustaining the reaction. The relatively cool exhaust from a diesel engine leads to delays in catalyst heating when compared to an engine running at or near stoichiometry. Various methods have been developed to shorten the delay time before light-off as it is this period which contributes most to the cumulative emissions figures, especially for an SI petrol engine equipped with a well matched three way catalyst. An intuitive step is to move the catalyst closer to the exhaust manifold. The conventional position for the catalyst is beneath the vehicle floor. The gas stream will cool down considerably in this distance. Moving the assembly to just after the turbine output (or even integrating it within the turbine housing/exhaust manifold assembly) leads to significantly quicker light-off. Problems can occur at high power outputs. If the catalyst temperature rises beyond around 450^oC any sulphate particles in the gas stream will oxidise causing an undesirable increase in particulates. There are also durability problems associated with excessive temperatures and rapid temperature gradients.

A second technique is to preheat the catalyst either electrically or chemically, raising it to the light-off temperature more quickly. Kaiser (1995) describes a system using a small electric heating element used to initiate oxidation of the excess fuel in the exhaust of an SI engine in the early stages of operation. The subsequent exothermic reaction quickly brings the rest of the monolith up to operating temperature. This arrangement is not easily applicable to the Diesel engine due to the lean operation, even on start up. The concentrations of unburnt fuel in the exhaust should be too low for combustion to occur in the manner required. Here other techniques such as secondary fuel injection or longer electric heating periods must be employed, reducing the attraction of the strategy.

The system used on the vehicle considered for this work is mounted in the underfloor position. **Figure 4.4a** shows the monolith temperature during a typical ECE15 + EUDC test. **Figure 4.4b** shows the corresponding conversion efficiency for uHC. It can be seen that the catalyst becomes partially effective after approximately 400 seconds, but the temperature repeatedly drops below 250^oC throughout the ECE15 cycle. As such the catalyst cannot be relied upon to remove the HC for significant portions of the test. This effect may be taken into account when formulating the IOL. The weighting attributed to HC may be reduced during periods where the catalyst is operational. This will have the effect of allowing optimisation for NOx at higher power conditions such as those encountered during the EUDC.

4.2.7 Implications for IOL generator design

The above discussion of catalyst performance and basic engine attributes leads the inclusion of several features in an IOL generator. These are:-

- User defined weights to prioritise the different pollutants
- Limited engine model including slow transients only
- Variable weights to account for effects such as varying catalyst performance

4.3 IOL generator structure

The structure of the IOP generator developed is shown schematically in **figure 4.5**. The function may be split into several sub functions as discussed below.

4.3.1 Operating line interpolation

Upon receiving a power demand from the supervisory controller the optimiser interpolates along a pre-determined ideal optimum line (IOL) to return an ideal engine speed and torque (IOP) for the demanded power. These data are returned to the supervisory controller which may either move the plant to this condition or may decide to overrule the optimiser to achieve good transient drivability.

4.3.2 Operating line optimiser

This IOL is determined by a second module, the *operating line optimiser*. This function incorporates a model of the engine which is used to revise the IOL according to changing operating conditions such as coolant temperature.

Figure 4.6 shows graphically the steps taken to determine the minimum emissions point on a constant power curve. Initially a straight ‘optimum’ line is drawn between the fixed minimum and maximum power points. The line consists of a series of points spaced at 5kW intervals from minimum to maximum engine power. Lines of constant power at 5kW spacing have been superimposed on this and other similar plots for clarity. The routine finds the engine speed which gives the lowest predicted weighted sum of emissions for each 5kW power step.

The engine model is used to predict the emissions at point 1 where the initial line crosses the 5kW constant power curve. The various emissions are scaled according to pre-determined weightings and summed. This step is repeated at points 2 and 3 which lie on the same power curve but at lower and higher engine speeds respectively. The point with the lowest weighted sum is chosen as the new ‘ideal’ point for 5kW. This process is repeated for each power step up to 95% full power. The whole procedure is repeated at the next controller iteration, which will move the line again in the direction of reduced emissions.

If the ideal point moves in the same direction twice consecutively, the speed step between points 1, 2 and 3 is increased. If the middle point is chosen twice in succession, or if the optimum point is approaching the outer envelope of engine performance, the speed step is reduced. The minimum speed step is user defined. This simple procedure quickly settles on an ideal line for the chosen weightings. Thereafter the line will change slowly with engine water or catalyst temperature fluctuations as they affect the weightings for the emissions and, in the case of coolant temperature, the predictions of the engine model. This slow variation will continue throughout vehicle operation.

4.3.3 Modifications to weights

The various emissions are combined in a weighted sum according to weights which may be updated continuously to reflect the prevailing conditions.

The network outputs are normalised over the range 0-1. The scaling factors used to achieve this are shown in **table 4.1**. This process allows easy calculation of weightings. If figures were in engineering units BSFC would dominate the process as the numerical values are large in comparison with those for pollutants.

Inputs

Variable	Units	Minimum	Maximum
speed	rev/min	500	5000
torque	Nm	0	150
coolant temp	^{o}C	0	110

Outputs

Variable	Units	Minimum	Maximum
HC	g/kWhr	0	12
NOx	g/kWhr	0	16
PM	g/kWhr	0	24
Smoke	Bosch	0	6
BSFC	g/kWhr	150	1000

table 4.1 - Optimiser network scale factors

Weights are set for each of the pollutants considered (HC's, NOx, PM, CO_2 and smoke). The most basic setting would be each weight having a equal value, say unity. The user determined weights are applied to the normalised values to set their relative importance. These weights may be modified according to coolant temperature and catalyst temperature. In this way the operating point may be varied to give less NOx after the catalyst has become operational. There will be a degradation in HC and PM performance due to this action, but the catalyst should minimise the impact of these increases.

The weights may be varied with power demand, for example, it may be desirable to optimise for HC at low power outputs as seen during the ECE15 cycle, but as higher power levels are demanded, as in the EUDC, the emphasis could be switched to minimise NOx production. To evaluate this approach the cycle test results may be analysed to relate the emissions production to power output. If a correlation between power output and emissions levels may be determined then the above approach may be beneficial.

4.3.4 The engine model

The model used to predict the emissions at each step is a neural network of the form shown in **figure 4.7**. It is similar to the transient engine model, but since the optimiser is only useful for steady conditions (as discussed above) the rapidly changing inputs have been ignored. Water temperature has been retained as an input as it varies sufficiently slowly to be taken into account and its effect on emissions is quite marked.

The network has been trained using a subset of the data gathered for the transient engine model. In addition to the *on design point* data the maps generated at low and high water temperatures are included. It may seem attractive to include all of the data generated for the transient model but this would detract from the accuracy of the resulting representation as the variations in emissions may be due to any of the independent variables being perturbed. If insufficient inputs are included to reflect this the variations will just appear as noise in the training data set, masking the true relationships under investigation.

Training & validation

The validation task for this model is relatively straightforward compared to the fully transient engine model. The model only has three inputs, as such the main two inputs (speed and torque) may be varied across their ranges while holding the third (water temperature) fixed. The resulting data may be presented conveniently as a two dimensional map or three dimensional surface. A more objective method of evaluation is to calculate the RMS and standard deviation of the error in the network prediction compared to the training data. These figures are shown in **table 4.2**.

BSuHC	BSNOx	BSPM	Bosch	BSFC
7.80	6.74	4.64	9.87	2.43

table 4.2 - Optimiser Network Training Errors

BSNOx

An example of the optimiser network output is shown in **figure 4.8**. Experimental data points for BSNOx are superimposed on a surface mesh generated by the network. It can be seen that the network has generalised quite successfully to represent the real data. **Figure 4.9** shows how the shape and position of this surface changes with varying coolant temperature.

Contrary to the observations of Belardini (1991) the BSNOx production falls as coolant temperature falls. Belardini found the higher friction at low temperatures caused more NOx to be produced for the same brake power. This difference is likely to be due to the elevated oil temperature during the training data tests. Although held to the same temperature as the water this is not quite representative of a real cold start. In vehicle installations oil temperature will rise due to the heating effect of the pressure relief valve, shearing in the various bearings and heat transfer from the coolant in the heat exchanger. The oil will however be cooler than the water for a few minutes. Once this point is reached the drop in viscosity will reduce the effect of oil drag to less significant levels more consistent with the model predictions. Future studies would benefit from the inclusion of oil temperature as an independent variable.

BSFC

Figure 4.10 shows the pattern learnt by the network for BSFC production at the nominal 85^{o}C water temperature. Again the pattern appears to be representative of the superimposed experimental data. **Figure 4.11** shows the same network's response at a water temperature of 25^{o}C. Here there is less experimental data available but the pattern seems to be representative.

Figure 4.12 shows the variation of BSFC with water temperature. One of the key advantages of the Diesel engine is apparent in this illustration. It can be seen that BSFC is relatively insensitive to water temperature except at very low power outputs. This is due to the nature of the fuel preparation mechanism and the minimal heat losses in a DI engine. The equivalent data for an SI petrol engine would show a more marked drop in fuel economy at low temperatures. It can also be seen that the major part of the operating map is relatively flat, showing the more uniform characteristics of the engine when compared to an SI engine.

The relative insensitivity to coolant temperature and flat characteristic of the DI Diesel are important factors in achieving good all round performance as discussed by Charbonnier (1993). To the control engineer the same characteristics are a little frustrating as the difference between an engine running at an optimum condition and one running a little further from optimum is not as great as may be hoped. The benefits are clear, although more radical improvements may be expected in an SI powertrain.

BSHC

Figure 4.13 shows the nominal pattern for HC production and the superimposed network data. The shape of the surface is similar to that for BSFC. There is a strong link here as uHC and BSFC performance are both strongly influenced by the efficiency of combustion. The BSHC map is less flat than the BSFC map, worse figures are noted at higher engine speeds and low loads as the time available for complete mixing and combustion becomes limited. **Figure 4.14** shows the worsening HC performance as water temperature falls.

Bosch smoke

Of all the emissions considered here smoke is the least well represented by the network. There are two reasons for this. The experimentally determined smoke map (**figure 4.15**) is the most irregular of all the maps considered. In addition there is a large noise component to the data. The network training process will tend to reject noise and settle on a representative mean value.

A more complex network structure may be employed to represent the smoke levels more accurately. This would allow more discontinuous relationships to be described. This has not been done for two main reasons. The first is that network interrogation times would be increased by adding to the structure. This is important in a model intended to be used in an on board controller the shape of the surface learnt is representative and this is more important then its absolute error.

The second reason is that smoke is not directly measured during the legislative test. Smoke levels will only influence the test result by adding to the PM gathered on the filter paper. Since PM is well represented by the network smoke is not used in the weighting procedure and is hence superfluous for legislative purposes. Its inclusion in the model is to allow the effect of different operating lines to be studied in simulation. Poor smoke levels are displeasing to the driver and other road users and this consideration may be investigated using the network. If a more detailed investigation were required it would be sensible to train a new, more complex, network to represent smoke levels.

BSPM

Figure 4.16 shows the nominal pattern for PM production and the superimposed network data. The pattern is similar to that observed for BSHC. This is not surprising since most of the PM will be composed of partially burnt fuel. The correlation developed by Greeves and Wang (1981) shows that HC concentration in the exhaust is around four times as important as the soot concentration when predicting PM levels. Similarly, **figure 4.17** shows the increasing PM levels as coolant temperature falls.

4.4 Results

The optimiser can be used to generate a vast number of subtly different IOLs by varying the weightings. In the first instance it is instructive to investigate the optimum lines for each of the pollutants in isolation. This is easily achieved by setting all the weightings to zero with the exception of the pollutant under investigation.

4.4.1 BSFC line

Figure 4.18 shows the resulting ideal operating line for fuel consumption at a water temperature of 100^{o}C. All weightings are set to zero except for BSFC (which is analogous to CO_2 production). Superimposed is a map of the network prediction for BSFC at the same water temperature. The IOL generated lies in the region intuitively seen to be the optimum. It can be seen that for the calibration of engine used here the optimum line follows the LTC for much of the power range. The calibration used was de-rated from a peak torque of 180Nm to the 130Nm shown here. This was to safeguard the transmission used. The shape of the BSFC map would be identical in the region shown. Hence, the BSFC IOL for the 180Nm engine would be higher than the de-rated LTC.

4.4.2 BSHC line

Figure 4.19 shows the ideal line generated if uHC alone is considered. Here the ideal line is some way beneath the LTC. When operating at a steady low power condition the controller can react quickly to any driver demand for more power by raising the torque to the IOL. This attribute diminishes at higher power demands until the IOL and LTC converge at a demand of around 45kW. Any demand for increased power at this condition would necessitate an increase in engine speed, which takes longer to accomplish than an increase in torque.

4.4.3 BSNOx line

A similar line for BSNOx is shown in **figure 4.20**. Here the ideal line is much lower than the LTC. At lower power demands an extra 15-20kW may be developed very quickly by raising the fuelling to the LTC. The IOL does not meet the LTC until the maximum power point of 50kW. Since this can only be achieved at one speed/load combination with this engine all the IOLs will terminate at this point. Similarly the IOP for zero power is defined as the idle condition of 0Nm and 800rev/min in all cases.

4.4.4 BSPM line

As may be expected the IOL for BSPM (**shown in figure 4.21**) is nearly identical to that for BSHC. The inclusion of PMs and HC in an optimising algorithm would add extra importance to the area of the operating envelope which is good for both emissions. This needs to be considered since it may be detrimental to the NOx performance of the calibration.

4.4.5 Bosch smoke line

The smoke line (**figure 4.22**) must be regarded with caution due to the inaccurate representation of the smoke learnt by this simple network. By comparison with the experimental Bosch smoke map (**figure 3.28**) it can be seen that the IOL seems to be a realistic approximation. This is due to the network surface being the correct shape, with its minimum region correctly positioned. The absolute values are not well represented but this does not affect the optimising procedure.

4.4.6 Weighted lines

In normal operation the weightings for each of these products will ensure a line which is a compromise between these five extremes. Some possible examples are presented below. **Figure 4.23** shows a line generated with unity weightings for HC, NOx, BSFC and PM. The line moves as temperature changes. **Figure 4.24** shows a line optimised for NOx and HC.

The weighting for NOx varies from 0 at 0kW to 1 at 50kW . The weighting for HC varies from 1 at 0kW to 0 at 50kW.

4.5 Use in the vehicle

When used as an on board operating point predictor the network takes some inputs from transducers on the engine. Others such as power requirement are taken from the powertrain control routine. The optimiser is implemented as a C function within the control code. The IOL is updated continuously during operation to allow for varying conditions.

Tests were undertaken at Ford’s Research and Development Centre at Dunton, Essex. Their chassis dynamometer facility was used to evaluate various control algorithms and several different IOLs over the standard ECE15 + EUDC test. **Table 4.3** summarises the tests undertaken and the results.

Two baseline tests were conducted. The first used a development Ford Escort vehicle K-AX5891 fitted with a comparable 1.8DI TCi and a Ford MTX manual 5 speed transmission. As far as possible the calibration of this vehicle was the same as the project vehicle. The major exception was the LTC. K-AX5891 used the full 180Nm calibration. This should have limited effect on the results as the ECE15 + EUDC test does not use high engine torques. **Figure 4.25** shows the points visited during such a test on the torque/speed plot.

The second baseline vehicle was the project Ford Orion. In baseline configuration it was tested with the same powertrain and engine calibration as in the following tests. Here however the VDT continuously variable transmission was controlled by the maker’s strategy. Engine torque was directly related to the pedal position, as in a manual vehicle. It must be recognised that the calibration supplied by VDT was only a first iteration based upon the supplied torque characteristic of the engine. The resulting controller tends to use very high engine speeds for any given power. This may be regarded as a *drivability* IOL as discussed below.

Test	Optimised for	Controller	HC	NOx	HC+NOx	PM	FUEL
Bath - D16104 Bath - D16090	BSFC	Trantab Fuzzy	0.15 0.14	0.66 0.69	0.81 0.83	0.08 0.09	5.99 6.14
Bath - D16119 Bath - D16098	NOx	Trantab Fuzzy	0.16 0.12	0.43 0.51	0.59 0.64	0.10 0.11	6.47 6.60
Bath - D16083	-	VDT	0.15	0.59	0.75	0.21	8.09
H157MJN K-AX5891 K-AX5891		- - .-10% EGR	0.19	0.48	0.67 0.49 0.61 (+26%)	0.12 0.13 0.08 (-43%)	6.54
96 Limits Stage 3 Limits					0.70* 0.50	0.08 0.04	

* 0.90 for DI until 30/9/99

% improvements over H157MNJ

Test No	Optimised for	Controller	HC	NOx	HC+NOx	PM	FUEL
D16104 D16090	BSFC	Trantab Fuzzy	20.63 27.21	-38.19 -44.10	-21.25 -23.62	30.25 26.36	8.41 6.12
D16119 D16098	NOx	Trantab Fuzzy	16.00 35.11	9.40 -7.08	11.45 5.08	11.86 6.78	1.07 -0.92
D16083	-	VDT	19.84	-23.81	-11.20	-81.02	-23.70

% improvements over K-AX5891 (June 94)

Test No	Optimised for	Controller	HC	NOx	HC+NOx	PM	FUEL
D16104 D16090	BSFC	Trantab Fuzzy			-67.61 -70.89	37.65 34.17	
D16119 D16098	NOx	Trantab Fuzzy			-22.40 -31.21	21.21 16.67	
D16083	-	VDT			-53.72	-61.82	

% improvements over VDT controller

Test No	Optimised for	Controller	HC	NOx	HC+NOx	PM	FUEL
D16104 D16090	BSFC	Trantab Fuzzy	0.98 9.19	-11.61 -16.39	-9.04 -11.17	61.47 59.32	25.96 24.10
D16119 D16098	NOx	Trantab Fuzzy	-4.79 19.04	26.82 13.51	20.37 14.64	51.31 48.50	20.02 18.42

Table 4.3 - Chassis dynamometer tests undertaken

Two Bath developed control strategies were used. The development of these is the subject of a PhD thesis by Deacon (1996). The first strategy used a fuzzy logic routine to relate the driver and plant inputs the desired output values. The second used a tabular method to modify the torque and speed demands from those suggested by the optimiser.

Two IOLs were tested with each of the Bath controllers. The first was the BSFC line shown in **Figure 4.18**. The second was the BSNOx line shown in **figure 4.20**. These lines were chosen as they represent the extremes of possible IOLs. The NOx line uses very high engine speeds, the BSFC line follows the LTC for most of the operating range. For the purposes of this work the temperature input to the network was artificially held to 85^{o}C and the catalyst input was set to zero. This was in order to guarantee that the IOL was not subject to wide variation during the cycle, allowing the results to be analysed more easily.

The results are shown graphically in **Figure 4.26**. **Table 4.3** shows some more data and compares the strategies. It can be seen that the IOL is more influential on the outcome of the test than the choice of controller. The results from the controllers using the same IOL are grouped closely. This is due to the common feature of both controllers which brings the engine onto the IOL during periods of steady or near steady running. Most of the ECE15 and a significant proportion of the EUDC fall into this category. The influence of the different controllers is more noticeable during transient manoeuvres.

The controllers optimised for NOx produced more PM than allowed by the EEC stage 2 limit due to the relatively high engine speeds used. NOx was reduced, however, giving an HC + NOx figure comfortably the stage 2 requirement.

The controllers optimised for BSFC produced worse HC + NOx figures than the baseline manual vehicle but significantly better PM results were observed. This is due to the lower engine speeds used on the BSFC line. More NOx is produced at the associated higher loads this increase outweighs the improvement in HC production. PM is reduced, also due to the lower engine speeds. Overall the improvement in performance was sufficient to move the vehicle into the pass region for the EEC directive Stage 2. BSFC improvements of around 7% were achieved relative to the 5 speed manual. This is another consequence of the low engine speeds.

Better fuel consumption can be achieved with the manual vehicle by reducing the final drive ratio to reduce engine speed. Changes to final drive ratio are generally limited by the need for hill start capability or acceptable acceleration in first gear. The gaps between gears may be increased but this has a detrimental effect on drivability. More ratios may be provided, indeed, some vehicles are now equipped with six forward speeds. This allows a very deep overdrive sixth speed to be specified. More than six speeds are difficult to accommodate as a conventional H pattern shift can only have six positions (plus a reverse engaged by a secondary action such as lifting a collar).

The same argument is valid for a CVT. Lowering the engine speed will improve fuel consumption. Here also the ratio range of the unit must be increased to accommodate this. The ergonomics are easier to allow for since there is no requirement for extra selection lever positions as in the manual vehicle.

Both IOLs performed better than the VDT strategy, particularly if PM and fuel consumption are considered. This is primarily due to the higher engine speeds selected by this strategy.

Plots of the transient emissions data show that the worst part of the test for NOx (**Figure 4.27**) is the EUDC where higher power demands are required. HC production (**Figure 4.28**) falls as the catalyst becomes more effective but also has some large spikes during the EUDC. PM production (**Figure 4.29**) is worst in the EUDC where high engine speeds are used. Here a larger ratio range would be particularly advantageous as the transmission is on the overdrive endstop during high speed steady running. CO_2 production (**Figure 4.30**) is analogous to fuel usage and as such is indicative of the power usage at each point in the cycle.

4.6 Implications of drivability requirements

Figure 4.31 shows a stylised representation of an ‘ideal’ powertrain controller. During steady operation the operating point falls on the IOL. During moderate acceleration the operating point moves from a low power to a higher power. The IOL could be followed, ensuring minimum production of pollutants, but in this case the rate of power increase would be limited by the slowest transient in the system. This is the transmission ratio control. If the engine is accelerated too rapidly there will be insufficient excess torque to accelerate the vehicle. This will appear as an undesirable hesitation to the driver. In the extreme, if the engine speed slew rate is too rapid the engine will accelerate by absorbing momentum from the vehicle. This will result in a slowing of the vehicle which is to be avoided at all costs following a foot down input. To avoid these adverse effects the engine slew rate must be limited to rates determined empirically. Following the ideal line during a transient would give sluggish performance.

The solution is to deviate from the ideal line as shown in **Figure 4.31**. Extra torque is developed, an action which can be taken very quickly in a Diesel engine. This provides extra power, some of which may be used to accelerate the engine, the remainder accelerates the vehicle. Progressively higher acceleration demands will result in more excess torque being produced. A 'kick down' input will result in the engine torque demand rising steeply to ensure saturation at the LTC. The engine speed may then be increased at the desired rate.

The same strategy is followed in 'foot off' manoeuvres. If the engine power decayed along the IOL the rapid reduction in engine speed required would fight the vehicle brakes by transferring momentum to the vehicle and may even cause the vehicle to accelerate. The solution used was to reduce torque rapidly by cutting the fuel, leaving the engine speed to decay at a slower rate.

4.6.1 Stand-alone transmission controllers

It will be noticed that the above strategy results in a strong correlation between accelerator pedal position and engine torque demand. It is this synchronicity which allows transmission control engineers to design systems which retain the direct link between accelerator pedal and engine throttle. As may be expected these systems are able to exhibit good drivability behaviour. Even though the VDT controller performed poorly in terms of emissions it is generally considered to possess good drivability characteristics, even though the VDT calibration engineers were unable to test the strategy in the vehicle.

4.6.2 Integrated control

The controller developed in this project broke the direct link between pedal position and engine torque. A supervisory controller was used to control both the transmission and the engine. This allows much more flexibility in the control algorithm. Consider, for example, the BSFC line (**figure 4.18**). Much of this line is on the LTC of the engine. A transmission controller with a direct link between pedal position and torque would be unacceptably non-linear if this IOL were followed. A ninety five percent pedal displacement would cover only fifty percent of the available power, with the remaining fifty percent power being accessed in the last five percent of pedal movement.

The BSFC line has other drivability problems, however, even if an integrated control architecture is used. The IOL shown in **figure 4.31** is some way below the LTC. A rapid increase in power can be achieved by jumping from the IOL to the LTC. The BSFC line does not facilitate this manoeuvre as the IOL already lies on the LTC for much of the power range. This results in sluggish performance as the only way to develop more power is to change the engine speed. As previously discussed this cannot be done very rapidly.

4.6.3 Engine design for CVTs

The task of the supervisory controller is thus quite demanding if drivability is considered in addition to emissions performance. This difficulty can be addressed partly by revised engine design. An engine can be designed which has the same power output at lower speed. If the revised engine is of the same capacity this is achieved by operating at a higher BMEP. The compromises in the engine design necessary to achieve this are likely to reduce the viable maximum speed of the engine. This is not such a problem in a CVT vehicle as it would be in a manual vehicle. In a manual the limited speed range of even the best Diesel engines is a perceived problem when compared to the speed range (and consequently higher maximum power output) covered by a comparable petrol engine. In fact, most drivers rarely use the rated power of the engine, a more important consideration is shift busyness. If an engine has a relatively narrow useful power band the driver needs to shift more often to keep the engine speed in this region. This has an adverse effect on perceived drivability. A Diesel engine is advantageous in this regard as its relatively even torque delivery across the engine speed range provides good flexibility.

A more serious concern when increasing the maximum BMEP of an engine is the effect on emissions, especially NOx. New technologies and better control techniques are being employed to overcome these problems. A highly rated engine may also require a heavier structure to maintain its durability, an area which needs to be carefully evaluated. Good power to weight ratios are required to reduce vehicle weight (and hence reduce fuel consumption and/or improve acceleration).

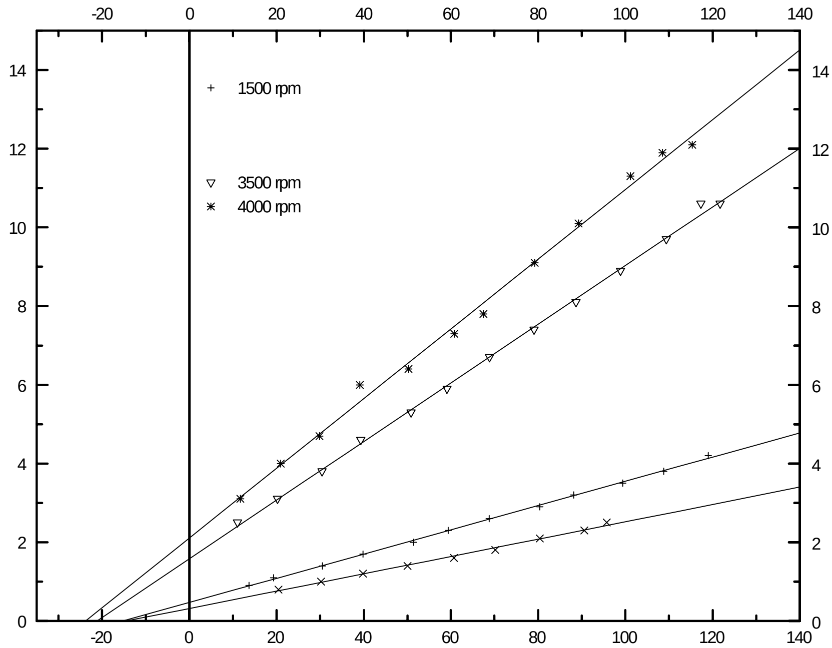

Figure 4.1 - Willans line for 1.8DI TCi

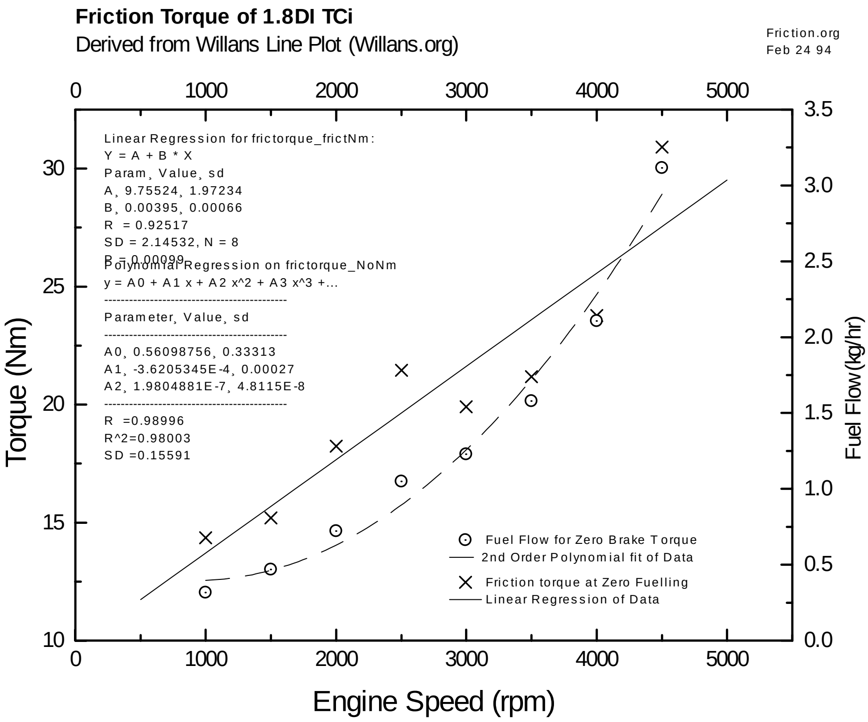

Figure 4.2 - Friction for 1.8DI TCi

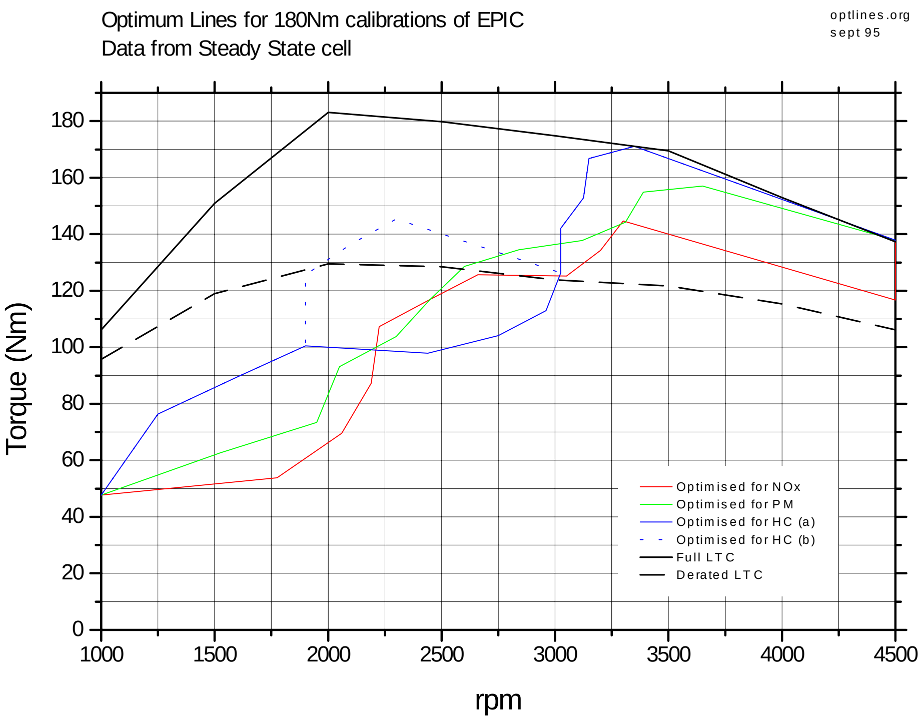

Figure 4.3 - IOLs for 180Nm rating of 1.8DI TCi

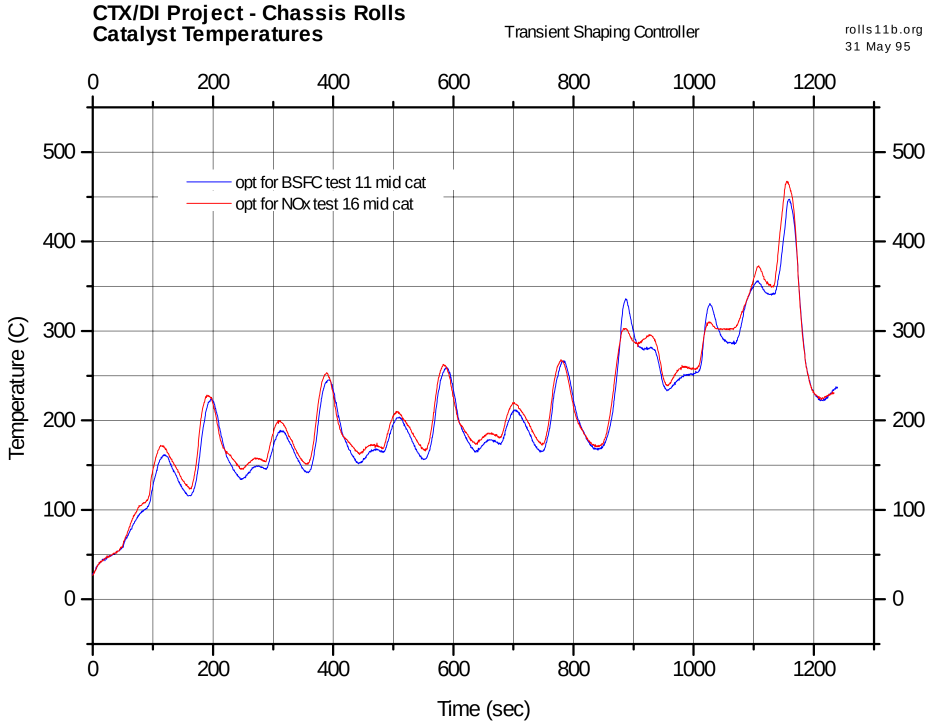

Figure 4.4a - Catalyst temperatures during ECE15 + EUDC

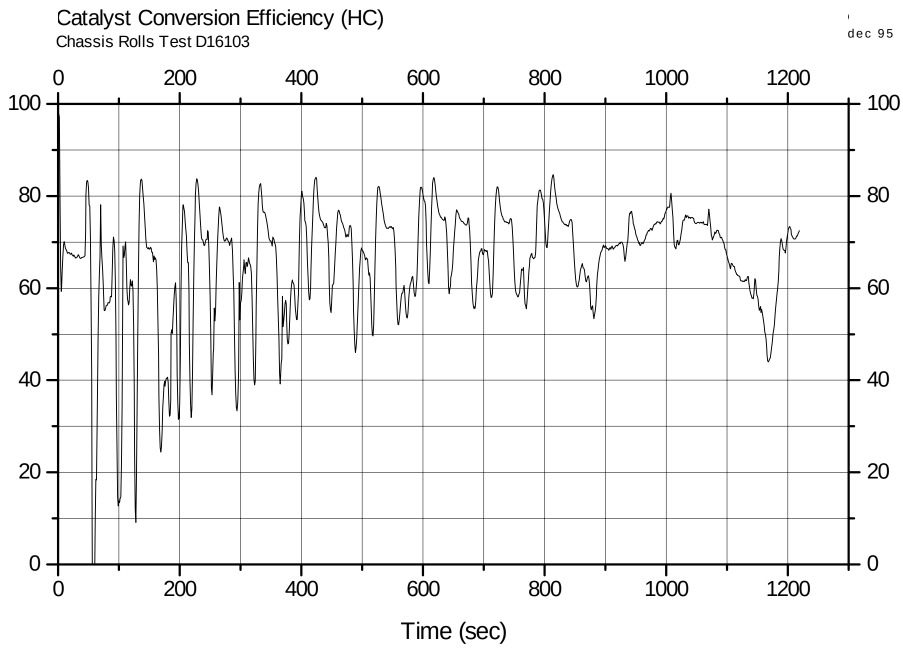

Figure 4.4b - Catalyst conversion efficiency for HC during ECE15 + EUDC

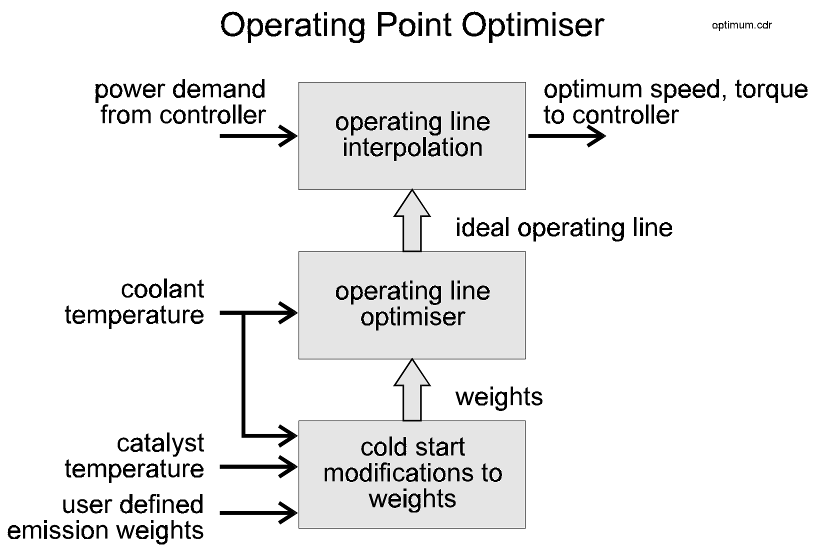

Figure 4.5 - Operating point generator - schematic

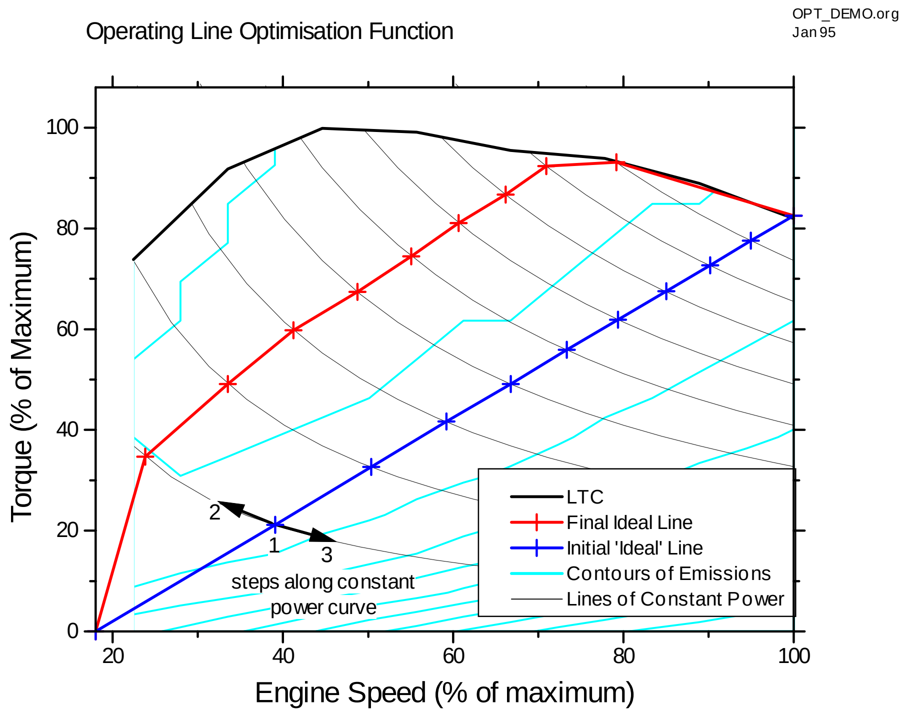

Figure 4.6 - IOL generator strategy

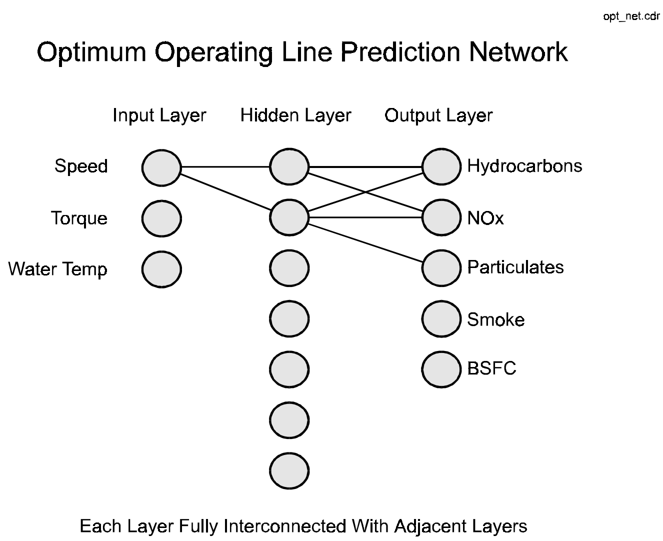

Figure 4.7 - Engine model structure

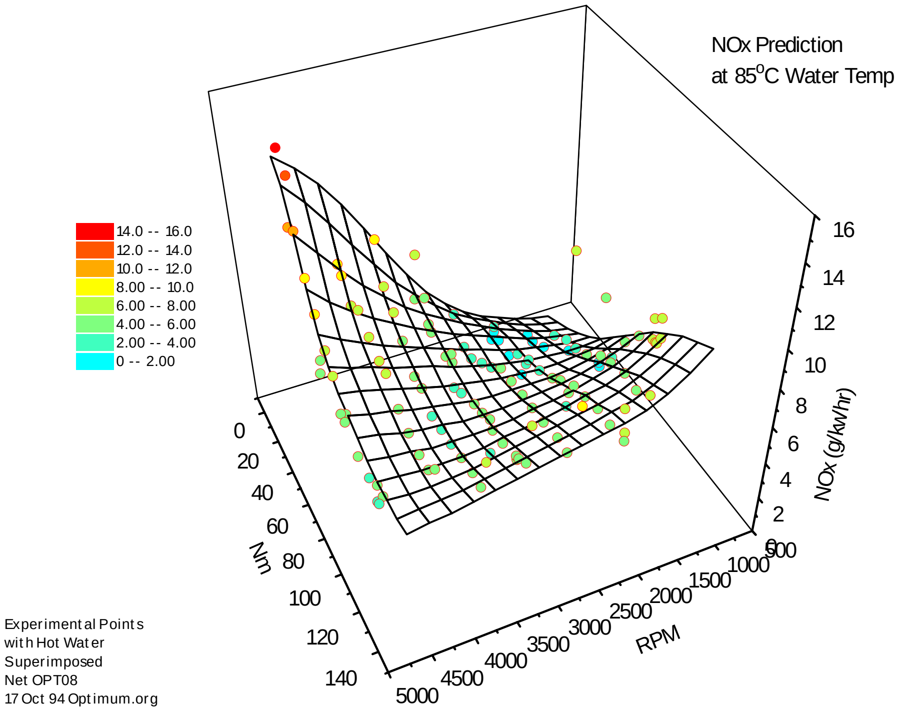

Figure 4.8 - Network predictions for NOx

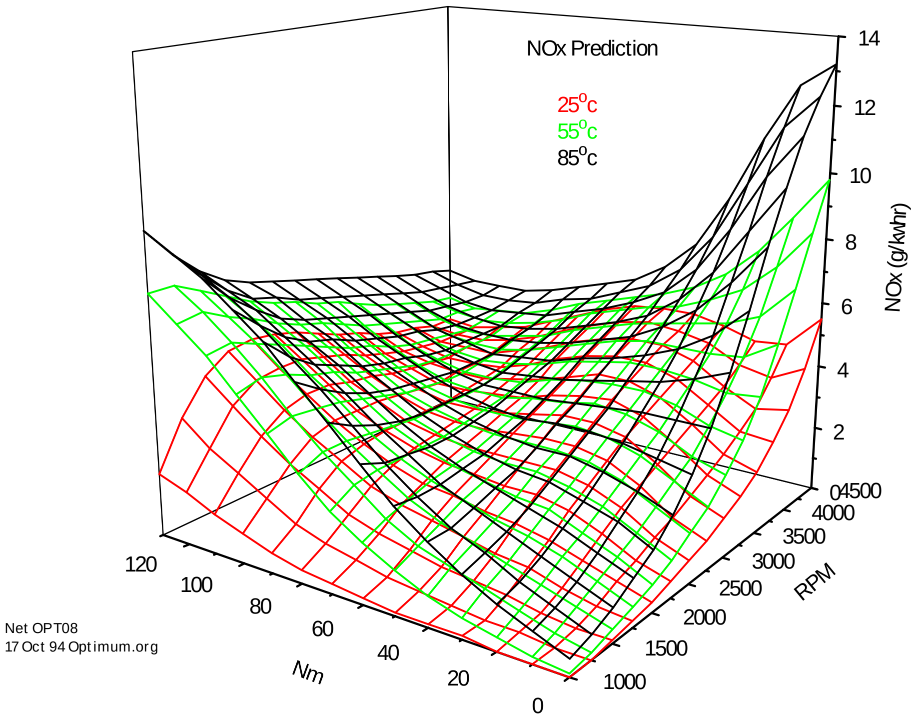

Figure 4.9 - NOx predictions at various temperatures

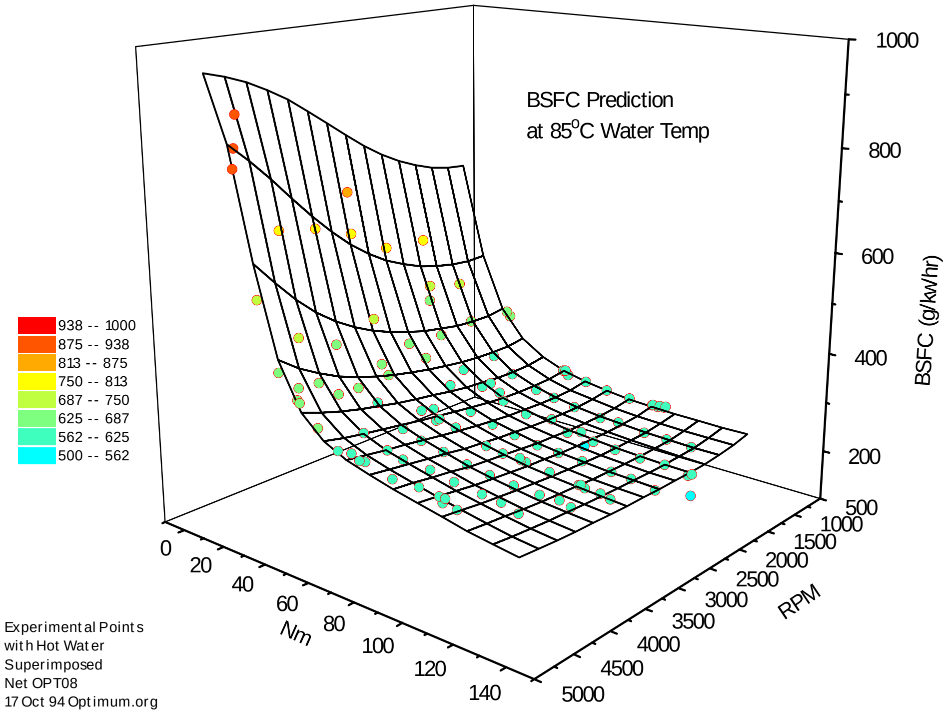

Figure 4.10 - BSFC predictions 85°C water temp

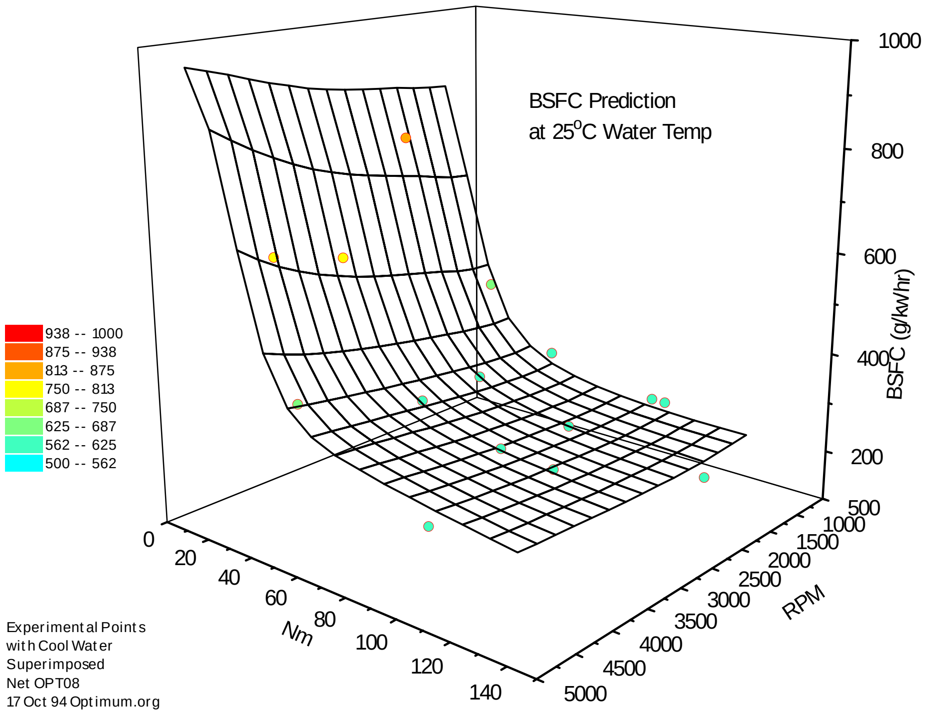

Figure 4.11 - BSFC predictions at 25°C water temp

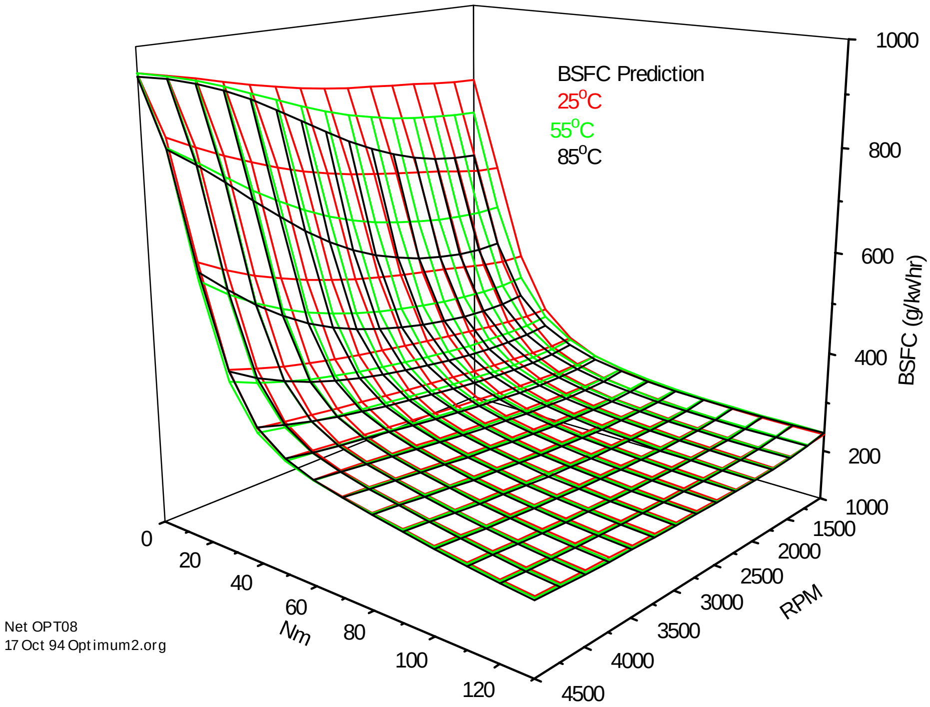

Figure 4.12 - BSFC predictions at various temperatures

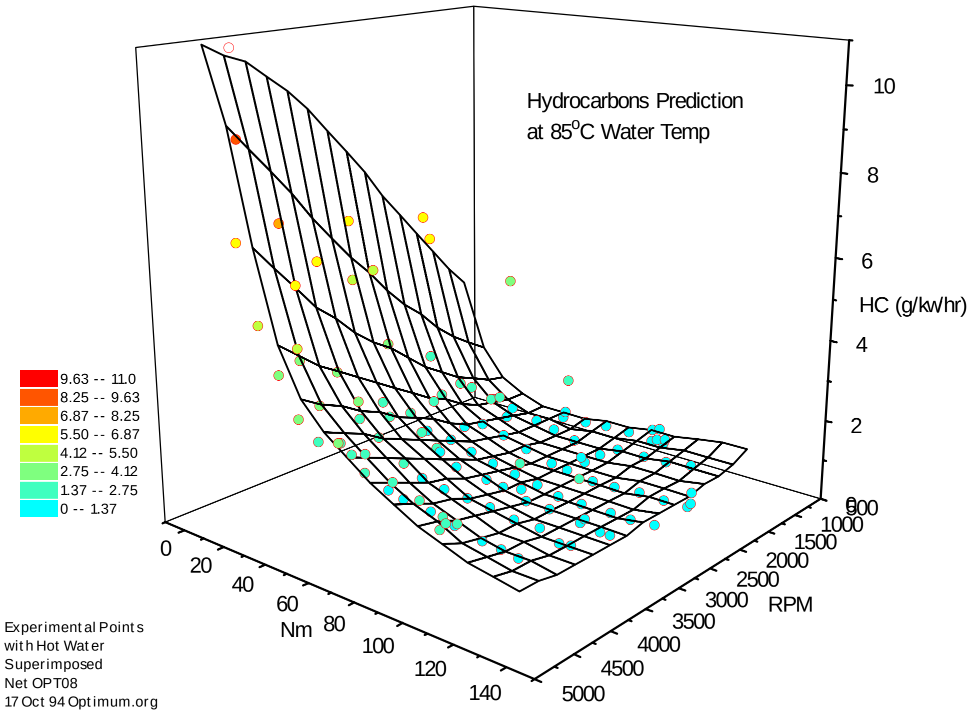

Figure 4.13 - HC predictions 85°C water temp

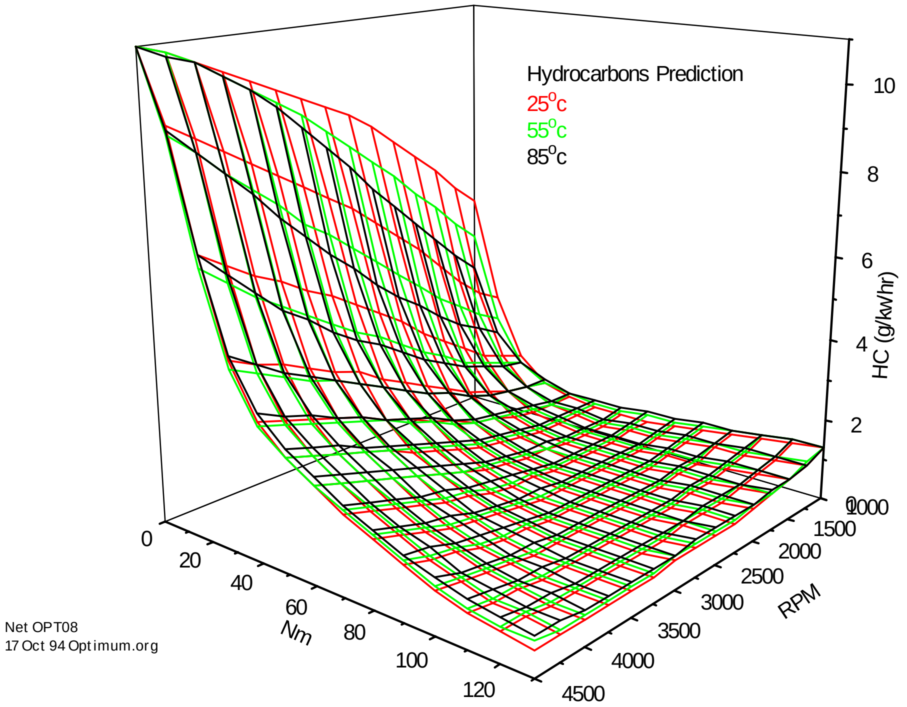

Figure 4.14 - HC predictions at various temperatures

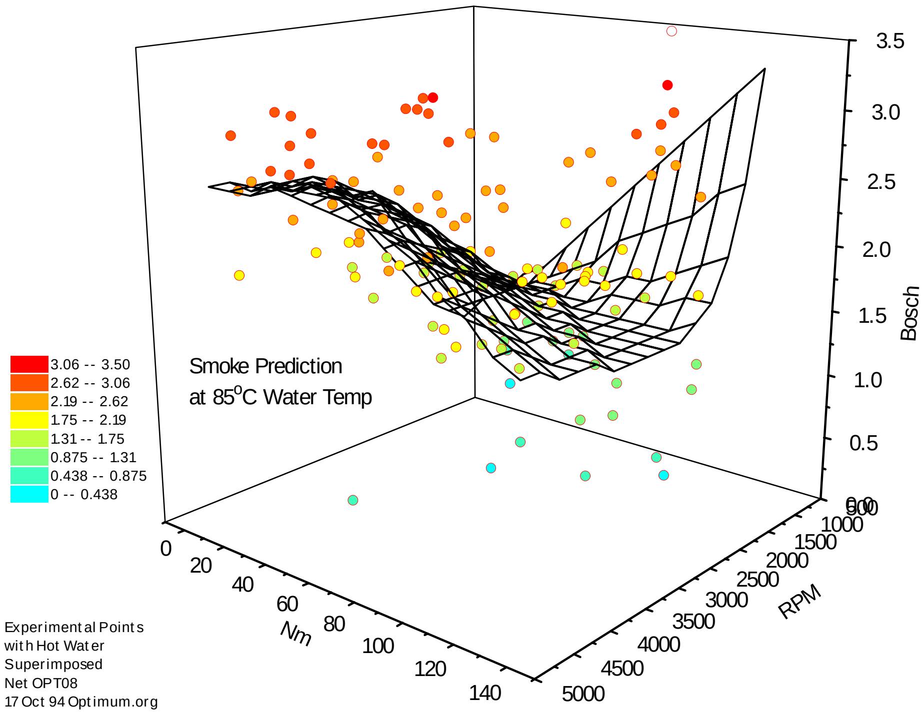

Figure 4.15 - Smoke predictions 85°C water temp

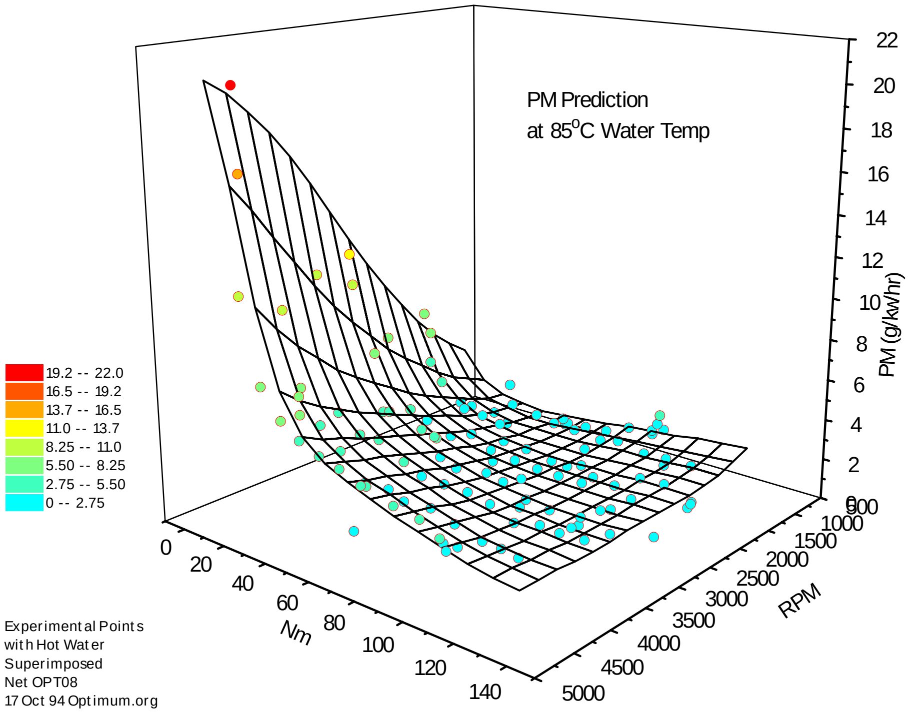

Figure 4.16 - PM predictions 85°C water temp

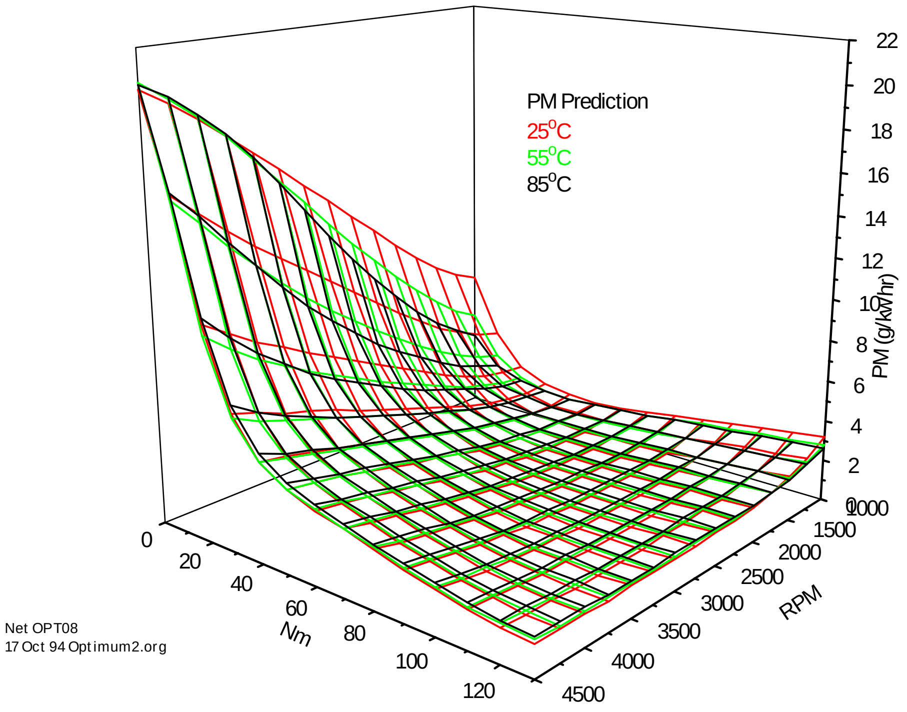

Figure 4.17 - PM predictions at various temperatures

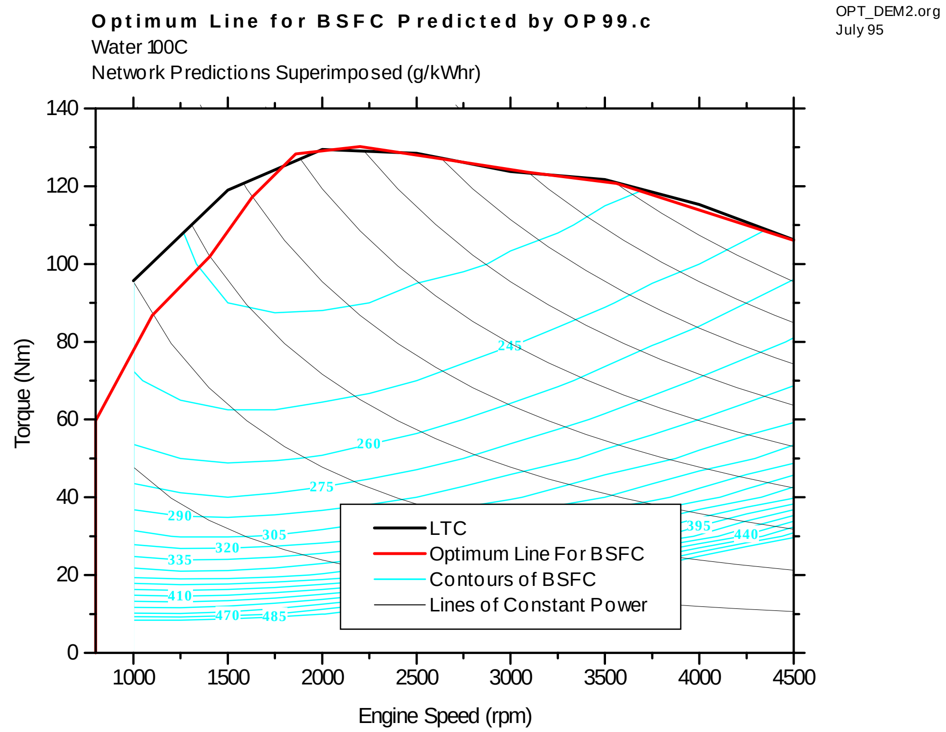

Figure 4.18 - Optimum line for BSFC

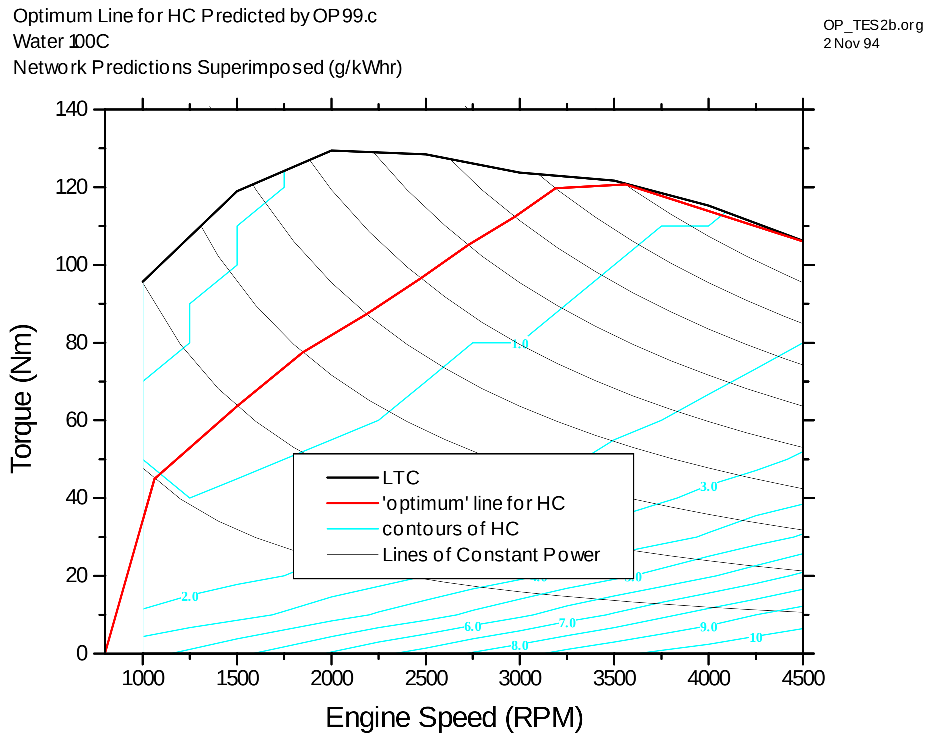

Figure 4.19 - Optimum line for HC

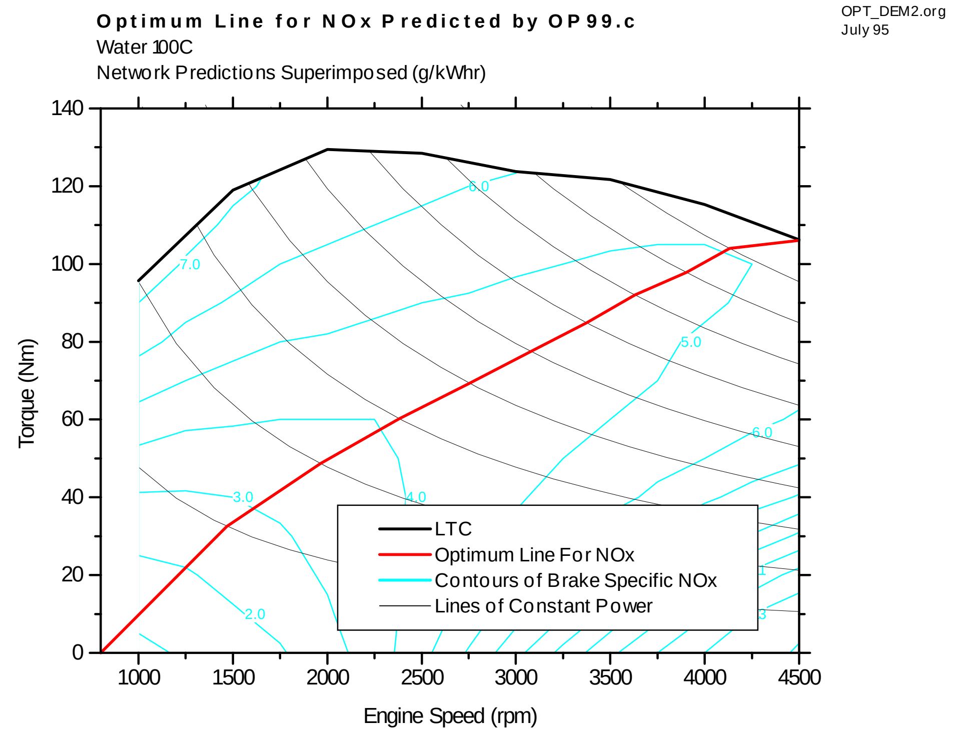

Figure 4.20 - Optimum line for NOx

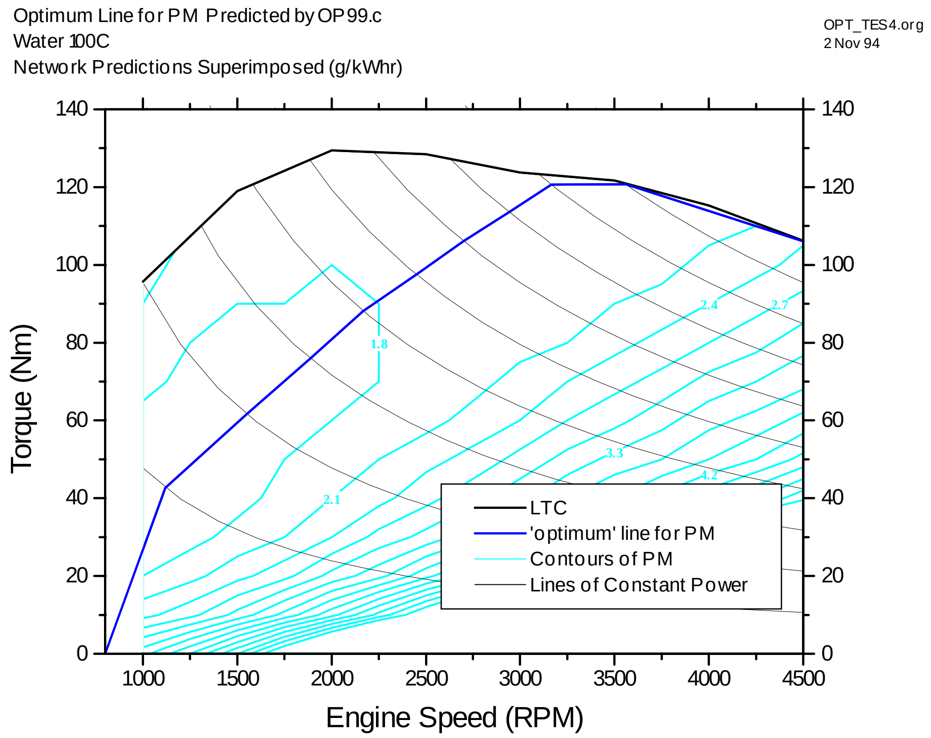

Figure 4.21 - Optimum line for PM

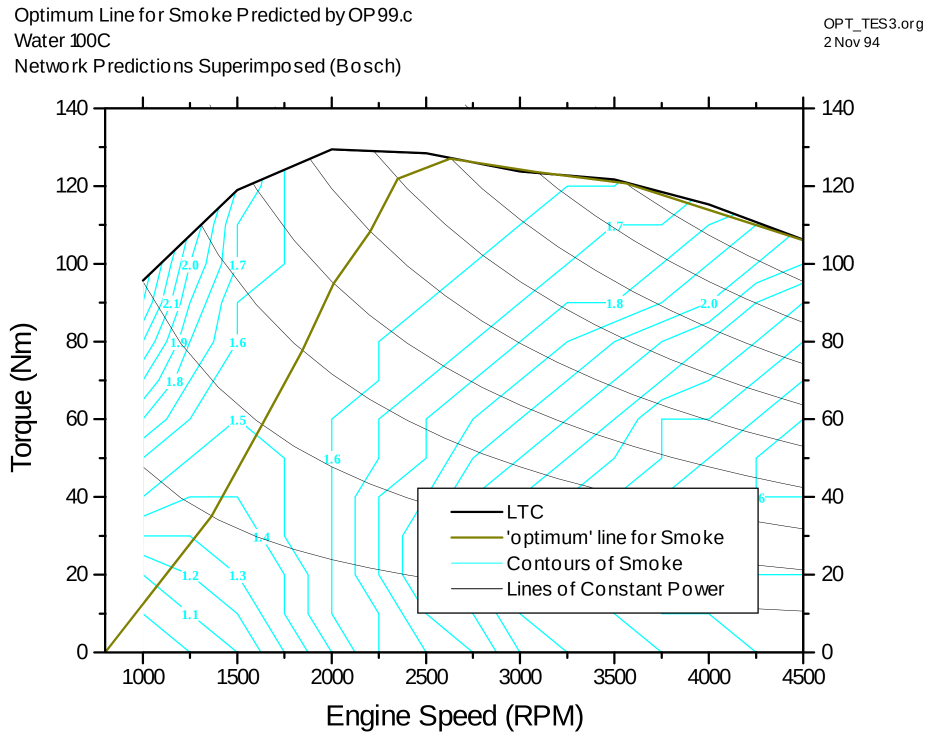

Figure 4.22 - Optimum line for Smoke

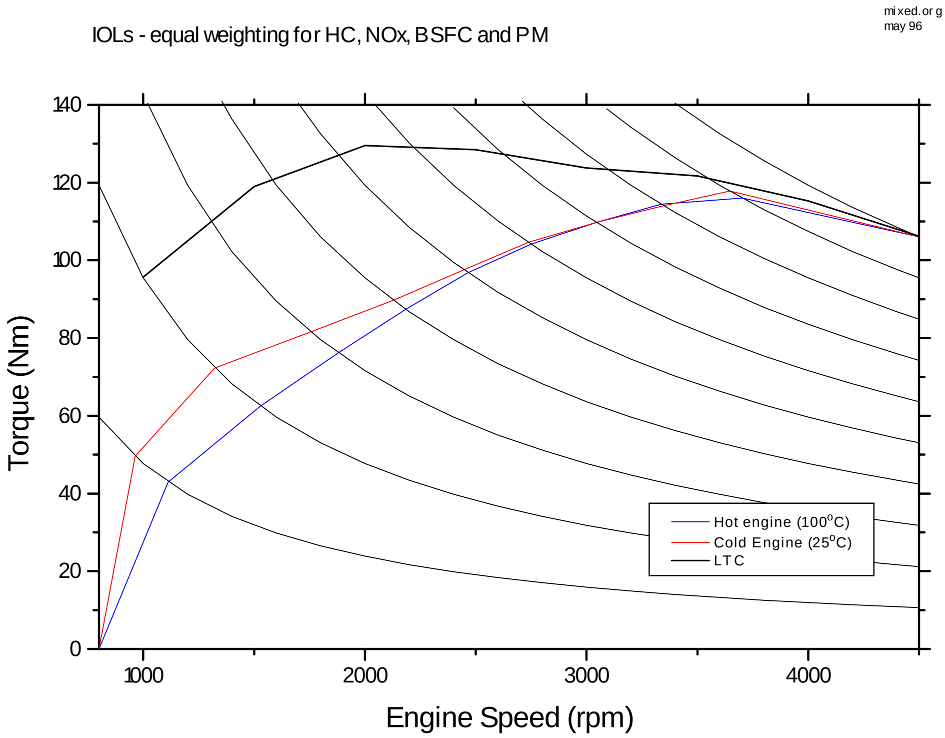

Figure 4.23 - Optimum line with unity weightings for HC, NOx, BSFC & HC

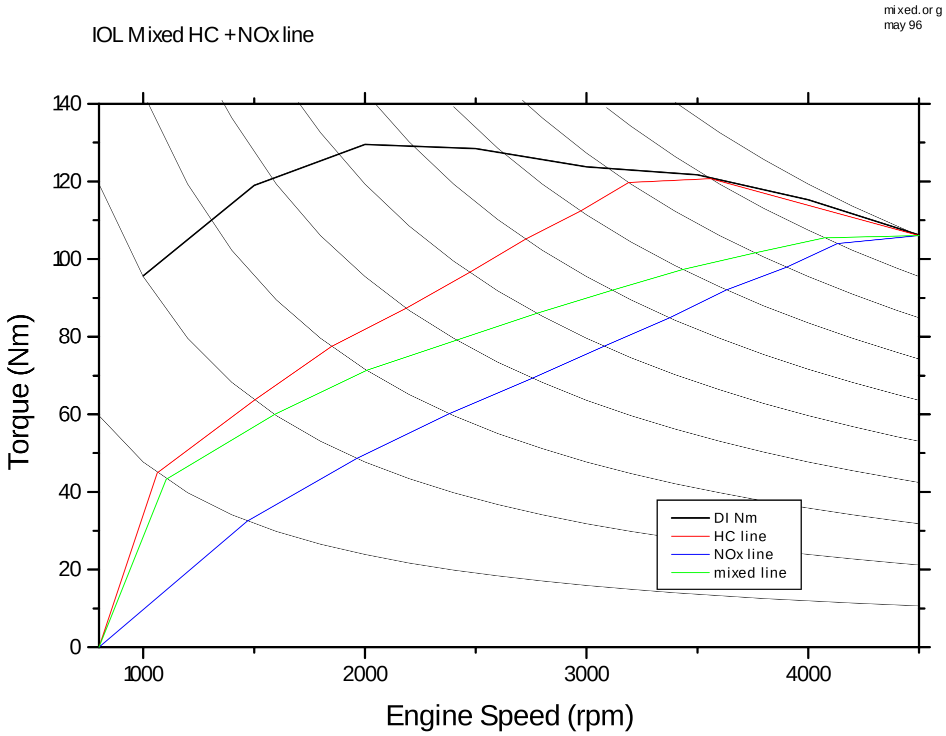

Figure 4.24 - Optimum line for HC & NOx

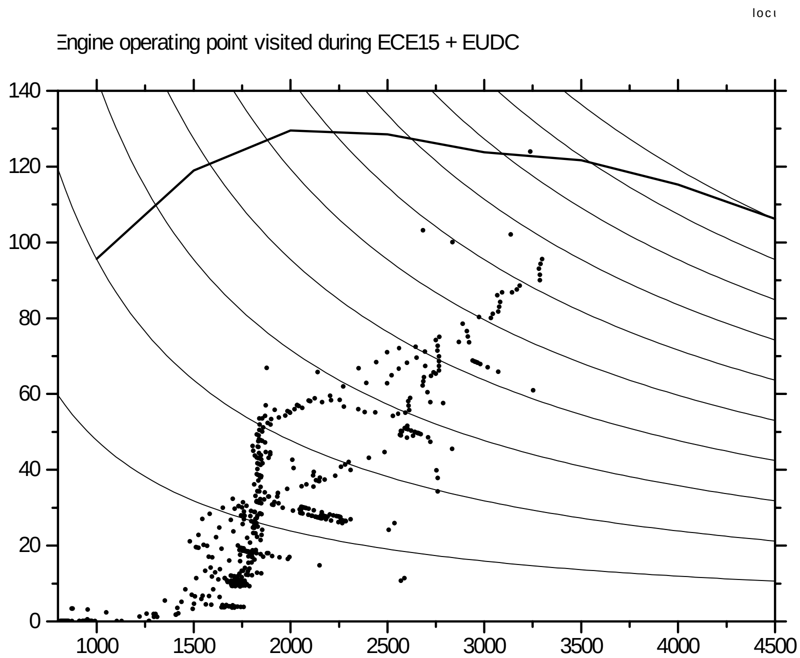

Figure 4.25 - Points visited during ECE15 + EUDC

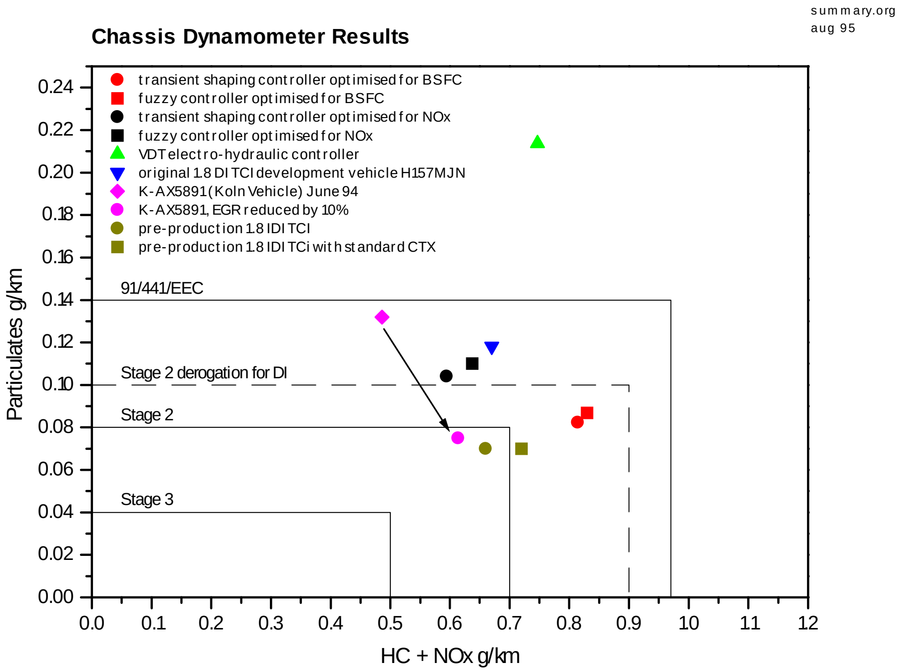

Figure 4.26 - Chassis dynamometer results

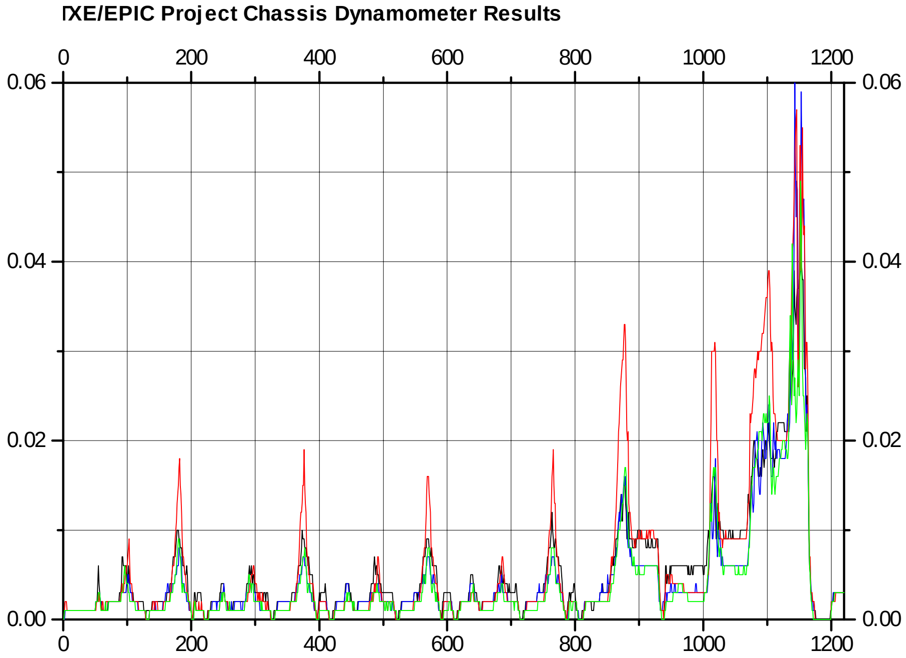

Figure 4.27 - NOx production during ECE15 + EUDC

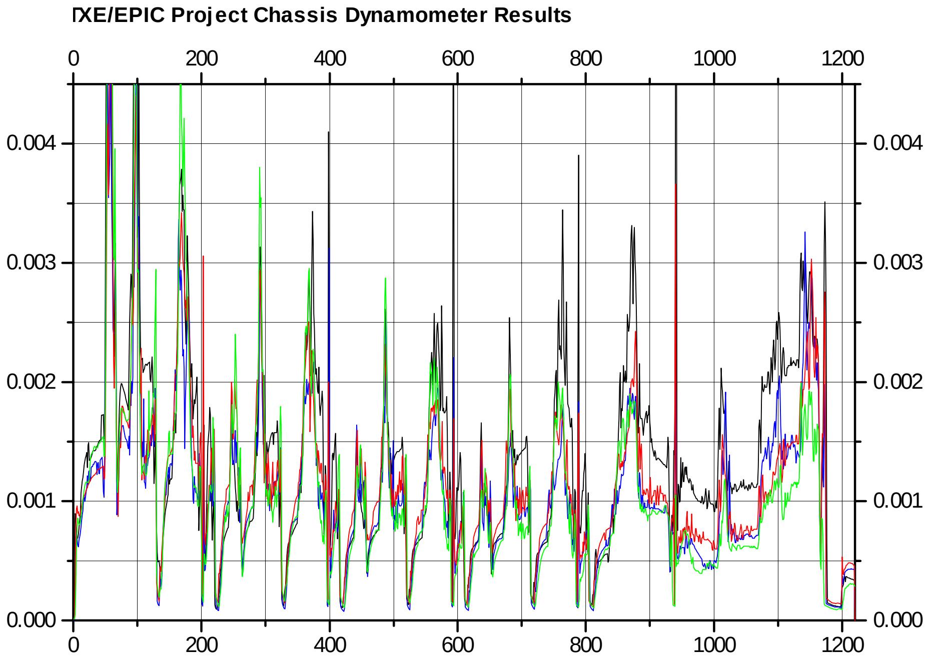

Figure 4.28 - HC production during ECE15 + EUDC

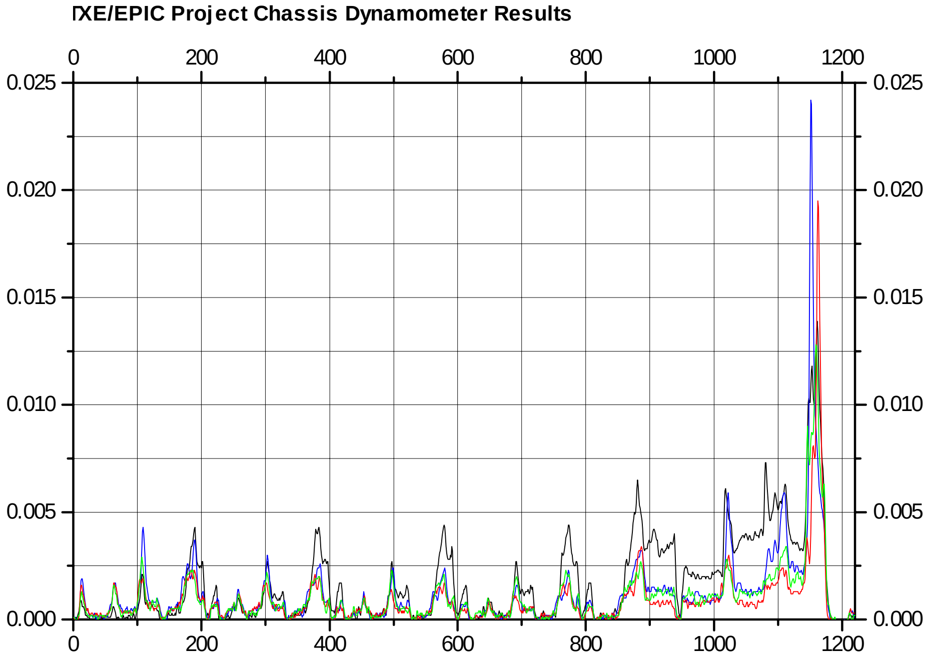

Figure 4.29 - PM production during ECE15 + EUDC

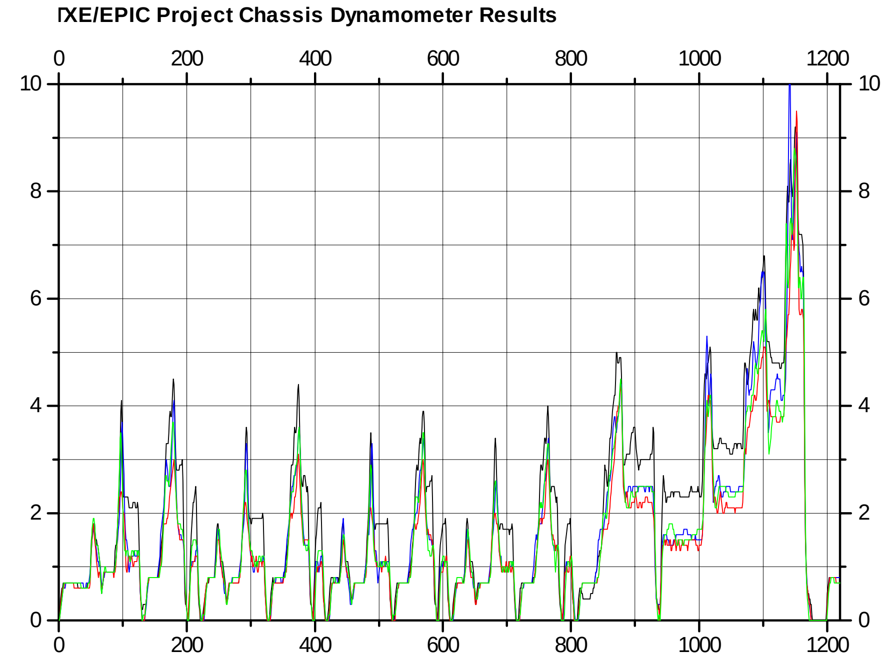

Figure 4.30 - CO_2 production during ECE15 + EUDC

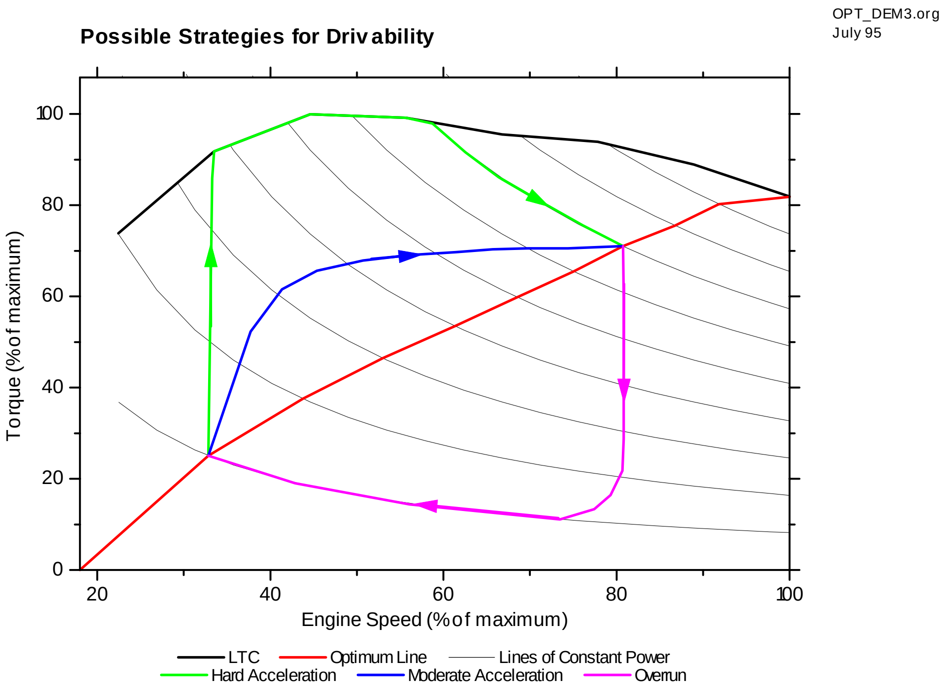

Figure 4.31 - 'Ideal' transmission control strategy for drivability