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Load transient between conventional diesel operation and low-temperature combustion

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Abstract

The operation of diesel low-temperature combustion engines is currently limited to low-load and medium-load conditions. Mode transitions between diesel low-temperature combustion and conventional diesel operation and between conventional diesel operation and diesel low-temperature combustion are therefore necessary to meet typical legislated driving-cycle load requirements, e.g. those of the New European Driving Cycle. Owing to the markedly different response timescales of the engine's turbocharger, exhaust gas recirculation and fuelling systems, these combustion mode transitions are typically characterised by increased pollutant emissions. In the present paper, the transition from conventional diesel operation to diesel low-temperature combustion in a decreasing-load transient is considered. The results of an experimental study on a 0.51 l single-cylinder high-speed diesel engine are reported in a series of steady-state 'pseudo-transient' operating conditions, each pseudo-transient test point being representative of an individual cycle condition from within a mode transition as predicted by the combination of real-world transient test data (for fuelling and load) and one-dimensional transient simulations (for intake manifold pressure and exhaust gas recirculation rate). These test conditions are then established on the engine using independently controllable exhaust gas recirculation and boost systems. The results show for the first time that the intermediate cycle conditions encountered during combustion mode change driven by the load transient pose a significant operating challenge, particularly with respect to control of carbon monoxide, total hydrocarbon and smoke emissions. A split-fuel-injection strategy is found to be effective in mitigating the negative effects of the mode change on smoke emissions without significantly increasing oxides of nitrogen or decreasing fuel economy; however, unburned hydrocarbon emissions are increased. Additional experimental testing was also conducted at selected intermediate cycles to understand the sensitivity of key fuel injection parameters with the split-injection strategy on engine performance and emissions.

Keywords

Diesel, exhaust gas recirculation, low-temperature combustion, transient, Extra-Urban Driving Cycle, New European Driving Cycle, dual-mode combustion, split injection

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Introduction

Reducing the engine emissions, especially of oxides of nitrogen (NO_x) and particulate matter (PM), remains a major challenge for the automotive industry. Reductions in engine-out emissions would enable simplification of exhaust after-treatment requirement and hence reduce vehicle cost. One strategy that has been demonstrated to reduce emissions from diesel engines substantially is low-temperature combustion (LTC), where the combustion temperature is kept so low that the formation of both NO_x and soot precursors are essentially near zero. One way to achieve this is to use very high levels of exhaust gas recirculation (EGR), at times in excess of 60%. This reduces local flame temperature and increases ignition delay, increasing the mixing of the fuel with the oxidiser, therefore

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effectively minimising formation of both NO_x and PM.[1–5] One of the main drawbacks of this strategy is that high levels of EGR reduce the amount of oxygen available, which limits the amount of fuel that can be burned and, correspondingly, the torque that can be generated. Despite significant research, high-EGR LTC operation remains limited to low-load and mid-load operating regimes. To achieve higher loads, the engine would need to revert to conventional diesel operation. In a passenger car application in real-world driving, this would lead to frequent transitions between conventional diesel and high-EGR LTC modes. This paper investigates the impacts of one such transition on engine performance and emissions.

The requirement for frequent mode switching represents a potential barrier to the adoption of LTC operating strategies. In a typical legislative driving cycle, in general, the transients are major contributors to the total cycle pollutant emissions. For example, one recent work on a conventional diesel engine reported that 53% of soot emissions and 16% of NO_x emissions during a particular city driving condition were the result of transient 'spikes'.[6] Typically, poor transient control in conventional diesel operation is due to the markedly different response timescales of the engine's air (turbocharger and EGR) and fuelling systems.[6–8] Adding the complexity of switching between combustion modes that operate with substantially different injection timings, intake pressures and most notably EGR rates represents an even greater technical challenge.

Combustion mode switching between high-EGR LTC and conventional diesel combustion is reported to pose substantial problems in both emissions and driveability.[4,5,9–15] In a recent paper,[14] the present authors considered the emissions and performance characteristics of a high-EGR LTC to conventional diesel mode change prompted by a constant-speed increasing-load transient. Modelling work indicated that the boost system was unable to generate sufficient intake pressure to match the demanded load during some intermediate cycles during this increasing-load transition. As a result it was suggested that a practical dual-mode (LTC–conventional diesel) engine would likely require a new turbocharging system, probably incorporating a twin-stage turbocharger or an electrically assisted turbocharger (e-turbo). It was also noted that smoke emissions within the transient were dominated by the response of the EGR system. High smoke emissions were found for cycles in which the EGR levels were greater than the levels required for effective NO_x control in conventional diesel mode but less than the levels required to inhibit soot formation in LTC mode.

The present work now considers the reverse case: a constant-speed decreasing-load transient that requires a combustion mode shift from conventional diesel operation to high-EGR LTC operation. In this decreasing-load scenario it seems reasonable to expect that intra-transient cycle performance will be less influenced by the available boost pressure and more by slow transient response in the EGR level. In this case, the EGR supply needs to transition rapidly from about 25% EGR for conventional diesel operation to about 60% EGR for diesel LTC operation. The response of the EGR system and its interaction with the turbocharger often lead to unpredictable and unfavourable EGR rates which may cause high NO_x or smoke emissions. Note that the EGR flow rate is controlled through the modulation of the EGR valve and the setting of the vane positions of the variable-geometry turbocharger (VGT).

This study considers the cycle-by-cycle changes in operating conditions experienced during a specific decreasing-load transient encountered in the Extra-Urban Driving Cycle (EUDC) phase of the New European Driving Cycle (NEDC) procedure. Individual cycle conditions from this transient are replicated in steady-state conditions on a well-instrumented single-cylinder research engine to obtain corresponding emissions and performance data. The potential of a split-fuel-injection strategy as a means of mitigating high smoke emissions in intermediate cycles is investigated. A sensitivity study showing the effects of variations of injection parameters on engine emissions and performance is also performed and reported in this paper at selected intermediate cycles during the transient.

Research methodology

Experimental data obtained over the NEDC test procedure on a four-cylinder 2 l turbocharged diesel production vehicle was used to specify the target transient behaviour of the single-cylinder research engine. The production vehicle data for engine speed, fuelling quantity and intake pressure during the EUDC phase of the NEDC procedure are shown in Figure 1. Clearly, the necessity for a combustion mode transition at any point within the driving cycle is dependent on the load that can be achieved in the high-EGR LTC mode of operation.

In this work the present authors take the fuelling quantity to be a measure of load. The limiting fuelling quantity for stable high-EGR LTC operation on the single-cylinder research engine has been shown previously to be 16 mg/cycle in naturally aspirated operation.[5] Reference may be made to Figure 2(a), where it is shown that the boost pressure at the beginning of the transient event (cycle 13) with the LTC–conventional diesel dual-combustion mode was approximately 102 kPa (absolute) only. Therefore, a fuelling quantity of 16 mg/cycle was selected as the high-EGR LTC fuelling limit for the present work also. Although this was a rather arbitrary limit, it can be seen from Figure 1 that similar transitions would be required at any other reasonable upper bound of LTC fuelling. With this upper limit imposed and with reference to Figure 1, it

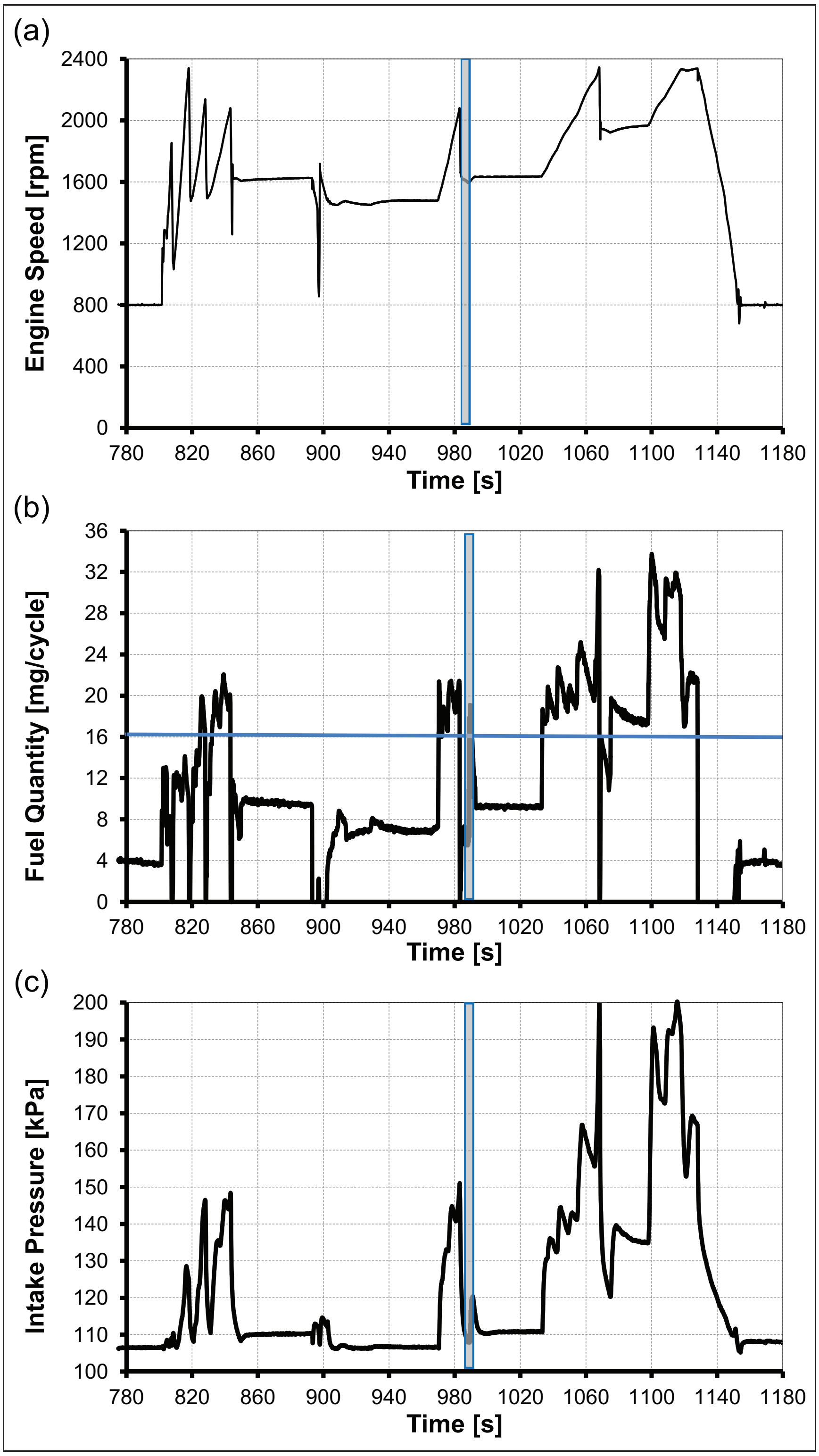

Figure 1. (a) Engine speed, (b) fuel quantities and (c) intake pressure during the EUDC phase of the NEDC procedure derived from a four-cylinder turbocharged diesel engine.

is evident that frequent combustion mode transitions would be required in order to meet the full range of the driving cycle's speed and load requirements. In the EUDC shown in Figure 1, there are three speed and load transients that would require a combustion mode shift:

(a) load and speed transients crossing combustion modes between 820.21 s and 834.21 s (LTC to conventional diesel operation);
(b) load transient crossing combustion mode at a constant speed between 969.9 s and 970.5 s (LTC to conventional diesel operation);[14]

(c) load transient crossing combustion mode at a constant speed between 984.9 s and 991.9 s (LTC to conventional diesel operation and vice versa).

Among the different transient operations, a load transient at a constant engine speed is considered to be the most demanding and most representative of the engine response;[7] increasing-speed transients generally experience less turbo-lag and are considered to be easier to control.[16,17]

In this work, we consider the entire transient involving transition from LTC to conventional diesel operation and then back again to LTC mode that occurs in the transient spike shown in Figure 1 between the elapsed cycle times of 984.9 s and 991.9 s. Note that this is a particularly challenging transient for dual-mode operation as the change from conventional diesel mode to high-EGR LTC mode that is the subject of the present study is immediately preceded by a switch from the high-EGR LTC mode to the conventional diesel mode. Although there are other similar transient spikes in Figure 1, they were associated with gear changes, involving both speed transients and load transients. Further details of the selected transient spike are given in Table 1, which involved a load transient (corresponding to a fuelling quantity from 6 mg/cycle to 19 mg/cycle and then back to 12 mg/cycle) while the engine speed varied between 1618 rpm and 1630 rpm. During this transition period (7 s), the intake pressure initially increased from 108 kPa to 118 kPa (during the increasing-load transient) and then decreased to 112 kPa (during the decreasing-load transient). Note that these boost pressures correspond to load transients in conventional diesel combustion only.

Modelling of the transients and selection of the operating conditions

The cycle-by-cycle changes in the operating conditions of a four-cylinder 2 l turbocharged diesel engine over the selected transient were predicted using Ricardo WAVE one-dimensional (1D) simulations. Details of the present authors' WAVE simulations have been reported in earlier publications.[4,14] However, for completeness, a brief description of the model is provided below.

The inputs to the model were the initial and final fuel quantities, the engine speed, intake pressure, EGR rates and the transient duration. Note that the bounding values of fuel quantity, intake pressure and the transient duration were specified from the multi-cylinder test data described in the previous section. The initial and final EGR rates for high-EGR LTC operation were selected on the basis of previous single-cylinder test results,[4,5,15] and the EGR levels in conventional diesel operation were selected to represent EGR levels used in current production engines. A production engine's turbocharger map was used in the model; however, no attempt was made to optimise the turbocharger for LTC operation. The VGT used in this model was a standard turbocharger typically used in a production 2 l diesel engine; however, it had the lowest inertia in its class[18] and therefore was considered suitable for this investigation where turbo-lag is expected to be long in the LTC–conventional diesel dual-mode operation owing to the reduced exhaust enthalpy. It is expected that, when LTC strategy is employed in a production diesel engine, the existing turbocharger may not be suitable to meet the demanded intake pressure. High boost pressure can be achieved by resizing the turbine; however, resizing the turbocharger may not necessarily meet the full-load requirement (i.e. conventional diesel operation). Therefore, advanced turbocharger technology such as a twin-stage turbocharger (parallel or series sequential) or an e-turbo may be required in an LTC–conventional diesel dual-mode engine, although it may also add cost, complexity, weight and packaging challenge to the engine. Because only a 3–5 kPa higher intake pressure was considered, it is not unreasonable to assume that such a system would be available for the dual-mode engine.

Moderate advanced injection timing (single injection) was used for high-EGR LTC operation, whereas retarded main-injection timing with a pilot injection was used for conventional diesel combustion. The model was initially allowed to run for 20 s in the steady state before the transient simulation started. This was done to ensure that the model had converged before the start of the transient simulation. Similarly, at the end of the transient simulation, the model was allowed to run until the final transient point reached steady-state conditions.

In the absence of experimental cylinder pressure data, the energy release rates were not available to be used in the combustion model. Instead, WAVE's in-built diesel Wiebe model (dependent on engine speed, air flow rate and fuel injection quantity) was used, and no attempt was made to tune the Wiebe constants to match the experimental energy release rates. Due to this relatively crude combustion model, no claim is made regarding the prediction of the engine performance in this work. For this reason, the load specified at each cycle in the 'pseudo-transient' experimental study was obtained from the single-cylinder engine with conventional diesel operation (using the operating conditions obtained from the NEDC test data). These loads were taken as the representative load during the combustion mode transition. It is expected that during the combustion mode transitions the cycle-by-cycle predictions of intake pressure and EGR rates, the key output of these simulations, are little affected by the failure of the combustion model to predict ignition delay, combustion duration and energy release accurately, and that therefore the model predictions are valid in this regard.

The model outputs the predicted EGR rate and intake pressure during the transient on a cycle-by-cycle basis at the specific engine speed. These parameters together with the experimental data for fuelling quantity (load) are used to define the operating conditions for the single-cylinder research engine at selected cycles during the transient event under consideration. The objective of this work was to determine the cycle-by-cycle changes in EGR rate and intake pressure likely to be encountered during a load-change-induced mode transition from conventional diesel to high-EGR LTC operation relative to those expected during the same load change in the conventional diesel mode. Throughout this paper, these cycle-resolved operating conditions, derived from the combination of the driving-cycle test data and 1D simulations, are referred to as 'pseudo-transient test points' or 'pseudo-transient operating conditions'. Pseudo-transient test points were used in this work to allow steady-state evaluation of the conditions typically encountered during transient operation. This concept is based on the assumption that the in-cylinder conditions, between the closure of the intake valves and the opening of the exhaust valves, have the dominant effect on combustion and pollutant formation. If the charge composition, including the intake air mass, EGR mass and the injected fuel quantity, can be matched for a specific cylinder in a specific cycle, it does not matter whether the engine is in the steady state or is undergoing a transient. The effects of factors such as changes in the cylinder liner temperature and residual gas fraction are presumed to be secondary to the principal influence of the charge composition. It should be noted that the engine coolant and oil temperatures were maintained at 80 °C and 90 °C respectively, similar to the values obtained from the EUDC phase of the driving-cycle test. Therefore, the effects of the thermal boundary conditions on emission formation are minimised.

The previously identified constant-speed load spike (see Figure 1) was simulated in its entirety. Thus, the change from diesel to high-EGR mode which is the subject of the present study was immediately preceded in the simulation by a high-EGR LTC to conventional diesel mode change. The solid lines in Figure 2(a) and (b) show the predicted variation in the intake manifold pressure and the EGR levels during the transient with a combustion mode shift. The dashed lines in Figure 2(a) and (b) represent the intake pressure and the EGR levels determined from experimental data during a load transient in conventional diesel operation only. The measured driving-cycle data for the fuelling quantity are shown in Figure 2(c). The crosses within the figures identify the conventional diesel to high-EGR pseudo-transient operating conditions for the single-cylinder

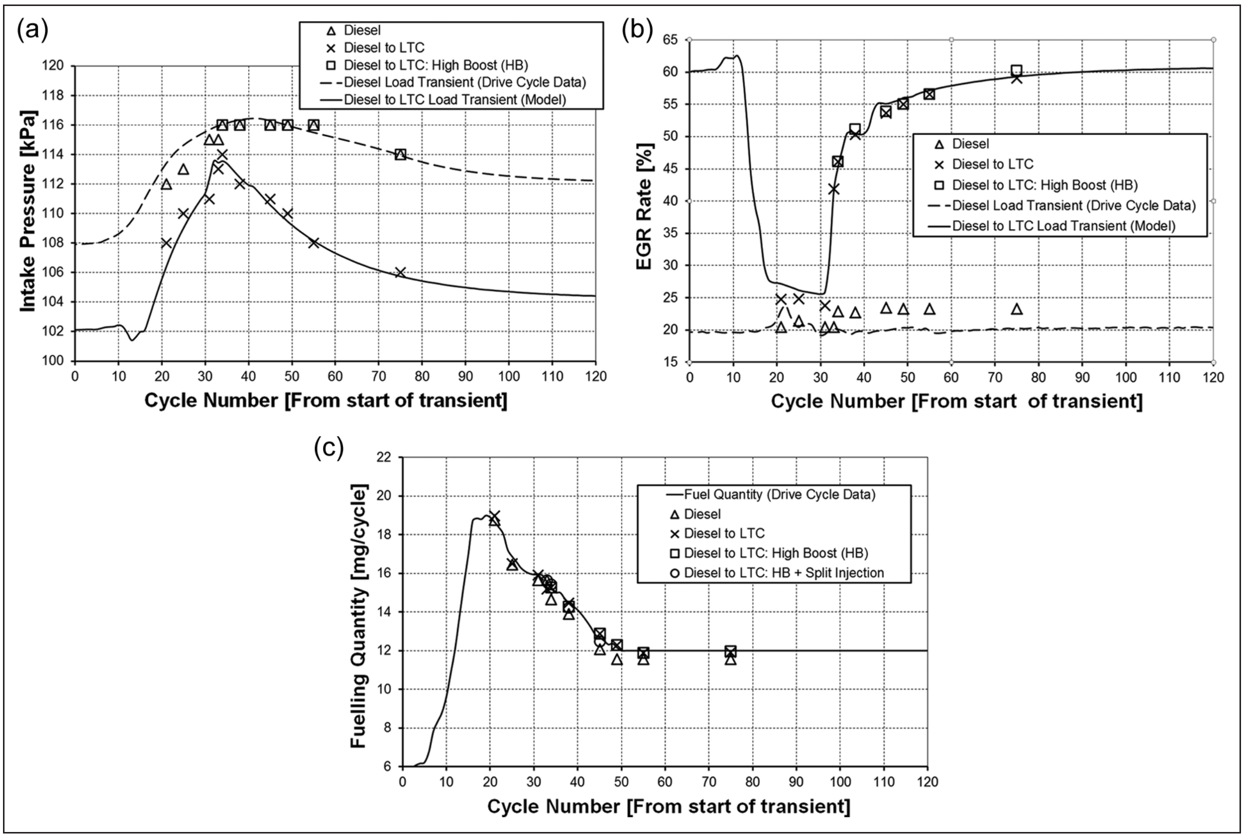

Figure 2. Pseudo-transient operating conditions ((a) intake pressures (b) EGR rates and (c) fuelling quantities) during the selected combustion mode transition. The values for the 'diesel driving cycle' are from the experimental results provided by the project partner.
LTC: low-temperature combustion; EGR: exhaust gas recirculation.

Table 1. Combustion mode transitions (LTC to conventional diesel and vice versa).

	Initial value	Final value
Elapsed cycle time (s)	984.9	991.9
Engine speed (r/min)	1618	1630
Fuel quantity (mg/cycle), high-EGR to conventional diesel	6	19
Fuel quantity (mg/cycle), conventional diesel to high-EGR	19	12
Intake pressure (kPa)	115	118

EGR: exhaust gas recirculation.

Table 2. Pseudo-transient operating conditions (conventional diesel to LTC).

Transition case	Value for the following operations	
	Conventional diesel to LTC[a]	Conventional diesel[b]
Engine speed (r/min)	1600	1600
Fuel injection pressure (MPa)	120 to 80 (decreased with decreasing fuelling quantity)	120 to 80 (decreased with decreasing fuelling quantity)
SOI (deg CA ATDC)	4 to –21	Main injection, ≈5 Pilot injection, –10
Intake charge temperature (°C)	60 ± 20	40 ± 10
Coolant temperature (°C)	80	80
Oil temperature (°C)	90	90

LTC: low-temperature combustion; SOI: start of injection; CA: crank angle; ATDC: after top dead centre.
[a]Data are derived from single-cylinder operating conditions that were found in previous work[4,5] to provide a reasonable trade-off between fuel efficiency and emissions.
[b]Data are derived from the driving-cycle test data of a production engine.

engine experiments. The open triangles represent the experimental test points for the identical transient occurring in conventional diesel operation only. Further details of the pseudo-transient operating conditions are given in Table 2. The open squares refer to the experimental test points for a strategy involving an increasing boost pressure at some of the intermediate cycles during the transient. This strategy will be described in detail later in this paper.

The basis of selection of a series of pseudo-transient operating conditions representing a load transient is as follows. Two pseudo-transient operating conditions were selected corresponding to the start of the transient and the end of the transient. Typically the intake pressure did not reach the desired steady-state value by the end of the transient, although the desired fuel quantity and EGR rates were achieved. This highlights the slow response of the air system, consistent with previous publications.[9,11] It is also due to the feedback controller configuration in the model used in this work which was configured to supply first the targeted EGR levels and then control the VGT position to achieve the targeted boost pressure. Note that the WAVE model used in this work was specially adapted for high-EGR operation and did not replicate the operating strategy to control the EGR levels and intake pressure in the production engine. Some intermediate cycles likely to impose difficulties in terms of achieving the desired engine performance and emissions were also selected for this investigation so that the entire transient event can be represented by a set of pseudo-transient operating conditions. Note that the intermediate operating conditions were selected on the basis of previous results[5,19] which had serious issues of high PM, total unburned hydrocarbon (THC) and carbon monoxide (CO) emissions and increased specific fuel consumption with an EGR sweep at a constant fuelling quantity and at a constant intake pressure.

It is interesting to see that the predicted intake pressure during the combustion mode transition is lower by 5–10 kPa than the corresponding transient in conventional diesel operation. This is a result of reduced exhaust enthalpy as LTC requires a substantial amount of exhaust gases to be routed through the EGR loop. What is also notable in these results is the length of the load transient, particularly with respect to the responses of the air and EGR systems. As indicated by the decrease in fuelling shown in Figure 2(c), the decreasing-load phase of the transient spike starts at cycle 20. However, as shown in Figure 2(a) and (b), overshoots from the earlier increasing-load phase are predicted for the air and EGR systems, resulting in a delay of approximately 10 cycles before the intake manifold pressure and the EGR rates respond to the reduced load demand. Thereafter, Figure 2(b) shows the predicted EGR delivery to increase from approximately 25% to approximately 50% over the next 10 cycles of the transient. It then takes a further 10 cycles for the EGR delivery to reach 55%, and another 30 cycles to reach the steady-state LTC condition of about

Table 3. Single-cylinder research engine and fuelling system specifications.

Research engine	
Engine type	AVL 5402 single-cylinder diesel
Bore	85 mm
Stroke	90 mm
Connecting-rod length	148 mm
Swept volume	0.51 l
Compression ratio	17.1:1
Rated speed	4200 r/min
Maximum brake mean effective pressure	1.3 MPa
Swirl ratio	1.78
Combustion chamber geometry	Re-entrant bowl
Intake ports	Tangential and swirl
Fuelling system	
Injection system	Bosch common rail
Maximum rail pressure	135 MPa
Nozzle type	Valve-covered orifice
Number of holes	5
Diameter of holes	0.18 mm
Spray included angle	142°
Fuel[a]	
Density at 15 °C	840 kg/m^3
Polycyclic aromatic hydrocarbon content	9%
Sulphur content	8 mg/kg
Cetane number	52
Lower heating value	42.6 MJ/ kg

[a]Values provided by the supplier.

60%. With high-EGR LTC, even a small change in the EGR level can have a significant negative impact on the emissions, especially of PM.[20] As a result, the 40 cycles between 50% EGR and 60% EGR have an unfavourable intermediate EGR rate; this is significantly more complex than the 4 cycles observed for the increasing-load transient examined previously by Sarangi et al.[14] Therefore, handling this decreasing-load transient (conventional diesel to high-EGR LTC) is more complex than the increasing-load transient (high-EGR LTC to conventional diesel combustion).

Experimental apparatus

The experimental research facility used in this work was based on an AVL 5402 single-cylinder high-speed diesel engine. This engine is a single-cylinder version of a typical 2 l four-cylinder light-duty high-speed direct-injection diesel engine. It features a central fuel injector and three valves (two inlets and one exhaust) with a double-overhead-camshaft valvetrain. One of the exhaust valves was removed and deactivated to obtain borescopic access to the combustion chamber, although this feature was not used in the current work. The fuelling system was a Bosch common-rail CP3 injection system consisting of a production-type high-pressure common-rail fuel pump supplying fuel to the injector up to a maximum pressure of 135 MPa. A fuel-conditioning unit maintained a constant-temperature fuel supply to the injector. An automotive-grade sulphur-free diesel fuel that meets the current British Standard BS EN 590 and complies with the current requirements of the UK ‘Motor fuel (composition and content) regulations’ was used in all tests. Detailed specification of the engine, the fuelling system and the fuel used in this work are given in Table 3.

The intake charge pressure, temperature and EGR rate were controlled using a custom-built air exchange system, as illustrated schematically in Figure 3. The EGR temperature was controlled between 50 °C and 150 °C to have an intake manifold temperature varying between 40 °C and 80 °C (see Table 2) depending on engine load (fuelling quantity) and the EGR rate. The research facility was fully instrumented to measure the air temperature and pressure, fuel and air flow rates, exhaust emissions and in-cylinder pressure. Detailed specifications of the engine, the fuelling system, the fuel and the air-exchange system used in this work have been given in previous publications.[14,19]

Experimental results

The recorded in-cylinder pressure data were averaged over 200 consecutive cycles. Subsequently, the averaged data were used to derive the gross indicated mean effective pressure (GIMEP) and the energy release rate. The energy release rate was used to identify the start of combustion (SOC) corresponding to the main combustion event and the midpoint of the cumulative energy release (CA50). Note that the cool-flame reactions that are generally considered to be representative of LTC and the energy release from pilot combustion in the conventional diesel mode were ignored while identifying the SOC from the energy release rate. This is referred to as the ‘main combustion event’ in this work, to distinguish it from the initial cool-flame reactions and any diesel pilot combustion event. The reported emissions data were averaged over a minimum of 3 min in steady-state engine operation. The estimated percentage error for each of the reported quantities is given in Table 4. Note that error bars are not plotted in the line graphs showing the experimental results in this work as, in the majority of cases, the error bars would be smaller than the data marker.

The results were subdivided into two subsections to provide a more focused investigation of the effects of the load transient on performance, emissions and related combustion parameters: first, the effects of the load transient on cycle-by-cycle performance, emissions and combustion characteristics; second, the sensitivity of the split-fuel-injection strategy to performance, emissions and combustion parameters in intermediate cycles.

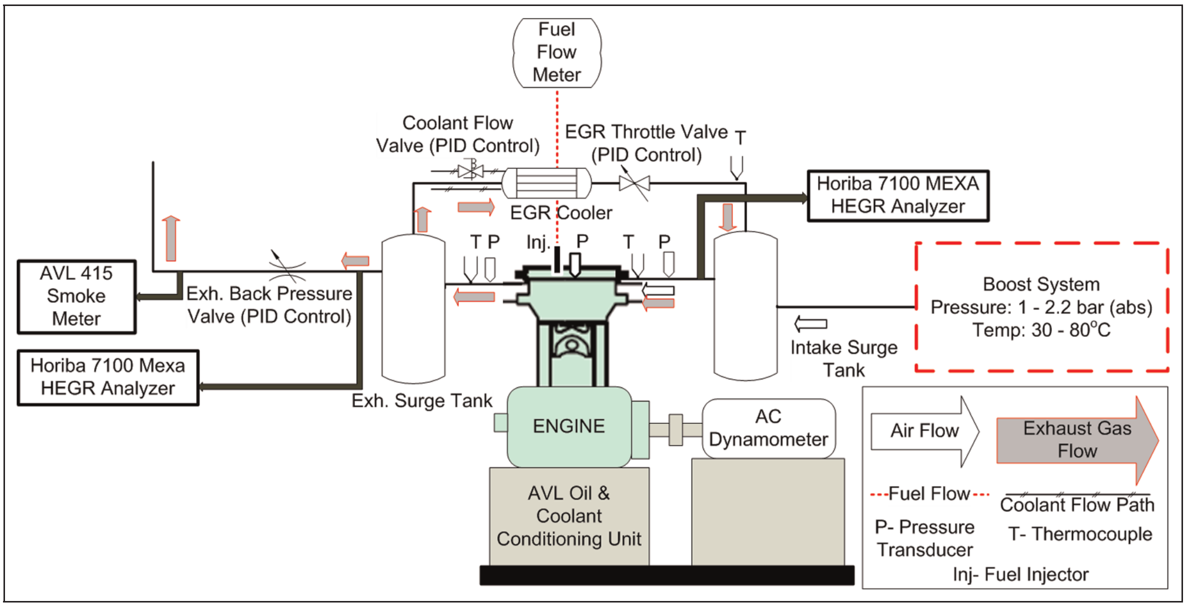

Figure 3. Schematic diagram of the research engine facility and the air-exchange system.
EGR: exhaust gas recirculation; PID: proportional–integral–derivative; HEGR: high-efficiency exhaust gas recirculation: Exh: exhaust; AC: a.c.

Table 4. Estimated percentage errors in the experimental results.

Reported quantity	Percentage error
Gross indicated specific fuel consumption (g/kWh)	2.8%
Smoke (FSN)	3.5%
Carbon monoxide (g/kg fuel)	3.5%
Nitrogen oxides (g/kg fuel)	9.1%
Total unburned hydrocarbons (g/kg fuel)	9.1%
Gross indicated mean effective pressure (kPa)	3%

FSN: filter smoke number.

Effects of the load transient on cycle-by-cycle performance, emissions and combustion characteristics

Combustion progression, including the ignition delay (SOC–start of injection (SOI)) and the CA50 together with the GIMEP are shown in Figure 4 for the pseudo-transient operating conditions detailed in Figure 2. Data are also shown for modified test conditions corresponding to two potential emissions reduction strategies: increased boost pressure, and increased boost pressure with split-injection timing. For the split-fuel-injection strategy, the ignition delay (SOC–start of first injection (SOI1)) for the fuel from the first injection event and the timing of the second injection relative to the SOC (SOC–start of second injection (SOI2)) are also shown for selected intermediate cycles (see Figure 4(b)). The data shows the combustion mode transition from conventional diesel to LTC to occur over a period of approximately 16 cycles, between cycle 33 and cycle 49 of the transient spike. Note that the cycles are numbered consecutively from the commanded start of the transient. The combustion mode transition is evidenced by a substantial increase in the ignition delay (SOC–SOI); the increase is from approximately 4° CA in conventional diesel operation to approximately 12° CA in high-EGR LTC mode.

The data also shows that CA50 moves from a retarded (approximately 16° CA ATDC) timing in conventional diesel operation to a timing near top dead centre (TDC) (approximately 2° CA ATDC) as a result of the mode switch. This is consistent with the engine operating strategy: retarded main-injection timing was used in conventional diesel operation to reduce the NO_x emissions, whereas advanced SOI timing was used in high-EGR LTC to maximise the combustion efficiency. Figure 5 details the experimental results for the NO_x, smoke, THC and CO emissions from the single-cylinder engine. The gross indicated specific fuel consumption (GISFC) of the engine is also shown. Diesel LTC is a low-NO_x operating strategy and, as expected, the results presented in Figure 5(a) show the transition from conventional diesel to LTC within the transient to be accompanied by a substantial reduction in NO_x emissions. In contrast, NO_x emissions for the load transient in conventional diesel operation increased through the transient despite the use of substantially retarded injection timings.

Figure 5(b) shows the experimental results for smoke emissions. It can be seen that there is a general decrease in smoke emissions between the initial and final states of the transient for both the mixed mode and the conventional diesel operating strategies. However, whereas the decrease in the smoke emissions is monotonic in the conventional diesel transient, substantial increases in smoke emissions are observed in the intermediate cycles

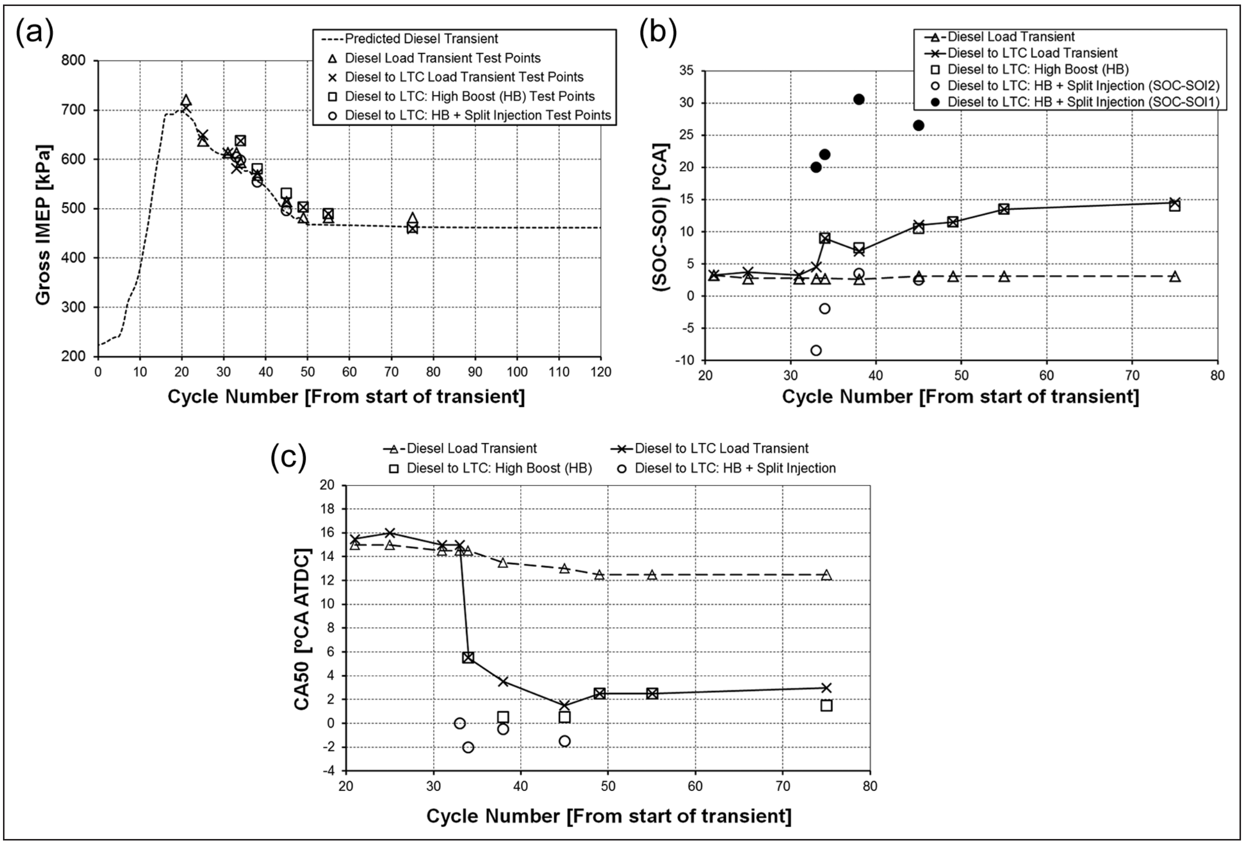

Figure 4. (a) GIMEP, (b) ignition delay and (c) combustion phasing corresponding to the pseudo-transient operating conditions during the diesel-to-LTC phase of the selected transient event.
IMEP: indicated mean effective pressure; LTC: low-temperature combustion; SOC: start of combustion; SOI; start of injection; CA: crank angle; SOI2; start of second injection; SOI1: start of first injection; CA50: midpoint of cumulative energy; ATDC: after top dead centre.

(approximately, cycles 30 to 50) of the conventional diesel to LTC combustion mode transition. By reference to Figure 2(b), it can be seen that these high smoke producing cycles correspond to cycles that see intermediate levels of EGR, i.e. EGR levels that are somewhere between the 25% required for conventional diesel operation and the approximately 60% that is required for high-EGR LTC operation. The transition from conventional diesel combustion to LTC leads to increased PM emissions; as has been shown in previous work by the present authors, there is a clear maxima in PM emissions at EGR levels in between conventional diesel and LTC conditions. The use of split-injection strategies can reduce this PM peak, although this is highly dependent on the details of the injection strategy.[21,22] During these cycles there is then a relatively high oxygen-based equivalence ratio and a short ignition delay, i.e. the in-cylinder conditions that result from the demanded load change are not suitable for conventional diesel operation nor are they suitable for diesel LTC. Later in the transient, the EGR rate reaches the required level for LTC; the ignition delay is increased, the in-cylinder temperature is decreased and smoke emissions fall to a similar level as found in the conventional diesel load transient. However, it should be noted here that neither the conventional diesel operating strategy nor the LTC strategy used on the single-cylinder engine were optimised. As a result, the conventional diesel emissions data presented in Figure 5(b) are higher than would be expected from a typical Euro 5-compliant production engine.

The substantial spike in smoke emissions produced during the transition from conventional diesel operation to high-EGR LTC operation is obviously undesirable. Therefore, two potential mitigating strategies, i.e. an increase in boost pressure and a split-injection strategy, were applied to the respective cycles and evaluated. First the effects of a small (approximately 5–10 kPa) increase in the boost pressure during the combustion mode transition were considered. The data for this strategy are shown in all relevant figures by open squares and are referred to as the high-boost-pressure strategy. The boost pressure was increased while holding the EGR rate constant. As a result, both fresh air and the EGR mass flow rate increased. In this work, the increase in the boost pressure was achieved through the use of the electrically driven compressor. As shown in Figure 6, the increased intake charge pressure causes only a minor reduction in the oxygen-based equivalence ratio over the cycles of interest, which results in only marginally lower smoke emissions in cycles 38 and 45 (see Figure 5(b)), and a negligible effect on NO_x

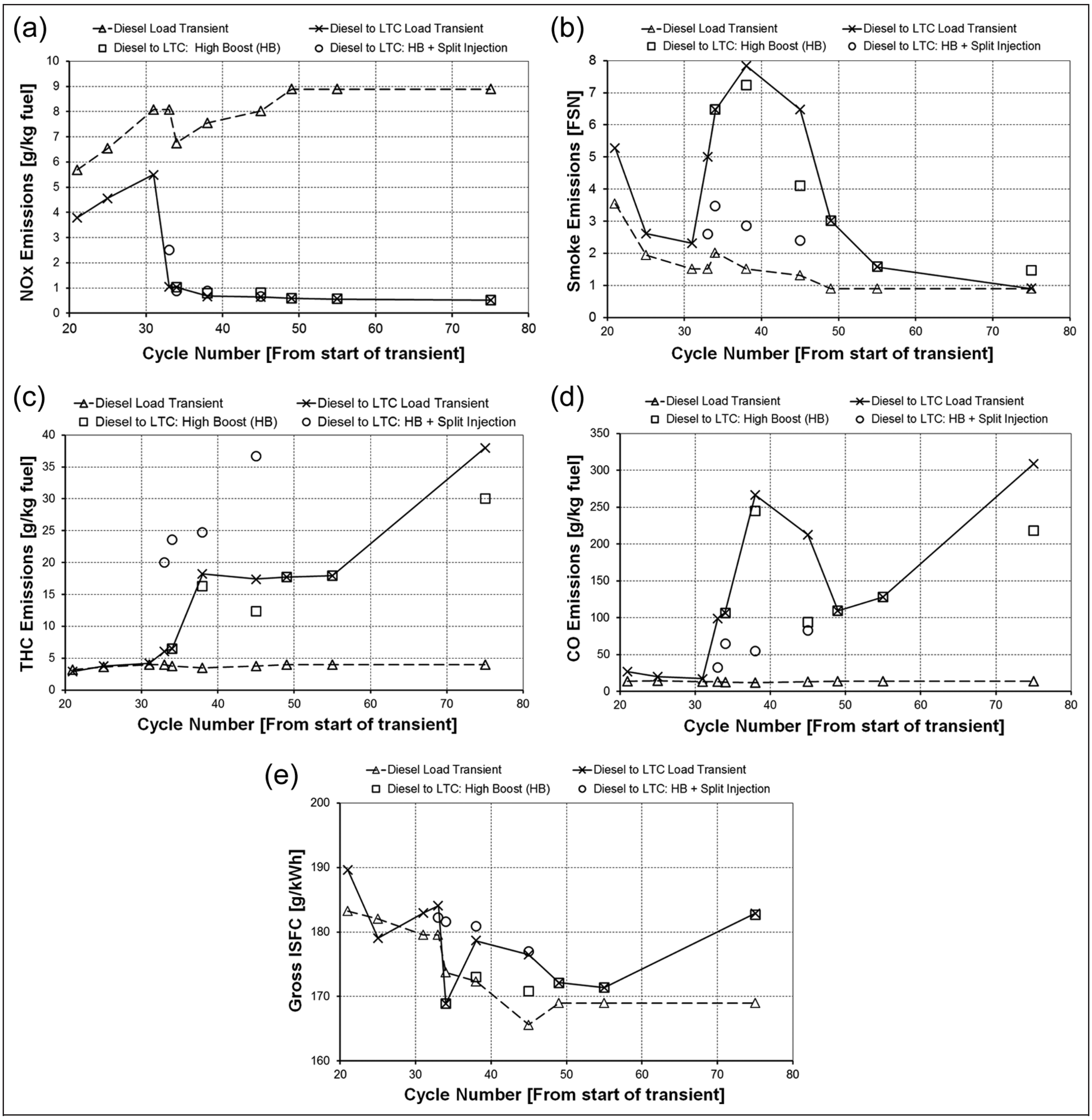

Figure 5. (a)–(d) Emissions and (e) GISFC corresponding to the pseudo-transient operating conditions.
LTC: low-temperature combustion; NOx: oxides of nitrogen; FSN: filter smoke number; THC: total hydrocarbon; CO: carbon monoxide; ISFC: indicated specific fuel consumption.

emissions (see Figure 5(a)). Similarly, no significant changes in the combustion parameters (see Figure 4) were seen; these observations were consistent with previous work evaluating small changes in the boost pressure.[14] It should be noted that in the single-fuel-injection strategy the increased boost pressure did not influence the GIMEP; however, in the split-fuel-injection strategy, GIMEP was reduced marginally. Therefore, a small increase in the boost pressure was investigated in this work to compensate for any possible reduction of GIMEP as a result of the split-fuel-injection strategy.

High emissions of partial combustion by-products (THC and CO) is a known issue in diesel LTC; it is therefore expected that these emissions towards the end of the conventional diesel to LTC transient will be substantially higher than the emissions in the corresponding cycles during a transient in the conventional diesel mode only. Figure 5(c) and (d) confirm this to be the case. Significantly, the experimental results indicate that the increase in THC emissions and CO emissions over the combustion mode switch is not smooth. There is an initial sharp increase and plateau in the THC emissions and an evident spike in the CO emissions

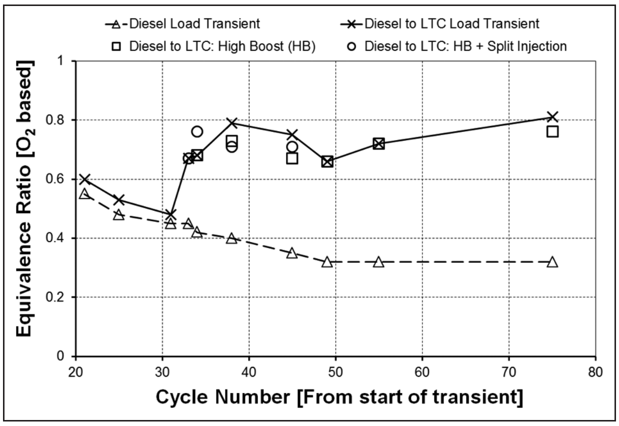

Figure 6. Oxygen-based equivalence ratios derived from experimental data corresponding to pseudo-transient operating conditions.
LTC: low-temperature combustion; O_2: oxygen.

within the transient. Again these features occur between cycle 30 and cycle 50 corresponding to the cycles with intermediate EGR levels. In previous publications,[5,19] the present authors have demonstrated that there is a narrow range of EGR levels where low NO_x and low PM emissions are achieved before THC and CO emissions substantially increase. At the conditions tested here, this was found to be in the intake oxygen mass fraction range 13–14%. As discussed in previous publications,[14,19] the high THC and CO emissions of LTC operation may be attributed to the low in-cylinder oxygen concentration and lower flame temperature of the combustion mode compared with those for conventional diesel operation. However, the high-boost-pressure strategy investigated in this work had a minor effect on the oxygen-based equivalence ratio, as shown in Figure 6. Therefore, it may be expected that the high-boost-pressure strategy would not lead to any significant reduction in THC and CO emissions through the mechanism of increased oxygen availability. In general, the relatively modest increase in boost pressure examined in this work has a negligible effect on the THC and CO emissions (see Figure 5(c) and (d)).

It might also be expected that high emissions of partial combustion by-products would be indicative of poor combustion efficiency and hence correlate directly to poor specific fuel consumption. However, as shown in Figure 5(e), there is no clear correlation between the THC and CO emissions presented in Figure 5(c) and (d) and the GISFC data shown in Figure 5(e). Moreover, with the exception of the initial and final states, there is little difference between the diesel to LTC transient and the transient within conventional diesel combustion only with respect to the GISFC. This is particularly the case when the high-boost-pressure strategy is applied during the conventional diesel to LTC mode switch. During the transition from conventional diesel to high-EGR LTC, combustion phasing was maintained close to TDC. This helped to maximise the work extracted during the expansion stroke and recovered a part of the lost combustion efficiency associated with LTC operation. Conversely, the retarded combustion phasing used in this study acts to increase GISFC during the load transient in conventional diesel operation.

Although the use of an increased boost pressure during the combustion mode change was found to be beneficial in reducing emissions during the transient, the magnitude of the emissions reduction was shown to be small. Smoke emissions, in particular, remained high during those cycles that see an intermediate level of EGR. Therefore, in an effort to reduce the smoke emissions further, the use of a split-fuel-injection strategy in tandem with the previously described high-boost-pressure strategy was investigated.

In previous publications the present authors have demonstrated the effectiveness of a 50:50 split-injection strategy in reducing the smoke emissions at reduced (approximately 52%) levels of EGR.[23,24] This split-injection strategy was also implemented in the intermediate cycles during a combustion mode transition from high-EGR LTC to conventional diesel operation.[14] In this last referenced work, it was found that, although smoke emissions were reduced significantly in the intermediate cycles, GIMEP was also reduced, albeit marginally compared with the single-fuel-injection event. Any reduction in GIMEP is undesirable from a driveability point of view. With this in mind, and based on experience gained earlier,[5] a fuel split ratio of 35:65 was selected for use in the present work. The use of this split ratio was expected to have less of an impact on GIMEP and, therefore, smoke emissions could be reduced without any significant effect on the driving-cycle load requirement. It should be noted that optimising the fuel split ratio was not a specific objective of this work; the aim was only to elucidate the effect of this parameter on engine performance and emissions. It was shown earlier[5] that, in the 35:65 fuel split ratio case, the amount of fuel going into the squish region and impinging on the piston top is reduced in comparison with that in the 50:50 fuel split ratio case, thereby reducing the THC emissions and combustion efficiency penalty. This, in turn, helps to maintain the driving-cycle load. However, since the degree of premixing is reduced with lower proportion of fuel in the first injection event (35:65), smoke emissions increase.

The high-boost-pressure + split-injection strategy was applied to cycles 33, 34, 38 and 45 within the transient. Figure 7 details the range of fuel injection timings for both split-injection and single-injection studies. With respect to the high-boost-pressure + split-injection strategy, the SOI1 timings are represented by open circles, whereas the SOI2 timings are represented by full circles. The commanded pulse width of the first-injection fuel event in the split-injection cases is 3° CA. The effects of the split-injection strategy on engine performance and emissions are shown in Figure 4 and

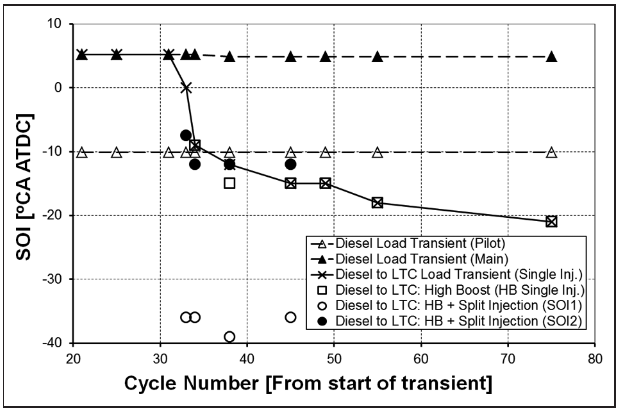

Figure 7. SOI timings corresponding to the pseudo-transient operating conditions during the transient. The commanded pulse width of the first injection fuel event in the split-injection cases is 3° CA.
SOI: start of injection; CA: crank angle; ATDC: after top dead centre; LTC: low-temperature combustion; Inj.: injection; SOI1: start of first injection; SOI2: start of second injection.

Figure 5 respectively. The results show the NO_x emissions within the transient to be largely unaffected by the split-injection strategy, the exception to this being cycle 33 in which the emissions are increased relative to those in the single-injection case. On detailed inspection, it was found that cycle 33 was characterised by an advanced CA50 timing and a high intake oxygen mass fraction ($Y_{O_2} > 14\%$) compared with the other intermediate cycles examined, which have much lower ($Y_{O_2} \leqslant 12\%$) intake oxygen mass fractions. This is consistent with the previous investigation by Tanabe et al.[11]

With respect to the soot emissions within the transient, the results presented in Figure 5(b) show the split-injection strategy to be a very effective means of reducing the soot spike that is typically observed with a single-injection event. Note the increased ignition delay (SOC–SOI1) of the fuel from the first injection event (see Figure 4(b)). Although the delay between the 2nd fuel injection event and the start of combustion (SOC-SOI2) is reduced in intermediate cycles, the increased premixing time for the fuel from the first injection event and the better distribution of the injected fuel between the piston bowl and the squish region resulted in less soot formation and higher soot oxidation than in the single-injection event. CO emissions are similarly reduced; however, as shown in Figure 5(c), there is a significant increase in THC emissions from the intermediate cycles. Note that the increase in THC emissions with the split-injection strategy that was observed in this work is consistent with results published in an earlier work by the present authors.[14] The simultaneous increase in THC emissions and the decrease in the CO emissions suggest that there may be an increase in piston–wall wetting with the split-injection strategy, which then reduces the amount of reacting or partially reacting fuel. The increase in the utilization of air in the squish region also helps to reduce CO emissions. However, there is a corresponding minor increase in GISFC.

Sensitivity of the split-fuel-injection strategy to performance, emissions and combustion parameters in intermediate cycles

The results presented in Figure 5(b) clearly indicate the potential of split fuel injection as a means of reducing smoke emissions in intermediate cycles within a conventional diesel to LTC load transient. However, as evidenced by the results presented in Figure 5(c), there is a considerable THC emissions penalty associated with the specific split-injection strategy applied in the reported tests. Two of the intermediate cycles (cycles 33 and 38) were noted to pose a particular challenge in this respect. In light of this, additional experimental testing was performed at the pseudo-transient conditions corresponding to these two cycles so as to understand better the influence of key fuel injection parameters with the split-injection strategy. Table 5 details the six combinations of fuel injection parameters examined in this phase of the work. The CA50 results obtained from these investigations are also listed in the table. The fuel injection parameters were selected on the basis of fuel

Table 5. Split-fuel-injection commanded parameters at selected pseudo-transient operating conditions.

Cycle	Fuel quantity (mg/cycle)	EGR rate (%)	P_{int} (kPa)	Injection timing (°CA ATDC)			P_{inj} (MPa)	Fuel split ratio (commanded)	Case	CA50 (°CA ATDC)
				SOI1	SOI2	Commanded pulse width of first injection event (°CA)				
33	15.4 ± 0.2	41.5	117	–36	–3.75	2.5	120	50:50	A	2
				–36	–7.5	2.5	100	40:60	B	0
				–37.875	–7.5	3	100	50:50	C	–1
38	14.6 ± 0.2	50.5	116	–39	–12	2.5	90	35:65	D	–0.5
				–42	–12	2.5	90	35:65	E	0.5
				–36	–15	3.5	70	50:50	F	–3.5

CA: crank angle; ATDC: after top dead centre; EGR: exhaust gas recirculation; P_{int}: intake pressure; SOI1, start of first injection; SOI2: start of second injection ; P_{inj}: injection pressure; CA50: midpoint of cumulative energy.

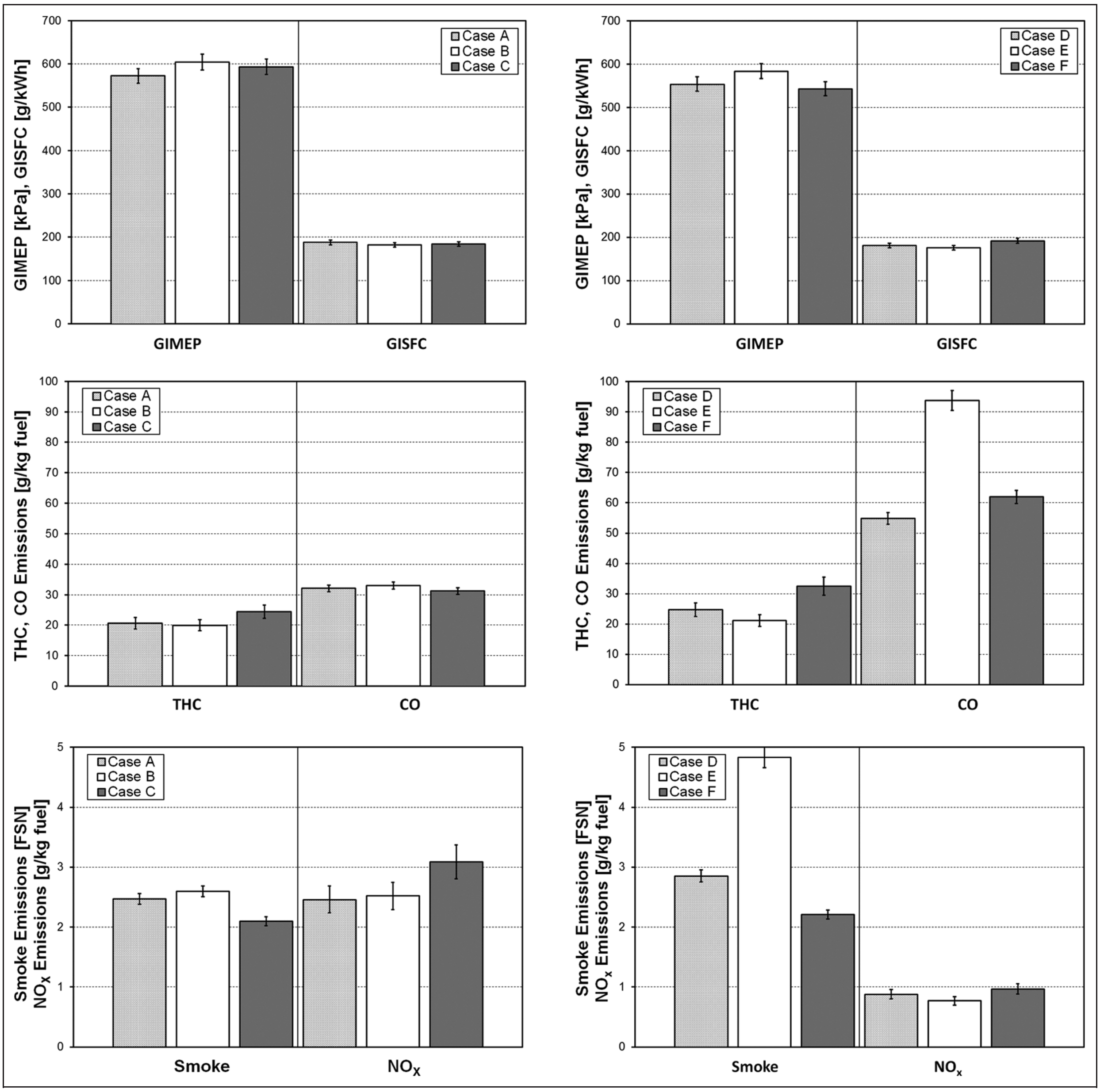

Figure 8. In-cylinder performance and emissions with variable split-fuel-injection parameters (left-hand column, cycle 33; right-hand column cycle 38).
GIMEP: gross indicated mean effective pressure; GISFC: gross indicated specific fuel consumption; THC: total hydrocarbon; CO: carbon monoxide; FSN: filter smoke number; NO_x: oxides of nitrogen.

injection pressure affecting spray penetration, fuel injection timing affecting spray impingement and split ratio affecting the level of mixing. It should be noted that the injection parameters investigated here were selected arbitrarily within close proximity to each other so as to determine their sensitivity on engine performance and the emissions. No attempt was made to optimise these parameters as the optimization would involve detailed design of experiments techniques and was beyond the scope of this work. Figure 8 presents the experimental results for the in-cylinder performance and emissions of the engine in these six test cases together with the baseline single-fuel-injection case where the left-hand column of plots in the figure refer to cycle 33, and the right-hand column to cycle 38. Note that the error bars shown in Figure 8 correspond to the estimated experimental errors reported in Table 4.

It is clear from the results presented in the left-hand column of graphs in Figure 8 that the cycle 33 emissions and performance were largely insensitive to the investigated changes in the split-injection operating strategy. No statistical difference was observed between the GIMEP, GISFC and CO data for the three split-injection operating conditions examined. Smoke emissions were reduced by approximately 0.5 filter smoke number (FSN) in the case of a 50:50 split injection at 100 MPa injection pressure (case C) compared with the other two cases. For a fixed 50:50 split injection, a minor increase in THC emissions was seen when the injection pressure was decreased from 120 MPa to 100 MPa. As described previously, THC emissions from the split-injection strategy are substantially higher than those recorded within the same cycle with a single-injection fuelling strategy (approximately 20 g/kg fuel for a split injection compared with approximately 6 g/kg fuel for a single injection). There is similarly small increase in NO_x emissions associated with the decrease in the injection pressure for the 50:50 split-injection case because of an advanced CA50. The NO_x emissions increased by up to 200% with the split-injection strategy compared with the single-injection case owing to advanced CA50 at low EGR levels ($Y_{O_2} > 14\%$).

For cycle 38 (the right-hand column of graphs in Figure 8), the performance and emissions data is shown to be more readily influenced by the effects of fuel injection parameters. Smoke and CO emissions are statistically different for all three split-injection points. By comparison of cases D and E, the smoke and CO emissions are shown to be very sensitive to the timing of the first injection event (SOI1). For case E, SOI1 is advanced by 3° CA with respect to case D. This results in an approximately 70% increase in smoke and CO emissions. For case F, the THC emissions are greater than those of either case D or case E, and the smoke emissions are markedly reduced. However, these results are confounded as both the split ratio and the injection pressure of case F differ from those in cases D and E. As for the cycle 33 data, the trend of GISFC, which is influenced by both the combustion efficiency and the combustion phasing does not show significant sensitivity to the variations in the injection parameters (within the uncertainty limits of the experimental data). Finally, it should be noted that the relative sensitivity of the results for cycle 38 to the injection timing parameters compared with the results for cycle 33 is consistent with the results of previous studies from the present authors' research group.[19,23,25] In these studies, it was shown that the sensitivity of the engine's performance and exhaust emissions to variations in fuel injection strategy is increased as the intake oxygen mass fraction is reduced. The EGR rate is approximately 51% in cycle 38 as opposed to approximately 42% in cycle 33, yielding intake oxygen mass fractions of approximately 11.95% and 14.8% respectively.

Conclusions

By means of a modelling-led experimental study, the authors have considered the performance and emissions of a 2 l production diesel engine in dual-mode operation during a specific decreasing-load transient encountered in the EUDC phase of the NEDC procedure. The results of this study show the difficulty of managing transient-driven combustion mode changes. The new results have yielded the following important observations.

1. Smoke emissions were substantially increased during the combustion mode switch transient owing to the relatively slow response of the EGR system. This results in EGR levels that are not well matched to the requirements of either conventional diesel combustion or LTC diesel combustion for a significant fraction of the transient event.
2. THC and CO emissions were, in general, very high in high-EGR LTC operation. They also increased during the intermediate-EGR level cycles.
3. Split fuel injection at higher intake pressures was effective in reducing the smoke and CO emissions from the intermediate cycles during combustion mode transition. However, THC emissions were increased.
4. The sensitivity of the engine, in terms of the performance and emissions, to fuel injection control parameters in the split-injection LTC mode is highly dependent on the intake oxygen mass fraction (proportional to the EGR rate). Thus, cycle 33 within the transient (approximately 42% EGR) was found to be insensitive to injection pressure and the split ratio, whereas the smoke, THC and CO emissions in cycle 38 (approximately 51% EGR) were affected. This suggests that optimisation of a combustion mode switch which includes high-EGR LTC operation will require high-frequency monitoring of the engine conditions and a rapid response from the engine controller.
5. Combustion mode transition needs to be coupled to a similar combustion phasing (CA50) transition (from a retarded CA50 in conventional diesel to a near-TDC CA50 in high-EGR LTC) strategy so that a high combustion efficiency can be achieved with high-EGR operation. However, the transition of CA50 should be gradual and, accordingly, the injection timings and injection pressures need to be adjusted.

Considering the significance of these results, it should be recognised that the NEDC procedure is not necessarily a true representation of the transient conditions in real-world driving conditions in Europe, nor is it the regulatory driving cycle in other jurisdictions. The real-world transients and the driving cycles in other jurisdictions are typically more rapid, aggressive and different from the NEDC procedure. The NEDC is a relatively low-load slow-acceleration driving cycle which is therefore more conducive to LTC operation than other certification driving cycles and real-world driving conditions. It is expected that a more aggressive driving cycle will be 'more challenging' for combustion mode transitions in an LTC–conventional diesel dual-mode engine. The substantial technical challenges associated with diesel–LTC combustion mode changes as illustrated in this work demonstrate the importance of minimising the need for such transitions by significantly increasing the LTC load limit.

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Declaration of conflict of interest

The authors declare that there is no conflict of interest.

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Appendix 1

Notation

Y_{O_2}	intake oxygen mass fraction

Abbreviations

ATDC	after top dead centre
CA	crank angle
CA50	midpoint of cumulative energy
CO	carbon monoxide
EGR	exhaust gas recirculation
EUDC	Extra-Urban Driving Cycle
FSN	filter smoke number
GIMEP	gross indicated mean effective pressure
GISFC	gross indicated specific fuel consumption
LTC	low-temperature combustion
NEDC	New European Driving Cycle
NO_x	oxides of nitrogen (nitric oxide and nitrogen dioxide)
PM	particulate matter
SOC	start of combustion
SOI	start of injection
SOI1	start of first injection
SOI2	start of second injection
TDC	top dead centre
THC	total unburned hydrocarbon (C1)
VGT	variable-geometry turbocharger
HB	high boost