Natural Gas Vehicles Conference

Buenos Aires October 2004

Dual fuel allows Euro II without a catalyst on 1960's engine designs

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1 Abstract

Diesel engines designed in the 1950's and 1960's fail to meet modern emissions standards. A dual fuel engine management system, mechanical design changes and application engineering processes were developed and applied to two engine types of this vintage with the aim of meeting Euro II 1998 emissions standards using the R49 test cycle in an uncertified test cell that allowed comparative testing.

To allow the Daimler Benz OM355 and OM314 diesel engines to meet Euro II standards under these conditions required the innovative application of modern electronic controls, few mechanical changes and the application of alternative engine management and control processes to ensure success. No exhaust gas catalyst was required. 90% substitution at full load was achieved while the power and torque matched that of the diesel engines. Fuel efficiency on dual fuel at full load was improved by more than 10%.

2 Background

Dual fuel diesel engines using compressed natural gas (gas) as the alternative fuel involve the use of complex and varying combustion processes as the engine transitions between a genuine compression ignition (CI) combustion cycle at idle and Otto cycle pilot diesel ignition combustion at full load. Taking advantage of the benefits of each combustion type in various parts of the engine operating cycles has allowed the engine to meet emissions requirements.

The ECE R49 Euro II 13 mode test cycle does not cater for dual fuel engine testing and so some changes have been made to accommodate the mixed fuelling combustion. Using the Euro standards to measure the performance of a dual fuel engine means that the carbon monoxide (CO) limit is difficult to meet. USA standards generally allow about 4 times higher CO levels, maybe because of the greater use of large Otto cycle engines. Hydrocarbons (HC) are also difficult to meet and so for these trials only non-methane hydrocarbons (NMHC) have been included (similar to Euro III R49 requirements).

This paper presents some of the experience gained, innovative methods applied and test conditions used to achieve Euro II emissions standards on the OM314 engine.

3 Principles for dual fuel engine optimisation

3.1 Diesel and dual fuel combustion

Dual fuel engines combine diesel cycle CI combustion and Otto cycle combustion. The diesel engine under development uses excess air, high compression ratio (17:1) and a direct injection combustion chamber design. During dual fuel operation the gas is injected into the inlet air immediately prior to the inlet manifold, the air-fuel ratios are controlled within the lean burn range (21:1 to 27:1 air fuel ratios) and ignition is achieved by pilot injection of diesel.

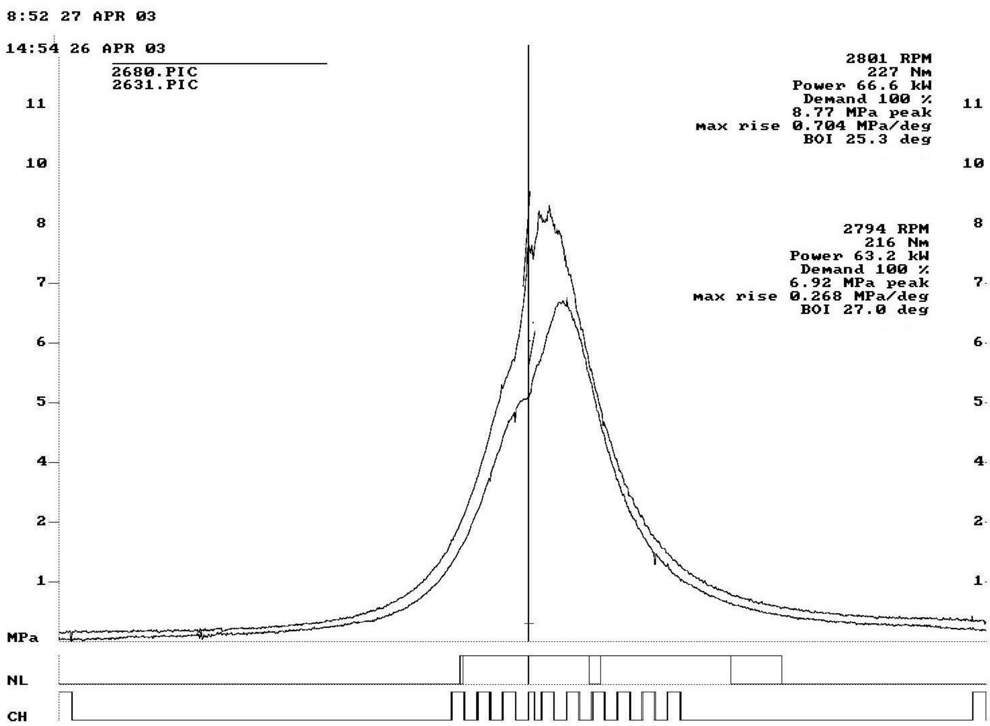
3.2 Combustion manipulation to achieve emissions targets

3.2.1 Nitrogen oxides (NOx)

The slower combustion rates of methane gas, at high substitution levels, achieves reduced NOx. See Figure 1 Diesel & dual fuel combustion pressure traces below.

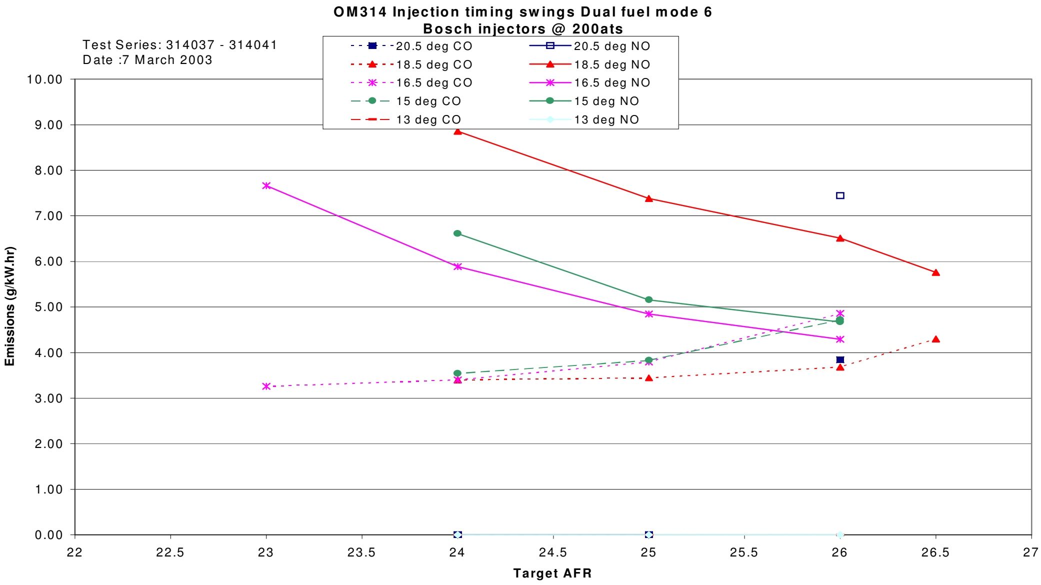
For the same power and torque the diesel combustion pressure peak and rise rate is higher than dual fuel reflecting the slower combustion of gas. The graph reflects lower base pressures on dual fuel caused by air throttling.

Figure 1 Diesel & dual fuel combustion pressure traces



NOx was shown to decrease on dual fuel as air-fuel ratios become leaner and the timing was retarded. Advancing the injection timing and reducing the air-fuel ratios saw an increase in NOx. See Figure 2 Injection timing swings on page 4.

Figure 2 Injection timing swings



3.2.2 Non-methane hydrocarbons (NMHC)

High diesel substitution allows lower NMHC in the exhaust gases at high loads as there is much less NMHC in the fuel.

It is possible for the NMHC to increase at light load during dual fuel operation if air throttling to below atmospheric pressure allows the low inlet manifold air pressure (IMAP) to draw oil down the valve guides or past the piston rings. For this project a minimum IMAP of 60kPaA was used and no increase in NMHC was noted.

3.2.3 Carbon monoxide (CO)

CO was shown to increase on dual fuel as air-fuel ratios become leaner and the timing was retarded. Advancing the injection timing and reducing the air-fuel ratios saw a reduction in CO. Figure 2 Injection timing swings above.

CO at idle, 10% and 25% load required either a reduction in the IMAP below 60kPaA or operating on diesel to achieve Euro II compatible CO emissions. Because operating this engine below 60kPaA IMAP was not considered a viable option the engine was operated on diesel at these 13 mode test points.

3.2.4 Particulate matter (PM)

Pre-conversion, particulate matter was very high on diesel. See Table 1 Pre-conversion diesel emissions test result (314010) below. Dual fuel operation saw more than a 95% reduction in PM. See Table 2 Dual fuel Emissions test result (314095) below.

Table 1 Pre-conversion diesel emissions test result (314010)

Weighted Results		BS NO		BS THC	BS NMHC		BS CO		CO2	P.M. (Soot + 40%)		BSSoot
Result	(g/kW.hr)	7.85	Meets Euro I	0.43		Meets Euro II	8.52	Fail	512.43	1.784	Fail	1.275
Euro I	1992	8.00	Pass	1.1	1.10	Pass	4.50	Fail		0.612	Fail	
Euro II	1996	7.00	Fail	1.1	1.10	Pass	4.00	Fail		0.250	Fail	

Table 2 Dual fuel Emissions test result (314095)

Weighted Results		BS NO		BS THC	BS NMHC		BS CO		CO2	P.M. (Soot + 40%)		BSSoot
Result	(g/kW.hr)	5.83	Meets Euro II	4.78	0.60	Meets Euro II	3.76	Meets Euro II	393.12	0.070	Meets Euro II	0.050
Euro I	1992	8.00	Pass	1.1	1.10	Pass	4.50	Pass		0.612	Pass	
Euro II	1996	7.00	Pass	1.1	1.10	Pass	4.00	Pass		0.250	Pass	

After cross checking the calculations against dilution tunnel results, the measurements for these tests were calculated. Ref:1 This method of calculating and reporting results was accepted as the readings were about 1/3 of the allowable limit.

3.2.5 Carbon dioxide (CO_2)

The replacement of more than 80% of the total diesel fuel used with gas, allowed a reduction of 24% in CO_2 emissions levels on dual fuel. See Table 1 Pre-conversion diesel emissions test result (314010) & Table 2 Dual fuel Emissions test result (314095) above

3.2.6 Emissions summary

Meeting NOx and PM is relatively straightforward using dual fuel. Euro II CO levels can be easily met by diesel engines but are difficult to meet with dual fuel Otto cycle engines.

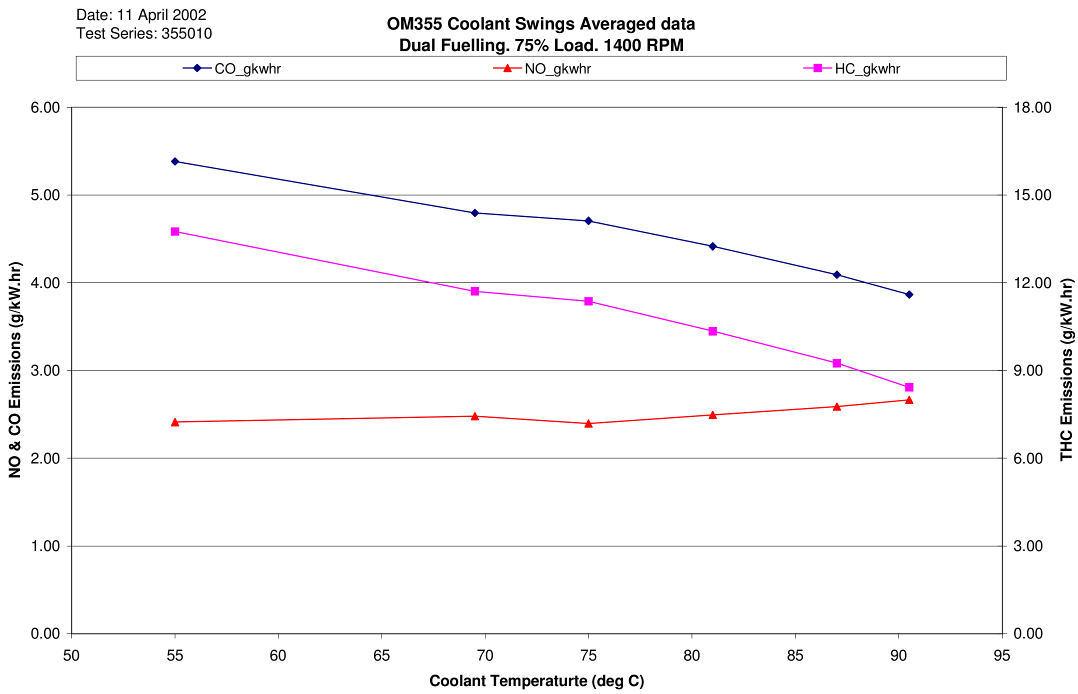
CO is the most difficult parameter to meet. It appears dual fuel Otto cycle type engines were not considered when the ECE R49 Euro II test cycle was developed, ECE R49 Euro III requirements allow for Otto cycle engines.

For the USA, where Otto cycle engines are often of a size used in commercial vehicles, emission standards have CO levels set at about 4 times that of the Euro standards. On this project about 80% of the emissions reduction development time was spent on reducing CO from 6g/kWhr to 4g/kWhr. This action also results in a major increase (about 50% to >7g/kWhr) in the level of NOx compared to a fuelling strategy required to achieve CO levels half that of the USA standard.

3.3 Coolant temperature

The pre-heating temperature of the combustion chamber is partially determined by the temperature of the engine coolant. Maintaining the coolant temperature at the higher of the two thermostat options available for this engine showed a general reduction in CO. See Figure 3 Coolant temperature analysis on page 6

Figure 3 Coolant temperature analysis



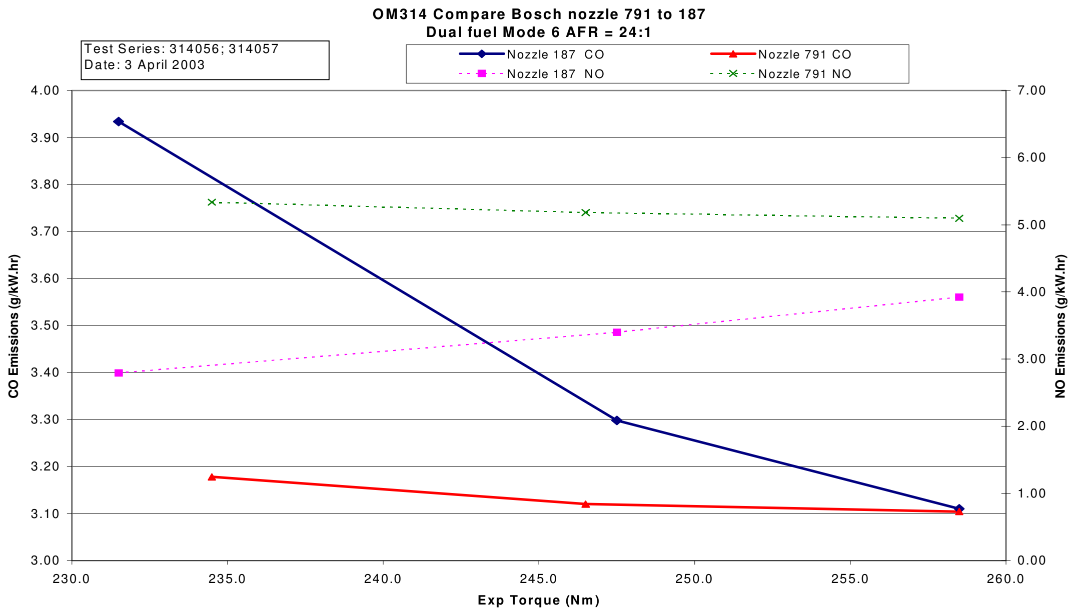
3.4 Injector nozzles type

The OM314 engine received for development was fitted with nozzle type DLLA150S187. This was original nozzle type fitted from the 1960's. Using the nozzles developed by Robert Bosch for the OM314 engine in 1980, part number DLLA142S791, allowed a reduction in CO. The obvious difference is a reduction of 8^{O} in the nozzle spray angle. This nozzle was used for the EURO II emissions trails.

A further nozzle developed by Robert Bosch for the OM314 has been identified and is used in Brazil. This nozzle appears to allow a further reduction in CO.

Modern diesel engine designs show reduced CO in diesel operation. Although dual fuel combustion shows an inherent increase in CO with each of the nozzle types, each of the more modern nozzles showed a continuing reduction in CO production on dual fuel. See Figure 4 Nozzle comparisons on page 7

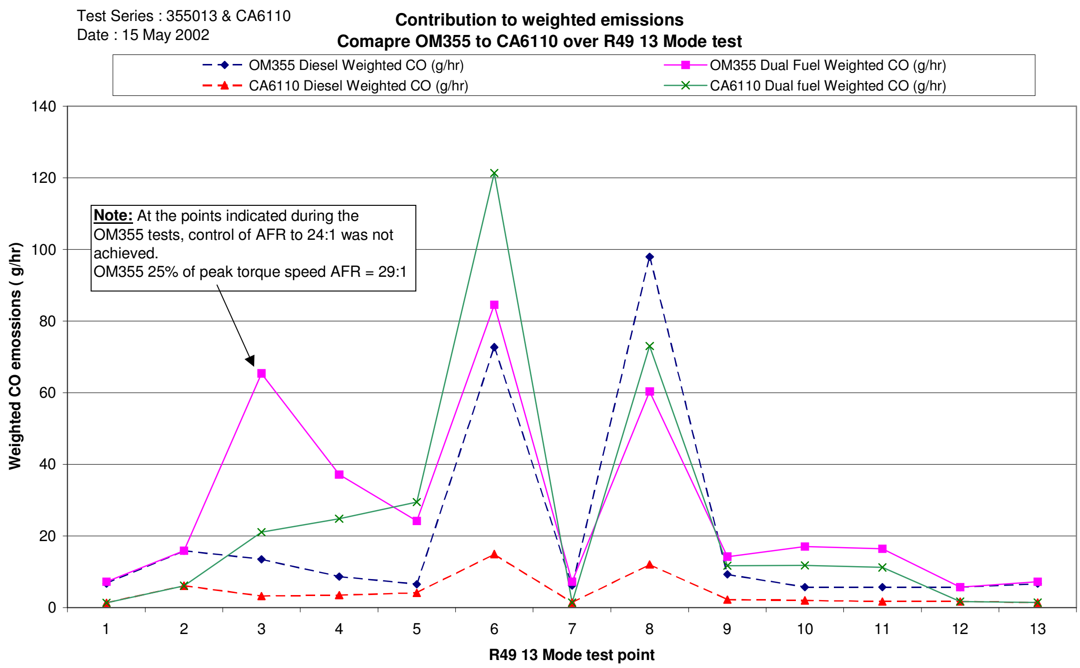
Figure 4 Nozzle comparisons



3.5 Combustion chamber shape

A review of two radically different combustion chamber shapes for various engines tested on dual fuel showed a large variation in CO production on diesel. The same engines on dual fuel did not exhibit the same degree of variation. See Figure 5 Combustion chamber comparison below

Figure 5 Combustion chamber comparison



3.6 Air fuel ratio control

3.6.1 Turbocharging

The OM314 engine on diesel at sea level had air-fuel ratios of 15.8:1 at peak torque and full load. On dual fuel at 100% load a reduced air-fuel ratio will be experienced because the gas injected into the air stream replaces about 10% of the air.

The requirement for 21:1 to 27:1 air-fuel ratios on dual fuel meant the use of a turbocharger was required.

3.6.2 Air throttle

At less than 100% load and above 1,000 rpm a reduction in airflow through the engine is required to maintain the correct air-fuel ratio for good fuel efficiency and emissions. This reduction in airflow was achieved by the use of a stepper motor controlled throttle plate.

Maintaining air-fuel ratios within +/- 0.25:1 of optimum allowed the balance between CO and NOx to be controlled and maintained to meet the emissions targets.

3.7 Combustion timing

Combustion timing is affected by two main parameters. Static diesel injection timing and ignition delay following injection of the diesel. Diesel Cetane number variation had a strong influence on ignition delay in this project.

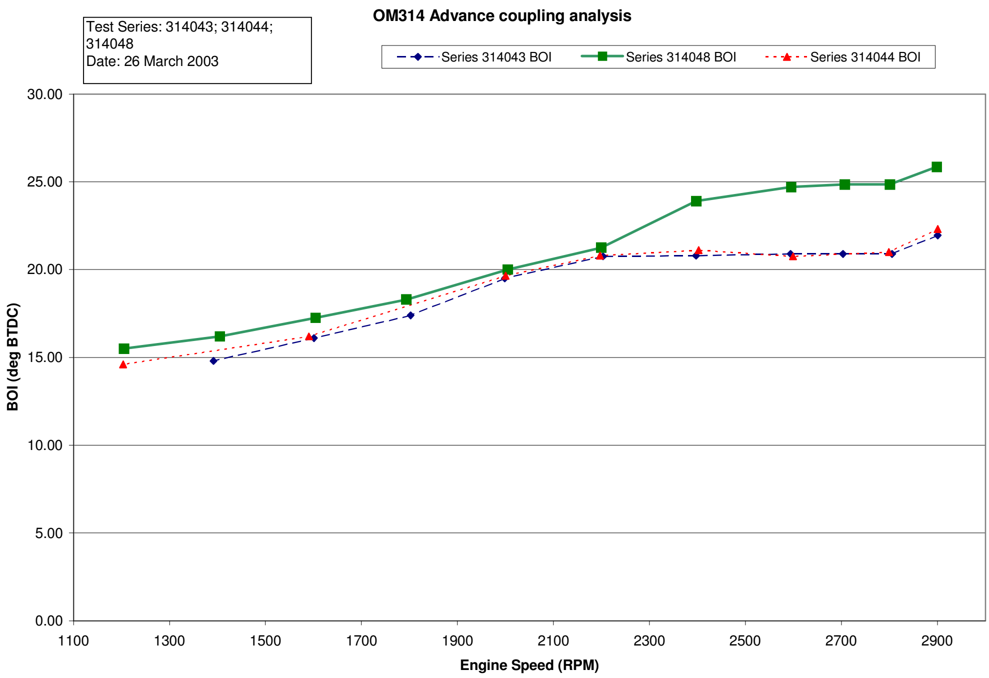
3.7.1 Static timing

After testing timing settings from 13^{O} to 19^{O} BTDC the static diesel injection timing was optimised at 16^{O} BTDC. 15^{O} BTDC is standard for this engine.

3.7.2 Dynamic timing

The automatic advance coupling was designed with a 9^{O} of advance capability. The Advance coupling analysis graph on page 9 shows a change in dynamic timing as the engine speed increases. The variation between the two curves shows the automatic advance coupling was initially incorrectly adjusted and after being correctly adjusted the engine achieved the correct dynamic advance. The extra advance at rated engine speed allowed reduced CO and improved fuel efficiency at this speed.

Figure 6 Advance coupling analysis



3.8 Diesel cetane number

The Cetane number specified for ECE R49 Euro II testing is 52 to 54. Ref 3. During OM355 engine development, diesel purchased in New Zealand showed cetane index variations from 49 to 61. These changes meant that, with the same engine set up, the ignition delay changed several degrees. As the cetane index increased the ignition delay was reduced and the timing advanced, as the cetane index reduced the ignition delay was increased and the timing retarded. These changes were sufficient to mean the engine failed to meet Euro II when the diesel was other than that specified. For the OM314 engine development sufficient diesel was purchased, with a cetane index of 54, to complete the project. Ref:4.

3.9 Substitution

Taking advantage of the differences between CI and Otto cycle combustion processes and choosing to use one or other cycle in each of the 13 modes meant the best advantage could be taken of each combustion cycle. Because the Otto cycle produces high CO at idle and low loads these modes were operated on diesel in a gas suppression mode. This accounts for modes 1, 2, 3, 7, 11, 12, & 13 as shown in Fuelling Configuration C. See Table 3 Fuelling configurations table on page 10.

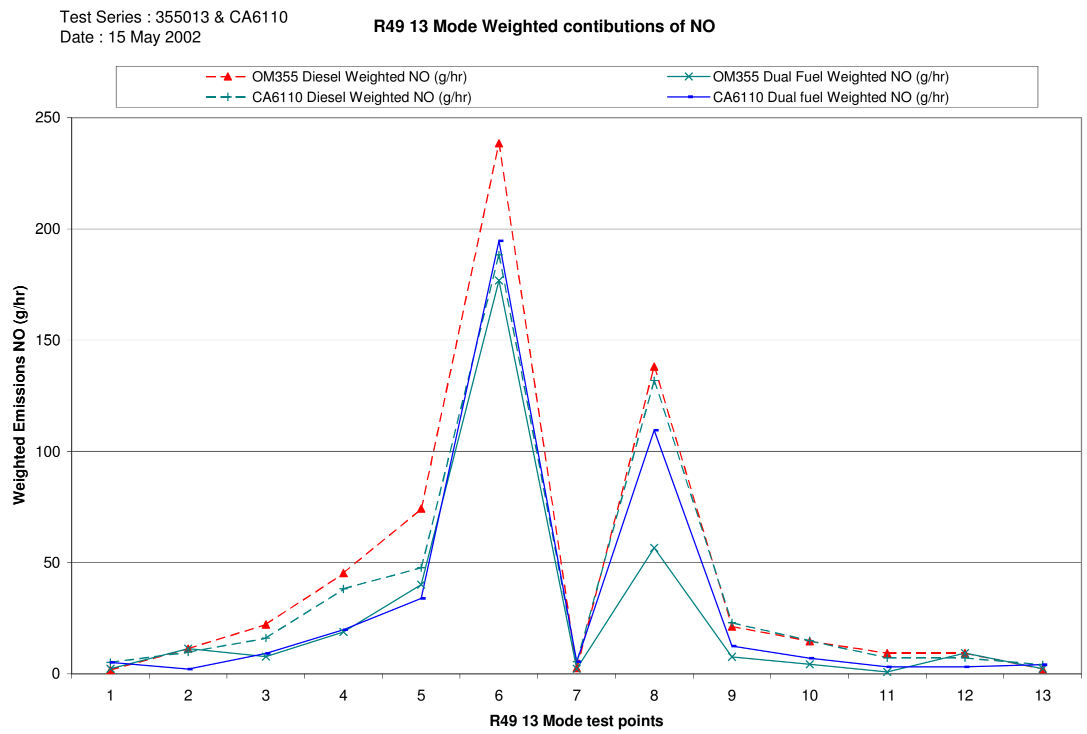
Table 3 Fuelling configurations

Fuelling:
d = Diesel Only
df = Dual Fuel

R49 Mode #	A	B	C	D	E	F	G	H	I	J	K	L
	Configuration											
1	d	d	d	d	d	d	d	d	d	d	d	d
2	df	d	d	d	d	d	d	d	d	d	d	d
3	df	df	d	d	d	d	d	d	d	d	d	d
4	df	df	df	df	d	d	d	d	d	d	d	d
5	df	df	df	df	df	df	df	d	d	d	d	d
6	df	df	df	df	df	df	df	df	df	d	df	d
7	d	d	d	d	d	d	d	d	d	d	d	d
8	df	df	df	df	df	df	df	df	df	df	d	d
9	df	df	df	df	df	df	d	df	d	d	d	d
10	df	df	df	d	df	d	d	d	d	d	d	d
11	df	df	d	d	d	d	d	d	d	d	d	d
12	df	d	d	d	d	d	d	d	d	d	d	d
13	d	d	d	d	d	d	d	d	d	d	d	d

In choosing the optimum cycle for each of the 13 mode test points a graph of emissions for the R49 13 mode test points for CO & NOx was used. This was found to be a very useful tool to plot the weighted emissions in g/hr against the number of the mode. It instantly identified the emission contribution from each mode and made comparisons between the different test conditions easy. After testing each configuration, configuration C was used to optimise emissions reductions. See Figure 6 Combustion chamber comparison on page 7 & Figure 7 NO Contributions below

Figure 7 NO Contributions



Modes 6 & 8 are at 100% load and produce the largest quantity of emissions. At 100% load the engine showed a plateauing of emission as the substitution was increased up to a max of 90%.

3.10 Other

3.10.1 Using ECE R49 test standards for dual fuel engines

The ECE R49 emissions standard specifies procedures for measuring and calculating the mass flow of pollutants for engines fuelled wholly by diesel and wholly by natural gas. Calculation procedures differ in the areas of wet vs dry concentrations, NOx humidity dependency, and final mass flow computation.

Because the standard does not consider the reality of dual fuelling, the calculation procedures were extended by performing a weighted combined calculation at each of the stages that are specified differently. The weight factor in each case is the proportion of diesel and natural gas by mass of total fuel. For example, in the area of wet vs dry concentrations, the chemistry of the two fuel types leads to a difference in the amount of water that will be produced during combustion. Hence the amount of water that is presumed to have been lost during drying of the exhaust gas has to be based on which fuel was used. Ref:5.

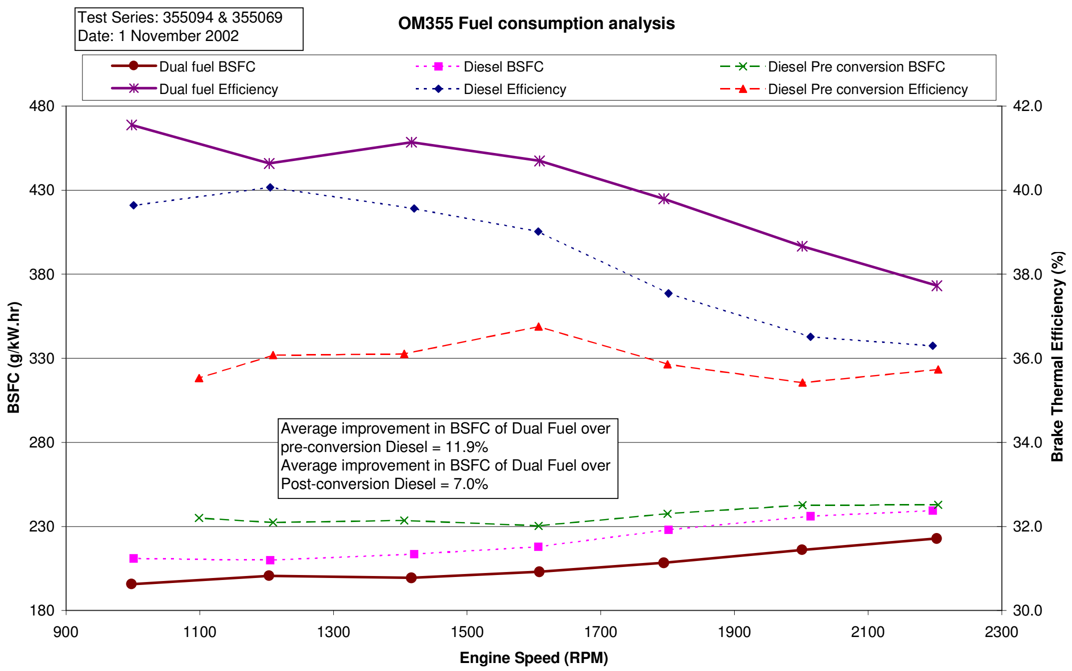
3.10.2 Indirect measurement of pollutants

The test cell measuring instruments used for these tests were not capable of providing the specified direct hydrocarbons and particulates, instead NO and total hydrocarbons were measured with a 5 gas analyser and soot calculated using the Bosch method. Each of these measurements was converted to an estimate of the specified quantity by aligning the measurement with results from specified instruments, then applying a linear correction factor that included a reasonable error margin.

3.10.3 Improved fuel efficiency

Fuel efficiency on dual fuel at full load improved more than 10% over diesel operation. See Figure 8 Fuel consumption analysis on page 12

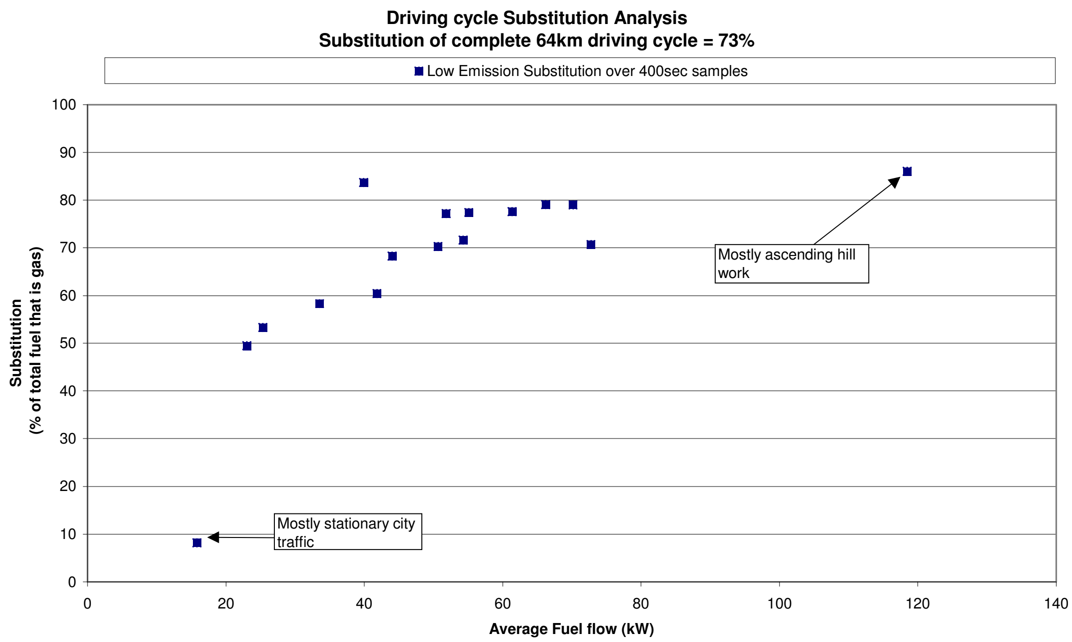
Figure 8 Fuel consumption analysis



3.10.4 Driving cycle substitution

Driving substitution for a mixture of city and urban driving using fuelling configuration C, showed a substitution of 73%. See Figure 9 Substitution analysis below

Figure 9 Substitution analysis



3.10.5 CO_2 reduction

CO_2 on dual fuel for the OM314 engine shows a reduction of 24%. See Table 1 Pre-conversion diesel emissions test result (314010) & Table 2 Dual fuel Emissions test result (314095) on page 5.

4 Process of emissions reduction

4.1 Baseline diesel

4.1.1 Naturally aspirated diesel engine

The pre-conversion diesel engine performance, see Table 1 Pre-conversion diesel emissions test result (314010) on page 5, shows very high CO and PM, NO met EURO I and THC met EURO II.

4.1.2 Turbocharged diesel

Emissions from the OM314 engine with turbocharger installed and wastegate setting at the diesel engine standard of 160kPaA saw an interesting change in emissions. NOx increased and CO decreased. See Table 4 Diesel emissions test result (314012) below

Table 4 Diesel emissions test result (314012)

Weighted Results		BS NO		BS THC	BS NMHC		BS CO		CO2	P.M. (Soot + 40%)		BSSoot
Result	(g/kW.hr)	9.09	Fail	1.13		Fail	4.64	Fail	546.75	1.618	Fail	1.156
Euro I	1992	8.00	Fail	1.1	1.10	Fail	4.50	Fail		0.612	Fail	
Euro II	1996	7.00	Fail	1.1	1.10	Fail	4.00	Fail		0.250	Fail	

4.2 Diesel injection timing swings

Diesel injection timing trials at 13^{O} 15^{O} 17^{O} & 19^{O} BTDC static timing showed reduced CO and increased NOx as the timing was advanced under dual fuel operation. As the timing was retarded CO increased and NOx reduced. See Figure 2 Injection timing swings on page 4

4.3 Substitution swings

Use of the same pilot diesel across load means an increasing diesel percentage of the total fuel across load. This meant 90% substitution at full load, 83% at 75% load and 75% at 50% load.

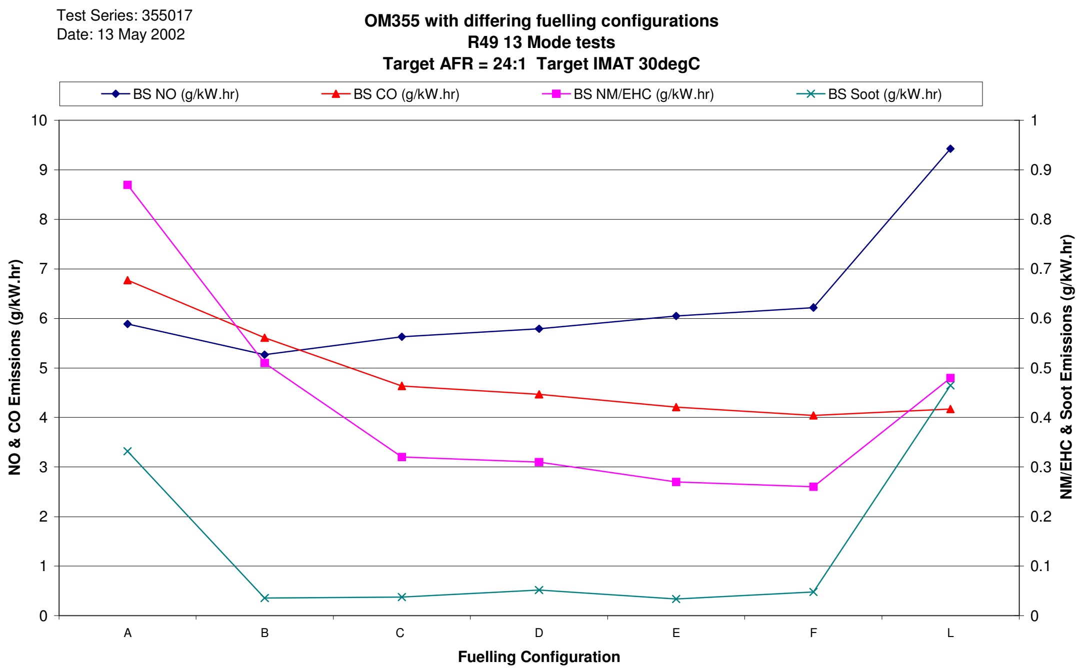
4.4 Oxidation catalyst

The results from the use of an oxidation catalyst showed a substantial reduction in CO and a small reduction in NOx. Using the oxidation catalyst CO emissions at 0.28 g/kWhr were less then 10% of a Euro II target of 4.00g/kWh. For most markets a catalyst is not a viable option because the high sulphur content in the diesel reduces the catalyst life.

4.5 Fuelling Configuration strategies

All Fuelling Configurations from A to L were tested. Figure 10 Fuelling configuration comparisons on page 14 demonstrates the trends in emissions performance for differing fuelling configurations. Refer to Table 3 Fuelling configurations on page 10 for the key to fuelling configurations.

Figure 10 Fuelling configuration comparisons



C Fuelling Configuration was the only configuration to achieve the emissions goal and on this vehicle gave satisfactory substitution levels for city 65%, urban 75% and highway 85% driving.

4.6 Bosch nozzle options

The original nozzles showed higher emissions whereas the 1980 nozzles showed a reduction in emissions and an increase in power. See Figure 4 Nozzle comparisons on page 7. The Brazil nozzles showed a further improvement in emissions reduction and increase in power.

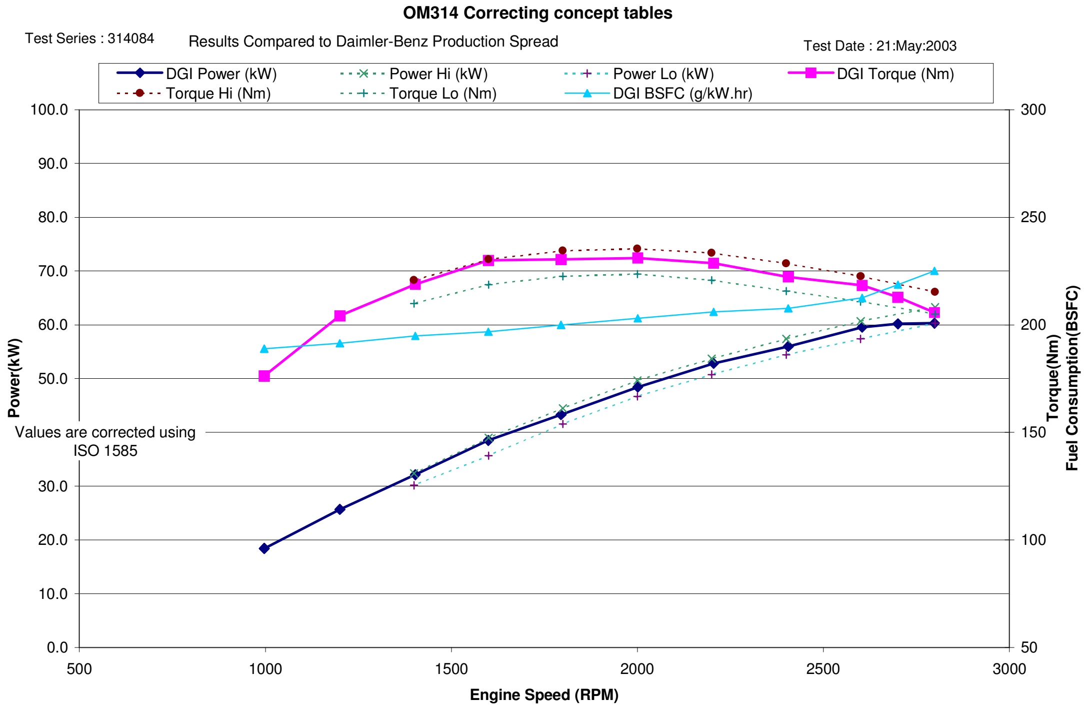
4.7 Dynamic advance

Correct dynamic timing of diesel injection is critical to emissions reduction and fuel efficiency. Identifying a fault in factory supplied advance coupling was an important step in the emissions reduction process.

4.8 Power & torque

Figure 11 Dual fuel performance curves on page 15 compares the dual fuel 100% load performance to the engine manufactures production spread curves for the diesel version of the engine.

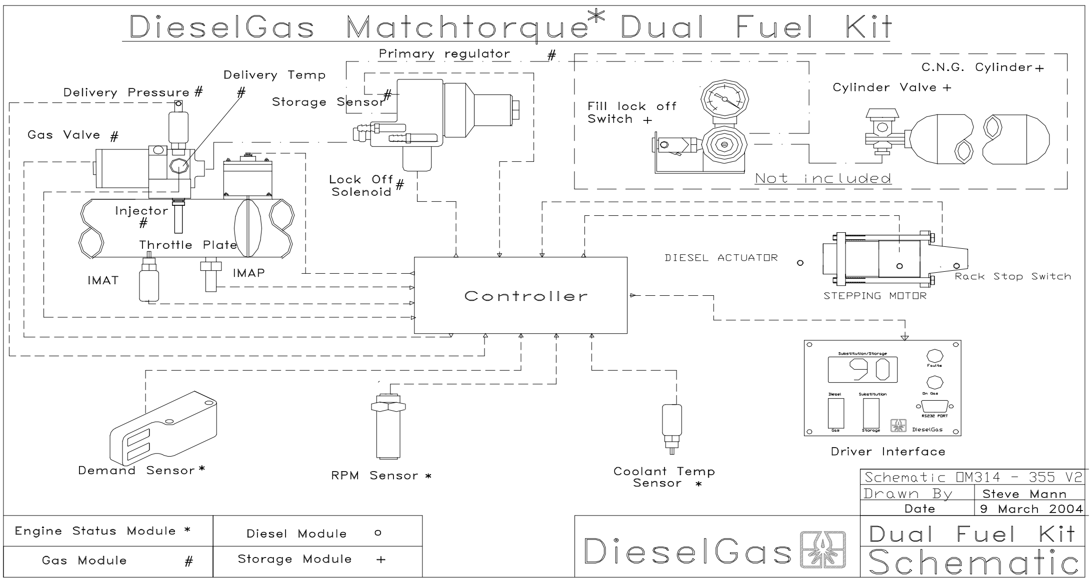
Figure 11 Dual fuel performance curves



5 Equipment & parameter choices

5.1 Hardware selection

No major changes were made to the engine mechanical hardware. The addition of a turbocharger and aftercooler was required to maintain the correct air fuel ratios.



The dual fuel engine kit contains the equipment set out in the schematic.

5.2 Tables & settings

The process of emissions reduction on the OM314 engine produced data that included over 4,500 engine test records.

From this data tables were developed to control diesel, gas, air and governor parameters at each of -10^{O}, 10^{O}, 30^{O}, 50^{O} & 70^{O} C IMAT. Each of the more than 400 parameter settings was established. These tables and settings were established with a strong focus on meeting the emissions standard, high substitution and allowing similar driveability to diesel.

5.3 Transient driveability testing

The OM314 engine is traditionally installed in small 20 seater type minibuses. The two installations undertaken are in chassis designs that are 40 years old. The installations proved a challenge in that they demonstrated “chuggle” on diesel and this negative feature was amplified on dual fuel. This required the application of targeted features to smooth the diesel to dual fuel transition and reduce “chuggle” to where dual fuel driveability was similar to diesel. Driving cycle testing on combined city and urban driving cycle showed over 73% substitution.

6 EURO II achieved

6.1 OM314 dual fuel engine specification

6.1.1 Mechanical hardware

No change in mechanical hardware components of the engine except

- Turbocharging and aftercooling added
- Injector nozzle change and pressure set to 200 Bar
- Injection timing changed from 15^{O} to 16^{O} BTDC
- Inlet air throttling on gas
- No catalyst required

6.1.2 Fuelling features

- Use of Fuelling Configuration C with dual fuel in modes 4, 5, 6, 8, 9, & 10 and diesel in the other 7 modes.
- Air-fuel ratios on dual fuel from 22:1 to 24:1

6.1.3 Performance

- Emissions met Euro II using DieselGas test cell and data calculations
- 90% substitution at full load
- Fuel efficiency 10% better than diesel at full load.
- Power and torque same as diesel and within Daimler Benz performance curves
- Driveability similar to diesel
- Driving substitution for a mixture of city, urban and highway driving showed substitution of 65% to 85%.

7 Conclusions

The OM314 engine achieved Euro II with few mechanical changes except for the addition of a turbocharger and aftercooler.

An extensive combination of engine testing, tables and settings development and on road adjustments has allowed an engine and vehicle performance similar to diesel.

The biggest challenge was the reduction of CO from 6g/kWhr to 4g/kWhr. This was estimated to have taken 80% of the emissions reduction development time. The value in local or global emissions terms of having CO at 4g/kWhr rather than 6g/kWhr for the markets these engines are used in is questionable especially when it comes at the expense of doubling the NOx emissions.

Not having to use a catalyst means this technology could be used on various engine types in countries where high sulphur diesel is used.

65% to 85% driving cycle substitution means this technology has applications and economic benefits in many markets.

Ref:1. Calculation of PM see DGI web site

Ref:2. Combustion design option paper see DGI web site

Ref:3. ECE R49 Euro II 1998 standard

Ref:4. Cetane effect See DGI web site and also:
http://www.med.govt.nz/ers/oil_pet/cetane/index.html

Ref:5. Variations to ECE R49 Emissions Standards – See DGI web site for calculations

DGI web address www.dieselgas.co.nz