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New EGR technology retains HD diesel economy with 21st century emissions

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New EGR Technology Retains HD Diesel Economy with 21st Century Emissions

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ABSTRACT

An EGR system for turbocharged (and aftercooled) heavy-duty diesel engines has been demonstrated on a 12 litre 315 kW engine with 4 valves per cylinder head and high pressure injection system.

In the EGR system exhaust gas is tapped off before the turbine, run through a cooler and mixed with the intake air after the compressor and aftercooler. This is done with a minimum of disturbance to the pressure balance across the engine by combining a very efficient venturi-mixer unit with a VGT turbocharger. The venturi-mixer is positioned between the aftercooler and the intake manifold and provides a suction power to the EGR gas.

Optimization of EGR quantity and injection timing reduced the NO_x emission over the European 13-mode test by almost 60% to 2.4 g/kWh. Particulate emissions were 0.107 g/kWh and the BSFC penalty 2.5%. Initial tests demonstrate acceptable transient behaviour when using a dedicated control strategy. The expected EURO4 emission requirements are 3 g/kWh NO_x and 0.10 g/kWh particulates (approximate time of implementation is 2004).

INTRODUCTION

Over the last decades emissions by light duty vehicles have reduced significantly, primarily as a result of the introduction of the three-way catalyst on otto engines. Consequently, the relative contribution of heavy duty vehicles to the atmospheric pollution has become more important. This has resulted in increasingly stricter emissions legislation for these vehicles over the past ten years.

Table 1 presents the current and future European emission requirements for heavy-duty (HD) engines. According to this table the emphasis in Europe is on a further reduction of NO_x and, to a lesser extent, of particulate (PM) emissions. The EURO3 and EURO4 emission values mentioned are projections and, as indicated in Table 1, the test procedure itself will change with the introduction of the EURO3 legislation. Final EURO3 legislation however will be equivalent to the values mentioned in Table 1. There is more uncertainty on the EURO4 levels and their time of introduction. The EURO4 values mentioned in Table 1 are therefore indicative only, but not without ground: similar values have been suggested as a possible long term standard [1,15].

TABLE 1.
European emission standards for HD vehicles (> 3.5 tonne GVW; > 85 kW) - ECE R49 13-mode test procedure

	EURO1	EURO2	EURO3*	EURO4
Effective date	1992	1995	1999	2004
NO_x (g/kWh)	8.0	7.0	< 5.0	< 3.0
PM (g/kWh)	0.36	0.15	< 0.10	< 0.10
HC (g/kWh)	1.1	1.1	0.7	0.5
CO (g/kWh)	4.5	4.0	2.5	1.0

* with introduction of EURO3 legislation the test procedure will change

Furthermore, as shown in Table 2, similar legislative action is expected in the US and Japan. The question then is, how will this affect the engine configuration ?

TABLE 2.
Estimated Emission Targets for HD Trucks at the Start of the 21st Century

	Europe	United States	Japan
Source	- *	Wilson [2]	Joko et al. [3]
Test procedure	ECE R49	US HD Transient	Japan 13-mode
Units	g/kWh	g/bhp.h (g/kWh)	g/kWh
NO_x	2.5	2.0** (2.68)	4.5
PM	0.10	0.1 (0.134)	0.25
HC	0.5	0.5** (0.67)	1.1
CO	1.0	15.5 (20.8)	unknown

* These are the EURO4 target emission levels used in this project

** Or, alternatively, a (NO_x + HC)-limit of 2.4 g/bhp.h

In Europe almost all heavy-duty vehicles use diesel engines, and at present EURO2 diesel engines are appearing on the market [4]. They are the result of the introduction of new fuel injection equipment (FIE) allowing higher injection pressures, sometimes in combination with speed and load dependent injection timing control. Further combustion system optimisation (4V per cylinder head, lower swirl, high pressure FIE, central injector) and electronic fuelling control together with advanced turbocharging will bring the diesel engine close to meeting the EURO3 emission targets, be it with some fuel consumption penalty [5]. In order to achieve emission levels that are substantially below these EURO3 values, other measures need to be taken, i.e. new/additional technology will have to be introduced. Already some time ago a number of these potential key-technologies had been identified [6,7,8].

At that time (beginning of '93) activities were started at TNO, aiming at determining and demonstrating the technologies that would give the diesel engine emission levels similar to the EURO4 values mentioned above while keeping its fuel consumption comparable to that of its EURO2 predecessors.

A literature study soon revealed that the two most promising key technologies were selective catalytic NO_x reduction (SCR DeNOx) and exhaust gas recirculation (EGR). However, it became clear also that the progress in the development of a HC-based DeNOx-catalyst was slower than originally projected due to problems with low NO_x reducing efficiency and catalyst poisoning. It was therefore decided that in this project exhaust gas recirculation would be the key technology for achieving EURO4 targets. (Since then however in-house developments have shown very promising results with a urea-based SCR DeNOx system [16].)

This paper describes the first stage of this project and focuses on hardware development and steady state engine optimization. In a companion paper [9] the activities in control system development and transient engine behaviour optimization are described.

OBJECTIVES

At the start of the project the following set of goals were put forward :

- the design and development of a practical EGR system,
- the development of an engine management control system,
- demonstration of this technology on a multicylinder TCA HD diesel engine.

This demonstrator engine should meet the projected EURO4 R49 emission levels (as mentioned in Table 2), with competitive fuel economy and present-day diesel engine transient behaviour.

For the first stage of this project it was decided that the combustion system optimization efforts would be limited in order to concentrate on the design of the EGR hardware and control system. A HD diesel engine with EURO3 potential would be procured and on this engine the above-mentioned technology would be implemented and target emission levels demonstrated.

EGR STRATEGY FOR LOWEST EMISSIONS WITH HD DIESEL ENGINES

Exhaust gas recirculation is essentially a NO_x-reducing technology. In engines NO_x-formation is largely due to a reaction between N_2 and O_2 at temperatures above 2000 K, with NO_x-formation rate increasing (almost exponentially) with temperature. Hence in diesel engines most NO_x forms in or around the flame of the burning spray because temperatures are highest there. Recirculation of exhaust gas to the cylinder reduces these temperatures and therefore also the amount of NO_x in the exhaust.

Early investigations showed that EGR at the same time increases PM emissions [7,10] especially at high load conditions where air/fuel ratio (AFR) is lowest. These investigations also indicated that the PM increase could be minimised by keeping AFR constant. This explains why EGR is quite commonly applied to light-duty engines. In these engines EGR is applied under light and medium load conditions only, i.e. where AFR is large and PM increase due to EGR is negligible. This strategy is effective because during the official light-duty test cycles the engines run the majority of the time at light or medium load conditions.

This is different for medium or heavy duty engines. As mentioned before, in Europe, emissions characteristics of HD engines are determined according to the ECE R49 (13-mode) test procedure. This test procedure attributes a large weighting factor to the maximum torque and rated power working points (Figure 1). For an EGR system to be effective it should enable sufficient exhaust gas re-circulation at high load in the intermediate to rated speed range.

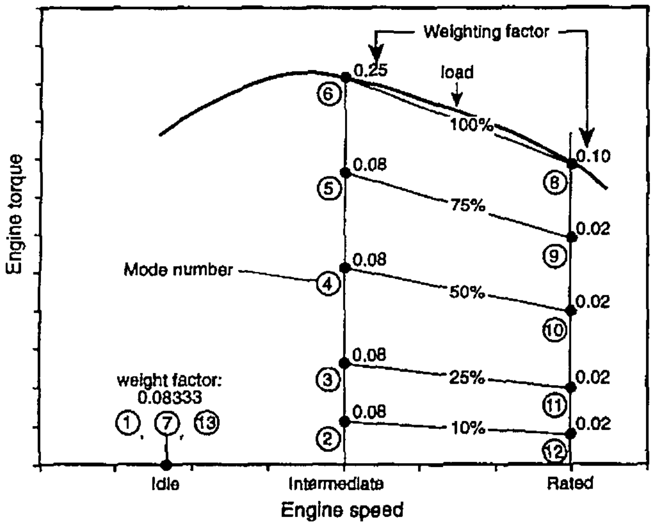


Figure 1: European 13-mode test cycle (ECE R49, EEC/88/77, EEC/91/542).

Compared with the EURO3 levels, the EURO4 levels mentioned in Table 1 require a 50% decrease in NO_x at constant PM. In a pilot study on a 5.7 L 6-cylinder ISUZU diesel engine a 50% NO_x reduction was achieved with 10% EGR admission at full load in the maximum torque to rated speed range. EGR admission is defined as mass fraction of exhaust gas in the intake manifold flow and calculated from:

$$EGR\ (\%) = \frac{CO_{2,intake} - CO_{2,ambient}}{CO_{2,exhaust}} * 100$$

With a conventional TCA HD diesel engine typically around 60% of the NO_x in the ECE R49 test comes from the full load points. With this in mind and looking at the results from the pilot study it was decided that the EGR system should be able to realize 15% EGR or more in these points.

Although the ECE R49 test procedure will be replaced by a new test procedure in the near future, it is not likely that this will significantly influence the above mentioned requirements. In the US, similar EGR-ratio's will be necessary in the rated speed, above 60% load range, if a substantial reduction in NO_x through EGR is to be achieved in the US HD Transient test procedure [6,12].

Finally, when applying EGR at full load it is important that the original air mass flow through the engine is maintained at its baseline (no EGR) level. Thus the original torque curve can be maintained without excessive smoke penalty. This means however that the system lay-out should be such that it realizes "additional" EGR rather than "replacement" EGR.

EGR SYSTEM CONFIGURATION

In the past several ways of exhaust gas recirculation have been identified. They can be classified into two groups:

a) "internal" EGR: via exhaust valve actuation or a special valve, combustion products are prevented from leaving the cylinder during the exhaust stroke or fed back into the cylinder during the intake stroke.
b) "external" EGR: exhaust gases are fed back to the inlet system of the engine via an external route.

For turbocharged and aftercooled engines there are two important options for this external route :

- "ATBC": exhaust gas from after the turbine is fed back to a place before the compressor,
- "BTAC": exhaust gas from before the turbine is fed back to the intake manifold after the compressor (and aftercooler).

Table 3 summarizes the challenges that oppose EGR implementation on HD TCA diesel engines. After analysing each of these challenges we selected a BTAC design for cooled exhaust gas recirculation.

TABLE 3.
Challenges to EGR Implementation.

Challenge	Remedy
Fouling	BTAC
TC-size limitation	BTAC + EGR cooling
Negative pressure diff. across engine	VGT + venturi-mixer
Pressure loss in EGR circuit	New venturi-mixer design
Accurate EGR metering	EGR control system
"Additional" EGR	VGT

The BTAC option was selected because it has the considerable advantage that the compressor and aftercooler are not exposed to particulates, hydrocarbons and sulphate present in the exhaust gases. Thus problems with fouling and corrosion are limited, and the mass flows through the turbocharger remain more or less unchanged. The decision to cool the EGR was taken because it reduces the volume of exhaust gas flowing into the intake manifold and therefore limits the boost pressure increase necessary for realising the "additional" EGR. Cooling also limits the temperature rise with EGR of the trapped gases, thus assisting NO_x reduction.

Another important aspect when applying EGR to a heavy-duty engine is that over an important part of the engine operating range, the exhaust manifold pressure is lower than the inlet manifold pressure. This means that exhaust gas does not flow by itself from exhaust manifold to intake manifold.

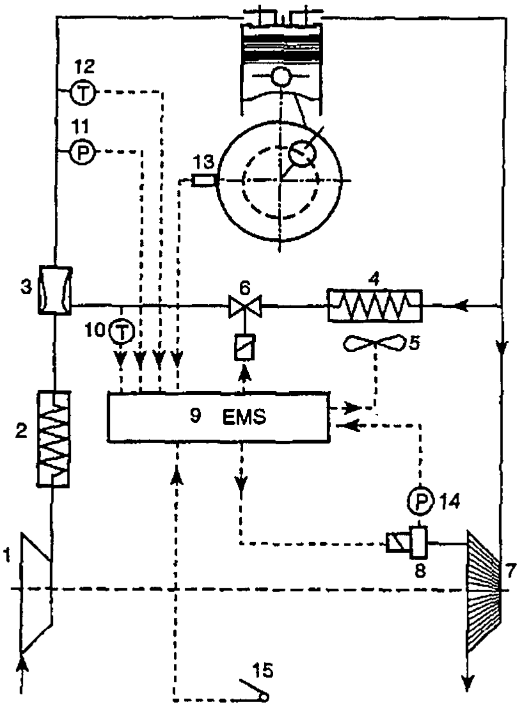
This problem can be overcome in a simple manner by introducing a back pressure valve in the exhaust system (or a valve in the intake system), and for small EGR ratios in a few selected operating points this might prove to be acceptable [12]. This approach will however penalize engine efficiency, since positive pumping power is lost.

A different approach was followed by combining a venturi in the intake manifold with a variable geometry turbine (VGT) turbocharger. A schematic of the EGR system is shown in Figure 2. In this system exhaust gas is taken from the exhaust manifold before the turbocharger. The exhaust gas is then led through an EGR cooler and an EGR control valve to a venturi mixer unit, where it is mixed with the inlet air. As the VGT is closed both the power taken up by the turbine and the pressure before the turbine increase. As soon as this pre-turbine pressure exceeds the pressure at the venturi throat section an EGR flow builds up. The EGR control valve gives the possibility to modulate air flow and EGR admission independently. It can also be used to shut off EGR in certain parts of the engine operating field.

BASELINE ENGINE/FUEL SPECIFICATION

Table 4 gives the main specifications of the 2 engines used in this project. The specifications mentioned are prior to implementation of EGR.

Engine 2V was used for preliminary testing (and confirmation of the potential) of the EGR system design. It is a EURO2 production engine with 2 valves/cylinder and an inclined injector. The combustion system is swirl-supported with reentrant piston bowl geometry. In standard production build it has an in-line fuel injection pump with no timing modulation and a fixed geometry turbocharger. Charge cooling is provided by an air-to-air cooler. In preparation for the EGR work a Bosch Hubschieber (H-series) fuel injection pump and variable geometry turbocharger were fitted to this engine. The pump enables injection timing to be varied through the position of a sliding sleeve which controls the spill cut-off. Although injection nozzle geometry had been re-optimised for this pump, injection pressures were rather similar to those of the production pump.



1. Compressor
2. Aftercooler
3. EGR venturi mixer
4. EGR cooler
5. Ventilator EGR cooler
6. EGR control valve
7. Variable Geometry Turbine (VGT)
8. VGT actuator
9. Engine management system
10. EGR temperature sensor
11. Air manifold pressure sensor
12. Air manifold temperature sensor
13. Engine speed sensor
14. VGT actuator pressure sensor
15. Accelerator pedal position

Figure 2: Schematic of EGR system with Engine Management System

To demonstrate the potential of the EGR system to achieve EURO4 emission levels, a different engine with EURO3 emissions potential was used. This engine has the same bore and stroke as the 2V engine but has 4 valves per cylinder and a vertical injector. In original build this engine combines a conventional turbocharger with a unit pump fuel injection system. As for the 2V engine, combustion is swirl-supported and piston bowl geometry is reentrant. Again a VGT was fitted to this engine before testing with EGR. Other changes are discussed in the following sections.

In all of the tests on engine 2V and in the development work on the 4V engine a conventional diesel fuel (0.2% mass S) was used. In Europe a maximum limit of 0.05% mass S will be mandatory from October 1996 onwards. Hence, for the 13-mode validation on the 4V engine a conventional but low sulphur fuel (< 0.05% S) was used.

TABLE 4.
Baseline/original engine specifications prior to EGR.

	2V ENGINE		4V ENGINE	
	Production build	Changes prior to EGR	Original Build	Changes prior to EGR
Bore / Stroke (mm)	130 /146		130 / 146	
No. cylinders	6		6	
Displacement (litre)	11.63		11.63	
Compression ratio (nom.)	16		16	15.3
FIE type	Bosch RP48	Bosch RP39	Bosch PLD	
	In-line pump	Hubschieber	Mk 1	
Turbocharger geometry	Fixed	VGT	Fixed	VGT
Charge cooling	air-to-air		air-to-air	
Max. power	315 kW at 2000 rpm		315 kW at 2000rpm	
Max. torque	1740 Nm at 1500 rpm		1740 Nm at 1500 rpm	

EGR SYSTEM DESIGN

COMBUSTION SYSTEM MODIFICATION - For engine 2V, changes to the combustion system were limited to the fuel injection equipment i.e. the Hubschieber pump and injector nozzles. Changes to the combustion system were also deliberately limited on the 4V engine. However, attention was given to the implications of adding EGR to the normal air charging. A revised compression ratio was determined which would allow the required charging to be achieved within the cylinder pressure limit. Although reducing the compression ratio would conventionally risk some compromises in part-load emissions and combustion noise as well as cold starting, it was felt that the provision of an EGR circuit would ultimately offer some scope to address these effects. In addition it was considered that, since the reduction in compression ratio would be achieved by increasing the volume of the piston bowl, the air utilisation could be improved to benefit the particulate emissions.

Several alternative bowl designs were considered. A deeper version of the standard bowl was selected for testing since this would result in the least changes to swirl in the bowl at the end of compression. In-cylinder charge densities were predicted for various EGR levels at different engine operating points for the revised compression ratio. Fuel spray characteristics were predicted for a range of nozzles using data from previous work [13]. A selection was made that was judged to give similar spray characteristics to that of the original build.

VENTURI-MIXER DESIGN - The following requirements can be defined for a venturi-mixer for EGR :

- a minimum of pressure loss across the venturi-mixer
- a maximum of suction power to the admitted gas
- a fast, homogeneous mixing performance.

The mixing of the EGR flow with the inlet air is very important for getting a homogeneous mixture and a good EGR gas distribution over the cylinders. TNO has considerable experience with mixing gaseous fuels and EGR to inlet air, because of its long standing experience with gaseous fuelled engines. This technology was developed further, leading to a new venturi design, for which a patent application has been filed.

TURBOCHARGER SELECTION - Together with the turbocharger manufacturer a candidate VGT turbocharger was selected. For this, simple matching calculations were done based on projected EGR and air mass flows and on data from the manufacturer. The VGT selected uses a Varable Nozzle Turbine with adjustable guide vanes to vary the power output of the turbine. It was estimated that this design would allow sufficient range of turbine operation to support the required compressor operation with EGR.

EGR COOLER DESIGN - For the demonstrator engine air was chosen as the coolant medium (in this way EGR temperature could easily be varied). The design calculations were carried out within TNO with a dedicated computer program. As a first step in this design, demands and/or limitations on cooler size, corrosion resistance and pressure drop across the cooler were determined.

As with all losses in the EGR circuit, the pressure loss across the cooler necessitates higher pre-turbine pressures for a given EGR flow. This in turn increases fuel consumption. With this in mind, the design target for the pressure loss across the cooler was set at 5 kPa at the maximum EGR flow (in "fouled condition"). Similarly a target was set for the EGR temperature upon leaving the cooler. It is clear that NO_x emission would benefit from an as low as possible EGR temperature. On the other side of the trade-off are however increasing cooler size and fouling and corrosion due to condensation of sulphate and possibly water. In the end, cooler-out temperature was primarily chosen such that condensation of corrosive gases would be prevented. Finally, when designing the heat exchanger the effect of fouling had to be taken into account. Rather than taking data from the literature it was decided to set up a separate study in which the influence of fouling (e.g. soot layer build-up) on heat transfer and pressure loss was experimentally determined for different design choices. It appears that with the right choice for the hydraulic diameter of the EGR passages, the thickness of the soot layer stabilizes at an acceptable level.

EGR SYSTEM DESIGN CALCULATIONS - The system design calculations made use of a mean value model which predicts steady-state and transient behaviour of heavy-duty diesel engines [14]. Originally this engine model was developed to aid the design of engine management control strategies and the evaluation of different engine concepts. In this project the mean value model was first validated using data from preliminary tests on the 2V engine without EGR. The engine was then tested with a first prototype EGR system that used a gas engine type venturi-mixer. A generic description of the venturi-mixer (and EGR system) was added to the mean value model. With this model then, the effects of venturi-mixer geometry (e.g. throat size and EGR admission area) and of venturi efficiency (e.g. pressure recovery) on turbocharger operation and engine performance were evaluated. From the results a venturi-mixer design was selected for the 4-valve engine. Similarly, specifications were examined for the EGR control valve [9].

STEADY STATE TESTS WITH EGR

PREPARATION OF ENGINE 4V FOR EGR - Preparation for EGR meant running the 4V engine at the boost pressure levels expected with EGR and defining appropriate hardware and settings such that EURO4 emission levels are likely to be achieved when EGR is applied.

First, the engine was prepared with the low compression ratio pistons, the VGT and the preferred nozzle hole size determined from the spray growth calculations. Three engine operating points were selected from the target torque curve for testing, corresponding to the maximum and minimum full-load speeds and maximum torque speed. Using a selected timing and boost pressure for each key point, injector intrusion was varied to establish an optimum for the new piston bowl.

Then, a limited optimization of the combustion system was undertaken using fractional factorial testing. The timing and boost settings were combined with nozzle hole size, fuel cam static timing and engine speed (associated with the 3 key points) to construct an experiment comprising 5 factors at 3 levels. Analytical models were then derived, which describe the measured engine responses (e.g. NO_x, PM and BSFC) to the various factors. Sequential quadratic programming (SQP) methods were then used to determine the engine settings that enable lowest BSFC for given weighted emissions and/or maximum smoke levels. Projecting the responses of NO_x and particulates to EGR as measured on the 2V engine to the 4V engine, suggested the need for pre-EGR emission targets for this engine of 5 g/kWh NO_x and 0.05 to 0.07 g/kWh PM. Thus the fuel injection hardware was selected to enable these targets based on the assumption of typical contributions from the maximum power and maximum torque points.

Additional tests at part-load however indicated a higher smoke and therefore particulate level than expected. To cater for this a higher intake swirl was realized by mounting the intake manifold in the reverse direction to that normally employed. After re-optimizing injector intrusion, significant reductions in the part-load PM were found at the expense of a moderate increase in NO_x. The new arrangement of the manifolding also had a practical advantage because it created additional space to install the EGR venturi-mixer later.

Figures 3 and 4 indicate the relative status of the engine at this stage prior to introducing EGR. The NO_x/BSFC and NO_x/PM trade-offs are shown for the full-load and 50% load conditions at maximum torque speed. The solid lines show the responses to injection timing for the baseline engine build and the low compression ratio engine with VGT, but prior to revising inlet swirl and injector intrusion. The BSFC penalty associated with the use of injection retard is evident. PM reduces with retard due to the increase in injection pressure which results from the rising rate fuel cam profile. The PM response with the low compression ratio build at 50% load differs from that of the baseline engine and may reflect a bad smoke reading used in the PM prediction. The single test point measured with revised swirl and intrusion shows the reduction in part-load PM achieved. The reduction is less at full-load where injection pressures are higher. The remaining scatter plots derived from EGR tests with the 2V engine suggested how the 4V engine could respond to EGR.

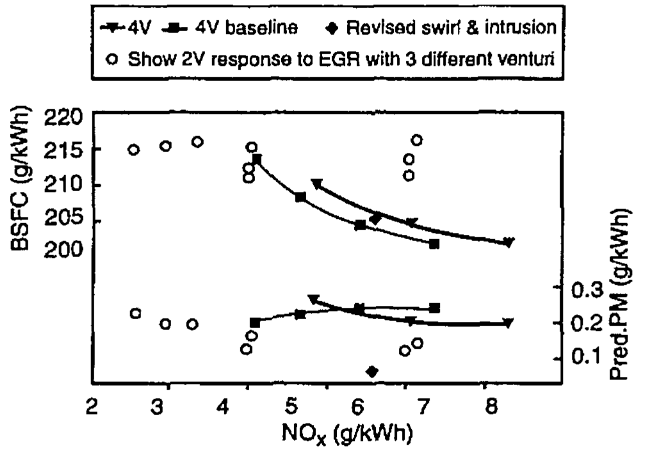


Figure 3: NO_x versus BSFC and PM. 4V engine prior to EGR: 1500 rpm 50% load.

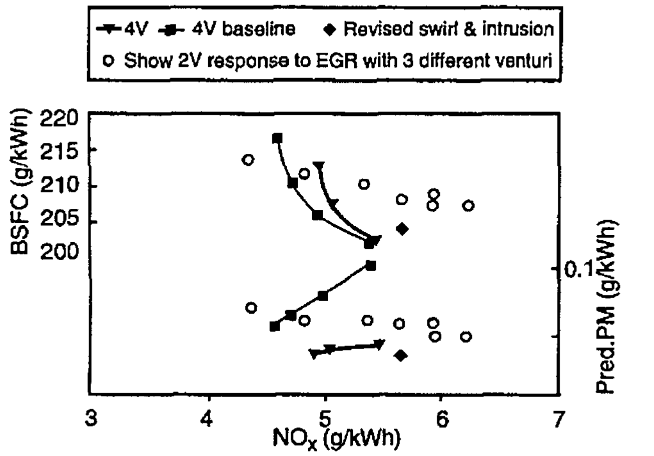


Figure 4: NO_x versus BSFC and PM. 4V engine prior to EGR: 1500 rpm 100% load.

STEADY STATE TESTS WITH EGR - First, tests were conducted to check EGR admission and mixing characteristics with the new venturi-mixer design. These tests showed that the target of 15% EGR could be achieved in most of the engine operating range.

Figures 5 and 6 show some characteristics of the venturi-mixer at 1500 rpm and 10% EGR both for 50% and 100% load. Figure 5 shows the pressure level at the venturi throat and the venturi exit relative to that at the venturi entrance. The difference between pressure at venturi entrance and throat section is the so-called suction pressure. Figure 6 shows the pressure loss across the venturi and its suction pressure versus mass flow. The measured suction pressures are in line with calculations using Bernouilli's equation. Obviously, this venturi has some interesting features. It realized a suction pressure of up to 20 kPa and pressure recovery is about 60%. This corresponds to a pressure loss of only about 2% of the boost pressure. Mixing of EGR with the inlet air was assessed and found to be satisfactory.

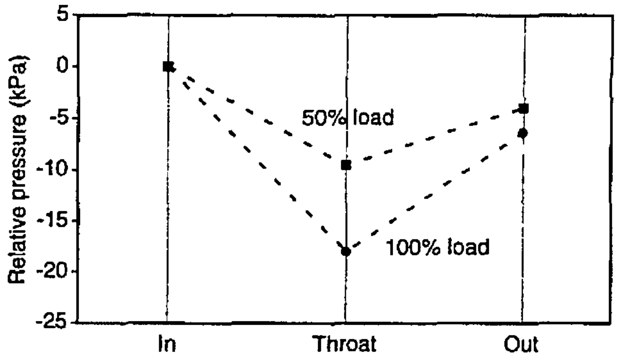


Figure 5: Pressure across EGR venturi at 1500 rpm, 10% EGR.

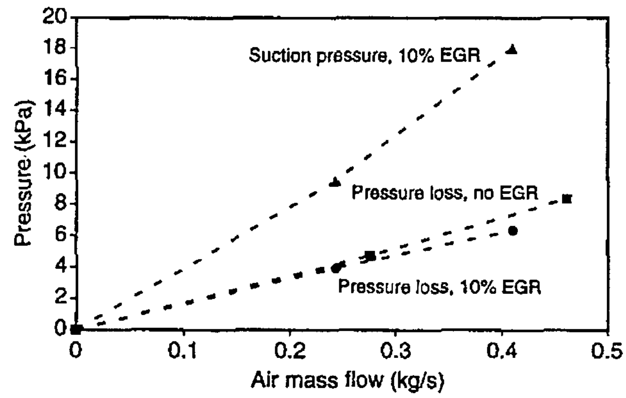
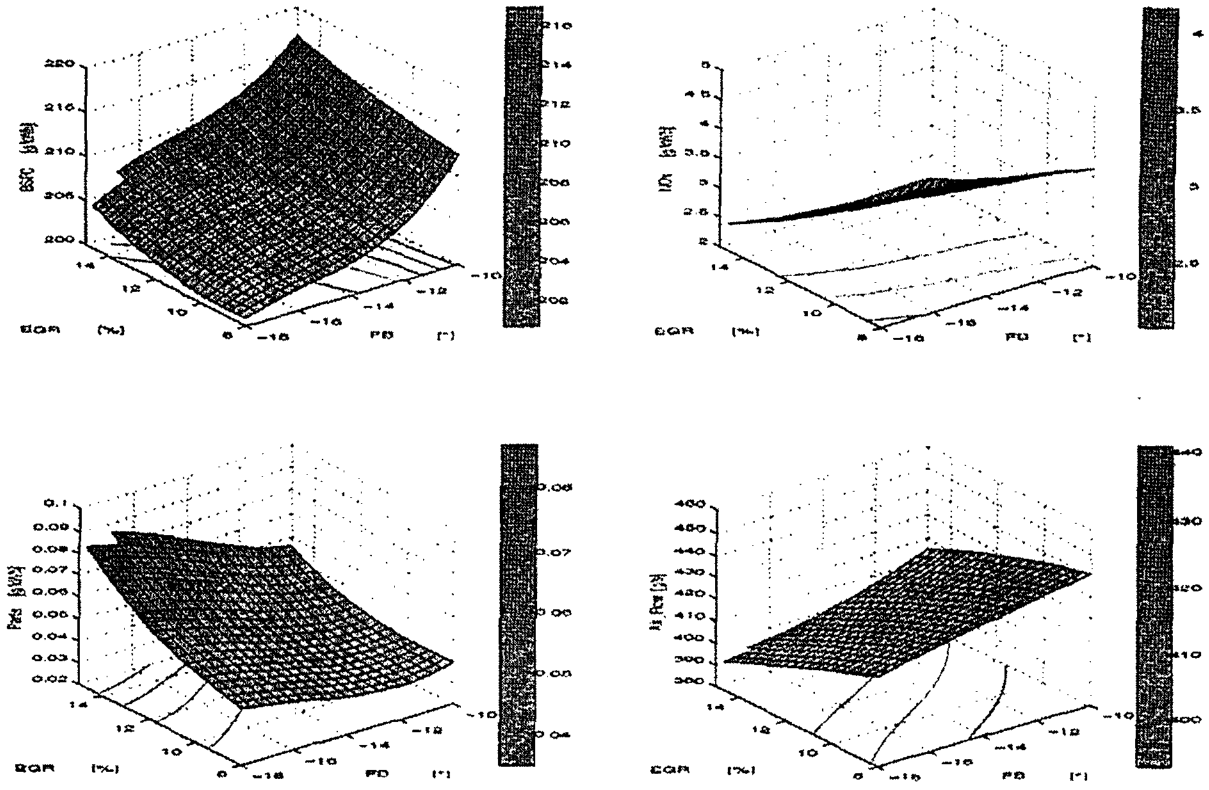
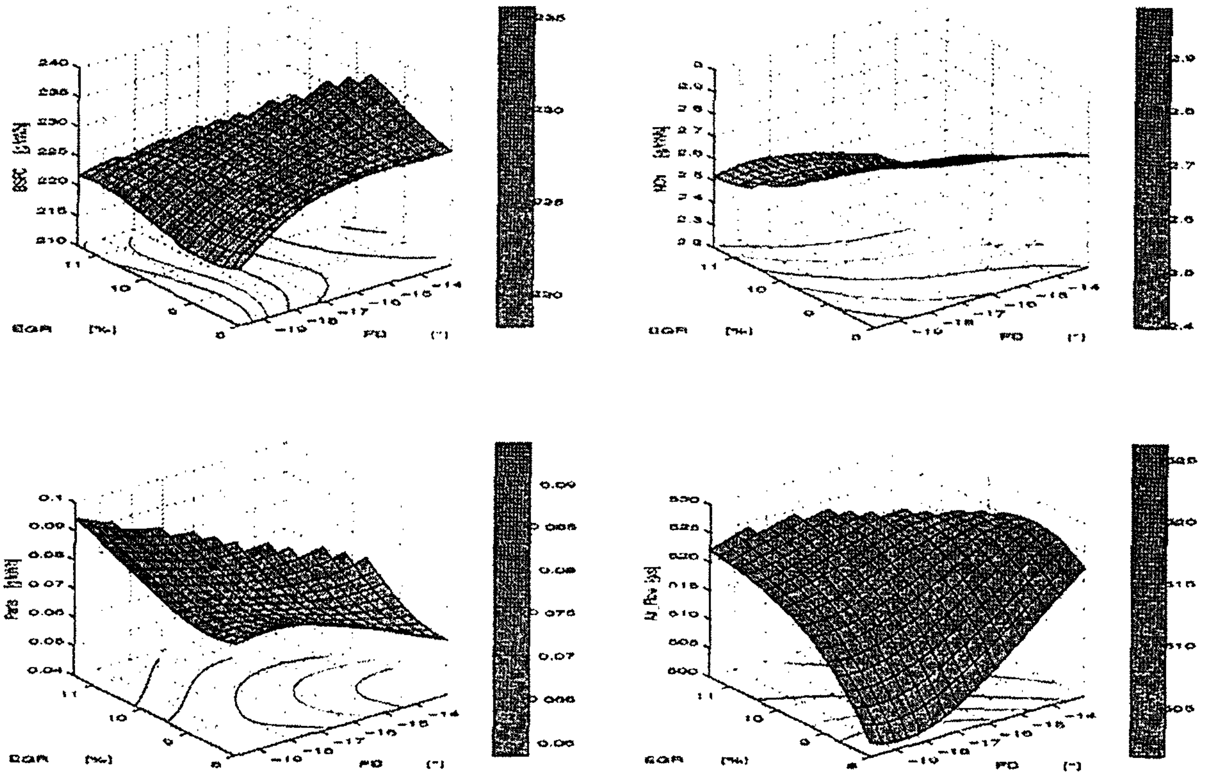


Figure 6: EGR venturi-mixer characteristics (1500 rpm, 10% EGR).

A set of experiments was defined and conducted to enable analytical models of the main engine responses to EGR, AFR and injection timing to be constructed. When determining engine settings, it was found that optimum results were generally achieved with the EGR valves fully open. Figures 7 and 8 show examples of the responses to EGR and injection timing (FB = solenoid-triggered start of delivery at the injection pump) at full-load. The data shown were obtained with the EGR valves fully open. Since EGR rate is then controlled through turbine geometry, in these figures airflow can be considered as a response. Of special note is the fact that airflow increased with EGR at 2000 rpm but reduced at 1500 rpm. This is partly due to the working point to which the turbine is driven in order to support EGR. Figure 9 shows how intake and exhaust manifold pressure and EGR change with VGT guide vane position (VGT position). As the VGT closes exhaust manifold pressure rises faster than intake manifold pressure. In combination with the suction power of the venturi, at some point this is sufficient to create an EGR flow. The different pressure curves at these two operating points reflect the way the turbocharger is matched across the engine speed range (at full load). At 1500 rpm, even with the assistance of the venturi the turbine must be driven to a relatively low efficiency before EGR flows. Once EGR flows the turbine is further deprived of energy and the compressor delivery falls away.



igure 7: 4V engine with EGR; response to EGR and injection timing (1500 rpm, 100% load).



igure 8: 4V engine with EGR; response to EGR and injection timing (2000 rpm, 100% load).

At 2000 rpm however, the turbine modulation required to achieve increasing EGR flows seems to favour high turbine efficiencies, such that compressor delivery is maintained or even increased with EGR.

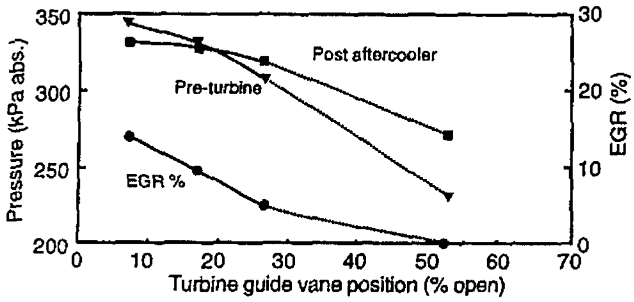


Figure 9a: Change of EGR% and exhaust and inlet pressure with turbine guide vane position. 4V engine: 1500 rev/min 100% load.

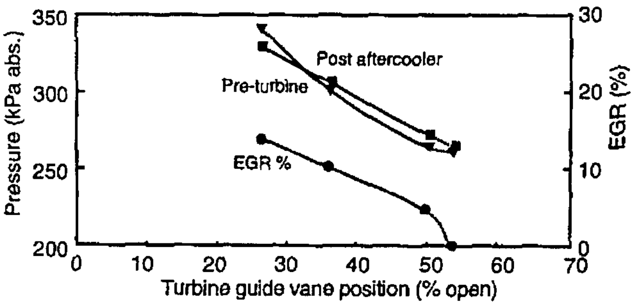


Figure 9b: Change of EGR% and exhaust and inlet pressure with turbine guide vane position. 4V engine: 2000 rev/min 100% load.

The use of fully-open EGR valves suggested a simplification of the control strategy and this was supported by tests at other engine operating points. This provided a simpler experiment for optimization since only turbine geometry and injection timing needed to be varied. Additional points were tested to provide more data for 13-mode optimization. From this data, EGR and timing strategies versus engine speed and load began to emerge which enabled settings at other operating points to be interpolated or extrapolated. Single measurements were made at these settings at remaining points in the 13-mode test such that final optimization (based on a combination of mapped operating points and single setting measurements) could be conducted.

Figures 10 and 11 show examples of the NO_x/BSFC and NO_x/PM trade-offs at one engine operating point. The solid line in bold shows the trends with increasing EGR (0-15%) established during the initial venturi testing. The similarity of the 0% EGR point with the single measurement made prior to EGR is an indication of good pressure recovery across the venturi-mixer. Additional lines have been constructed from the response models to give an indication of the effect of injection timing and AFR at various EGR levels. In addition, a single point is included taken from the final EURO4-targeted 13-mode test (discussed below). The position of this point indicates a well-targeted selection by the optimization routines.

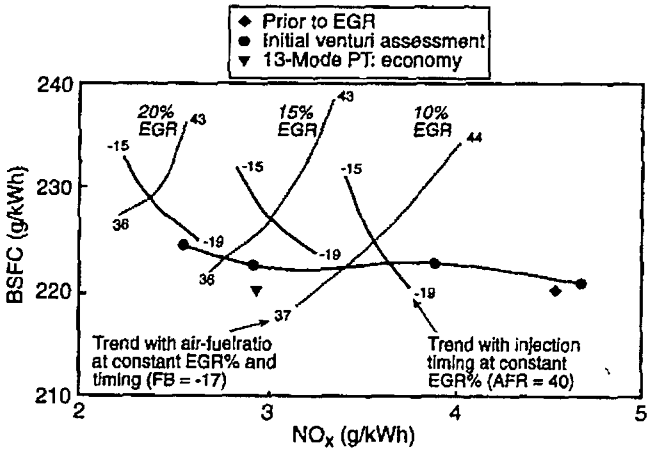


Figure 10: Effect of timing, EGR% and air-fuel ratio on NO_x/BSFC trade-offs. 4V engine: 2000 RPM 50% load.

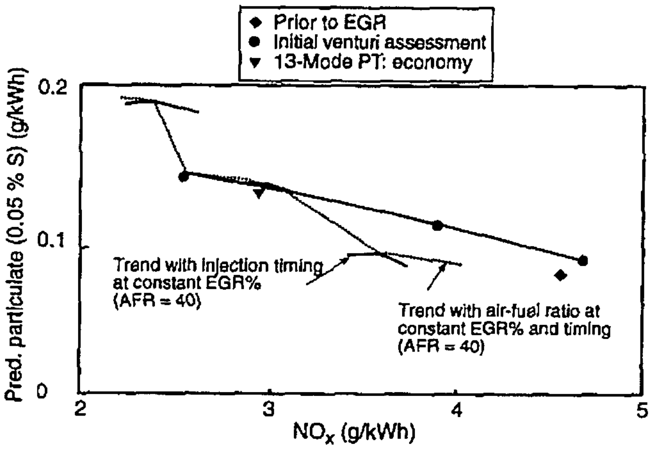


Figure 11: Effect of timing, EGR% and air-fuel ratio on NO_x/PM trade-offs. 4V engine: 2000 RPM 50% load.

TABLE 5.
Summary of ECE R49 test results (g/kWh).

	EGR	NO_x	PM (1)	PM (2)	HC	CO	BSFC
Reference EURO2	No	6.92	100	0.112	0.23	0.68	212.5
Baseline (3)	No	5.73	63	-	-	-	215
EURO 4 Low NO_x	Yes	2.42	82	0.107	0.2	0.6	220
EURO 4 Economy	Yes	2.88	88	-	0.18	0.58	216

(1) predicted PM as percentage of EURO2 Reference engine
(2) measured (NOVA Meßtechnik micro dilution tunnel)
(3) baseline (prior to EGR); values estimated from R49 test modes 4,6,8 and 10

13-mode tests were conducted with EGR and timing strategies, selected for EURO4 NO_x. The results are shown in Table 5 as "EURO 4 Low NO_x". Both NO_x and BSFC met the test targets but the PM exceeded the target by 7%. It was not judged practical in this test programme to reduce PM by additional optimization of EGR and timing. However, an additional EGR/timing strategy was tested aimed at relaxing the NO_x target to 3.0 g/kWh in the interests of improved economy (Table 5 "EURO4 Economy"). This test was also used to test the control strategies which had been developed during the programme [9] and the test was run under full ECM control.

EGR CONTROL SYSTEM

The EGR control system set-up for the 4V engine is shown in Figure 2. The EGR control system receives signals for engine speed, intake manifold pressure and temperature, gas pedal position, EGR temperature and VGT actuator pressure.

EGR quantity control takes place by controlling VGT guide vane position and the EGR control valve opening. The guide vane position is closed-loop controlled on the VGT actuator pressure. The EGR quantity is set according to engine maps for VGT actuator pressure and EGR control valve position, both (primarily) as a function of fuel flow and engine speed. EGR temperature is closed-loop controlled with a temperature sensor at the venturi inlet and a variable speed fan in front of the EGR cooler.

The EGR control system proved itself on the demonstrator engine. Actual EGR percentages over the R49 test were within 1% of the desired EGR percentages. For production engines, however, the addition of a closed-loop module with a good EGR quantity sensor is essential to accomodate for engine wear, fouling and production tolerances. A number of mass flow measurement principles for EGR flow are presently being evaluated.

PRELIMINARY TRANSIENT TESTING RESULTS

The rate at which a turbocharged engine can adjust to a sudden increase in demand is very dependent on the response of the turbocharger. Fuelling must be modulated such that the torque response best reflects the demand without unacceptable smoke (during transients the AFR is normally lower than during steady state operation leading to increased smoke). With EGR, the smoke tends to further increase. Also, applying EGR influences both the energy available at the turbine and the required compressor delivery. Preliminary work therefore was focused on the aspects of driveability and transient smoke. Other emissions were not measured.

A number of characteristic engine transients, simulating engine response in high and low gears, were selected and tested on a transient dynamometer. Performance in these tests was defined in terms of torque rise time and smoke opacity. Using the mean value model a transient control strategy was developed. For the reasons mentioned above, a strategy was selected in which EGR was withheld during a certain part of the transient. During that part of the transient the VGT position was closed-loop controlled on intake manifold pressure. With EGR the VGT position was closed-loop controlled on VGT actuator pressure. Figure 12 shows the load acceptance test results at 1500 rpm both for the engine with and without EGR. Differences in torque response and smoke opacity are small. For the situation without EGR an open-loop control strategy was used for the VGT position. This was felt to be acceptable as earlier measurements on the 2V (EURO2) engine demonstrated that the transient response with an open-loop controlled VGT was equivalent to or better than that with a fixed geometry standard turbocharger. According to Figure 12 the 90% torque rise time and the maximum smoke opacity are respectively 3 seconds and 10% opacity. These values are indeed typical for EURO2 engines. In the companion paper [9] more extensive transient test results are shown, including a strategy with continuous EGR during transients.

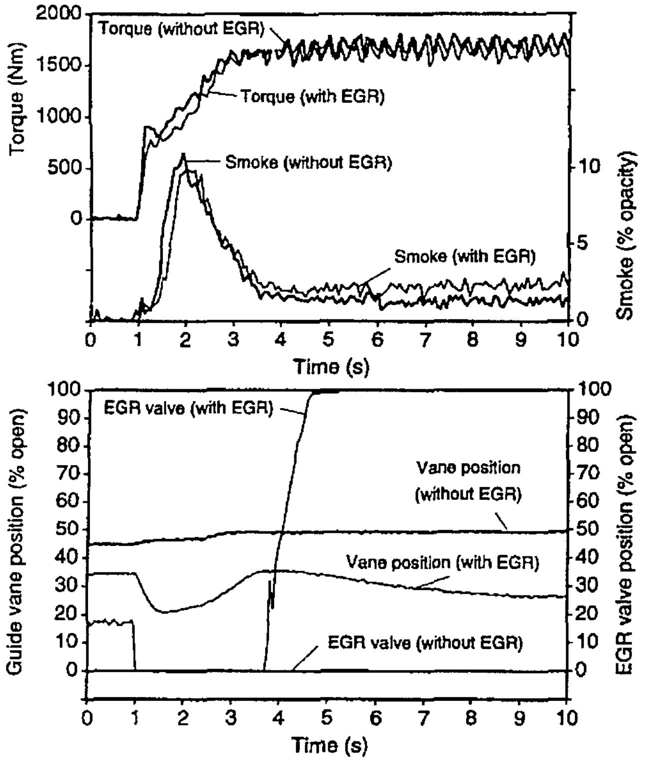


Figure 12: 4V engine: transient response with and without EGR.

DISCUSSION

In a recent publication by Havenith et al. [15] the potential of EGR for achieving EURO3 emission levels starting from a EURO2 baseline engine was demonstrated. For this they also selected a BTAC cooled EGR system design in combination with a VGT turbocharger; no venturi-mixer was used. Emission levels over the ECE R49 cycle were 4.8 g/kWh NO_x and 0.09 g/kWh PM with a 2% BSFC penalty. When trying to further reduce NO_x emissions however, PM and BSFC were found to increase rapidly.

The work reported here has demonstrated the potential for EGR technology to go one step further, i.e. to further reduce HD diesel engine emissions from a EURO3 level to a EURO4 level while keeping fuel consumption almost constant. Also, a control strategy was developed that took the engine through simple transients with baseline smoke and transient performance.

As indicated before, in this part of the project, the baseline combustion system was designed to meet EURO3-like emissions without applying EGR. There is therefore scope for improving the BSFC/PM trade-off while maintaining the 2.5 g/kWh NO_x level. This could be done by optimizing the combustion system for running with EGR, i.e. by changing injection rate shape, intake swirl ratio and/or mean effective injection pressure.

Although the EGR system has demonstrated encouraging potential there are aspects which warrant further investigation. The ability to modulate EGR and air-fuel ratio throughout the engine operating range is clearly very dependent on the turbocharger match. Also the type of geometry used to modulate the turbine appears to be important. In particular the reducing airflow characteristic at maximum torque when EGR is applied would give concerns with regard to the proximity to the compressor surge line in production.

One advantage of the venturi-mixer is the ability to limit the range over which the turbine and compressor must be operated. In this project a venturi-mixer configuration was selected to achieve a compromise between operation at maximum torque and maximum power. With a view to other engine torque curves and also to possible test cycles which place more emphasis on low speed torque, smaller venturi throat sizes will be examined in the future.

Of course, running an engine with EGR raises questions on wear and fouling, as they effect durability of the engine. The 4V engine has been running on EGR for over 300 hours without problems of either source. Cold starting behaviour and white smoke emissions have not yet been investigated.

CONCLUSIONS

1. Future emission legislation will require a further substantial reduction in NO_x levels by the beginning of the next century, necessitating new technology such as EGR or aftertreatment.

2. EGR hardware and control technology has been developed that enables such emission levels with a limited fuel consumption penalty. Characteristic components of the BTAC EGR system design are a combination of a VTG turbocharger with a new venturi-mixer design and an EGR cooler.

3. This technology has been demonstrated on a 12 liter 6 cylinder TCA HD diesel engine with 4 valves per cylinder and electronically controlled FIE. With the EGR system a strategy of "additional" EGR was adopted throughout the engine working range for engine speeds above 1000 rpm, allowing EGR ratios of up to 20% at low load and up to 15% at high load. In the absence of a direct EGR sensor, accurate steady-state EGR quantity metering was achieved through control of VGT position.

4. Over the ECE R49 cycle emissions were 2.5 g/kWh NO_x and 0.107 g/kWh PM. Fuel consumption was 220 g/kWh. This technology is expected to be also effective in reducing NO_x over other test cycles (outside Europe).

5. Using a transient model of the demonstrator engine a dedicated control system was developed. With this control system, performance of the engine in a selected number of engine transients was comparable to that of new EURO2 diesel engines presently coming on the market. This was achieved by withholding EGR admission during the initial part of the transients.

6. Future combustion system optimization and EGR system matching are expected to improve the BSFC/PM trade-off while keeping the low NO_x emission levels presented above.

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