The effects of early inlet valve closing and cylinder disablement on fuel economy and emissions of a direct injection diesel engine

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A B S T R A C T
The influence of EIVC (early inlet valve closure) on emissions, fuel economy and exhaust gas temperature of a turbocharged, 4 cylinder common rail direct injection diesel engine has been investigated and compared with the influence of deactivating two cylinders. IVC (inlet valve closing) timings were set at up to 60 CA (crank angle) degrees earlier than the production setting of 37° ABDC for the engine. At the earliest timing, effective compression ratio was reduced from 15.2:1 to 13.7:1. The effects on emissions were significant only for EIVC settings at least 40 CA degrees earlier than the production setting, and were sensitive to engine load. At 2 bar BMEP (brake mean effective pressure) and fixed levels of NOx, soot emissions were reduced but CO (carbon monoxide) and HC (hydrocarbon) increased unless fuel rail pressure was reduced. With increasing load, soot reduction diminished and was negligible at 6 bar BMEP; CO and HC emissions deteriorated further. At all conditions, EIVC raised exhaust gas temperature by >50 °C; the effect on fuel economy was negligible or a fuel economy penalty. Comparisons indicate cylinder deactivation is the more effective strategy for reducing engine-out emissions of HC and CO and raising exhaust gas temperature under light load operating conditions.

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1. Introduction

Diesel engines face increased competition from spark ignition engines with improved fuel economy achieved through downsizing, and will incur higher costs of emissions after treatment to meet Euro VI regulations coming into force in 2014 in the EU. Hence there is heightened interest in measures which can improve fuel economy, or reduce engine-out emissions or raise the effectiveness of after treatment. Variable valve timing is one candidate technology which, although widely employed on spark ignition engines to improve performance and fuel economy [1,2], has received relatively little attention for diesel applications. In an early study, Benajes et al. [3] looked at the effect of briefly opening the intake valve during the exhaust stroke to generate internal EGR (exhaust gas recirculation). The effects of early and late inlet valve closure timings on fuel economy, emissions and exhaust gas temperature have been explored in more detail in later papers covering applications to light and heavy duty diesel engines. A summary of results is given in Table 1. LIVC (late inlet valve closure) is commonly associated with the Atkinson cycle and has received most attention. When the inlet valve is closed later in the compression stroke, reverse flow into the inlet manifold reduces the in-cylinder trapped mass and lowers the effective compression ratio. This extends ignition delay and produces a more premixed type of combustion. EIVC (early inlet valve closure) is commonly associated with the Miller cycle and valve closure before the induction stroke is completed. The trapped charge then undergoes an expansion before being recompressed on the compression stroke, in effect reducing the compression stroke. Limits on valve train acceleration during valve closing restrict the maximum valve lift that can be used with the shorter valve opening durations, increasing flow resistance. Both a reduced valve lift and early inlet valve closure act to lower trapped charge and retard the crank angle at which cylinder pressure reaches intake manifold pressure during the compression stroke.

In Ref. [4], late inlet valve closure was shown to reduce soot and NOx, but increase HC and CO. Similar trends in emissions responses to early inlet valve closure were reported in Ref. [5], for a heavy duty V8 diesel engine. This similarity of responses to both EIVC and LIVC timing changes is evident in other work [6–13], with some exceptions which may be attributable to the varying range of MAP
local fuel-rich regions. The results of Walter et al. [16] are consistent with this interpretation. They also noted that lower in-cylinder temperatures reduced the level of EGR required to meet a target NOx concentration. This was shown in Ref. [18] to offer significant reductions in engine-out emissions of soot, CO and HC. Cylinder deactivation raised exhaust gas temperature, with the potential to raise the warm-up rate and operating temperature of after treatment components to the advantage of after treatment system performance at light loads. Subjectively, NVH (noise vibration harshness) qualities were judged to show only a minor deterioration.

The aim of the study reported here was to examine the benefits of early inlet valve closure timings over a range of loads and speeds, and compare these to the benefits of cylinder disabling used the same engine. The hardware available allowed the influence of only IVC timings to be examined; CFD simulations carried out before the experimental study and results in the literature suggest EIVC and LIVC produce similar trends in emissions responses. The experimental study was carried out using a Ford 4 cylinder, 2.2 l, turbocharged, common rail, direct injection diesel engine with a compression ratio of 15.5:1. The engine had an external EGR system with a gas-to-coolant EGR cooler. The engine was modified by fitting an experimental valve train with a cam shifting system produced by Schaeffler which allowed the IVC timing to be set to 37° ABDC, 7° ABDC, 3° BBDC or 23° BBDC. The standard IVC timing setting for the engine was 37° ABDC; the three other settings gave relatively early timings. Cylinder disablement was achieved by fitting new intake and exhaust camshafts which allowed the two inboard cylinders to be switched to zero-lift cams, leaving the intake and exhaust valves closed and the cylinders sealed. The fuel supply to the disabled cylinders was also cut off by directing the injector driving signal to two ‘dummy’ injectors placed outside the engine.

(Manifold absolute pressure), AFR (Air to fuel ratio) and EGR rate settings examined in the different investigations. The influence of LIVC on in-cylinder conditions was investigated by Murata et al. [11,14] through CFD (computational fluid dynamics) simulations. Plotting local states on an equivalence ratio–temperature map showed local peak values of temperature and equivalence ratio were lowered, moderating NOx and soot formation during combustion. Nevin et al. [15] examined the use of LIVC with early injection timings of 55° CA BTDC (before top dead centre) and 40% EGR at low load, observing that NOx was lower at high-speed medium-load conditions. In their experimental work, He et al. [12] suggest that the reduction in soot emissions achieved using LIVC is attributable to the extended ignition delay allowing greater time for mixing to reduce local fuel-rich regions. The results of Walter et al. [16] are consistent with this interpretation. They also noted that lower in-cylinder temperatures reduced the level of EGR required to meet a target value of NOx but increased HC and CO emissions. The observed effects of early inlet valve closing are consistent with the results presented in this paper. These are longer ignition delays, increased HC and CO, and a reduction in EGR requirement for the same NOx level.

The effects of changes in compression ratio produced by physically changing the bowl dimensions were examined in Ref. [17].

### Table 1
Summary of emissions, fuel economy and exhaust temperature responses for early and late IVC strategies in light and heavy duty diesel engines as reported in recent published studies.

| Authors          | HC&CO | NOx | Soot | FC | $T_{esh}$ |
|------------------|-------|-----|------|----|----------|
| De Ojeda [5]     | O     | +   | +    | +  | +        |
| Ehleskog et al. [6] | O     | +   | +    | O  | +        |
| Wang et al. [7]  | O     | +   | n/a  | O  | +        |
| Benajes et al. [9] | O     | +   | O    | n/a|          |

### Table 2
Ford 2.2 l Common-Rail, DI diesel engine specification.

|          |       |       |       |        |       |
|----------|-------|-------|-------|--------|-------|
| Bore [mm] | 86    |       |       |        |       |
| Stroke [mm] | 94.6  |       |       |        |       |
| Total displacement [litre] | 2.2     |       |       |        |       |
| Compression ratio [-] | 15.5:1 (geometric) |       |       |        |       |
| Injection system | Common-rail, seven-hole Denso centrally mounted solenoid injectors |       |       |        |       |
| Valves | Four valves per cylinder (two intake, two exhaust) |       |       |        |       |
| EGR system | High-pressure loop with gas-to-coolant EGR cooler |       |       |        |       |
| Turbocharger | Variable geometry turbine |       |       |        |       |
| Maximum power [kW] | 103 @ 3500 r/min |       |       |        |       |
| Maximum torque [Nm] | 350 @ 1800–2400 r/min |       |       |        |       |

**Key**

|          |       |       |       |       |       |
|----------|-------|-------|-------|--------|-------|
|           | Deterioration |       |       |        |       |
|           | Small Improvement |       |       |        |       |
|           | No Improvement |       |       |        |       |
|           | Improvement |       |       |        |       |
2. Test facility and methodology

2.1. Hardware

Details of the engine specifications are given in Table 2, and a schematic of the test installation is shown in Fig. 1. The only hardware change made to the engine, originally a Euro V compliant build, was to replace the standard valve train with a system supplied by Schaeffler. This was used to advance the timing of inlet valve closure by discrete amounts relative to the standard cam timing. The Schaeffler system is shown in Fig. 2. The switch between cams is made by shifting sliding elements with sets of cams along splined sleeves on the camshafts. The axial movement is produced by driving a pin into a groove in the sliding element using an actuating solenoid, shown in the detail image in Fig. 2. The axial movement along the camshaft is produced by the pin running in the path of the groove and the rotation of the camshaft. A run-out of the groove returns the pin to its original position, so the change in cam setting is completed in one rotation of the camshaft. The possible changes in inlet valve timing durations are illustrated in Fig. 3. Exhaust valve opening and closing times were fixed, as was inlet valve opening time, at the production settings of the engine. The high compression ratios and bowl in piston design of diesel engines limit the allowed changes in valve motion during the overlap period around top dead centre to avoid valve to piston contact.

The ranges of injection pressure, EGR, SOI (start of injection) and MAP settings covered at BSNOX levels around 1–2 g/kWh at the speeds and loads examined are summarised in Table 3. Within the specified range, rail pressure has higher values at high speeds and loads, and EGR levels are the highest when load is low and speed is high. The highest settings of MAP and SOI are used at high speeds. The MAP values cover a range that was restricted at the upper end by the EGR setting required to achieve a NOX level between 1 and 2 g/kWh and minimise pumping work.

The engine was instrumented with K-type thermocouples to measure coolant, oil and exhaust gas temperatures at various locations; these were recorded using ATI (Accurate Technologies Incorporated) Vision. In-cylinder pressure data was acquired using an uncooled Kistler 6125 piezoelectric pressure transducer fitted in
the glow plug seat and coupled to a Kistler 5011 charge amplifier. The pressure signal from these is normally converted to an absolute value by referencing the transducer signal at inlet BDC (bottom dead centre) to the intake manifold pressure [19]. In the current study, the reference pressure was taken as exhaust manifold pressure at TDC (top dead centre) on the exhaust stroke as this allowed the effect of advancing inlet valve closing on cylinder pressure to be examined. Cylinder pressure was recorded at 0.5 °CA intervals by the shaft encoder and ensemble averaged over 50 cycles. The data were recorded and post-processed using National Instruments LabView. BMEP (brake mean effective pressure) was determined from engine swept volume and measurements of brake torque. To determine ignition delay, heat release variations were computed from a First Law analysis and measured cylinder pressure data. Engine-out NOₓ, CO, HC and CO₂ emissions were measured using a Signal MaxSys 900 exhaust gas analyser while soot concentration was determined from an AVL (Anstalt für Verbrennungskraftmaschinen List) 415s smoke meter. This is a filter-type smoke meter which outputs FSN (Filter Smoke Number). Soot concentration (mg/m³) was determined from this using a correlation developed by MIRA (Motor Industry Research Association) [20]:

\[
\text{Soot concentration} \left( \text{mg/m}^3 \right) = 982 \times \text{FSN} \times 10^{(0.1272 \times \text{FSN} - 1.66)}
\]

Soot concentration is converted to a flow rate using the air and fuel flow rates together with the exhaust gas density. The engine brake power output is then used to calculate the brake specific soot value (g/kWh).

The exhaust gas analysers consist of a series of modules; their specification and the measuring technique employed are shown in Table 4. Fuel flow rate was measured using an AVL 733s fuel balance.

### Table 3
Operating conditions at BSNOx, levels around 1–2 g/kWh.

| Firing cylinders | 4 cylinders Standard IVC | Cylinder disablement | 4 cylinders EIVC |
|------------------|--------------------------|----------------------|------------------|
| Engine speed [r/min] | 1500–2500 | 2–10 | 1–4 | 2–10 |
| Engine load BMEP [bar] | 2 | 1–4 | 2–10 |
| Injection strategy | Single pilot + main (1.5 mm³ pilot and 500μs separation) | | |
| Rail pressure [MPa] | 50–140 | 50–135 | 45–140 |
| EGR [%] | 10–35 | 7–25 | 8–35 |
| SOI [°BTDC] | 0–10 | 1–10 | –3–19 |
| MAP [absolute kPa] | 103–205 | 102–130 | 103–205 |

### Table 4
Exhaust emission analysers.

| Emissions Module type | 7200M, Gas Filter Correlation Infra-Red (GFC IR) | 8000M, Paramagnetic Analyser (PARA) | 7200M, Gas Filter Correlation Infra-Red (GFC IR) | 7100M, Gas Filter Correlation Infra-Red (GFC IR) | 4000VM, Chemiluminescence (CID) | 3000HM, Flame Ionization Detector (FID) |
|-----------------------|-----------------------------------------------|-----------------------------------|-----------------------------------------------|-----------------------------------------------|-----------------------------------|-----------------------------------------------|
| CO₂ (intake)          |                                               |                                  |                                               |                                               |                                  |                                               |
| O₂ (exhaust)          |                                               |                                  |                                               |                                               |                                  |                                               |
| CO₂ (exhaust)         |                                               |                                  |                                               |                                               |                                  |                                               |
| NO (exhaust)          |                                               |                                  |                                               |                                               |                                  |                                               |
| NO₂ (exhaust)         |                                               |                                  |                                               |                                               |                                  |                                               |
| THC (exhaust)         |                                               |                                  |                                               |                                               |                                  |                                               |

### Table 5
Emissions and fuel economy uncertainty analysis; engine operating condition: 2000 r/min, 4 bar BMEP.

| Emissions | BSFC [g/kWh] | BSNOx [g/kWh] | BSCO [g/kWh] | BSHC [g/kWh] | BS soot [%] |
|-----------|--------------|---------------|--------------|--------------|-------------|
| Mean      | 251.59       | 0.76          | 5.23         | 0.205        | 0.036       |
| Std Deviation | 3.71        | 0.02          | 0.21         | 0.006        | 0.008       |
| COV [%]   | 1.48         | 2.72          | 4.06         | 3.03         | 23.45       |

The repeatability of engine performance and emissions measurements was monitored through tests carried out at a standard condition (2000 r/min, 4 bar BMEP) as part of each day’s schedule. Representative values of standard deviations and coefficients of variation are given in Table 5. The results are drawn from 6 repeat tests recorded over a period of 4 consecutive days. Over the duration of the study, no systematic long-term drift or trends attributable to uncontrolled changes in engine or instrumentation performance were observed.

2.2. Basis for comparisons and effect on operating condition

Fuel economy, emissions and exhaust gas temperatures have been compared for the standard and early inlet valve closing timings under fully-warm engine running conditions at engine speeds of 1500, 2000 and 2500 r/min and engine loads of 2, 4 and 6 bar BMEP. For car and light duty commercial vehicles, these engine speeds and loads are representative of the operating conditions covered in the NEDC (New European Drive Cycle) used in
mandatory emissions tests. At each load and speed, the fuel injection strategy comprised of a single pilot injection of fixed quantity (1.5 mm³) followed by a main, with a fixed separation of 500 μs. Although engine speed and brake load define engine demand, operating condition also requires settings of AFR, EGR rate, boost pressure (MAP), SOI and injection pressure. The values of AFR, EGR and MAP are constrained by engine hardware and not independent. These can have a different set of optimum values for baseline and advanced inlet valve closure cases, for example, so data covering the ranges of these noted in Table 3 are presented in plots of specific fuel consumption and emissions.

3. Results

3.1. Effect of EIVC on in-cylinder conditions

Advancing IVC to BBDC (before bottom dead centre) timings leads to lower cylinder pressure during compression as illustrated in Fig. 4. This is due to the reduction in trapped mass. The intake manifold absolute pressure was similar for all IVC settings and changes in cylinder pressure were due solely to IVC timing. The

![Fig. 6. Variation in volumetric and effective compression ratio as a result of advancing IVC timing at 1500 and 2500 r/min, 2 bar BMEP and 103 kPa MAP.](image)

![Fig. 7. In-cylinder and inlet manifold pressure for 23° BBDC IVC advance at 1500 r/min, 2 bar BMEP and 103 kPa MAP. Circle shows point in the compression stroke where in-cylinder pressure rises to the intake manifold pressure value. The effective compression ratio is defined as the ratio of cylinder volume at this point to cylinder volume at TDC.](image)

![Fig. 8. Cylinder Pressure variation at 5° BTDC with compression ratio calculated using cylinder geometric, IVC and effective definitions.](image)

![Fig. 9. Variation in effective compression ratio as a result of advancing IVC timing at 1500 r/min for 2 bar and 10 bar BMEP (103 kPa and 122 kPa MAP).](image)

![Fig. 10. Cylinder pressure variation at 5° BTDC with effective compression ratio at 1500 r/min for 2 bar and 10 bar BMEP (103 kPa and 122 kPa MAP).](image)

irreversibility of the expansion and recompression of the charge at the end of the induction stroke and start of the compression stroke produces a small net rise in-cylinder pressure, as evident in Fig. 5, but this is not enough to compensate for the reduction in mass.

The appropriate definition of compression ratio depends on the use. Three variations are plotted in Fig. 6. Geometric compression ratio, \( CR_g = \frac{(V_c + V_s)}{V_c} \approx 15.5:1 \), is simply the ratio of maximum to
minimum cylinder volume during the cycle and does not vary with valve timing. If compression ratio is calculated using the volume at IVC instead of the maximum cylinder volume, this compression ratio, CR_{IVC}, increases as IVC is advanced towards BDC and then falls, but is still based on geometry. It does not reflect other effects on gas exchange which influence charge state at IVC. More insightful and preferred in the current study, is an effective compression ratio, C_Re. In this case, the maximum cylinder volume is replaced by the volume when cylinder pressure during the compression stroke first reaches intake manifold pressure, as illustrated in Fig. 7. The effective compression ratio most generally represents the effect of geometry and breathing on the ratio of pressure at the end of compression to intake manifold pressure, as indicated in Fig. 8. In Fig. 6, the calculated values of effective compression ratio at an engine speed of 2500 r/min are consistently 1 point lower than at 1500 r/min and the same brake load. With increasing speed, the pressure drop across the intake valves increases and in-cylinder pressure during the intake stroke is lower. As a result in-cylinder pressure rises to match intake manifold pressure later in the compression stroke reducing the effective compression ratio further.

The calculated values of effective compression ratio also depend on MAP setting which rises with increasing brake load. In Fig. 9, the calculated effective compression ratio is lower for the higher MAP setting and more sensitive to the advance in IVC timing. At higher MAP values, cylinder pressure matches the MAP later in the compression stroke. Regardless of the reduction in effective compression ratio, in-cylinder pressure at the end of the compression stroke (Fig. 10) increases with MAP.

Volumetric efficiency, referenced to inlet manifold conditions, follows the same trend as compression ratio and is reduced as IVC timing is advanced. This reflects a reduced cylinder filling due to an earlier closing of the intake valve. However, unlike compression ratio, volumetric efficiency increased with engine load (and consequently MAP).

### 3.2. Engine-out emissions

The comparisons in this section cover a range of brake specific NO_x values including the 1–2 g/kWh which is typical of operating conditions at light engine loads. The results were taken under fully-warm operating conditions with engine-out emissions measured downstream of the turbine and without any after treatment. Advancing IVC timing to 7° ABDC showed a negligible change to the in-cylinder pressure; the changes in fuel economy and emissions were small to negligible. Advancing IVC timing to 3° BBDC and 23° BBDC produced a reduction in in-cylinder pressure of around 3–9 bar at the end of compression and a corresponding reduction in induced charge mass. At a given EGR...
The mass of air induced drops accordingly. To maintain the airflow required to avoid over-rich combustion, EGR must be lowered to reduce the displacement of air in the charge. Generally, at similar NOx values, less EGR was required when running with early inlet valve closure settings and AFR values could be kept similar to the baseline settings. Bulk gas temperatures calculated from in-cylinder pressure measurements were similar during compression, but typically peaked ~100 K higher during combustion due to the reduced thermal capacity of a smaller trapped charge.

Fig. 13. Emissions and exhaust gas temperature — BSNOx trends for standard and early IVC timings at 1500 r/min, 6 bar BMEP.

Fig. 14. Combustion efficiency and total fuel demand — BSNOx trends for standard and early IVC timings at 1500 r/min (top) 2 bar BMEP and (bottom) 6 bar BMEP.
Injection timing was adjusted to compensate for the effects of changes in EGR rate and AFR on combustion phasing. Phasing combustion to minimise BSFC (brake specific fuel consumption) required the location of the 50% mass fraction burned point to be at or close to 7 °CA ATDC (after top dead centre) in most cases. As EGR rate was changed, small adjustments to fuel mass injected were required to account for changes in pumping losses and thermal efficiency whilst maintaining a target brake torque output. Injection pressure was controlled by the ECU (engine control unit) according to the fuelling level demanded.

Fig. 15. Summary responses for a sweep of injection timing, standard v.s. early IVC timing. 1500 r/min, 4 bar BMEP, engine-out BSNOx 1 g/kWh.

Fig. 16. Summary responses for a sweep of injection timing, standard v.s. early IVC timing. 1500 r/min, 4 bar BMEP, engine-out BSNOx 1 g/kWh.
Data for three timings of inlet valve closing are given in Fig. 12 for an engine speed of 1500 r/min and an engine load of 2 bar BMEP. The apparent scatter of points reflects the range of MAP, EGR and SOI covered, not repeatability of results. In Fig. 12, the arrows indicate general trends: reducing boost tends to reduce both BSsoot and BSNOx. Increasing EGR tends to reduce BSNOx and BSsoot rises at an increasing rate. At light loads of 2–4 bar, earlier inlet valve closure produces an increase in ignition delay and significant reductions in soot of around 30%. CO and HC emissions are higher and for early IVC timings, reducing injection pressure brought these back to levels for baseline IVC timings, but soot emissions deteriorated significantly. Reducing injection pressure reduces the likelihood of fuel jet over-penetration and fuel impingement, and the possibility of over-leaning of the mixture [21] [22]. Results for a higher load (6 bar BMEP), presented in Fig. 13, show that the soot reduction from advancing IVC timing is lost at high engine loads, while the CO and HC penalty is lower. The corresponding combustion efficiency and total (pilot plus main) fuel demand results are presented in Fig. 14; the actual fuel injected was not recorded but in other data sets demanded and delivered fuel differences were small. Generally, fuel economy is unaffected until the most advanced IVC setting is reached when fuel consumption is penalised by poor combustion efficiency and the increase in CO and HC emissions.

The response of emissions levels to advancing IVC timing is in line with the observations of other researchers working on diesel engines with early and late IVC [4], or investigating the effect of compression ratio changes [17]: when the soot-NOx trade-off is improved, there is an increase in CO and HC emissions, particularly at light engine loads. In the current study, at a given speed, load and IVC setting, a range of combinations of EGR, AFR and SOI give similar values of NOx. The effect of injection timing is strong, as shown by the results given in Figs. 15 and 16 but differences in emissions associated with changes in IVC are maintained almost constant over a wide range of injection timing. At light loads, when available boost pressure is low, available settings of EGR and AFR were limited by their coupling through the displacement effect on proportions in the intake charge. When EGR was increased to
compensate for the effect of advancing SOI on NOx, the displacement of air in the intake charge caused a drop in AFR.

A comparison of ignition delays at 1500 r/min and 4 bar BMEP (Fig. 17) shows these are consistently between 0.5 and 1° CA longer for early IVC than for the standard IVC timing; the longer delay allows more time for mixing to the advantage of reducing soot formation. This is also supported by Ref. [23] who report that small increases in ignition delay are sufficient to cause measurable changes in soot. The extension of ignition delay is largest for the most advanced cam setting of 23° BBDC, when ignition delay varies between 2 and 3° CA. The increase has been associated with the reduction in in-cylinder pressure produced by EIVC. The results obtained are consistent with those of Crua et al. [24], who attributed the longer delay to poorer droplet break-up. Dec and Espey [25] support this conclusion, but add that the chemical delay is also increased as well as the physical delay.

3.3. Early inlet valve closing or cylinder disablement?

The benefits of advancing inlet valve closure by up to 60° CA or deactivating two cylinders by disabling intake and exhaust valve lift are similarly limited to light loads and diminish with increasing speed. Results taken on the same engine for three cases: standard valve timing (37° ABDC) and four cylinder operation, standard valve timing with two cylinders disabled, and early inlet valve closing (23° BBDC) and four cylinder operation, are compared in Figs. 18–21. EIVC timing resulted in a small fuel economy penalty independent of load. Cylinder disablement caused fuel economy to deteriorate at high loads. Both EIVC timing and disabling cylinders reduce soot emissions at light load. Cylinder deactivation produced an increase in soot at high loads. The main difference in emissions was in the responses of CO and HC. Whereas cylinder deactivation produced a substantial reduction in HC and CO, advancing IVC produced a significant increase. For nominally the same brake torque, when operating with two cylinders disabled the fuel delivered to the two firing cylinders is approximately doubled. This raises in-cylinder bulk gas temperatures and surface temperatures, improving mixture preparation and the oxidation of HC and CO. An increase in injection pressure also reduces soot while maintaining the advantage in CO and HC emissions. Within the range of IVC timings of up to 60° CA earlier than the standard IVC timing, advancing IVC was less effective than cylinder deactivation and compromised by opposite trends in soot and HC and CO for a target NOx value. Furthermore, cylinder deactivation produced higher exhaust gas temperatures as shown in Fig. 18, which is beneficial to achieving early light-off of the after treatment system and a higher final, steady state operating temperature. This is attributed to the combination of lower heat losses from burning in half the surface area and the increase in fuel burned per unit mass of exhaust flow. The increase in exhaust temperature with IVC is also the result of a reduction in total trapped charge but is not as pronounced as with cylinder disablement. A summary comparing responses to advancing IVC and cylinder deactivation is given in Table 6.

4. Conclusions

The effects of EIVC and cylinder disablement on fuel economy and emissions have been examined using the same engine operating over a range of speeds and loads typical of urban driving conditions. Intake valve timings were advanced by up to 60° CA, from a value of 37° ABDC used in production for the test engine, to 23° BBDC. When operating with cylinder disablement, the standard IVC setting was used and the two inboard cylinders were disabled. In both cases, the most benefits of EIVC and cylinder disablement were recorded at low engine speeds and light loads. Both EIVC and cylinder disablement raised exhaust gas temperature.

The effects of EIVC were only significant for advances greater than 30° CA. At a nominally fixed level of brake specific NOx emissions of 1–2 g/kWh, soot levels were reduced and exhaust gas temperature raised. However, CO and HC emissions increased and fuel economy deteriorated.

The effects of cylinder disablement at the same level of brake specific NOx were sensitive to engine brake load. Soot and CO emissions were reduced at light load but raised at higher loads; fuel economy deteriorated significantly with increasing load. HC was reduced at all operating conditions.

At light engine loads, cylinder disablement was the more effective strategy for reducing soot, CO and HC at a given NOx value. It also produced the more significant increase in exhaust gas temperature, which is more advantageous to promoting early light-off and maintaining operating temperature of the after treatment system. The change from improvement to deterioration in emissions as load increases is attributable to the lack of boost pressure available when operating on two cylinders, which limits the induction of charge air. This might be overcome if the air charging system was modified to make higher boost pressures available at higher loads.

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