Fuel Economy Analysis of Part-Load Variable Camshaft Timing Strategies in Two Modern Small-Capacity Spark Ignition Engines

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Highlights

- Valve timing strategies are explored at part-load in two small-capacity SI engines
- Simplified graphical fuel-economy VCT strategies are provided for the two engines
- Engine design specifics exert a profound influence upon optimal valve timings
- The theoretical best strategy determines BSFC reduction above 8% for the PFI engine
- Maximum BSFC reduction was about 5% for the more efficient GDI platform

ABSTRACT

Variable Camshaft Timing strategies have been investigated at part-load operating conditions in two 3-cylinder, 1.0-litre, Spark Ignition engines. The two small-size engines are different variants of the same 4-valve/cylinder, pent-roof design platform. The first engine is naturally aspirated, port fuel injection and features high nominal compression ratio of 12:1. The second one is the turbo-charged, direct injection version, featuring lower compression ratio of 10:1. The aim of the investigation has been to identify optimal camshaft timing strategies which maximise engine thermal efficiency through improvements in brake specific fuel consumption at fixed engine load.

The results of the investigation show that the two engines demonstrate consistent thermal efficiency response to valve timing changes in the low and mid part-load envelope, up to a load of 4 bar BMEP. At the lower engine loads investigated, reduced intake valve opening advance limits the hot burned gas internal recirculation, while increasingly retarded exhaust valve opening timing favours engine efficiency through greater effective expansion ratio. At mid load (4 bar BMEP), a degree of intake advance becomes beneficial, owing mostly to the associated intake de-throttling. In the upper part-load domain, for engine load of 5 bar BMEP and above, the differences between the two engines determine very different efficiency response to the valve timing setting. The lower compression ratio engine continues to benefit from advanced intake valve timing, with a moderate degree of exhaust timing retard, which minimises the exhaust blow-down losses. The higher compression ratio engine is knock-limited, forcing the valve timing strategy towards regions of lower intake advance and lower hot gas recirculation. The theoretical best valve timing strategy determined peak fuel economy improvements in excess of 8% for the port fuel injection engine; the peak improvement was 5% for the more efficient direct injection engine platform.

1. INTRODUCTION

In 2011, the International Energy Agency, in conjunction with the Global Fuel Economy Initiative, published a report which addresses their long term goal as a 50% reduction in global average emissions by the year 2030 [1]. Policy changes, enhanced technologies, revised fuels and reduced vehicle and engine size are indicated as possible ways to improving vehicles fuel economy, which in turn reduces pollutant emissions. In the last three decades, advanced technologies such as direct injection, turbo-charging and Variable Camshaft Timing (VCT), have contributed significantly to the evolution of engine design. Combinations of these technologies, along with engine down-sizing, have simultaneously delivered positive effects on fuel consumption and emissions [2, 3], as well as on driving pleasure [4].

Today's engine technical developments are primarily focused on improved part-load driving performance and fuel consumption with stoichiometric operation [5, 6]. One way of targeting these goals is through the implementation of VCT systems. The evaluation of the extent to which the VCT

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60 technology impacts on fuel savings, especially within the framework of current down-sized engines, is
61 of paramount importance. This evaluation, which enables the relative importance of the VCT system
62 to be assessed, is the primary objective of the present study.
63
64 **1.1 Variable Camshaft Timing**
65 In engines with fixed camshaft timing, the exhaust valve is normally closed 15 to 30 Crank Angle (CA)
66 degrees After Top Dead Centre (ATDC), whereas the intake valve is opened 10 to 20 CA degrees
67 Before Top Dead Centre (BTDC). These timings represent a compromise between several functions,
68 intended to provide sufficient valve overlap duration for good cylinder scavenging, but avoiding at the
69 same time excessive backflow from the exhaust port. By contrast, the flexibility associated to the VCT
70 technology has the potential to improve fuel economy, performance, as well as emission levels of
71 gasoline engines over wide ranges of running conditions [7]. The advantage of wide valve overlap
72 occurs especially at higher engine load and increasing engine speed, owing to improved volumetric
73 efficiency. At low engine speed, wide overlaps are generally detrimental, yielding higher residual gas
74 fraction [8] which may significantly degrade the combustion stability. Improvements in Brake Specific
75 Fuel Consumption (BSFC) of up to 10%, as a result of early Intake Valve Opening (IVO) strategies,
76 have been reported in recent studies [9, 10]; these explain the observed changes referring to the
77 displacement of fresh mixture with residuals during valve overlap, ultimately reducing the *need for*
78 *throttling* [6]. If the intake cam profile is fixed, the effects of IVO are inevitably linked to those of Intake
79 Valve Closing (IVC) timing. Because of fresh mixture backflow at low engine speed, and inertia of the
80 incoming gas flow at higher speed [11], the IVC timing shows significant effects on cylinder trapped
81 mass, and hence on the resulting pumping (intake throttling) losses. Sizeable improvements in
82 thermal efficiency have been observed with early IVC strategies, owing to greater effective
83 compression ratio [12, 13].
84
85 In traditional, fixed camshaft timing engines, the exhaust valve timing setting is a compromise
86 between improved exhaust blow-down (achieved with early Exhaust Valve Opening, EVO) and
87 greater work per cycle, associated to greater effective expansion ratio (late EVO). The influence of
88 exhaust VCT on Internal Exhaust Gas Recirculation (I-EGR) is another very relevant factor. At part-
89 load, the beneficial influence of late Exhaust Valve Closing (EVC) on cylinder scavenging, normally
90 seen at high engine speed, tends to reduce; as speed and load are reduced, retarded EVC enables
91 increasingly larger backflow from the exhaust to the intake system, which increases the fresh charge
92 burned fraction. A similar though generally smaller effect is associated to early EVC taking place
93 before Top Dead Centre (TDC), due to trapping large amounts of burned gases within the cylinder at
94 the end of the exhaust stroke.
95
96 As of today, the exhaust-only and intake-only variable camshaft timing schedules remain the most
97 practical and cost-effective in use. Typically, at part-load running conditions, intake-only strategies
98 entail significantly advanced intake valve timing and asymmetric valve overlaps extended well into the
99 exhaust stroke. In terms of benefits on fuel consumption, Leone et Al. [14] found that due to increased
100 charge dilution, a higher intake manifold pressure is required to maintain a given load, which in turn
101 reduces the pumping work. Similar reasoning, along with increased expansion work, justifies the use
102 of retarded exhaust valve timing at part-load, in the case of exhaust-only strategies. The Dual or Twin
103 Independent VCT systems, where the two camshafts can be phased continuously and independently
104 of one another, would synthesize the benefits of both exhaust and intake-only schedules; however,
105 DI-VCT mechanisms are bulkier, more expensive and more complicated to operate. Kramer and
106 Philips [15] found that, using a DI-VCT strategy, the fuel economy was improved due to the following
107 three, concurrent phenomena: de-throttling due to late IVC (backflow of mixture into the intake
108 manifold); de-throttling due to late EVC (internal Exhaust Gas Recirculation); increased effective
109 expansion ratio due to late EVO (more work extracted in each engine cycle).
110
111 **1.2 Engine Down-sizing**
112 Internal combustion engine down-sizing is becoming increasingly common because of the inherent
113 advantages of high specific power and torque, lower fuel consumption and emissions, as well as
114 improved driveability [4, 16]. Due to lower displacement volume, the down-sized engine tends to work
115 at higher load with reduced throttling/pumping losses [17]. The potential reduction in engine fuel
116 consumption, averaged upon the New European Drive Cycle (NEDC), has been shown to increase
117 exponentially with the factor of down-sizing [18]. High specific outputs are often achieved by means of
118 turbo-charging. Currently, single-stage mechanical turbo-charging is the standard method, but more
119 efficient systems such as multi-stage mechanical or electric super-chargers are being developed, with

120 sizeable improvements expected in torque output in the low engine speed range [16, 19]. Large low-
121 end torque, which naturally leads to improved fuel economy, is a distinctive characteristic of down-
122 sized gasoline engines featuring a combination of turbo-charger and VCT technologies [20].
123
124 Improvements of part-load specific fuel consumption from a turbo-charged gasoline engine with down-
125 sizing factor of 40%, have been found to be of the order of 12% over the NEDC, when compared to a
126 traditional naturally aspirated engine [21]. In the recent collaborative Ultra-Boost project, Turner et Al.
127 [22] found that an extreme down-sizing factor of 60%, applied to a naturally aspirated, 8-cylinder
128 engine, resulted in 23% reduction in part-load specific fuel consumption over the same drive-cycle.
129 The purposely developed engine platform incorporated Direct Injection (DI), Variable Camshaft
130 Timing and multi-stage forced induction technologies. The extent to which each single system
131 contributed towards the overall fuel economy improvement of 23% was not reported.
132
133 One significant efficiency limitation of down-sized engines is the knock tendency, associated with high
134 specific power output. Due to its marked effect on cylinder charge cooling, the introduction of fuel DI
135 minimises the risk of knock, enabling the implementation of comparatively higher compression ratio,
136 with measurable benefits in terms of engine thermal efficiency [23, 24]. Baêta et Al. [25] suggest the
137 use of pure ethanol fuel in direct injection spark ignition engines may enable a *more aggressive*
138 approach to down-sizing. The maximum part-load brake thermal efficiency recorded in their 1.4 litre
139 twin-stage-compressor VCT prototype engine was 44%. Fuel direct injection also improves volumetric
140 efficiency across all operating conditions, and this effect is amplified by parallel implementation of
141 VCT technology. Shinagawa et Al. [20] have shown how the tuning of the exhaust valve timing and
142 opening duration may lead to reduced exhaust gases blow-back pulsations, and hence improved
143 volumetric efficiency at low speed.
144
145 The availability of advanced, fast-response VCT systems has in fact opened up several different
146 potential avenues for improving the fuel economy of gasoline engines, including the implementation
147 and/or optimisation of alternative engine cycles not viable until recently. Among others, the Miller
148 cycle has potentials for improved fuel economy across a wide engine load range, both by reducing
149 pumping work and mitigating knock tendency [26, 27]. Very early or very late intake valve closing
150 timings can be used to control the amount of intake air, with a corresponding reduction of the effective
151 compression ratio. Li et Al. [26] reported an 11% improvement in net Indicated Specific Fuel
152 Consumption when early Miller cycle IVC timing was used at low speed and variable load between
153 4.5 and 9.0 bar IMEP. They suggest the optimisation of early IVC timing and intake cam profile brings
154 a concrete prospect for further improvement of gasoline turbo-charged engines.
155
156 The present study is specifically concerned with the influence of Dual Independent Variable Camshaft
157 Timing technology upon the fuel economy of modern small-size gasoline engines operating upon
158 traditional spark ignition cycles. Of particular interest have been the complex relationships which
159 establish between valve timing and BSFC, as the engine operating conditions are varied across
160 relevant points of the part-load envelope. The importance of the present work lies in that it
161 investigates and compares the results of two engine platforms of the same size and geometry, but
162 featuring different injection technology and different compression ratio. While most recent studies [20,
163 21, 22] report overall efficiency gains from a range of combined technology, this work aims to explore
164 the specific extent to which VCT technology can effectively improve the fuel economy of state-of-the-
165 art gasoline engines. Importantly, in this evaluation process the limitations of current VCT technology
166 remain identified. Another novelty of this work is that the comparison between the two engines unveils
167 how the selection of the optimal valve timing is intimately correlated to the effects of compression ratio
168 as well as those of fuel injection.
169
170 **2. METHODOLOGY**
171 Variable camshaft timing strategies for fuel economy are investigated in two modern small-capacity
172 test engines. Test engine A is a naturally aspirated, port fuel injection, 1.0 litre, 3-cylinder, 4-stroke
173 engine, with high nominal compression ratio of 12:1. Test engine B is the turbo-charged, direct-
174 injection variant of the same engine, bearing a smaller nominal compression ratio of 10:1. According
175 to the prevailing classification, engine B is a down-sized engine. The VCT mechanism used in test
176 engine A allows independent intake and exhaust valve phase variation, whereas the valve lifting
177 profiles remained fixed. The system allows phase variation over a range of 45 CA deg for both cams,
178 with a resulting maximum valve overlap of 90 CA deg. The VCT mechanism used in test engine B is
179 essentially the same; the only difference is a slightly shorter intake cam variability. The technical

specifications of the two test engines and associated VCT system operation are given in Table 1. The basic schematic of the variable cam phaser and its main components is presented in Figure 1 [28]. Figure 2 shows the schematic of the test room including engine and dynamometer assembly, test rig control and data acquisition systems.

	Engine A	Engine B
Total Displacement (cm^3)	999	999
Stroke (mm)	82	82
Nominal Compression Ratio	12:1	10:1
Connecting Rod Length (mm)	138.7	137
Engine Type	Naturally Aspirated, 3-Cylinder In-line, DOHC	Turbo Charged, 3-Cylinder In-line, DOHC
Combustion Chamber	4-Valve, Central Spark Plug, Pent-Roof Design	4-Valve, Central Spark Plug, Pent-Roof Design
Engine Cycle	4-Stroke Spark Ignition	4-Stroke Spark Ignition
Fuel Injection System	Port Fuel Injection	Direct Injection Common Rail; Centrally-Mounted Injector (Spray-Guided System)
Test Programme Fuel	E22 (78% Gasoline, 22% Ethanol by Volume)	95 RON Gasoline
Maximum Rated Power	62.5 kW at 6300 rev/min	88 kW at 6000 rev/min
Maximum Rated Torque	105 Nm at 4500 rev/min	170 Nm, 1500-4000 rev/min
IVO Pin-Lock Position (CA deg ATDC)	0 (TDC)	+ 5
IVO Variability (CA deg)	45	40
Inlet Valve Opening Duration (CA deg)	240	228
EVC Pin-Lock Position (CA deg BTDC)	0 (TDC)	0 (TDC)
EVC Variability (CA deg)	45	45
Exhaust Valve Opening Duration (CA deg)	236	228

Table 1. Technical specifications of test engines A and B
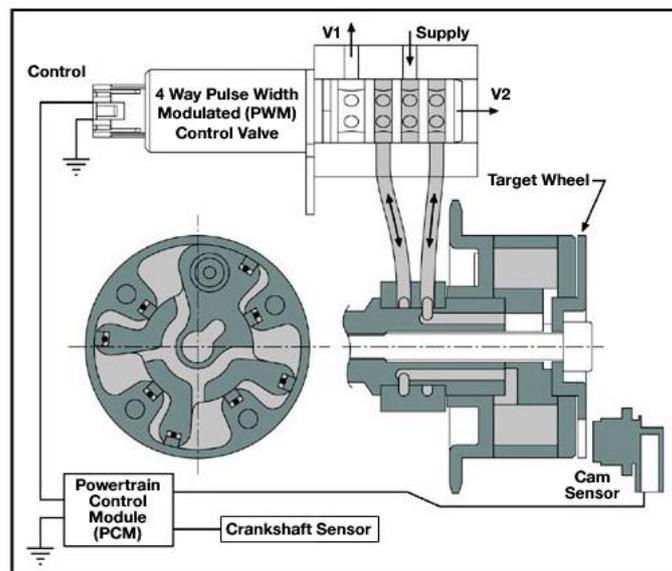
Figure 1. Variable cam phaser system used for intake and exhaust valve timing control; includes oil control valve, control module, crank and cam sensors [24]
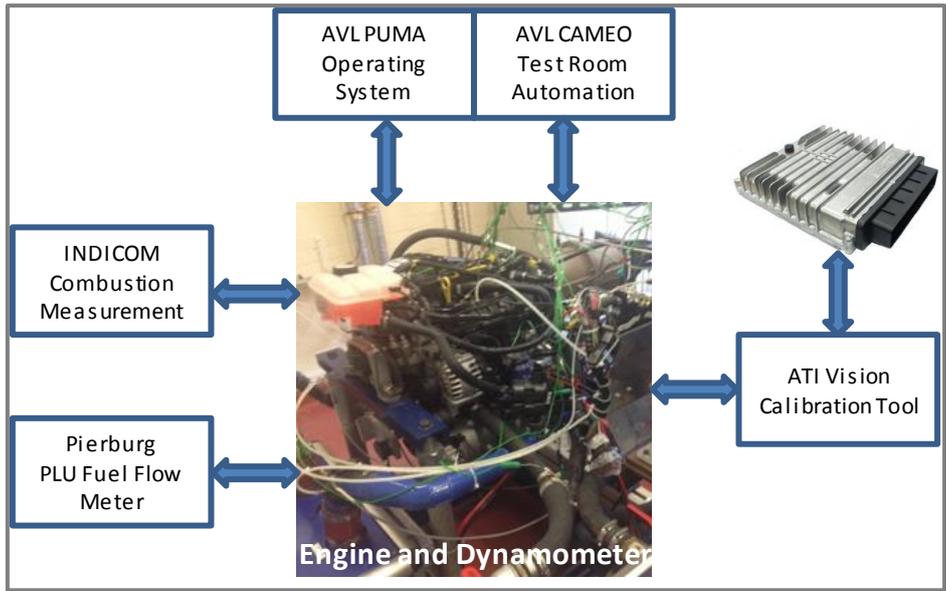
Figure 2. Schematic of engine test room including rig control and data acquisition systems.

The two engines were tested in steady-state conditions, over relevant portions of the part-load operating envelope, so as to represent typical urban and cruise driving conditions. As shown in Figure 3, 13 speed and load combinations were selected for engine A, based on emissions and fuel economy residency information relevant to the NEDC and Federal Test Procedure (FTP) drive cycles [29]. For engine B, 14 points were selected. Engine speed and load were varied in the range 1500 to 4000 rev/min, and in the range 1 to 7.5 bar Brake Mean Effective Pressure (BMEP), respectively. At each engine speed, dynamometer torque variation as a result of the VCT operation was annulled via throttle adjustment.
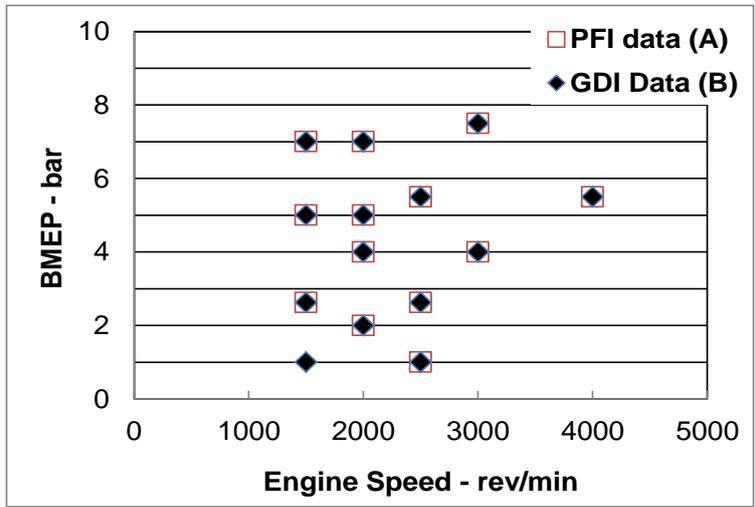
Figure 3. Part-load operating conditions investigated.

Engine A was tested in stoichiometric conditions using E22 fuel blend (78/22 gasoline/ethanol volume ratio). The gasoline in the blend was pump-grade 95 RON fuel. The blend stoichiometric AFR was 13.32. The proportion of oxygenate compounds, enabling the use of higher compression ratio, was dictated by the specific market destination for engine A. Engine B was also operated in stoichiometric conditions, employing early direct fuel injections in the intake stroke to ensure theoretically homogeneous cylinder gas mixture at the time of ignition. In this case, the fuel used was straight 95 RON gasoline. The whole test programme for the two engines was carried out in fully-warm, steady-state conditions, and the ignition timing advance was adjusted to Maximum Brake Torque (MBT) timing for each speed and load combination. A MBT-Limited rule was applied to some higher load cases to avoid the inception of knock. Here, the ignition timing was retarded of 2 CA degrees from the location where knock was first detected. At each test point, fundamental cycle-average quantities were recorded to enable the VCT schedule optimisation. Improvements in thermal efficiency as a

5

217 result of the VCT operation were signalled via decreased Brake Specific Fuel Consumption. Parallel
218 measurements of CA-resolved in-cylinder pressure were taken on one cylinder to enable the analysis
219 of combustion and its variability. A Kistler 6125 piezo-electric pressure sensor was used to acquire in-
220 cylinder pressure data. The measurement resolution was variable: 0.2 CA degree during the rapid
221 burning angle; 1 CA degree along the remainder of the engine cycle [30]. In order to maximise the
222 beneficial effects of variable valve timing without compromising combustion stability and hence
223 vehicle driveability, the fuel economy optimisation process was conditioned to the Coefficient Of
224 Variability (COV) of Gross Mean Effective Pressure (GMEP) being always smaller or equal to 5% [31].
225
226 At each speed and load point, engine data were recorded at 5 CA degree regular increments of the
227 exhaust and intake valve timing, following a classic/uniform grid of points. The behaviour of the data
228 was summarised by using a cubic polynomial model, yielding response surfaces for ease of
229 visualisation. This approach carried the additional benefit of reducing the noise in the experimental
230 measurements [32, 33]. The full range of variability of intake and exhaust valve timing gave a 10x10
231 grid of points. Nevertheless, in some engine conditions smaller experimental grids were sufficient to
232 identify the optimal VCT setting for best thermal efficiency. This approach minimised the need for
233 testing, avoiding expensive experiments especially in regions of high cyclic variability.
234
235 **3. RESULTS AND DISCUSSION**
236 The experimental results regarding the BSFC response to variations of the intake and exhaust VCT
237 setting are presented and discussed in this section. Other experimental and calculated data such as
238 cycle-by-cycle variability, intake manifold pressure and CA location of 50% Mass Fraction Burned
239 (MFB50) are used to aid the discussion. Contour-plot type maps have been used for the analysis of
240 the BSFC response metrics. The experimental records of BSFC are interpolated, as described in
241 Section 2, over a uniform grid of IVO and EVC timing points. Same approach is used for the COV of
242 GMEP. For ease of visualisation, COV data above 8% have been given a single-colour pattern in the
243 relevant contour-plot maps. In the following sections, the results are conveniently arranged by fixed
244 engine load intervals and increasing engine speed.
245
246 **3.1 Influence of VCT setting in the low part-load region**
247 Figures 4 and 5 show the BSFC and COV response metrics for the two test engines at a load of 1 bar
248 BMEP and increasing engine speed. For the PFI engine (engine A), only data at 2500 rev/min were
249 available. At the lowest speed and load investigated, the two engines show consistent response to
250 changes of the VCT setting, with minimum BSFC gained for late intake opening (close to TDC) and
251 limited degree of EVC retard (20 to 25 CA deg ATDC). As clearly suggested by the COV data, a
252 strong intake-to-exhaust pressure differential enables considerable internal burned gas recirculation
253 and high levels of cylinder charge dilution; virtually, any valve overlap duration in excess of 30 CA deg
254 results in degraded combustion and worsened fuel economy. As early IVO and late EVC are reported
255 to exert similar influence upon charge dilution at low engine load [6], the experimental data suggest
256 the effects of increased effective expansion ratio (more work extracted per cycle) would be more
257 significant than those associated to increased effective compression ratio. As the engine speed is
258 increased at low load for engine B (Figure 5), the overlap duration reduces in real time, limiting the
259 amount of internal recirculation; here combustion regains stability and further retarding the EVC timing
260 becomes beneficial. At 2500 rev/min, an EVC setting of 35 CA deg ATDC ensures minimum BSFC.
261
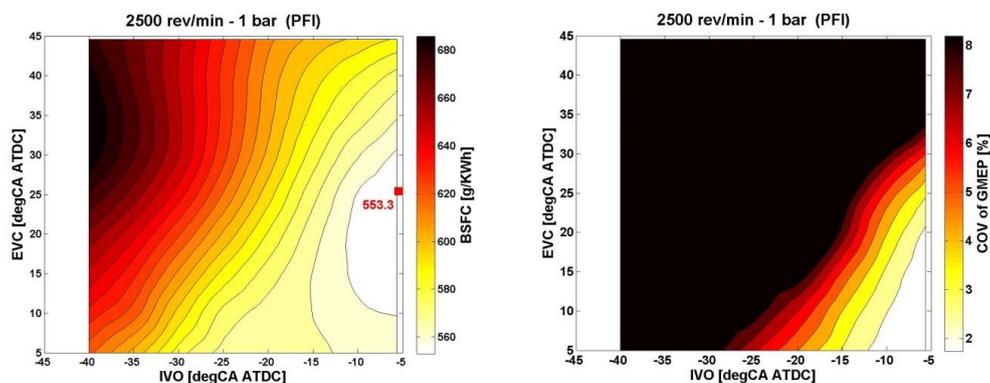
262 **Figure 4.** Contour-plots of BSFC (left) and COV of GMEP (right), as functions of IVO and EVC timing, for the PFI engine (A).
263 Operating conditions: engine load of 1 bar (BMEP) and engine speed of 2500 rev/min.

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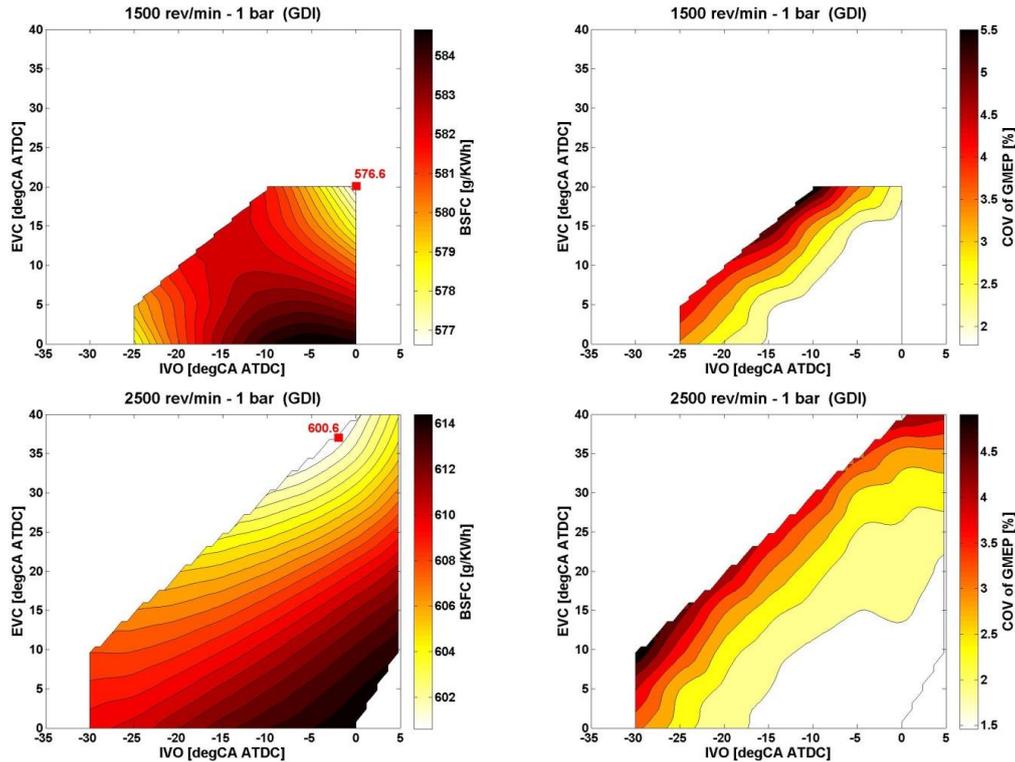
Figure 5. Contour-plots of BSFC (left) and COV of GMEP (right), as functions of IVO and EVC timing, for the GDI engine (B). Operating conditions: engine load of 1 bar (BMEP) and engine speed of 1500 to 2500 rev/min.

The BSFC response metrics for engine A and B, at load of 2 to 2.6 bar BMEP and increasing speed, are presented in Figures 6 and 7. The thermal efficiency of engine A, as indicated by the BSFC data, appears to be heavily influenced by combustion stability. Figure 4 shows that for engine speed up to 2000 rev/min, the unsuitably high cycle-by-cycle variability hides the effect of VCT on specific fuel consumption. The calculated BSFC minima require early IVO timing (25 to 35 CA deg BTDC), along with a moderate delay of the exhaust VCT setting (EVC of 5 to 10 CA deg ATDC). As engine speed rises to 2500 rev/min, engine A acquires increased stability and a much wider range of VCT variation becomes available; the influence of the VCT setting now assumes similar traits as the 1 bar BMEP case. Best engine thermal efficiency is associated to extremely retarded exhaust VCT setting (45 CA deg ATDC), in combination with a moderate degree of intake VCT advance (15 CA deg BTDC).

Figure 7 shows that at an engine load of 2 to 2.6 bar BMEP, the BSFC response metrics for engine B are significantly similar to the 1 bar BMEP case. At the lower engine speed investigated (1500 rev/min), IVO for best efficiency continues to be located close to TDC whereas, owing to the inherently reduced dilution, optimal exhaust VCT setting is retarded to 35 or 40 CA deg ATDC. Any changes to the intake VCT setting determine an unacceptable drop in combustion stability, with the COV quickly approaching the 5% threshold. As the engine speed increases, earlier intake VCT timings become affordable as expected. Remarkably, Figure 7 shows that, at 2500 rev/min and 2.6 bar BMEP, any IVO/EVC combination giving valve overlap duration between 50 and 55 CA deg ensures about 3.0% BSFC reduction compared to the reference setting (IVO= 5 CA deg BTDC; EVC= 5 CA deg ATDC). Any larger overlap combination becomes detrimental, most likely due to excessive cylinder charge dilution. At these running conditions, the optimal VCT setting is somewhat biased towards earlier IVO, rather than later EVC. Such BSFC response, typical of the mid part-load envelope as shown in the next section, suggests earlier IVO/IVC setting helps preventing excessive fresh-charge back-flow, ultimately increasing the volumetric efficiency.
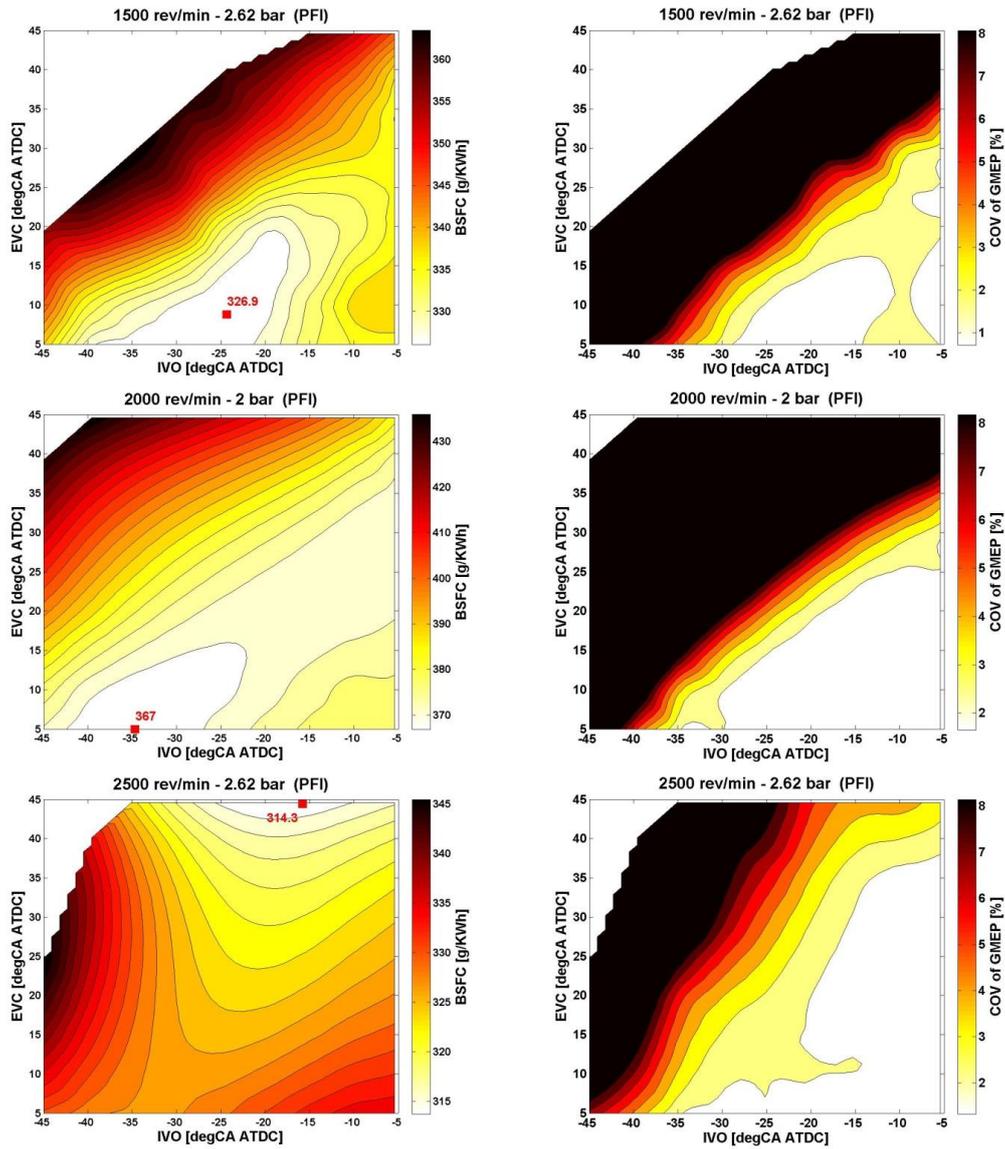
294 **Figure 6.** Contour-plots of BSFC (left) and COV of GMEP (right), as functions of IVO and EVC timing, for the PFI engine (A).
295 Operating conditions: engine load of 2 to 2.62 bar (BMEP) and engine speed of 1500 to 2500 rev/min.
296

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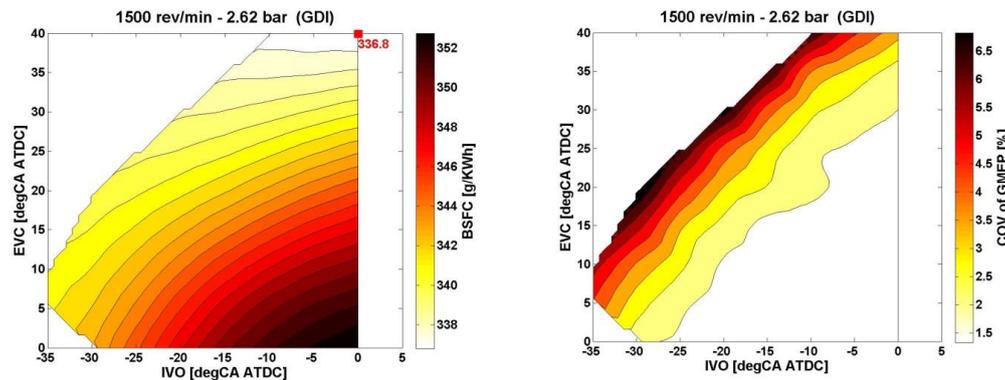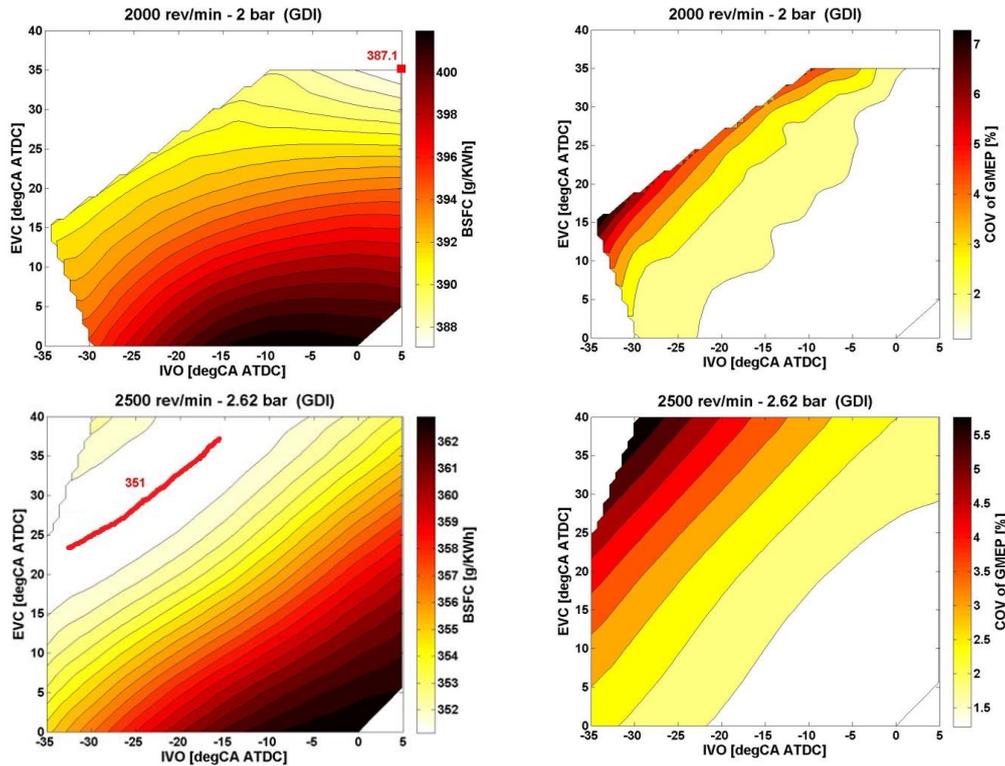
Figure 7. Contour-plots of BSFC (left) and COV of GMEP (right), as functions of IVO and EVC timing, for the GDI engine (B). Operating conditions: engine load of 2 to 2.62 bar (BMEP) and engine speed of 1500 to 2500 rev/min.

Figure 8 shows the variation of intake manifold pressure (MAP), MFB50 and, again, COV of GMEP, as a function of the VCT setting, at 2000 rev/min and 2 bar BMEP. These operating conditions were selected as they are representative of the engines' behaviour in the low part-load region. For completeness, data include not only experimental measurements, but also interpolated values to cover the full range of intake and exhaust VCT variation. Supporting data on Pumping Mean Effective Pressure (PMEP) and Indicated engine Torque (IT) are reported in Appendix 1.

Owing to the inherent increase in internal hot gas recirculation, and the consequent need for intake de-throttling, the intake manifold pressure grows with both the degree of IVO advance and the degree of EVC retard. The highest level of intake pressure is consistently associated to the widest symmetrical valve overlap duration. As evident, both MFB50 and COV of GMEP vary in a similar fashion as the intake pressure. Engine A suffers from volumetric efficiency losses, most likely due to the process of port fuel injection; by contrast, the process of direct injection in engine B favours cylinder filling, owing to cylinder charge cooling. At given VCT setting, engine A requires greater manifold pressure to balance a fixed dynamometer torque. In spite of this, the combustion process in engine A quickly degrades with increasing overlap duration, becoming slower and longer (hence, more unstable) as indicated by the MFB50 contour-plots. The latest generation, direct injection system used in engine B ensures a more precise fuel metering per cycle/cylinder, contributing towards increased stability. Figure 8 shows that, for given VCT setting, engine A is much less stable than engine B; in turn, this carries dramatic influence on the selection of an optimal VCT setting for fuel economy.
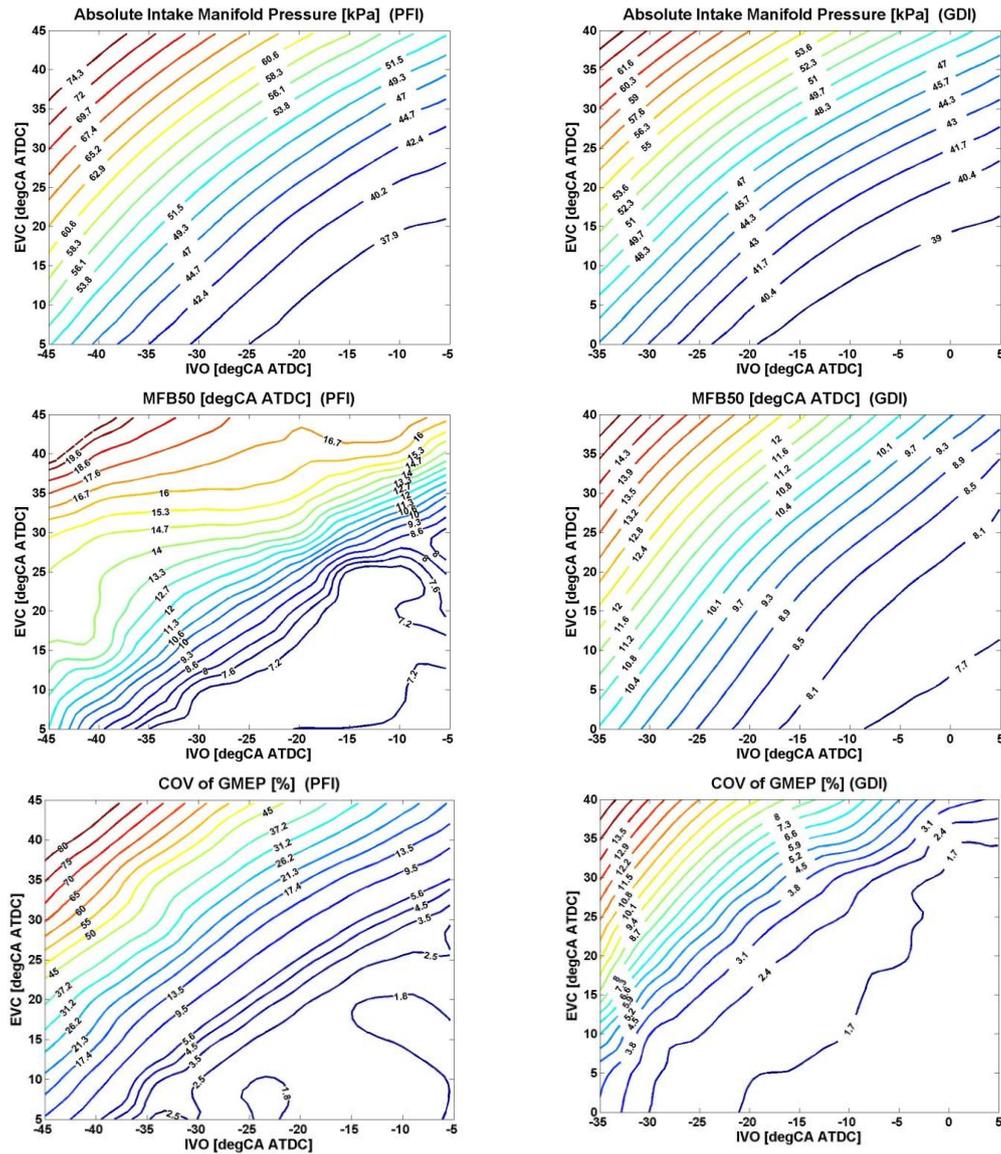
Figure 8. Contour-plots of intake manifold pressure, location of 50% MFB and COV of GMEP as a function of the VCT setting, for engine A (left) and engine B (right). Operating conditions: engine load of 2 bar (BMEP) and engine speed of 2000 rev/min

3.2 Influence of VCT setting in the mid part-load region

Figures 9 and 10 show BSFC as a function of intake and exhaust valve timing, at 4 bar BMEP and increasing engine speed. With the exception of the top left corner of the 2000 rev/min COV map in Figure 9, the measured cyclic variability is well within the 5% threshold, enabling a much more consistent behaviour across the two engine platforms. Experimental measurements of internal EGR were not performed; nevertheless, at an engine load of 4 bar BMEP and above, both amount and variation of in-cylinder charge dilution are expected to be rather limited [8, 9, 14]. This would enable the two engines to virtually exploit the full spectra of intake and exhaust VCT variation, with potential benefits associated to both early IVC/IVO (greater effective compression ratio; lower pumping losses), and late EVO/EVC (greater effective expansion ratio; again, lower pumping losses). The BSFC maps for engine A (Figure 9) show that while early IVO is always beneficial, EVC timing in excess of about 25 CA deg ATDC starts showing a detrimental effect. While stability deterioration may have constrained the EVC setting in the lower speed case, late EVC in the higher speed case is likely to suffer from a less effective blow-down process. The brake thermal efficiency of engine B, as indicated by the BSFC data in Figure 10, grows in a similar fashion with both EVC retard and IVO advance, reaching its peak for durations of 65 to 70 CA deg. The experimental data suggest at 4 bar BMEP,

341 intake VCT advance is somewhat more valuable than exhaust VCT retard due to the inherently less
342 effective blow-down process.
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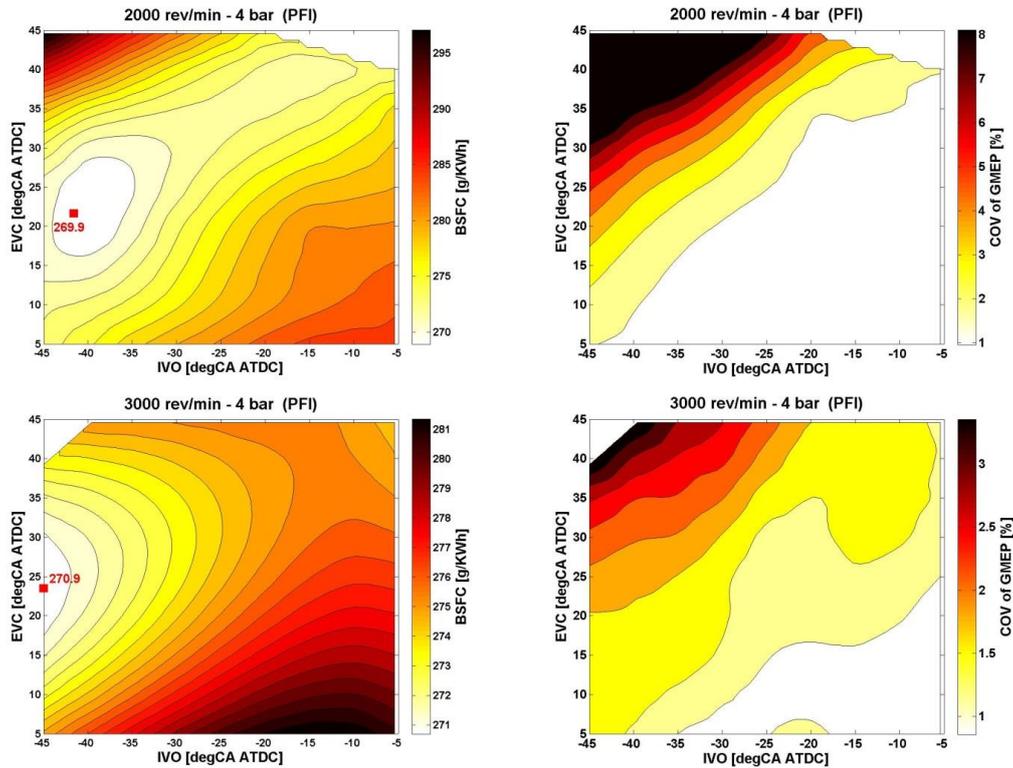
344 **Figure 9.** Contour-plots of BSFC (left) and COV of GMEP (right), as functions of IVO and EVC timing, for the PFI engine (A).
345 Operating conditions: engine load of 4 bar (BMEP) and engine speed of 2000 to 3000 rev/min.
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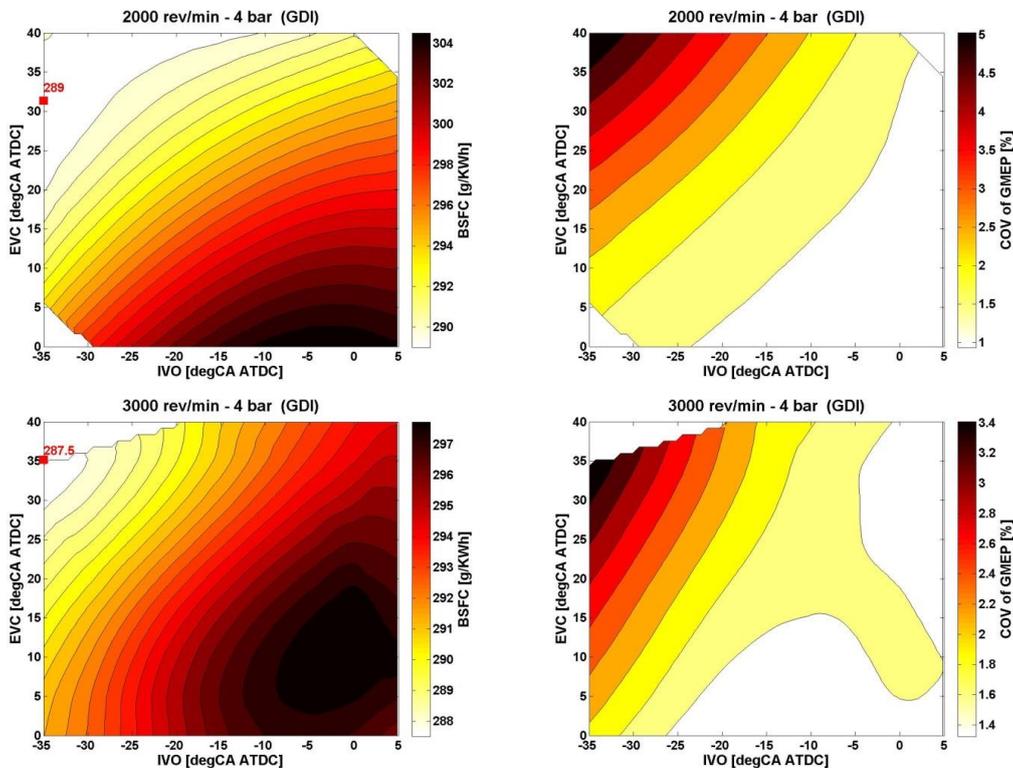
347 **Figure 10.** Contour-plots of BSFC (left) and COV of GMEP (right), as functions of IVO and EVC timing, for the GDI engine (B).
348 Operating conditions: engine load of 4 bar (BMEP) and engine speed of 2000 to 3000 rev/min.
349

350 **3.3 Influence of VCT setting in the high part-load region**

351 The variation of BSFC with changes to intake and exhaust VCT setting, at 5 to 5.5 bar BMEP, is
352 presented in Figure 11 for engine A and in Figure 12 for engine B. Only three operating points have
353 been shown for brevity and the corresponding COV variations not presented as within the 5% stability
354 threshold. In the upper portion of the part-load envelope (5 bar BMEP and above), the different make-
355 up of the two engines (namely, nominal compression ratio and fuel injection system) determines a
356 very different response to VCT changes. Figure 12 shows that the specific fuel consumption of engine
357 B declines with both intake advance and exhaust retard, with lowest levels invariably associated to
358 extremely advanced intake VCT setting. With only minor exceptions, engine A performs most
359 efficiently when late IVO timings between 10 and 5 CA deg BTDC are employed (Figure 11).
360 Consistently to one another, the two engines demonstrate highest fuel consumption for small or
361 negative valve overlap duration (bottom-right corner of the BSFC response maps).

362
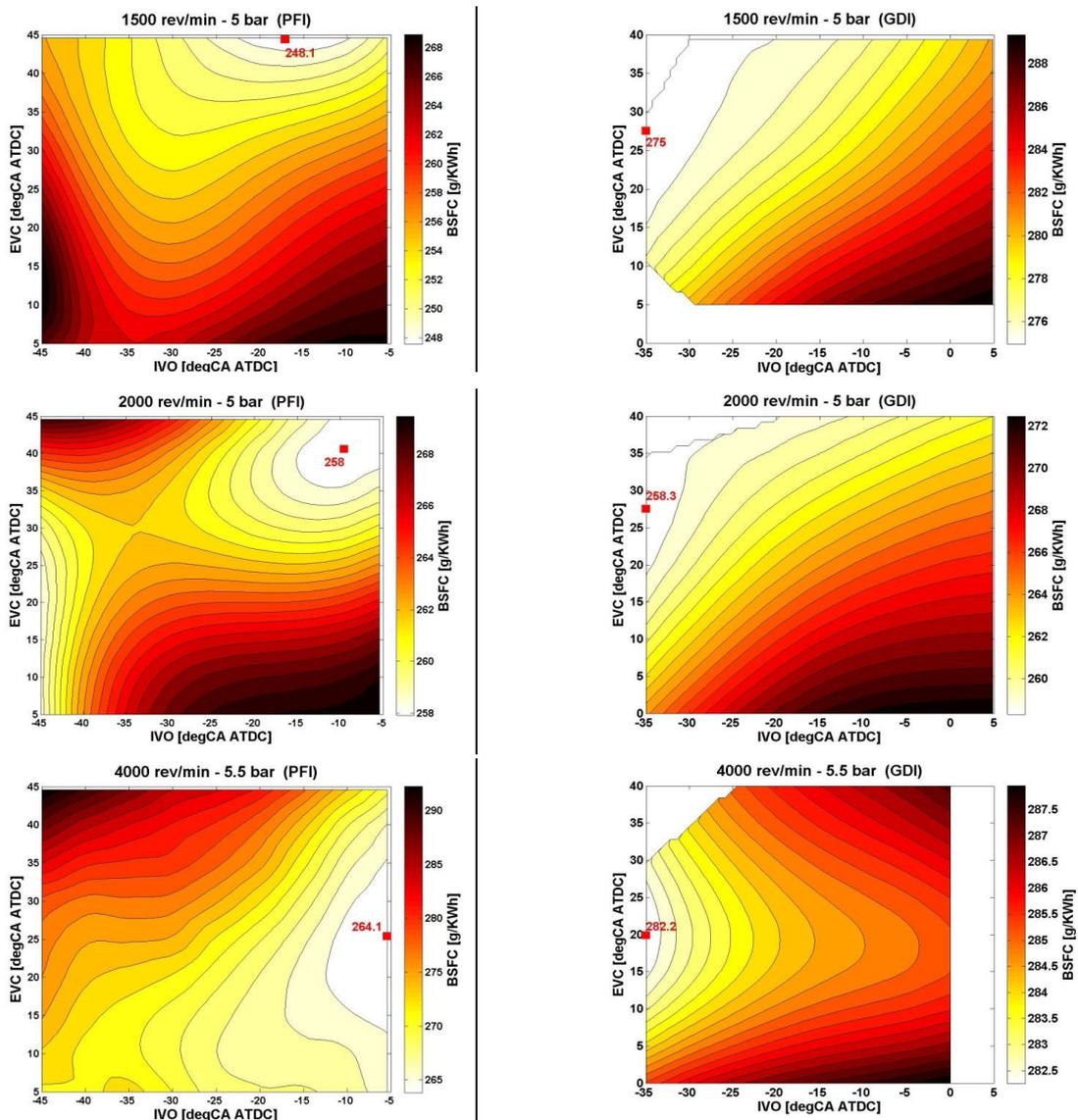
Figure 11. Contour-plots of BSFC as a function of IVO and EVC timing, for the PFI engine (A). Operating conditions: engine load of 5 to 5.5 bar (BMEP) and engine speed of 1500 to 4000 rev/min.

Figure 12. Contour-plots of BSFC as a function of IVO and EVC timing, for the GDI engine (B). Operating conditions: engine load of 5 to 5.5 bar (BMEP) and engine speed of 1500 to 4000 rev/min.

363

364 Engine B features lower nominal compression ratio (10:1) and this enables the implementation of
365 early intake VCT setting, without incurring into the inception of knock. The process of fuel direct
366 injection also contributes to delaying the occurrence of knock, due to its effect of cylinder charge
367 cooling. The absence of a constraint upon the ignition timing allows the use of as early IVO timing as

possible, with benefits associated to greater effective compression ratio (more work/cycle), lower fresh-charge back-flow (higher volumetric efficiency) and fresh-charge displacement (reduced pumping losses). Engine A, which features higher compression ratio (12:1), is knock-limited and ignition retardation must be introduced along with advanced intake VCT setting to suppress or delay the occurrence of knock. Figure 13 shows the CA location of 50% Mass Fraction Burned for both engines at representative operating conditions. The MFB50 is taken as an indication of the ignition timing strategy in the upper part-load envelope. As the IVO timing is shifted towards earlier events for engine A, the knock-limited MBT ignition strategy shifts combustion further into the expansion stroke, with consequent reduction of indicated work per cycle. Compared to engine B, this forces a very different response to VCT changes, with greatest efficiency almost invariably associated to late IVO operation. As shown in Figure 11, a moderate degree of intake VCT advance (IVO of 15 CA deg BTDC) is permitted at 1500 rev/min; the point of minimum BSFC quickly shifts towards later IVO events as engine speed increases, owing to an inherently greater knock tendency.
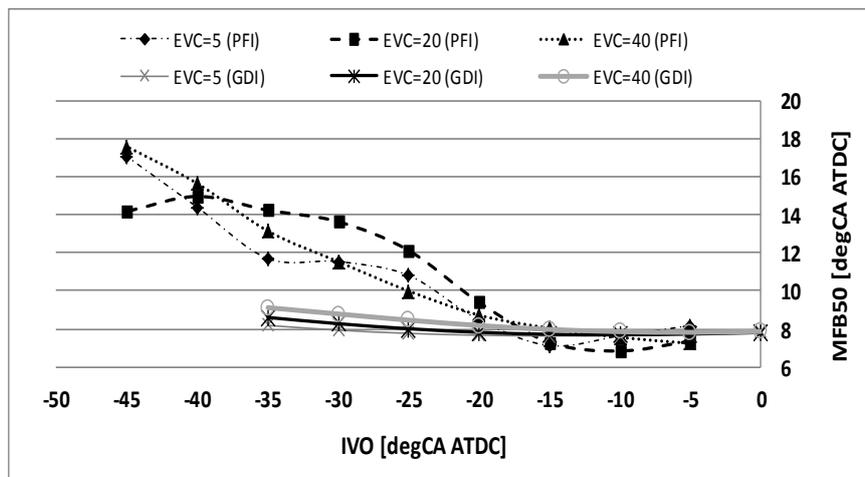
Figures 13. Location of 50% Mass Fraction Burned (MFB50) as a function of IVO timing, at three representative EVC timings. The operating conditions, 5 bar BMEP and 1500 rev/min, are indicative of the ignition timing strategy (as represented by the combustion process phasing) in the upper part-load envelope of both engines.

The fuel economy response to exhaust VCT changes is consistent across the two engines. The comparatively low cylinder charge dilution at these engine loads, allows the two engines to exploit the benefits of retarded EVC settings, both in terms of greater energy utilisation (extended power stroke) and lower pumping losses. For the PFI engine (A), the retarded intake valve opening enables the use of extremely retarded exhaust valve closing (40 to 45 CA deg ATDC), up to an engine speed of 2500 rev/min. At the highest engine speed investigated (4000 rev/min), lowest BSFC requires overlap duration of just 30 CA deg, with limited EVC timing of 25 CA deg ATDC. For the GDI engine (B), the advanced IVO setting limits the EVC retard to the range 20 to 27 CA deg ATDC, to minimise the losses from the less effective blow-down process.

Figures 14 and 15 show the BSFC response metrics at 7 to 7.5 bar BMEP and increasing engine speed. The behaviour demonstrated by the two engines in this load range is mostly consistent to the other upper part-load cases shown above. Engine A is knock-limited and lowest BSFC is gained using retarded intake VCT setting. Extremely advanced intake valve timing is favourable for the lower compression ratio, direct injection engine (B). In both cases, a moderate degree of EVC retard is permitted at lower engine speed, but causes blow-down losses at higher speed. Figure 14 shows that as engine speed reaches 3000 rev/min at 7.5 bar BMEP, some advancement of the IVO setting is advantageous also for engine A. At these operating conditions, lowest BSFC was recorded for EVC timing of 18 CA deg ATDC, and IVO timing of 22 CA deg BTDC. At engine speed is increased, the hot internal burned gas recirculation reduces [8], likely reducing the final cylinder charge temperature at IVC. As a result, the knock tendency also decreases, permitting the use of more advanced IVO setting without the need to aggressively retarding the ignition timing.

410 The overall theoretical fuel-economy Variable Camshaft Timing strategy for engine A and engine B, in
411 a simplified graphical form which disregards cases heavily affected by combustion stability, is
412 presented in the Conclusions section at the end of this paper.
413
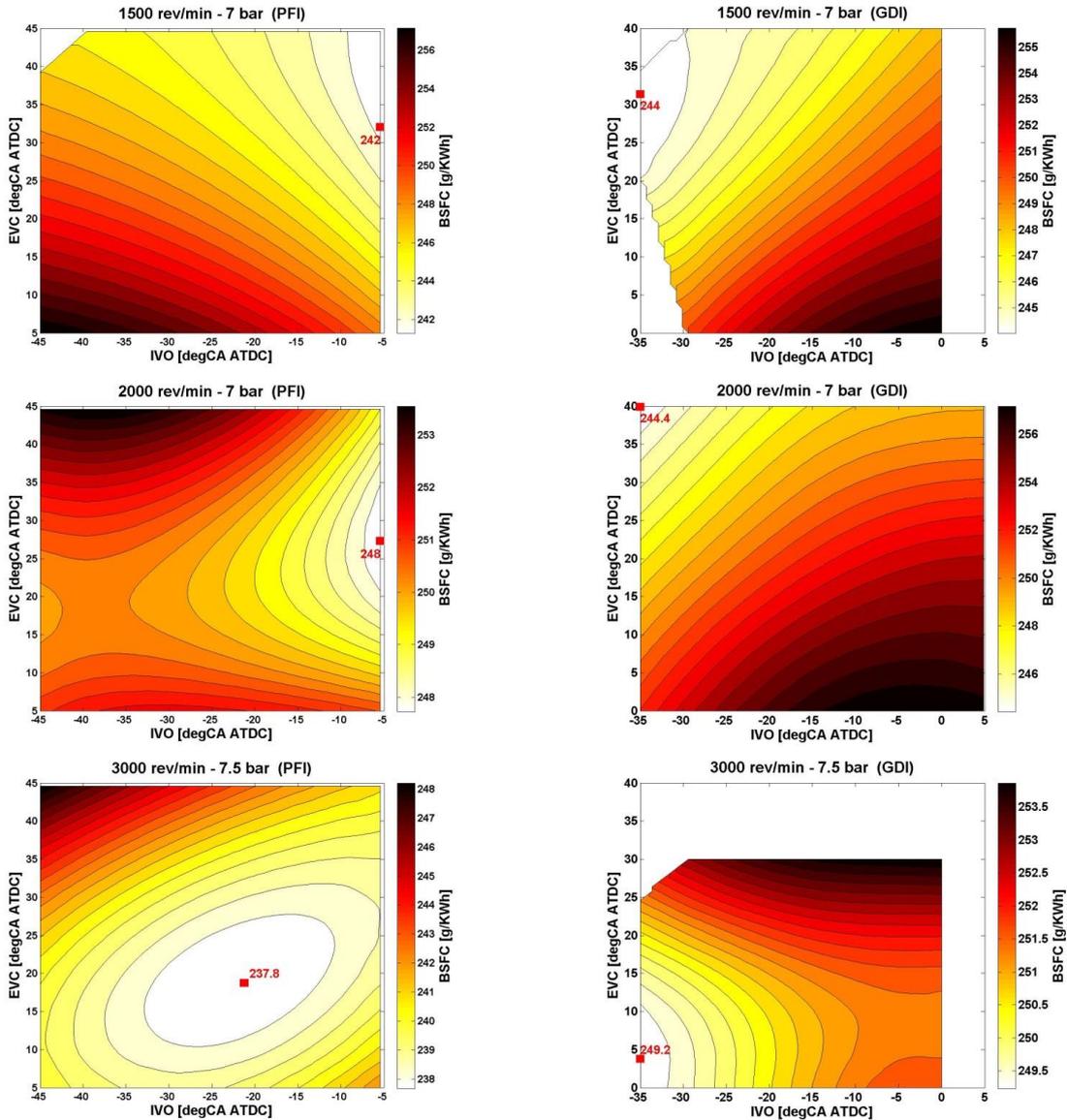
Figure 14. Contour-plots of BSFC as a function of IVO and EVC timing, for the PFI engine (A). Operating conditions: engine load of 7 to 7.5 bar (BMEP) and engine speed of 1500 to 3000 rev/min.

Figure 15. Contour-plots of BSFC as a function of IVO and EVC timing, for the GDI engine (B). Operating conditions: engine load of 7 to 7.5 bar (BMEP) and engine speed of 1500 to 3000 rev/min.

414
415 **3.4 Overall analysis of fuel economy in the part-load envelope**
416 The analysis presented above shows that as engine speed and load are varied in the part-load
417 envelope, changes to both intake and exhaust VCT settings are needed to gain optimal fuel economy.
418 Theoretically, both engines would benefit from a dual-independent valve timing strategy.
419 Nevertheless, dual-independent systems are more expensive, less practical, and show slower
420 response during transient operation when compared to one-variable-camshaft-only approaches. The
421 VCT strategy in a specific engine operating range must then carefully consider the trade-off between
422 potential fuel savings, and practical deployment of a complicated mechanical system.
423
424 Figure 16 summarises the BSFC percent variation between various potential VCT settings and a
425 reference setting where IVO= 5 CA deg BTDC and EVC= 5 CA deg ATDC. At each operating
426 condition, the "Dual VCT" horizontal bars refer to the best strategy identified above, whereas the other
427 bars refer to fixed IVO and best exhaust-only strategies. The choice of IVO settings was dictated by
428 the BSFC response metrics examined previously. Some fuel-economy bars are missing as

429 experimental records were not available. For both engine platforms, the dual VCT strategy invariably
430 enables greatest fuel economy; nevertheless, at some running conditions, specific exhaust-only
431 strategies determine very similar benefits. For engine A, the BSFC improvement was in excess of 8%
432 at 1500 rev/min and 5 bar BMEP and in excess of 6% at 2500 rev/min and 2.6 bar BMEP. These
433 values are in line with those reported recent literature [9, 10, 12, 13]. For the more efficient engine B,
434 the improvement associated to using variable valve timing was smaller, but extended to wider ranges
435 of running conditions. Compared to the reference VCT setting, BSFC decreased of almost 5% at 1500
436 rev/min and engine load between 5 and 7 bar BMEP, and at 2000 rev/min and load between 4 and 7
437 bar BMEP.
438
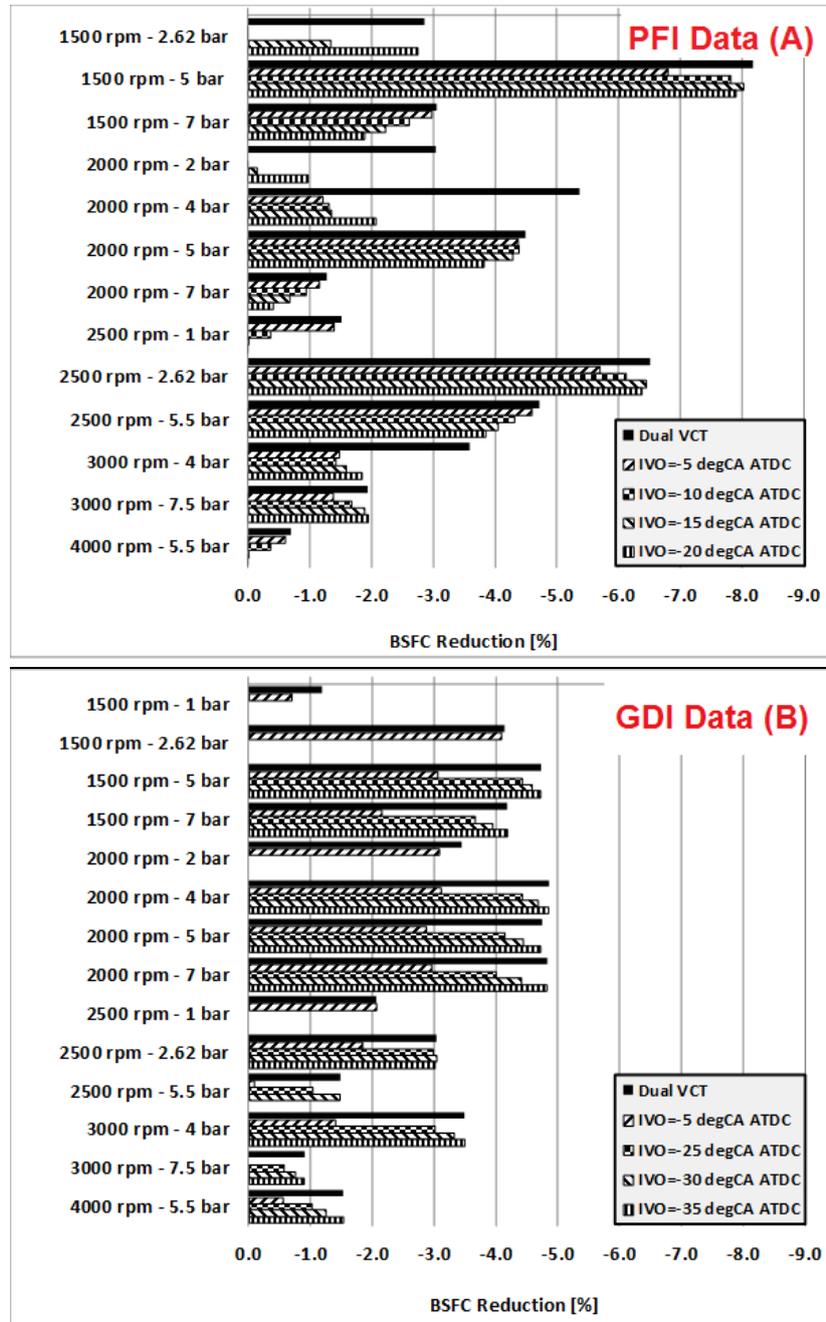
Figure 16. BSFC percent reduction between various VCT settings and reference setting (IVO= 5 CA deg BTDC and EVC= 5 CA deg ATDC), at all operating conditions investigated. Top plot: PFI engine (A); Bottom plot: GDI engine (B).

442 The BSFC percent reductions, averaged across all experimental conditions, are presented in Figure
443 17. The Dual VCT strategy enables average fuel saving in excess of 3.5% and 3.25% for engine A
444 and B, respectively. The best exhaust-only VCT strategy for engine A was based on fixed IVO timing

445 of 5 CA deg BTDC and caused average fuel saving of 2.4%. The best exhaust-only strategy for
446 engine B saw IVO timing fixed at 35 CA deg BTDC; the average saving was 3.25%. These results are
447 clearly aligned with the previous analysis which showed a knock-limited PFI engine (A), favoured by
448 retarded IVO setting; and, by contrast, a lower compression ratio GDI engine (B), favoured by early
449 IVO setting.
450
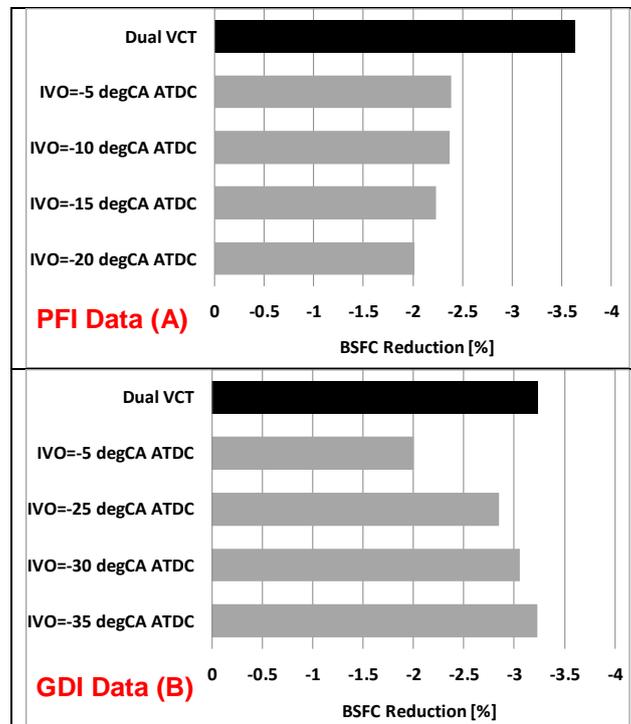
451
452 **Figure 17.** BSFC percent reduction between various VCT settings and reference setting (IVO= 5 CA deg BTDC and EVC= 5
453 CA deg ATDC), averaged upon all operating conditions investigated. Top plot: PFI engine (A); Bottom plot: GDI engine (B).
454

4. CONCLUSIONS

456 Variable camshaft timing strategies have been investigated in two modern, small-capacity, spark
457 ignition engines. Both engines featured modern-design, 4-valve/cylinder, pent-roof geometry and
458 same displacement volume of 0.333 litre/cylinder. The PFI engine featured higher compression ratio
459 of 12:1, whereas the GDI engine featured lower compression ratio of 10:1. The study focused on the
460 connection between valve timing setting and fuel economy at part-load operating conditions. The
461 study unveiled that engine specifics such as, notably, compression ratio and fuel injection technology,
462 have a profound influence upon the VCT strategy which maximises engine thermal efficiency,
463 especially in the upper part-load envelope.
464

465 The simplified traits of the fuel-economy VCT strategy for the two engines are presented in graphical
466 form in Figure 18. At low engine load both engines benefit from retarded IVO timing, which limits the
467 cylinder charge dilution and maximise combustion stability. As engine speed increases, increased
468 retard of the EVC timing becomes advantageous, most likely because it enables greater effective
469 expansion ratio.
470

471 At medium engine load, owing to the inherent reduction of internal charge dilution, both engines start
472 exploiting the beneficial influence of advanced IVO as well as retarded EVC timing. The experimental
473 data suggest that, for both engines but especially for the PFI platform, intake VCT advance is
474 somewhat more valuable than exhaust VCT retard due to a less effective blow-down process at this
475 engine load.
476

477 At the higher loads investigated, the two engines show a very different response to the valve timing
478 setting. The lower compression ratio GDI engine continues to benefit from advanced IVO timing,
479 whereas the degree of EVC retard must be moderated as engine speed increases to minimise the
480 exhaust blow-down losses. As shown in Figure 18, the higher compression ratio PFI engine is knock-
481 limited and this forces the VCT strategy to "come back on itself" towards regions of low IVO advance

482 and low hot gas internal recirculation. At the highest load investigated (7.5 bar BMEP), a moderate
483 degree of intake advance becomes again favourable as speed increases, owing to reduced hot
484 internal dilution and hence reduced knock tendency.
485
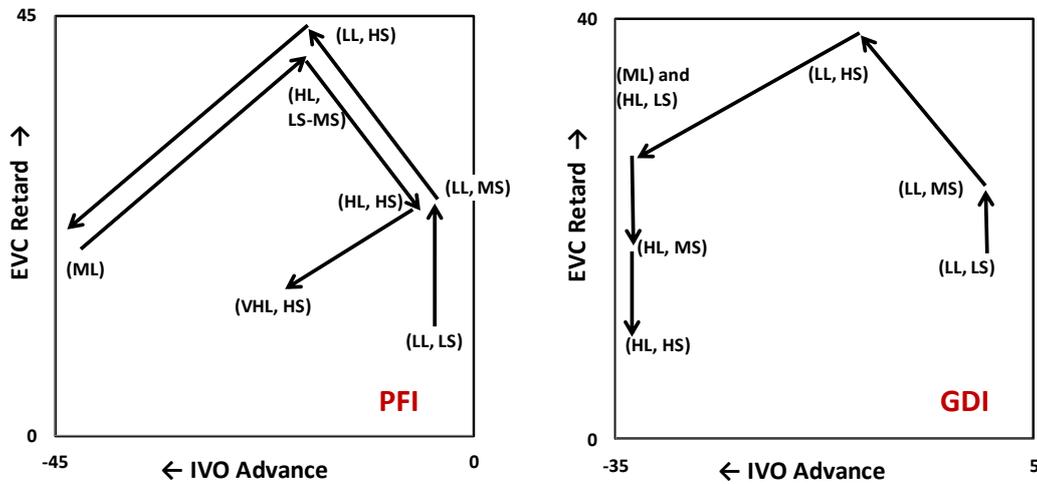
Figure 18. Simplified fuel-economy VCT strategy for the two small-size engines at part-load operating conditions. Left plot: PFI
engine (A); Right plot: GDI engine (B). Arrows describe a path between the low load/low speed and the high load/high speed
conditions investigated. Legend: LL=Low Load (1-2.6 bar BMEP); ML= Mid Load (4 bar BMEP); HL=High Load (7-7.5 bar
BMEP); LS=Low Speed (1500 rev/min); MS= Mid Speed (2000-2500 rev/min); HS= High Speed (3000-4000 rev/min); for the
PFI engine (A), a further condition is identified: VHL= Very High Load (7.5 bar BMEP).

492 The theoretical best dual-independent VCT strategy determines peak BSFC reduction in excess of
493 8% for the PFI engine; maximum reduction was just short of 5% for the more efficient GDI platform.
494 The BSFC reduction, averaged upon all part-load operating conditions investigated, was 3.5% for the
495 PFI engine, and 3.25% for the GDI one.
496
497 For both engine platforms, the upper part-load envelope benefits almost invariably from fixed IVO
498 setting. Consequently, specific exhaust-only VCT strategies may be conceived and deployed, which
499 enable similar fuel savings as the less practical dual-independent strategy.
500

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504

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618 **APPENDIX 1**
619 This section reports data of Pumping Mean Effective Pressure (PMEP) and Indicated Engine Torque
620 (IT) as a function of intake and exhaust valve timing. Figure A1.1 shows the corresponding contour-
621 plots for engine A (PFI) at engine load of 2 bar (BMEP) and engine speed of 2000 rev/min. As
622 anticipated in section 3.1, the operating conditions have been selected as they illustrate the engine
623 response which is typical of the lower part-load running envelope. The response of the GDI platform
624 (engine B) in this region was found to be very similar. The data shown in Figure A1.1 include
625 interpolated values to cover the full range of intake and exhaust valve timing variability.

627 Referring back to Figure 8, the PMEP shows as expected a strong degree of inverse correlation with
628 intake manifold pressure (MAP). As the valve overlap is increased, the intake de-throttling needed to
629 maintain constant engine brake load, induces higher intake manifold pressure; in turn, this lowers the
630 pumping work and hence the PMEP. While MAP increases, the PMEP is seen to decrease in equal
631 manner with both the degree of IVO advance and the degree of EVC retard. As shown in Figure A1.1,
632 the changes in pumping work are reflected by those in indicated torque, which tends to reduce with
633 increasing symmetric valve overlap to maintain constant brake load.
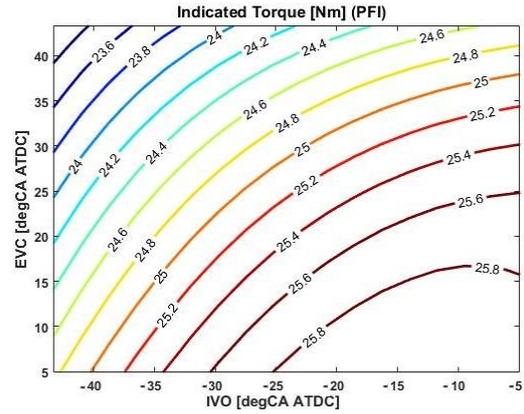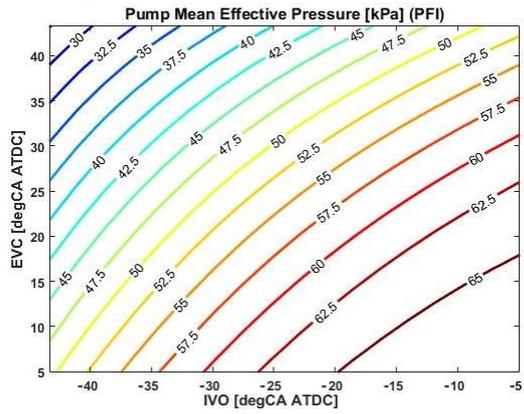
635
636 **Figure A1.1.** Contour-plots of Pressure Mean Effective Pressure and Indicated Torque as a function of the VCT setting for
637 engine A (PFI). Operating conditions: engine load of 2 bar (BMEP) and engine speed of 2000 rev/min.
638