

High Load Performance and Combustion Analysis of a Four-valve Direct Injection Gasoline Engine Running in the Two-stroke Cycle

Abstract

With the introduction of CO₂ emission legislation or fuel economy standards in Europe and many other countries, significant effort is being made to improve spark ignition (SI) gasoline engines because of their dominant market share in passenger cars and potential for better fuel economy. Amongst several approaches, the engine downsizing technology has been adopted by the automotive companies as one of the most effective methods to reduce fuel consumption of gasoline engines. However, aggressive engine downsizing is constrained by excessive thermal and mechanical loads as well as knocking combustion and low speed pre-ignition (also known as super-knock). In order to overcome such difficulties, a gasoline direct injection single cylinder engine was modified to run under the two-stroke cycle by operating the intake and exhaust valves around bottom dead centre (BDC) at every crankshaft revolution. The combustion products were scavenged by means of a reversed tumble flow of compressed air during the positive valve overlap period at BDC. The engine output was determined by the scavenging and trapping efficiencies, which are directly influenced by the intake and exhaust valve timings and boost pressures. In this research a valve timing optimization study was performed using a fully flexible valve train unit, where the intake and exhaust valves were advanced and retarded independently at several speeds and loads. A supercharger was used to vary the load by increasing the boost pressure. The effects of valve timing and boost pressure in the two-stroke cycle were investigated by a detailed analysis of the gas exchange process and combustion heat release. Gaseous and smoke emissions were measured and analysed. The results confirmed that the two-stroke cycle operation enabled the indicated mean effective pressure (IMEP) to reach 1.2 MPa (equivalent to 2.4 MPa of a 4-stroke cycle) with an in-

26 cylinder pressures below 7 MPa at an engine speed as low as 800rpm. The engine
27 operation was limited by scavenging inefficiencies and short time available for proper air-
28 fuel mixing at high speeds using the current fuel injector. The large amounts of hot residual
29 gas trapped induced controlled auto-ignition combustion at high speeds, and thus the
30 abrupt heat release limited higher loads.

31 **1. Introduction**

32 Two-stroke engines are well known for their superior power density and reduced
33 weight compared to equivalent four-stroke units and are employed to power handheld tools
34 to large marine engines [1][2]. Their use for high performance purposes is widely spread for
35 motorbikes, snowmobiles and outboard vehicles, with claimed power densities above 220
36 kW/litre [3]. However, these advantages, mainly related to crank-case scavenged two-
37 stroke engines, are often offset by drawbacks regarding gaseous emissions, thermal
38 efficiency and engine components durability [4].

39 On the subject of emissions, the fuel short-circuiting in mixture scavenged two-stroke
40 engines results in significant unburned hydrocarbon (uHC) emissions. The lubricant added
41 to the fuel has much less effect on emissions from crank-case scavenged two-stroke
42 engines according to [3], as modern units use proportions as low as 1% of oil in the fuel.
43 Regarding the thermal efficiency, conventional two-stroke engines usually lose expansion
44 work in favour of enhanced scavenging through early exhaust port opening. This procedure
45 uses the exhaust blow-down phase to reduce the levels of residual gas trapped prior to the
46 intake process, ensuring higher degrees of charge purity [5]. Lastly, the reduced
47 components durability (piston, rings and liner) of ported two-stroke engines can be
48 attributed to uneven thermal loads and reduced lubricant oil film when uHC emissions is a
49 concern [7]. It is important to keep in mind that all these disadvantages are related to cross-
50 scavenged and loop-scavenged two-stroke engines with intake and exhaust ports, where

51 the crank-case is employed as a pump for the air or air/fuel mixture and therefore lubricant
52 oil needs to be added to the air stream. Such problems can be avoided by the uniflow two-
53 stroke engine concept, in which externally compressed air is supplied through ports at
54 bottom dead centre (BDC) and the exhaust gas is forced out through conventional poppet
55 valves in the cylinder head. Greater scavenging efficiencies can be achieved with such
56 designs [1], but production complexity and packaging restrictions have limited its application
57 to large marine diesel engines so far though some attempts have been made to adopt such
58 an engine design for vehicular applications [8].

59 In the beginning of 1990 a new concept of two-stroke operation was proposed as a
60 possible solution to overcome the problems of conventional ported two-stroke engines.
61 Based on the design of modern four-stroke engines, the two-stroke scavenging process
62 was achieved through the overlap period of overhead intake and exhaust valves around
63 BDC at every engine revolution [2][7][9]. Because of the use of poppet valves higher power
64 outputs could be achieved with the same engine durability as four-stroke engines. The high
65 levels of uHC emissions due to fuel short-circuiting had been addressed by direct fuel
66 injection and air-assisted fuel injection [5]. The lubricant oil consumption, characteristic of
67 crank-case scavenged engines, had been eliminated by using wet sump and external
68 scavenge pump, mostly roots blower superchargers. When applying this concept to Diesel
69 engines, 40% higher torque and reduced combustion noise compared to an equivalent four-
70 stroke model was demonstrated by Toyota [7].

71 A reported problem of gasoline direct injection (GDI) two-stroke poppet valve
72 engines was the insufficient mixing between fuel and air in the cylinder, mainly attributed to
73 the shorter time available and the relatively lower injection pressures used at the time [7].
74 The poor charge mixing resulted in incomplete combustion and large emissions of CO, uHC
75 and soot, as studied by [10]. However, over the last few years significant advances have

76 been made in high pressure fuel direct injection systems and high efficiency boosting
77 devices (superchargers, turbochargers and e-boosters). In addition, flexible variable valve
78 actuation systems, particularly fast variable cam devices, have been developed for
79 production engines [11]. Such technological improvements have prompted renewed interest
80 in developing two-stroke poppet valve gasoline engines [12][13] and diesel engines
81 [14][15], considering their potential for aggressive engine downsizing with lower in-cylinder
82 pressures and less structural stresses than downsized four-stroke engines [16][17].
83 Moreover, the two-stroke poppet valve engine shares nearly all components from the
84 contemporary four-stroke engine architecture and hence can be produced from the same
85 manufacturing process.

86 In the previous study [13][18] it was demonstrated that controlled auto-ignition (CAI),
87 or homogeneous charge compression ignition (HCCI), combustion could be used to
88 improve the combustion stability and efficiency at part load conditions in the two-stroke
89 poppet valve engine. Higher efficiencies and near zero oxides of nitrogen (NO_x) emission
90 across a wide range of engine operation conditions at part load were achieved, using
91 gasoline and mixtures of gasoline-ethanol.

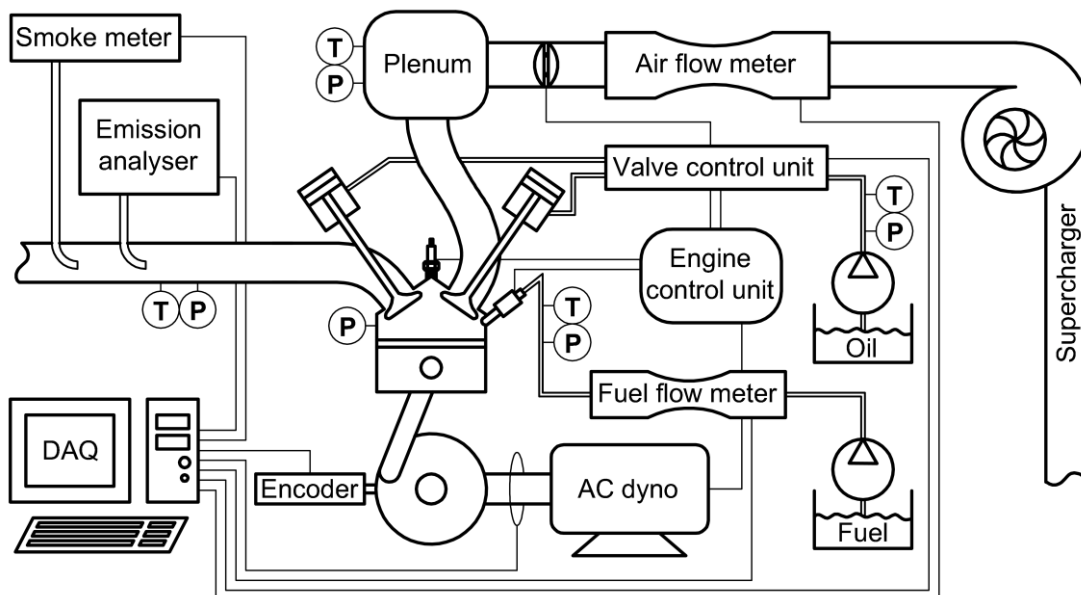
92 In order to evaluate the high load potential of the two-stroke direct injection gasoline
93 poppet valve engine operation, the present study was carried out at higher load conditions
94 with increased boost pressures at several engines speeds. The intake and exhaust valve
95 timings were varied and their effects on the scavenging process and engine performance
96 were investigated. Measurements of gaseous and smoke emissions, as well as the heat
97 release analysis, were performed to study the air-fuel mixing and combustion process
98 during the two-stroke cycle.

99 2. Experiments

100 2.1 Experimental setup

101 All the experiments were conducted on a single cylinder research engine mounted
102 on a transient test bed. The engine was equipped with an electro-hydraulic fully variable
103 valve train unit capable of independent control over the timings and lifts of each of the four
104 valves, enabling both two-stroke and four-stroke cycles operation [19]. The engine has an
105 81.6 mm bore and 66.9 mm stroke, with a geometrical compression ratio of 11.8:1 and a
106 pent roof combustion chamber. The valve control unit operated under closed loop control
107 over oil pressure, temperature and valve timing/lift, ensuring precise valve operation up to
108 3000 rpm in the two-stroke cycle. A Ricardo rCube engine control unit was used to manage
109 the throttle position, injection pulse width, spark timing and valve parameters. An AC
110 dynamometer enabled both motored and fired operations whilst an external cooling system
111 provided fully automated control over engine oil and coolant temperature. Gasoline (95
112 RON) was directly injected into the combustion chamber through a side mounted Denso
113 solenoid double-slit type injector [20]. The instantaneous fuel mass flow rate was measured
114 by an Endress+Hauser Promass 83A Coriolis flow meter, with a maximum error of $\pm 0.2\%$
115 for the flow range studied. The boosted air was supplied by an AVL 515 sliding vanes
116 compressor unit with closed loop control over the pressure. The air mass flow rate was
117 measured by a Hasting HFM-200 laminar flow meter with a maximum error of $\pm 1\%$. The
118 intake and exhaust pressures were measured by two Kistler piezo-resistive transducers
119 installed in the intake plenum (4007BA20F) and in the exhaust port (4007BA5F), with a
120 maximum error of $\pm 0.1\%$. The in-cylinder pressure was measured by a Kistler 6061B piezo-
121 electric sensor, with a maximum measurement error of $\pm 0.5\%$. To record the crank angle
122 positions a LeineLinde incremental encoder with a resolution of 720ppr was employed.
123 Averaged temperatures were measured at the intake plenum, exhaust runner, oil gallery,

124 coolant jacket and fuel rail by using K-type thermocouples with an accuracy of $\pm 1\%$. An
125 AVL 415SE smoke meter was used to measure the smoke levels, with repeatability better
126 than 3% of the measured value. Gaseous emissions were analysed by a Horiba MEXA
127 7170DEGR using the non-dispersive infrared principle for CO, a heated flame ionization
128 detector for uHC, a paramagnetic detector for O₂ and a heated chemiluminescence
129 detector for NO_x. The overall error attained to each gas measurement was smaller than
130 2%. The location of all instruments described above can be found in Figure 1, as well as the
131 temperature and pressure measurement points labelled as “T” and “P”, respectively.



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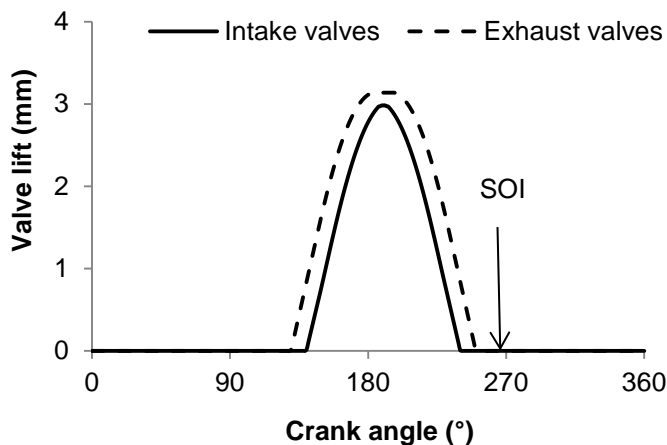
133 Figure 1 - Research engine and test cell facilities

134 A National Instruments 6353 USB X card was used for data acquisition (DAQ) and
135 an in-house software was employed for combustion analysis and specific emissions
136 calculations.

137 2.2 Test procedures

138 The two-stroke cycle was achieved by opening both the intake and exhaust valves
139 around BDC as presented in Figure 2. The long valve overlap period allowed the inlet
140 boosted air to scavenge the combustion products. The start of fuel injection (SOI) occurred

141 after all the valves were closed to avoid fuel short-circuiting to the exhaust. In addition, SOI
142 after the intake valve closing (IVC) prevented fuel from entering into the intake ports
143 through backflow, which may occur if the in-cylinder pressure becomes higher than the
144 intake port pressure. The fuel entrained in the intake port could then be carried back into
145 the cylinder and pass directly to the exhaust port in the following cycle, contributing to
146 increased uHC emissions.

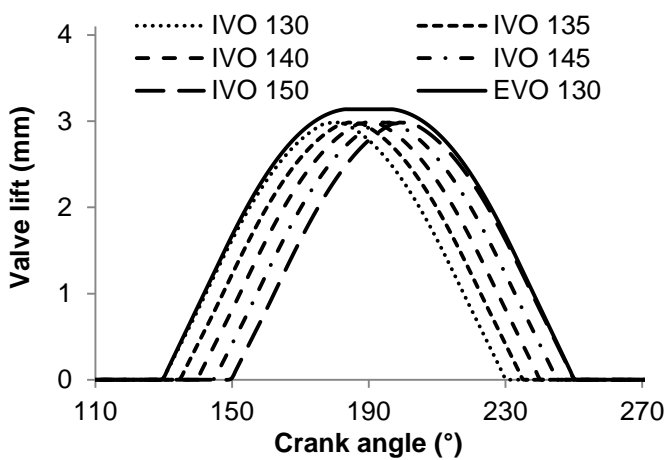


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148 Figure 2 - Two-stroke cycle operation principle

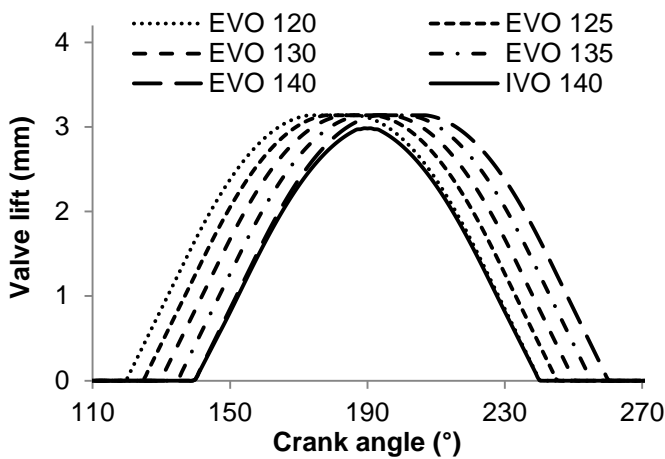
149 At each of the engine speeds studied, i.e. 800, 1500, 2200 and 3000±5 rpm, five
150 intake pressure levels were applied (where possible) as a way to control the engine load.
151 By increasing the boost pressures from 120±2 kPa to 280±3 kPa the scavenge ratio
152 increased and less residual gas was trapped, resulting in greater air mass in the cylinder
153 and higher engine power output. At some operation points stable combustion was not
154 achieved as the covariance of the indicated mean effective pressure (COV_{IMEP}) reached a
155 limit of 10%. This value seems high for four-stroke engines where a value around 5% is
156 usually considered [21]. However, bearing in mind the doubled firing frequency of two-
157 stroke engines the torque variation is reduced and the levels of vibration and harshness are
158 attenuated. In a previous study, stable operation in a two-stroke poppet valve engine was
159 claimed at COV_{IMEP} values as high as 35% [5].

160 At each engine speed, nine different combinations of intake and exhaust
 161 opening/closing timings were tested as shown in Figure 3 and Figure 4. The intake and
 162 exhaust valve durations were kept constant at 100° CA and 120° CA, respectively. At each
 163 engine speed and a given boost pressure, the exhaust valve timing was kept fixed and the
 164 inlet valve opening (IVO) was varied from 130° CA to 150° CA after top dead centre
 165 (ATDC), in steps of 5° CA. Then, the intake valve timing was fixed and the exhaust valve
 166 opening (EVO) was varied from 120° CA to 140° CA ATDC, also in steps of 5° CA.



167

168 Figure 3 - Intake valve timing optimization



169

170 Figure 4 - Exhaust valve timing optimization

171 In the previous research at part-load conditions [22] the EVC took place before the
 172 IVC to increase the residual gas trapped for CAI combustion. In this study, however, the
 173 EVC was delayed to obtain higher scavenging efficiencies. In addition, the exhaust valve

174 opened earlier during the expansion process to increase the exhaust blow-down period.
175 During the experiments the maximum advance of IVO was set to the EVO.

176 To ensure the same air-fuel mixing conditions for all the valve timings studied the
177 SOI was set to 260° CA ATDC, which was the latest EVC timing. During the engine
178 experiments, the fuelling rate was determined for a given engine speed and boost pressure
179 and kept constant when the valve timings were changed. The fuel injection pressure was
180 set to 15.0±0.5 MPa, and its temperature kept at 293±5 K.

181 The engine coolant and oil temperatures were kept at 353±3 K for all cases studied.
182 The intake air temperature was in the range from 295 K to 305 K, except for the maximum
183 intake pressure at 800 rpm when it reached 325 K due to insufficient air cooling.

184 The ignition timing was set to minimum spark advance for maximum brake torque
185 (MBT) or knock limited spark (KLS) at conditions when knocking combustion occurred. A
186 knocking combustion threshold of 1 MPa/°CA was set for the maximum rate of pressure
187 rise ($dP/d\theta$).

188 2.3 Data analysis

189 Based on the in-cylinder pressure and crank-angle measurements, the mass fraction
190 burnt was calculated according to the Rassweiler-Withrow method presented in [6]:

$$\frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \quad (1)$$

191 Where: Q_{net} is the net heat release, θ is the crank angle, γ is the ratio of specific
192 heats (considered constant and equal to 1.33), p is the in-cylinder pressure and V is the in-
193 cylinder volume.

194 Exhaust emissions were converted from parts per million (ppm) to g/kWh based on
195 the UN Regulation N49 [23]:

$$IS_{gas} = \frac{u_{gas} c_{gas} \dot{q}_{exh}}{P_i} \quad (2)$$

196 Where: IS_{gas} is the indicated specific gas emission (CO, HC or NOx), u_{gas} is the
 197 specific gas constant (CO = 0.000966, HC = 0.000499 and NOx = 0.001587) for gasoline
 198 fuelled engines, c_{gas} is the gas concentration in the exhaust stream, \dot{q}_{exh} is the exhaust
 199 mas flow rate and P_i is the indicated power output.

200 The combustion efficiency was calculated based on the emissions products not fully
 201 oxidized during the combustion:

$$\eta_c = 1 - \frac{\dot{m}_{CO} LHV_{CO} + \dot{m}_{HC} LHV_{HC} + \dot{m}_{H_2} LHV_{H_2}}{\dot{m}_{fuel} LHV_{fuel}} \quad (3)$$

202 Where: η_c is the combustion efficiency, \dot{m}_{CO} is the mass flow rate of CO, LHV_{CO} is the
 203 lower heating value (LHV) of CO (10.1MJ/kg), \dot{m}_{HC} is the mass flow rate of uHC, LHV_{uHC} is
 204 the LHV of uHC (44MJ/kg), \dot{m}_{H_2} is the mass flow rate of H₂, LHV_{H_2} is the LHV of H₂
 205 (120MJ/kg), \dot{m}_{fuel} is the fuel mass flow rate and LHV_{fuel} is the LHV of the fuel (44MJ/kg).

206 Emissions of hydrogen (H₂) were estimated based on the measurements of CO and
 207 CO₂ according to [24]:

$$[H_2] = \frac{0.5 y [CO] ([CO] + [CO_2])}{[CO] + 3 [CO_2]} \quad (4)$$

208 Where: $[H_2]$ is the exhaust concentration of hydrogen, y is the hydrogen to carbon
 209 ratio of the fuel (considered 1.87), $[CO]$ is the exhaust concentration of carbon monoxide
 210 and $[CO_2]$ is the exhaust concentration of carbon dioxide.

211 The air trapping efficiency, defined as the ratio of in-cylinder trapped air mass to the
 212 total intake air mass, was calculated based on the analytical method developed for fuel rich
 213 and stoichiometric combustion in two-stroke engines [25]:

$Trap_{eff(A)}$

$$= \frac{0.5[CO] + [CO_2] + 0.25 \left(\frac{yK[CO_2]}{[CO] + K[CO_2]} ([CO] + [CO_2]) \right) + 0.5[NOx]}{0.5[CO] + [CO_2] + [O_2] + 0.25 \left(\frac{yK[CO_2]}{[CO] + K[CO_2]} ([CO] + [CO_2]) \right) + 0.5[NOx]} \quad (5)$$

214 Where: $Trap_{eff(A)}$ is the air trapping efficiency, $[CO]$ is the exhaust concentration of
215 carbon monoxide, $[CO_2]$ is the exhaust concentration of carbon dioxide, y is hydrogen to
216 carbon ratio of the fuel (considered 1.87), K is the water-gas reaction equilibrium constant
217 (considered 3.5), $[NOx]$ is the exhaust concentration of oxides of nitrogen and $[O_2]$ is the
218 exhaust concentration of oxygen.

219 Due to the air short-circuiting from the intake to the exhaust during the valve overlap
220 period, the measured exhaust lambda value differed from the in-cylinder lambda. The in-
221 cylinder lambda was then calculated based on the air trapping efficiency and fuel trapping
222 efficiency [25]:

$$\lambda_{in-cylinder} = \lambda_{exhaust} \frac{Trap_{eff(A)}}{Trap_{eff(F)}} \quad (6)$$

223 Where: $\lambda_{in-cylinder}$ is the in-cylinder lambda, $\lambda_{exhaust}$ is the exhaust lambda,
224 $Trap_{eff(A)}$ is the air trapping efficiency and $Trap_{eff(F)}$ is the fuel trapping efficiency.

225 The fuel trapping efficiency (defined as the ratio of trapped fuel mass to the total
226 injected fuel mass) in a GDI engine is expected to be 100%, where no fuel short-circuiting is
227 supposed to happen. However, due to the high levels of fuel stratification resulted from the
228 short time available for air-fuel mixing at high speeds and loads, some of the fuel could not
229 take part in the combustion process and left the cylinder unburned. Thus, the fuel trapping
230 efficiency was introduced to take into account the short-circuited fuel from the previous
231 cycle, similar to that used in conventional ported two-stroke engines [25]:

$$Trap_{eff(F)} = \frac{[CO] + [CO_2]}{[CO] + [CO_2] + [uHC]} \quad (7)$$

232 Where: $Trap_{eff(F)}$ is the fuel trapping efficiency, $[CO]$ is the exhaust concentration of
 233 carbon monoxide, $[CO_2]$ is the exhaust concentration of carbon dioxide and $[uHC]$ is the
 234 exhaust concentration of unburned hydrocarbons.

235 The scavenging efficiency, described as the ratio of delivered air mass retained in
 236 the cylinder charge to the total in-cylinder charge, was used to indicate how efficiently the
 237 burned gases were displaced during the scavenging process. It can be calculated based on
 238 the air trapping efficiency and scavenge ratio, as follows:

$$Scvg_{eff} = Trap_{eff(A)} \left(\frac{m_{air}}{S_{W_{vol}} \rho_{air}} \right) \quad (8)$$

239 Where: $Scvg_{eff}$ is the scavenging efficiency, $Trap_{eff(A)}$ is the air trapping efficiency,
 240 m_{air} is the delivered air mass per cycle, $S_{W_{vol}}$ is the engine swept volume and ρ_{air} is the air
 241 density at ambient conditions. The term between brackets in equation (8) is the scavenge
 242 ratio (or delivery ratio), which compares the current delivered air mass per cycle to the
 243 reference mass in an ideal charging process.

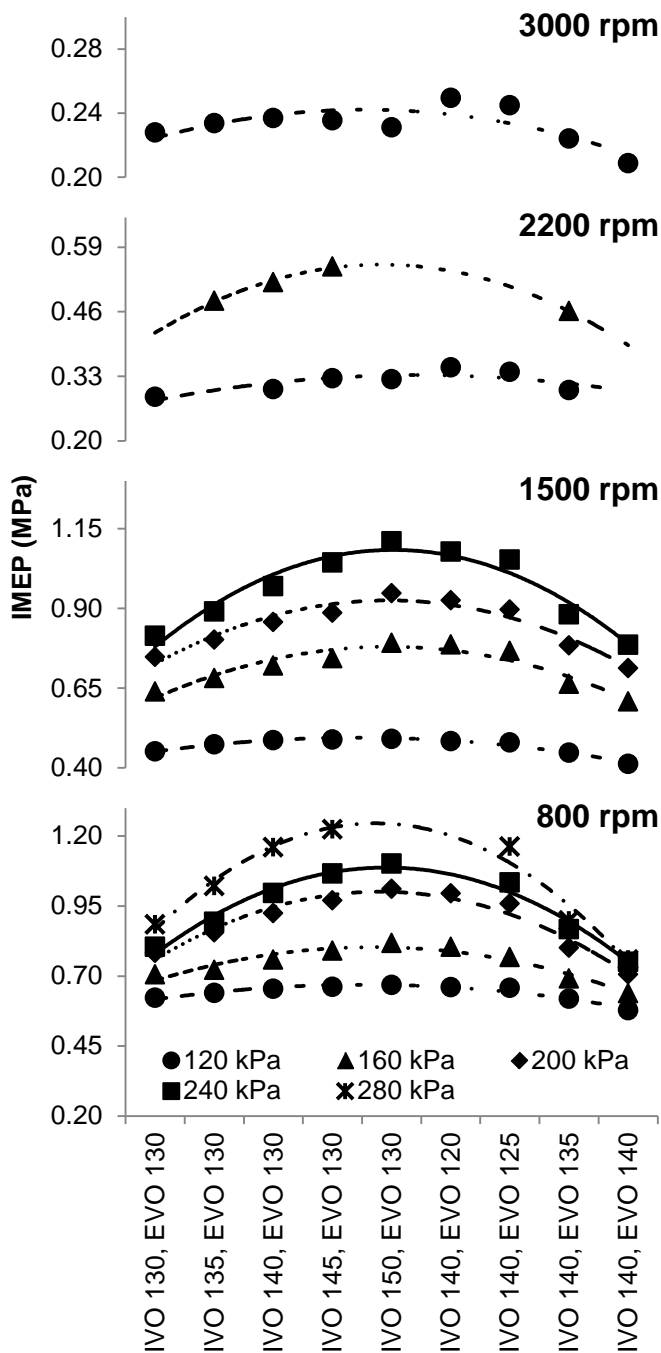
244 3. Results and Discussion

245 3.1 Performance and combustion analysis

246 The results presented here are averaged over 100 consecutive cycles and plotted as a
 247 function of valve timings at given engine speeds and intake pressures. The nomenclature of
 248 the different valve timings studied consists of the IVO and the EVO timings in °CA ATDC.
 249 The Y-axis is further divided into four parts according to the engine speed. When possible,
 250 second and third order polynomial curves were used to fit the data acquired.

251 Figure 5 shows the maximum IMEP values at different engine speeds and boost
 252 pressures. It is noted that higher boost operations were not possible at higher speeds (2200

253 rpm and 3000 rpm) due to violent combustion and unstable combustion. When the fuelling
254 rate was reduced to avoid excessive heat release rate at higher boost pressure, unstable
255 combustion occurred as measured by the higher COV_{IMEP} values. On the other hand, when
256 the fuelling rate was increased to avoid unstable combustion, the $dP/d\theta$ rose above the
257 knock limit. The occurrence of violent combustion or unstable combustion was likely related
258 to the large amount of hot residual gas trapped, resulted from insufficient time available for
259 scavenging at higher engine speeds. The presence of hot residual gas raised the charge
260 temperature and accelerated the occurrence of auto-ignition combustion in the unburned
261 mixture, resulting in rapid and violent heat release rate. In addition, since the SOI took
262 place at 260° CA ATDC (similar to that of the late injection stratified charge operation in
263 four-stroke GDI engines), significant fuel stratification could be present. If the fuelling rate
264 was reduced, the fuel stratification effect would become more prominent increasing the
265 cyclic variation of the mixture strength around the spark plug. Since the current fuel
266 injection and combustion system were not optimised for the stratified charge operation,
267 larger cycle-to-cycle variations could be expected to occur with greater fuel stratification.



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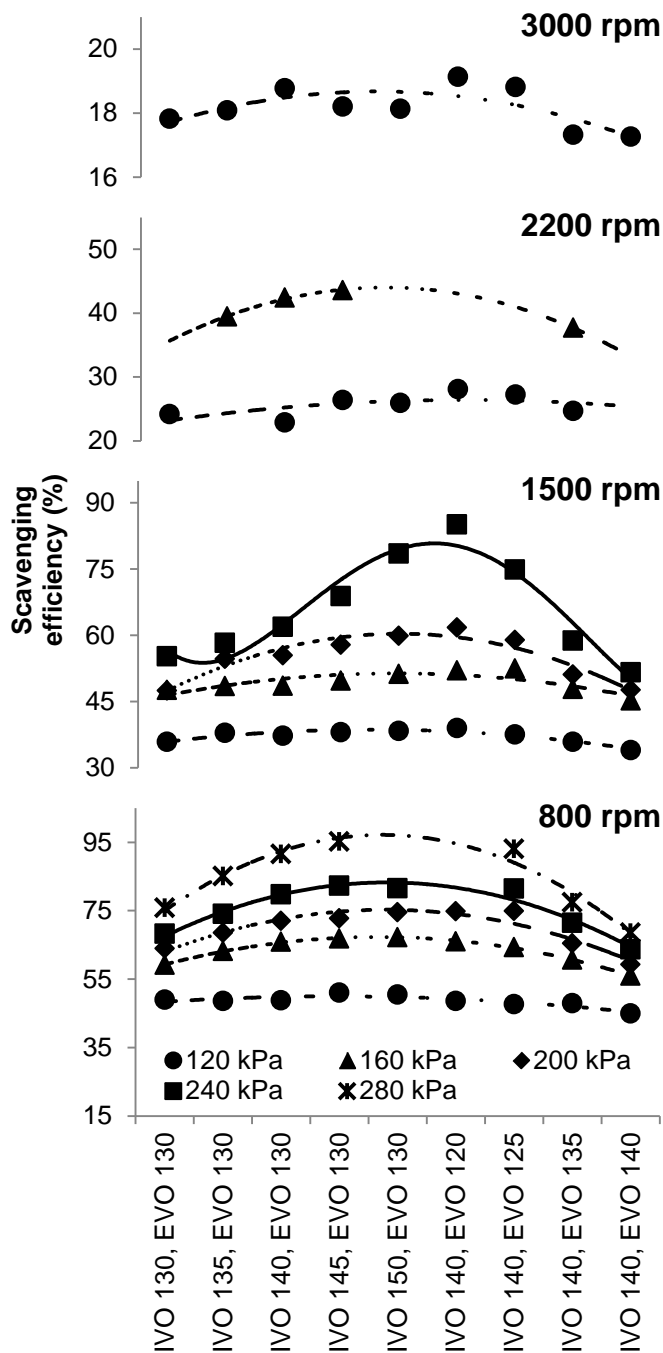
269 Figure 5 – Indicated mean effective pressure

270 At 800 rpm all the boosting levels could be tested throughout the valve timings
 271 studied except for the latest IVO (150° CA) and earliest EVO (120° CA), when combustion
 272 became excessively unstable. From the left to the middle point along the x-axis the IVO
 273 was retarded from 130 to 150° CA ATDC at a constant EVO of 130° CA ATDC. At the
 274 lowest boost pressure of 120 kPa the IMEP values varied little with IVO. When the boost

275 pressure was higher than 160 kPa, the IMEP increased with the retarded IVO and reached
276 its peak at IVO 150° CA ATDC. It is noted that the higher the boost pressure the more
277 pronounced is the change in IMEP with IVO. This can be explained by an increase in the
278 scavenging efficiency as presented in Figure 6, resulted from higher pressure difference
279 between the intake and exhaust ports. When the IVO was retarded, a more effective blow-
280 down event without intake air contamination was allowed. Such effect would be even more
281 pronounced at higher boost pressures. At 1500 rpm the IVO and EVO sweeps had similar
282 effects on the IMEP, but no stable combustion could be achieved at the boost pressure of
283 280 kPa.

284 From the right to the middle along the x-axis in Figure 5 and Figure 6, when the EVO was
285 advanced from 140 to 120° CA ATDC and the IVO kept at 140° CA ATDC, the scavenging
286 efficiency (and therefore the IMEP) changed little at lower boost pressures but rose steadily
287 to reach its peak at the middle of the graph. This behaviour mirrored the left part of the
288 curve and can be explained by the increased blow-down period and higher pressure ratio
289 across the exhaust valves at an earlier EVO. In addition, the difference between the intake
290 air pressure and the in-cylinder burned gases was greater at the same IVO as the in-
291 cylinder pressure had dropped to a lower value due to extended exhaust blow-down.

292 At 800 rpm the peak IMEP of 1.2 MPa was achieved at an intake pressure of 280
293 kPa, producing a specific torque of 195 Nm/l with the in-cylinder peak pressure as low as
294 6.8 MPa. To produce the same torque at the same speed in a four-stroke engine of the
295 same displacement, the engine would need to be operated at 2.4 MPa IMEP. This could
296 only be achieved with twice the in-cylinder pressure (13.6 MPa), assuming the engine
297 would not be limited by knocking combustion and/or low speed pre-ignition (LSPI) inducing
298 super-knock [26].

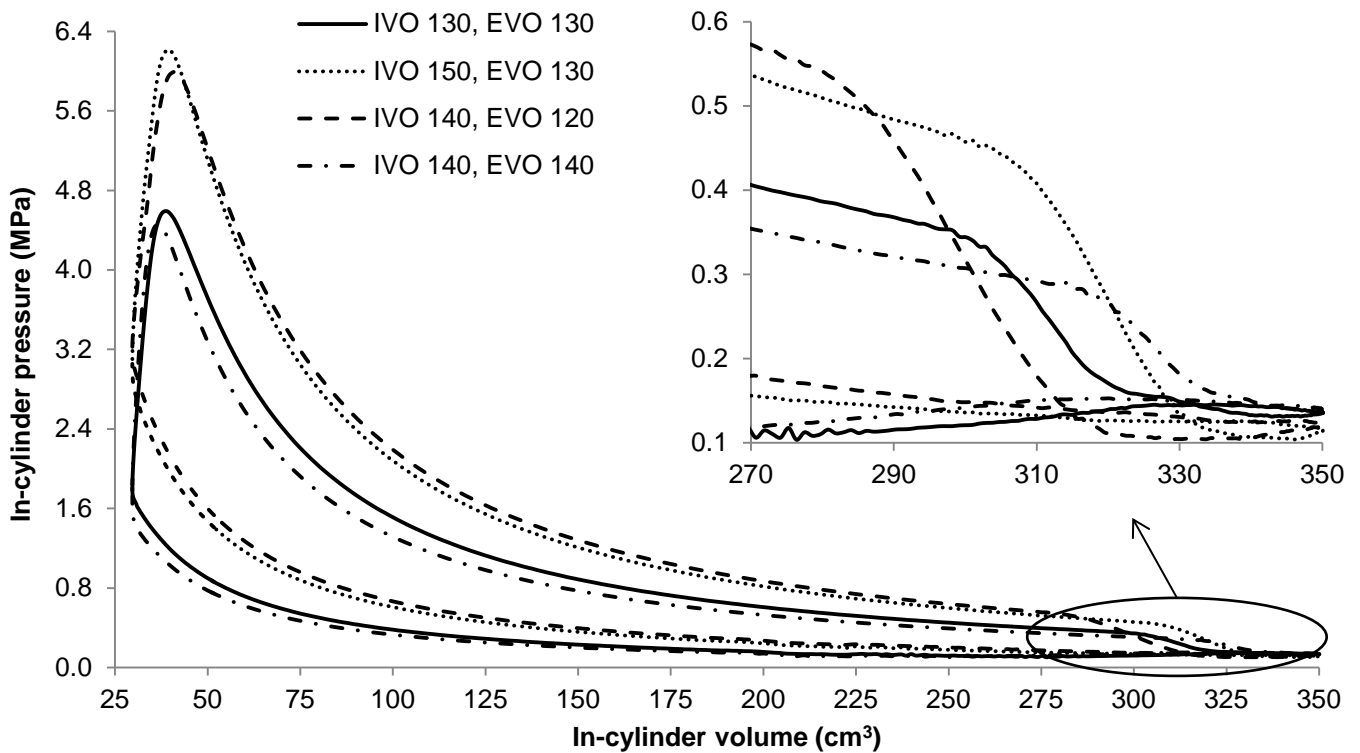


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300 Figure 6 - Scavenging efficiency

301 The results in Figure 6 illustrate that the maximum IMEP was a direct result of the
 302 most completed scavenging process achieved at the latest IVO (150° CA ATDC) and
 303 earliest EVO (120° CA ATDC). Because the fuelling rate was kept constant at a given
 304 intake pressure, it would have been possible to achieve even higher engine power outputs
 305 by increasing the fuelling rate at these valve timings at 800 rpm. However, it would have

306 been at the expense of poorer combustion efficiency and higher fuel consumption. At any
 307 given IVO and EVO timings the scavenging efficiency dropped with the increased engine
 308 speed because of the reduced time available for gas exchanging. For instance, at 2200 rpm
 309 and 120 kPa the residual gas level was found around 75%, whilst at 3000 rpm it reached
 310 82%. Furthermore, at each engine speed the scavenging efficiency decreased from the
 311 middle to the both sides of the x-axis, reaching a minimum when the valves opened at the
 312 same time, i.e. "IVO 130, EVO 130" and "IVO 140, EVO 140". In order to better understand
 313 the scavenging results, the pressure-volume (P-V) diagrams of four valve timings at 800rpm
 314 and 200kPa are plotted in Figure 7.



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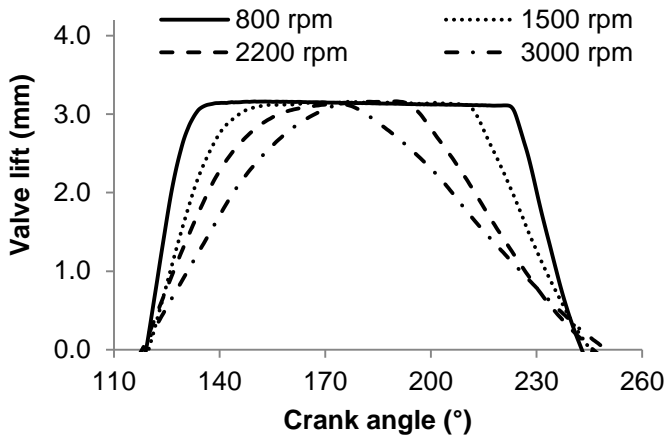
316 Figure 7 - Pressure-volume diagram for selected valve timings at 800 rpm and 200 kPa

317 It can be seen from the P-V diagram that the largest amount of useful work was
 318 achieved with the earliest EVO (120° CA ATDC) and the latest IVO (150° CA ATDC), when
 319 the scavenging process was optimized and less residual gas was trapped. As the valve
 320 timing was moved towards "IVO 130, EVO 130", the greater charge dilution promoted by
 321 the internal EGR reduced the heat release rate and hence the peak pressure. It can be

322 seen that in this case the intake and exhaust valves opened at the same time, part of the
323 burned gases mixed with the intake charge and thus compromised the scavenging during
324 the next cycle. The valve timing “IVO 140, EVO 140” was characterised with even lower in-
325 cylinder peak pressure as a result of greater amounts of residual gas trapped, as shown by
326 the lower scavenging efficiency (Figure 6). As shown by the zoomed part of the P-V
327 diagram in Figure 7, in this case the EVO was the most retarded and the expansion loop
328 was the longest amongst those shown. These two extreme valve timings also showed the
329 highest in-cylinder pressures around BDC, which caused the poor scavenging as the
330 pressure drop between intake and exhaust decreased. Moreover, the in-cylinder pressure
331 at the end of the compression phase for these two cases was about 50% lower than that for
332 “IVO 150, EVO 130” and “IVO 140, EVO 120”, resulted from less trapped fresh air mass
333 and higher levels of residual gas with larger heat capacity.

334 The two valve timings with the highest in-cylinder pressures, i.e.: “IVO 150, EVO
335 130” and “IVO 140, EVO 120”, presented similar peak pressures (less than 4% of
336 difference), although the early EVO case had reduced useful work and hence 2% lower
337 IMEP. At this speed it is possible to confirm that the exhaust blown-down phase can be
338 partially replaced by a later EVO (130°) with improved expansion work, without
339 compromising the purity of the charge. For this two valve timings the difference in
340 scavenging efficiency was less than 0.5% (Figure 6), whilst the IMEP increased by 2% with
341 later EVO (Figure 5).

342 The gas exchange process in this two-stroke poppet valve engine was also affected
343 by the actuation speed of the hydraulic valve train. As shown in Figure 8, the valve opening
344 and closing slopes became less steep as the engine speed increased, resulting in reduced
345 effective flow area. Such limitation of the camless system can be overcome by using a
346 conventional camshaft of higher lift driven by the crankshaft at the same speed.

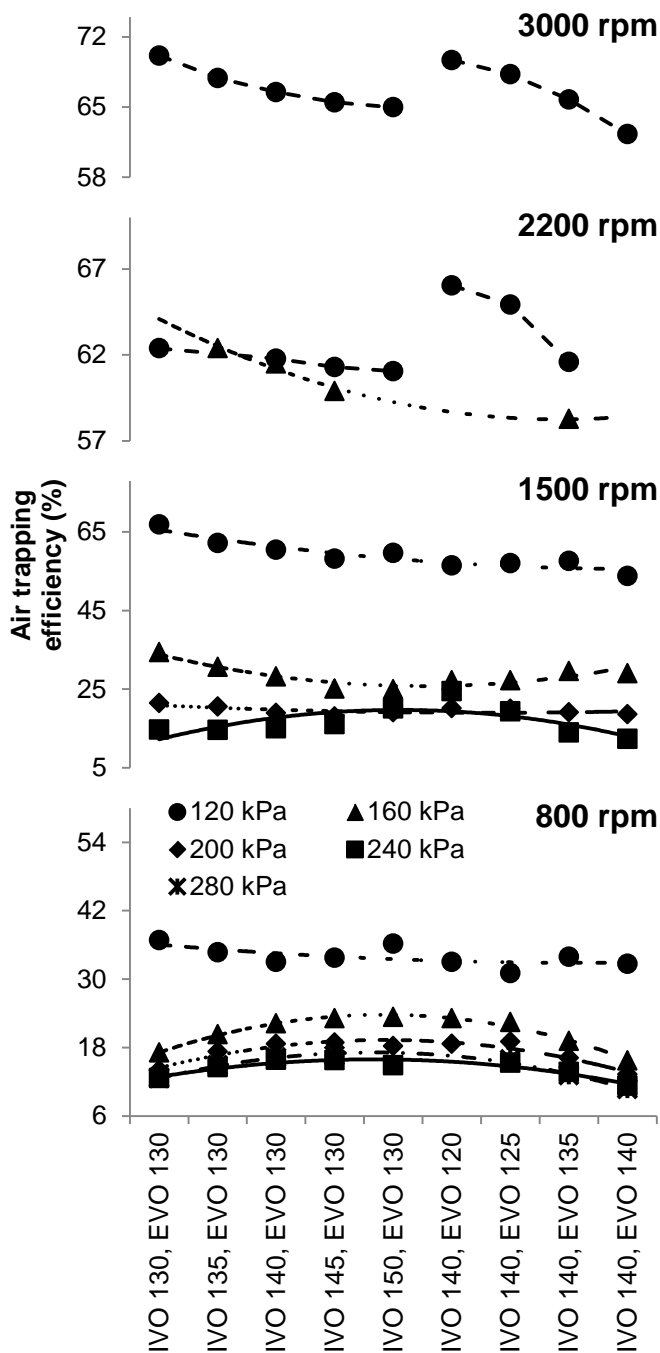


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348 Figure 8 - Effect of engine speed on valve opening and closing durations

349 Whilst the scavenging efficiency measured the effectiveness of the removal of
 350 burned gas, the air trapping efficiency was calculated to determine the air short-circuiting
 351 rate. As shown in Figure 9, the trapping efficiency rose steadily with the engine speed as a
 352 result of shorter time available for gas exchanging and consequent lower air short-circuiting
 353 rate. Higher trapping efficiencies were found for earlier EVO and hence earlier EVC,
 354 particularly at 2200 rpm and 3000 rpm, when the overlap period was reduced.

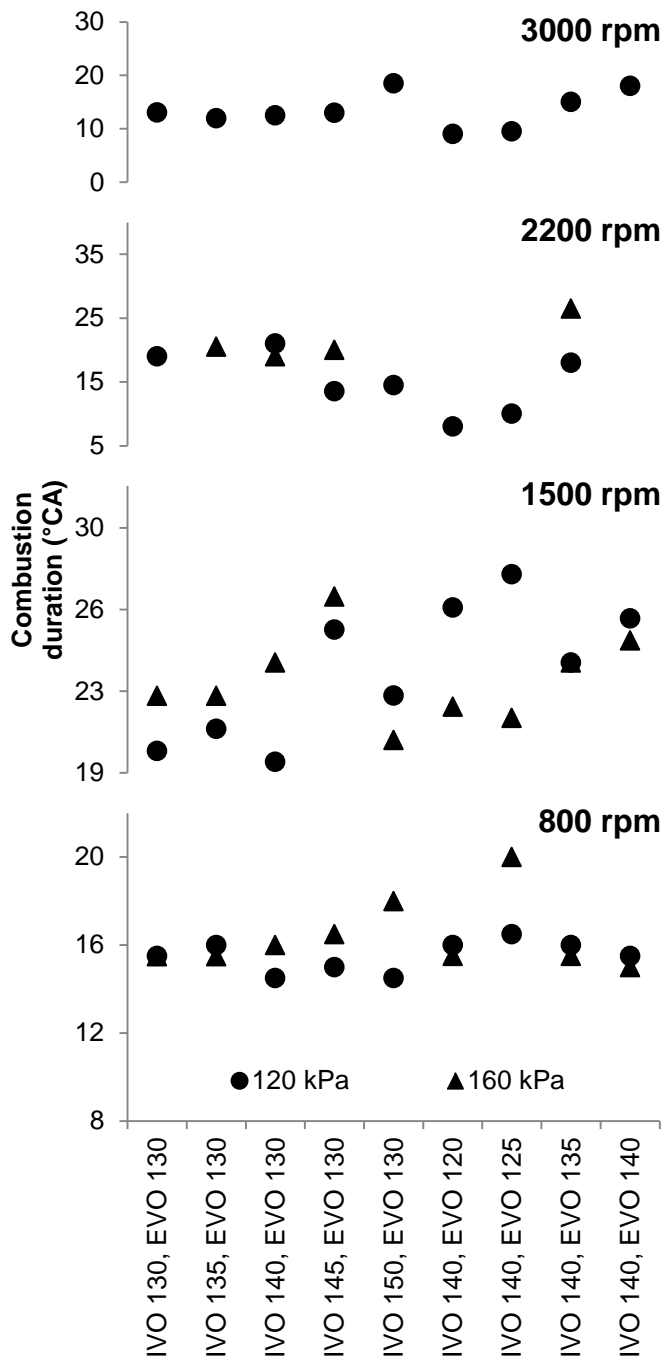
355 It is noted that when the intake air pressure was set at 120 kPa the air trapping
 356 efficiency at 800rpm and 1500 rpm exhibited different trends from the other pressures. This
 357 different pattern may be attributed to a transition from a displacement scavenging process
 358 to a mixing dominated scavenging process, as idealised by the Benson-Brandham two-part
 359 model for gas exchanging in two-stroke engines [27]. According to this theory the
 360 scavenging was firstly dominated by a displacement process until it reached a certain value
 361 of scavenge ratio, which in this case is around 1.5 at 800 rpm and 0.6 at 1500 rpm. After
 362 this point the fresh air and the burned gases were more prone to mix until the end of the
 363 scavenging process.



364

365 Figure 9 - Air trapping efficiency

366 The combustion duration, calculated from 10% to 90% of the mass fraction burned
 367 (MFB), is presented in two parts according to the intake pressures: the first part for 120/160
 368 kPa (Figure 10) and the second part for 200/240/280 kPa (Figure 11).

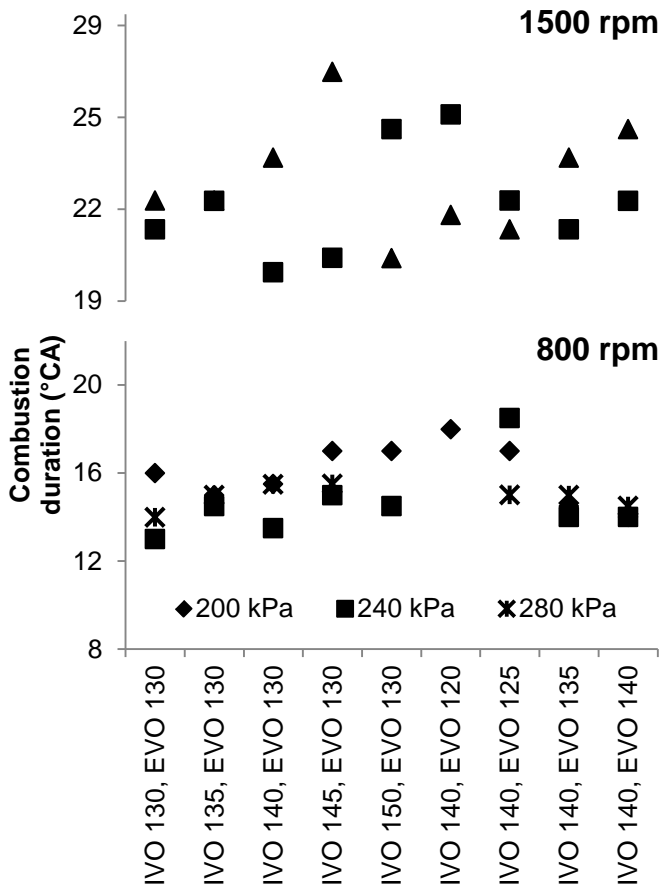


369

370 Figure 10 - Combustion duration for 120 kPa and 160 kPa intake pressure

371 At 800 rpm it is noted that the combustion durations decreased slightly as the boost
 372 pressure and load increased because of the higher charge temperatures and pressures. In
 373 addition, it can be seen from Figure 10 and Figure 11 that the combustion duration was
 374 between 13° CA and 19° CA at 800 rpm, which is much shorter than that of spark ignition
 375 (SI) combustion in four-stroke engines. This suggests that the heat release process might

376 have taken place in the form of a spark ignited flame around the spark plug and auto-
 377 ignition combustion of some premixed charge in the end-gas [28]. As the engine speed
 378 went up to 1500 rpm the combustion duration increased in terms of crank angles, but
 379 decreased slightly in absolute time as the flame speed was accelerated by the higher flow
 380 turbulence and the auto-ignition combustion was favoured by the hotter residual gas.



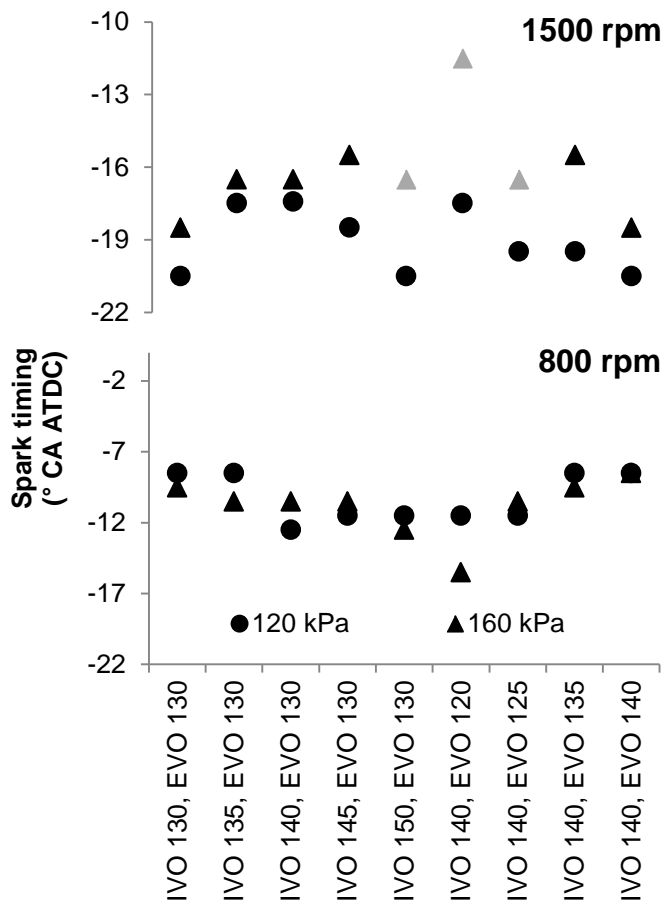
381

382 Figure 11 - Combustion duration for 200 kPa, 240 kPa and 280 kPa intake pressure

383 At 2200 and 3000 rpm stable engine operation was mainly limited to the boost
 384 pressure of 120 kPa. During such operation it was found that the spark timing had little
 385 effect on the combustion phasing and auto-ignition combustion became the dominant heat
 386 release process as evidenced by the very short combustion durations. The combustion
 387 duration remained nearly independent of the IVO variation when the EVO was set to 130°
 388 CA ATDC. In comparison, the EVO had a more pronounced effect on the combustion

389 duration as shown by the earliest EVO (120° CA ATDC) producing the shortest burning
390 duration.

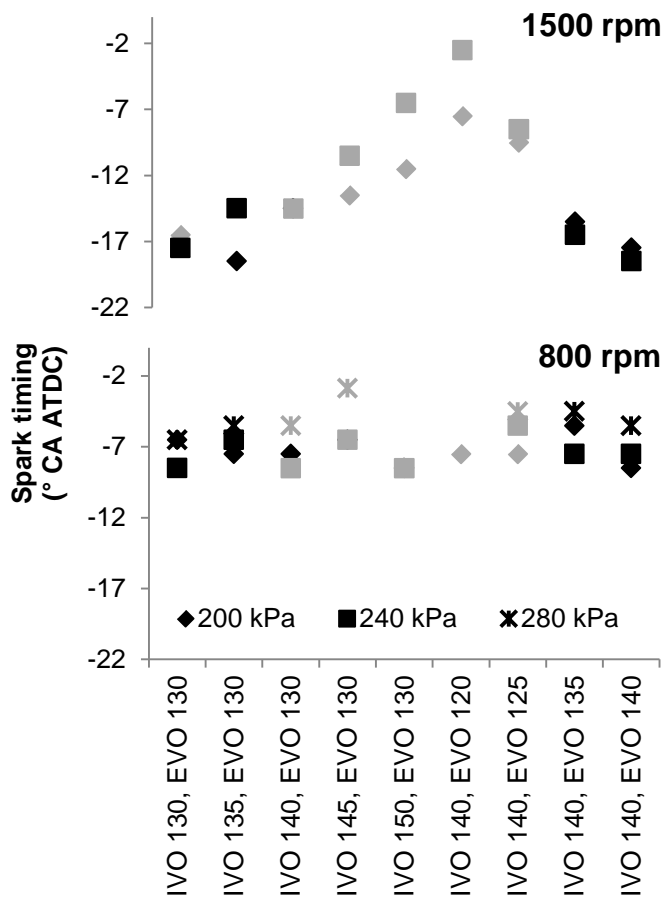
391 Figure 12 and Figure 13 show the spark timings set for MBT (black symbols) or KLS
392 (grey symbols) at 800 rpm and 1500 rpm, above which CAI combustion took place and the
393 spark timing had no effect whatsoever. It is noted that the presence of KLS at 1500 rpm
394 was about 50% greater than that at 800 rpm as a result of poorer scavenging efficiencies at
395 higher speeds. In addition, it can be seen that the most retarded KLS occurred at the
396 earliest EVO because of the minimum residual gas concentration as evidenced by the
397 highest scavenging efficiency (Figure 6). For the same reason, the KLS timing became
398 more retarded when the IVO was moved from 130 to 150° CA ATDC and less residual gas
399 was trapped. When the boost pressure was set to 120 kPa, MBT could be achieved for all
400 the valve timings and more advanced MBT timings were realized near the middle of the x-
401 axis, when both the scavenging efficiency and trapping efficiency were maximized.



402

403 Figure 12 - Spark timings set for MBT (black symbols) or KLS (grey symbols) for 120 kPa

404 and 160 kPa intake pressure



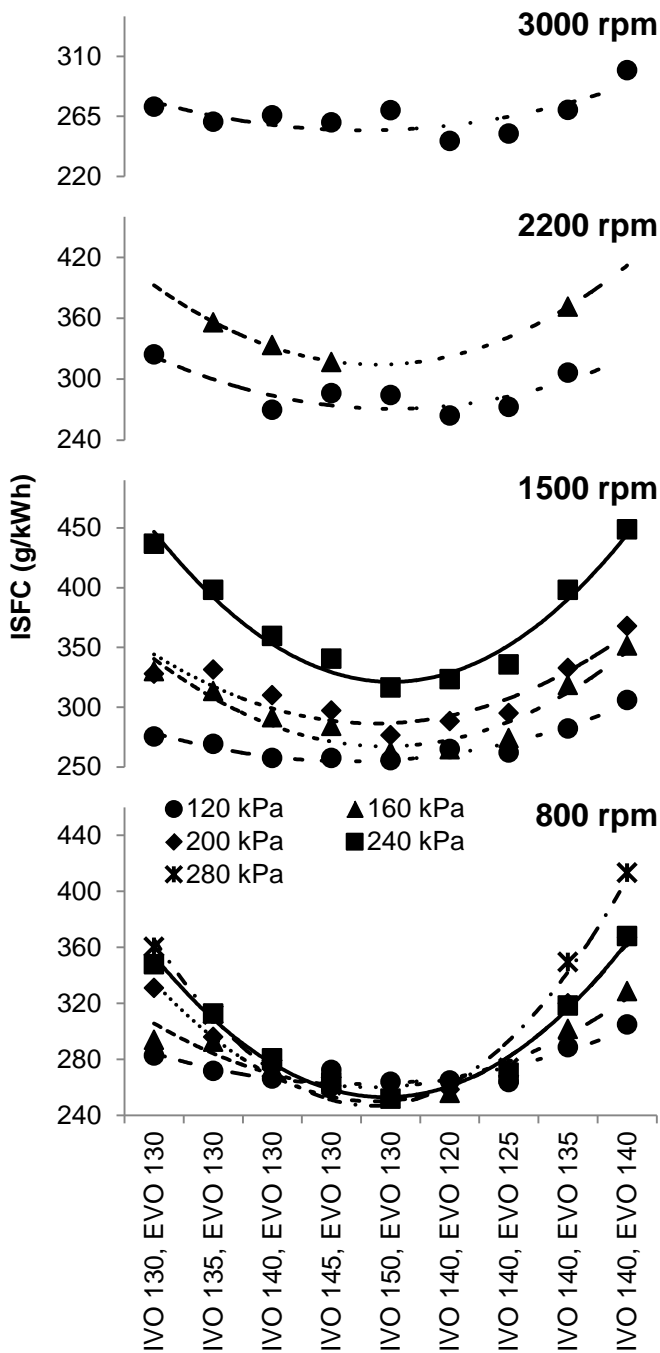
405

406 Figure 13 - Spark timings set for MBT (black symbols) or KLS (grey symbols) for 200 kPa,
 407 240 kPa and 280 kPa intake pressure

408 Figure 14 shows that a minimum ISFC of 255 g/kWh was achieved at 800 rpm and
 409 IVO 150° / EVO 120° CA ATDC for all the boost pressures. Although the minimum ISFC
 410 was achieved at the same valve timing at 1500rpm, its value increased with higher boost
 411 pressures. In order to better understand the ISFC results, it is necessary to look at the valve
 412 timing effect over the compression and expansion process, as well as the in-cylinder
 413 mixture composition and combustion.

414 Considering that the indicated specific fuel consumption is intrinsically linked to the
 415 expansion work, scavenging efficiency and combustion efficiency, there is a trade-off
 416 between higher scavenging rates through exhaust blow-down with early EVO and higher
 417 expansion works with late EVO. This effect is clearer in Figure 15, where the effective

418 compression ratio (ECR) and effective expansion ratio (EER) are plotted as a function of
 419 the valve timings.

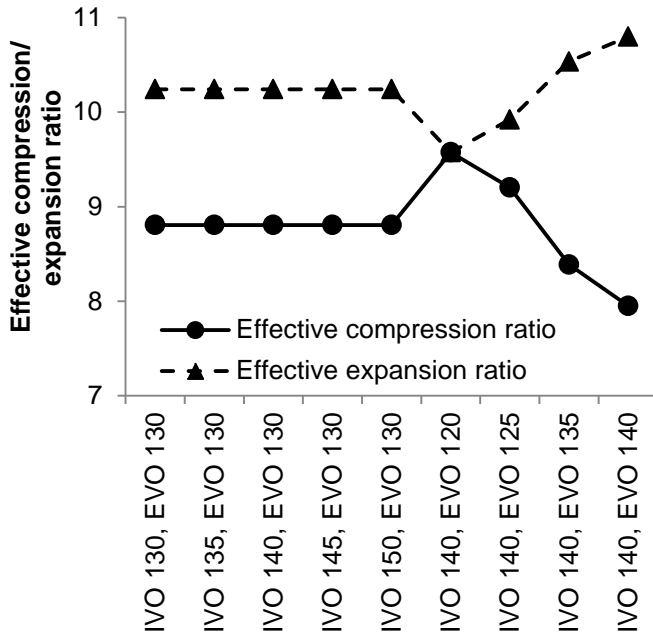


420

421 Figure 14 - Indicated specific fuel consumption

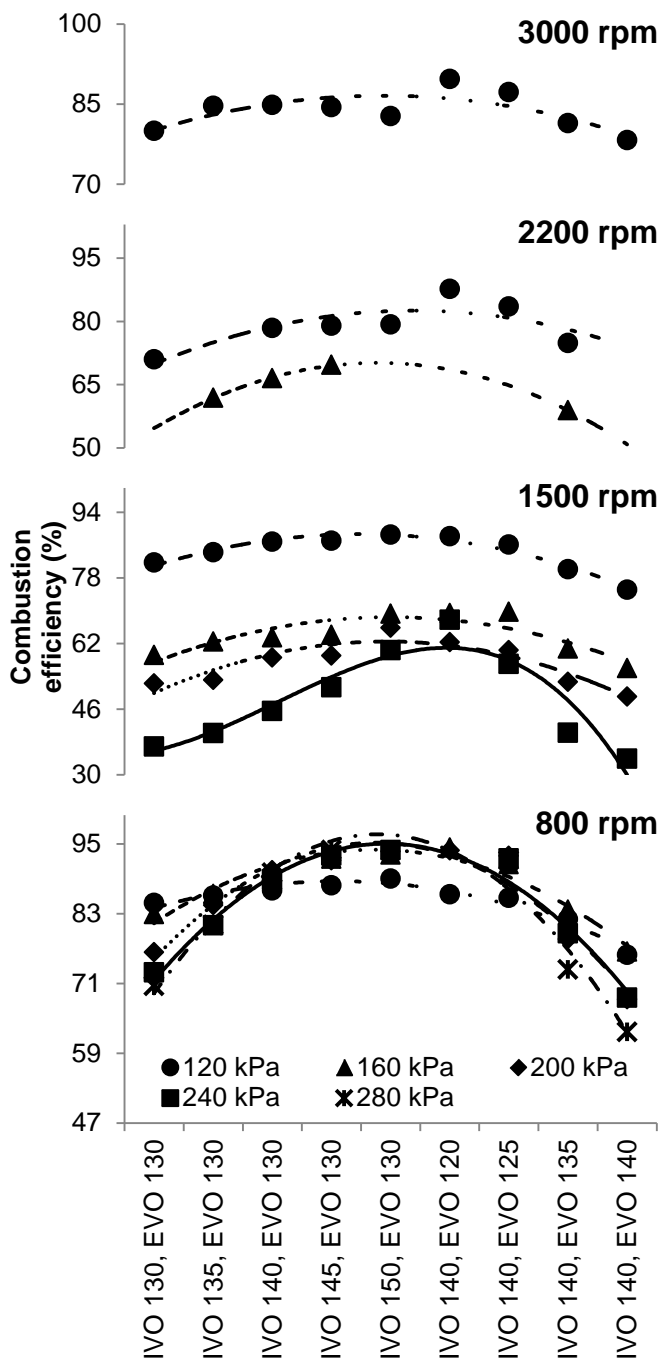
422 Figure 15 shows that for a given exhaust valve timing both the effective compression
 423 and expansion ratios were constant and the EER was higher than the ECR by about one.
 424 The effective compression and expansion ratios matched each other at "IVO 140, EVO

425 120", and when the EVO was retarded from 120° to 140° CA ATDC the EER increased and
 426 the ECR was reduced. The highest EER and hence highest expansion work was achieved
 427 with the most retarded EVO. However, such an increase in the useful work by the higher
 428 EER did not result in improved ISFC.



429
 430 Figure 15 - Effective compression and expansion ratios

431 The most significant cause for the change in ISFC as a function of valve timings was
 432 found from the combustion efficiency plots in Figure 16. It can be seen that the combustion
 433 efficiency results mirrored exactly those of ISFC presented in Figure 14. The highest
 434 combustion efficiencies and lowest ISFCs occurred in the middle of the graphs around "IVO
 435 150, EVO 130" / "IVO 140, EVO 120" and at 800 rpm. The combustion efficiency decreased
 436 with higher engine speeds at the same boost pressure, and it dropped with higher boost
 437 pressures at each engine speed.

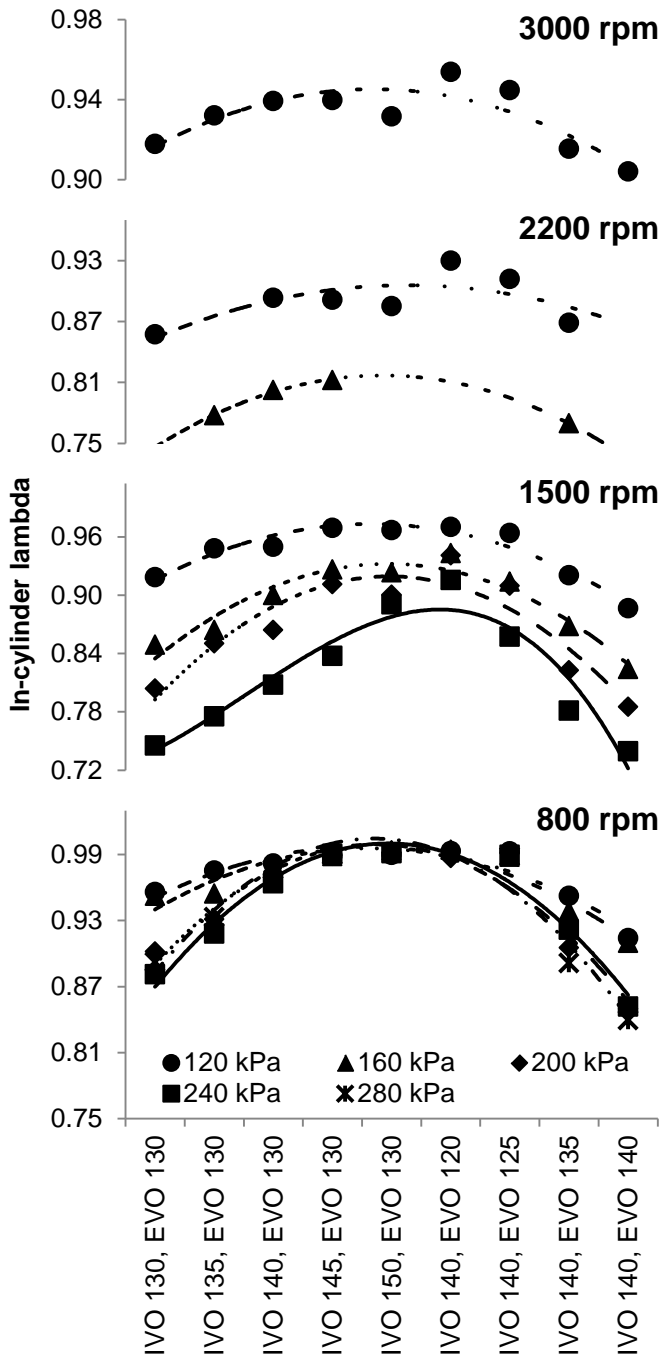


438

439 Figure 16 - Combustion efficiency

440 As shown by the in-cylinder lambda values in Figure 17, the change in combustion
 441 efficiency with valve timings can be attributed to the variation of in-cylinder air/fuel mixture
 442 with the gas scavenging process. The higher the relative air/fuel ratio (lambda) the more
 443 complete the combustion became. The leanest mixture of near stoichiometric air/fuel ratio
 444 was reached at 800rpm and resulted in a combustion efficiency of 94%. As the engine

445 speed was increased from 800rpm to 2200rpm, the decreased scavenging efficiencies led
 446 to richer air/fuel mixtures and lower combustion efficiencies. At the lowest boost pressure of
 447 120kPa, the combustion efficiency became higher at 3000rpm than 2200rpm mainly
 448 because of the leaner mixture and faster heat release rate (Figure 10).



449

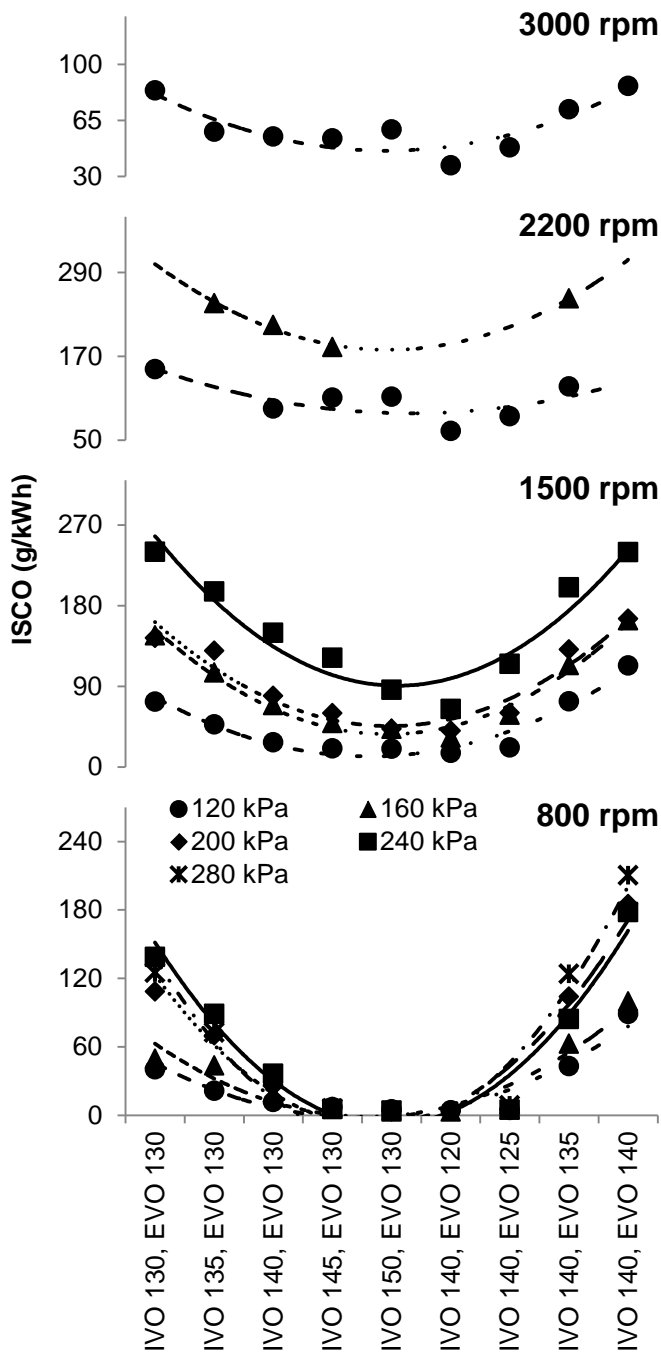
450 Figure 17 - In-cylinder lambda

451 3.2 Emission analysis

452 As shown in Figure 18, CO emission increased significantly as the mixture became
453 richer with more advanced IVO or retarded EVO at each engine speed. Figure 18 shows
454 that negligible CO emission was produced at 800 rpm when the scavenging efficiency and
455 lambda were maximised. Based on the estimated in-cylinder lambda results in Figure 17,
456 some noticeable CO emission was expected by combustion of the slightly overall fuel rich
457 mixture. The lower than expected CO emission could be caused by the oxidation of CO to
458 CO₂ by the short-circuited air mixed with the burned gas during the scavenging process. As
459 the engine speed was increased from 800rpm to 2200rpm, the poorer scavenging and
460 combustion of richer mixtures resulted in significant increase in CO and uHC emissions at
461 higher engine speeds. In addition, the mixture would be less homogeneous at higher
462 engine speed because of the reduced time available between the end of injection and the
463 beginning of combustion. This could have contributed to the very rapid rise in CO emissions
464 when the engine speed was changed from 800rpm to 1500rpm.

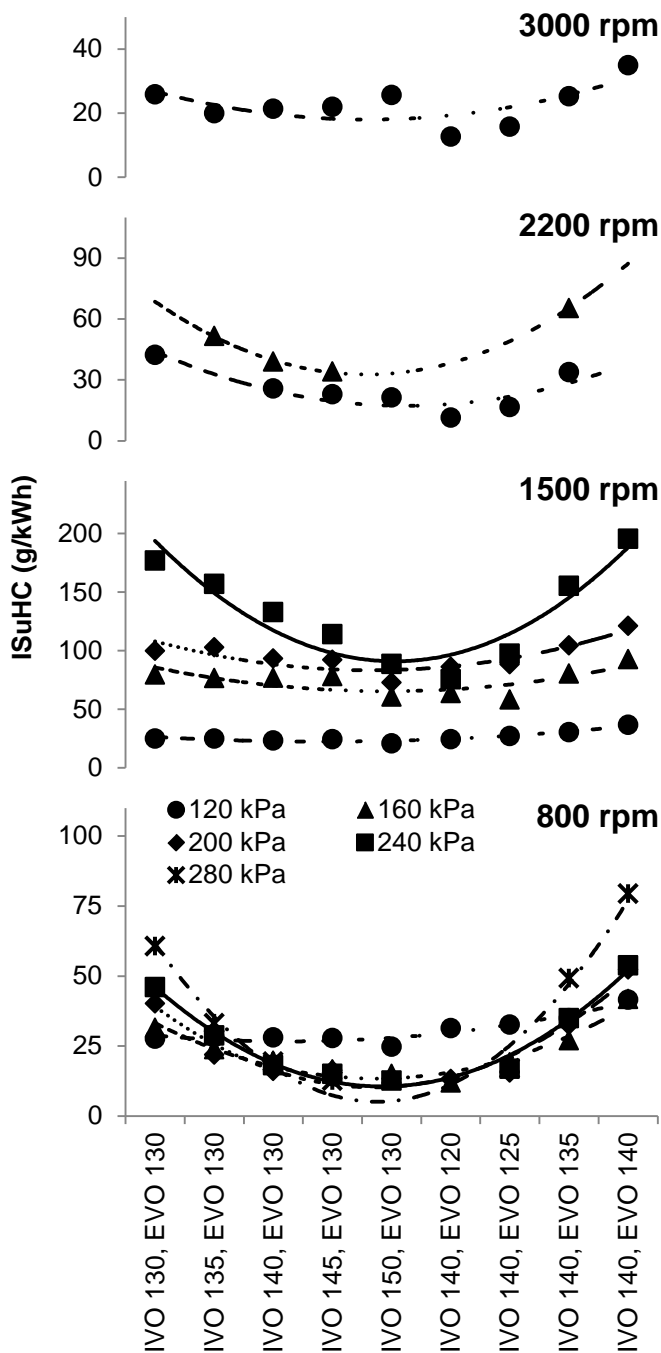
465 As shown in Figure 19, uHC emission showed less dependency on valve timings and
466 lower correlation with the scavenging efficiency and in-cylinder lambda. As late injections
467 were employed, most uHC emissions were likely produced by the fuel rich combustion as
468 well as fuel impingement due to retarded injection. The uHC emissions will not only
469 dependent on the overall air/fuel ratio but also its homogeneity. As injection took place after
470 260° CA ATDC, there was limited time available for a homogeneous mixture to form. Very
471 rich mixtures could be present in some regions producing uHC emissions. In addition, at
472 higher loads and boost pressures, the end of injection could be as late as 290 ° CA ATDC,
473 when the piston was only at about 25 mm from the cylinder head. Thus, the fan shaped
474 spray impinged onto the piston and formed pool fires on it top. For the same reasons, high
475 smoke emissions were observed as seen in Figure 20. Compared to uHC emissions, the

476 smoke emission was noticeably more affected by the load and speed as the fuel
 477 impingement increased with longer injection durations.



478

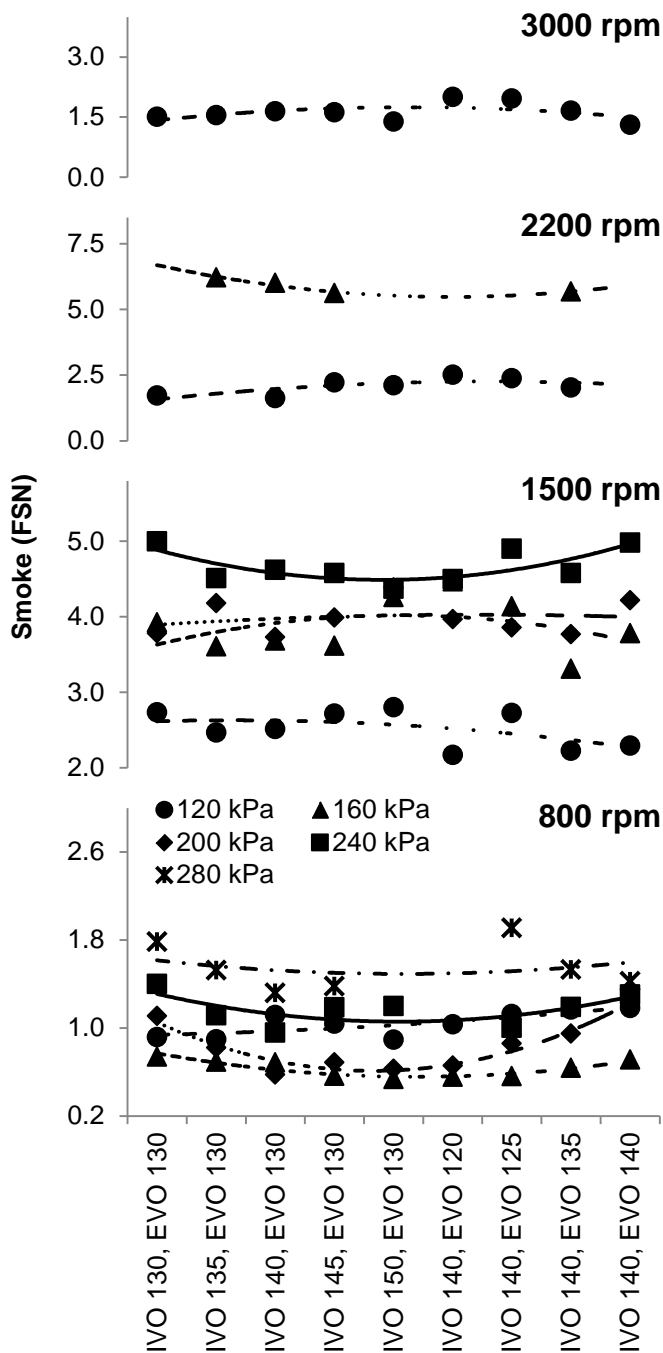
479 Figure 18 - Carbon monoxide emissions



480

481 Figure 19 - Unburned hydrocarbon emissions

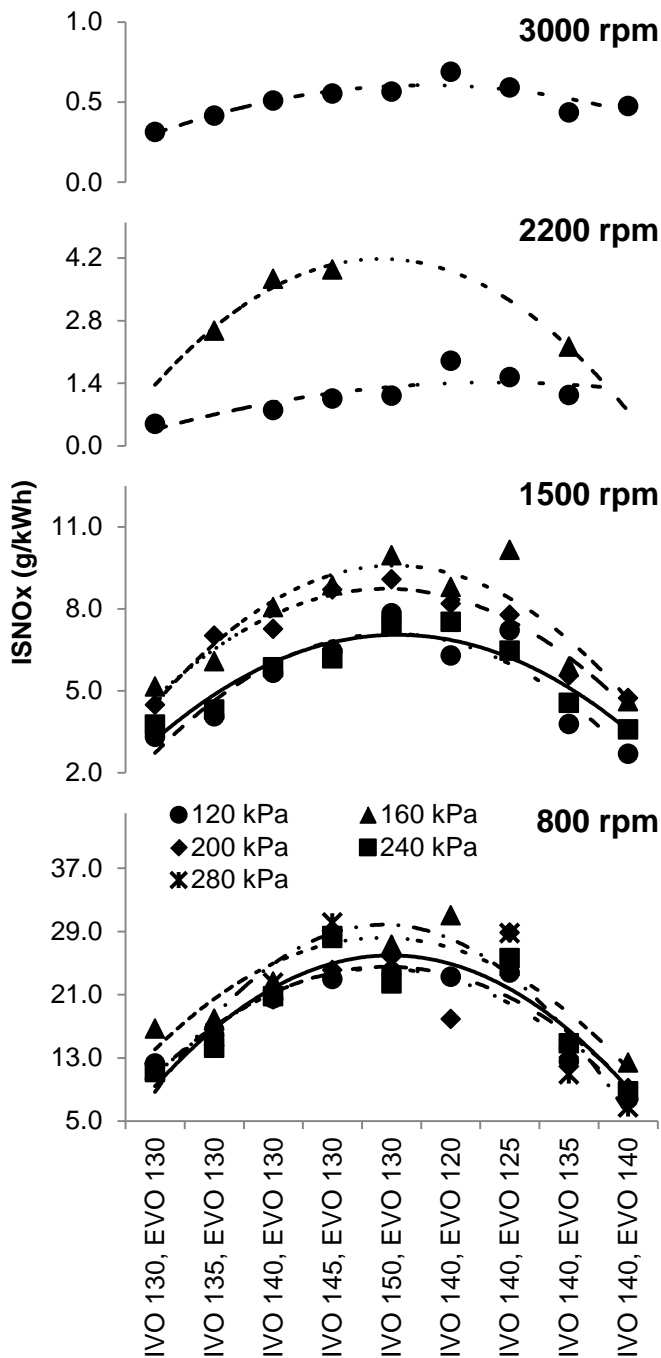
482 Furthermore, at the lowest boost pressure of 120kPa both CO and uHC emissions
 483 and smoke levels were less at 3000rpm than 2200rpm. It is attributed to the leaner mixture
 484 and faster heat release rate as shown in Figure 10.



485

486 Figure 20 - Smoke emissions

487 Finally, the NO_x emissions are presented in Figure 21. By moving along the x-axis
 488 from the middle to both sides, more residual gas was trapped resulted from lower
 489 scavenging efficiencies. Because of the increased heat capacity of CO₂ and reduced
 490 oxygen availability by the presence of recycled burned gases, the combustion temperature
 491 and hence NO_x formation were significantly reduced [29].



492

493 Figure 21 - Oxides of nitrogen emissions

494 At 800 rpm the early EVO raised the charge oxygen availability, increasing NOx
 495 emissions to levels of downsized four-stroke engines operating at similar conditions [30]. As
 496 the speed increased from 800 to 3000 rpm, the combustion mode progressed from SI
 497 towards CAI as a result of higher levels of hot residual gas trapped. Consequently, the NOx

498 emissions progressively decreased thanks to the higher charge dilution and lower
499 combustion temperature.

500 From Figure 21 it is also noted that the NO_x emissions were more sensitive to the
501 valve timings studied than to the load itself, especially at 800 rpm. At this speed the
502 emissions of oxides of nitrogen increased by 20% as the boost pressure was changed from
503 120 to 280 kPa (0.66 to 1.22 MPa IMEP). In comparison, by retarding the IVO in 10° CA
504 from 130° to 140° CA ATDC the NO_x emissions nearly doubled. The spark timing also
505 played an important role in NO_x emissions, as shown by the point “IVO 140, EVO 120” at
506 200 kPa boost. The ignition timing in this case had to be retarded to avoid knocking
507 combustion (Figure 13), reducing the in-cylinder peak temperature and NO_x production.

508 At 2200 and 3000 rpm and intake pressure of 120 kPa, pure CAI combustion took
509 place. At 2200 rpm the NO_x emissions rose rapidly as the boost pressure was increased
510 from 120 kPa to 160 kPa, as a result of both reduced residual gas concentration and
511 presence of high temperature flame in the spark assisted CAI combustion.

512 **4. Conclusions**

513 In this study, a four-valve direct injection gasoline engine was operated in the two-
514 stroke cycle mode by opening both the intake and exhaust valves around BDC. The
515 exhaust gas was scavenged by compressed air during the valve overlap period. At each
516 engine speed and boost pressure, the engine output was measured as a function of intake
517 and exhaust valve timings. The results can be summarised as follows:

- 518 • At 800 rpm the peak IMEP of 1.2 MPa was achieved at an intake pressure of 280
519 kPa, producing a specific torque of 195 Nm/l with the in-cylinder peak pressure as
520 low as 6.8 MPa. At each engine speed, the maximum IMEP was obtained with the
521 highest scavenging efficiency. As the engine speed was increased, the maximum
522 output was limited by the scavenging process and violent heat release rate.

- 523 • For the given valve duration and valve lift, the maximum scavenging efficiency of
524 95% could be achieved at 800rpm. At any given IVO and EVO timings the
525 scavenging efficiency dropped with the increased engine speed due to the reduced
526 time available for the gas exchange process, besides the reduced valve opening
527 area resulted from the hydraulically actuated valves.
- 528 • The trapping efficiency increased from about 35% to 70% with higher engine speeds
529 as the air short-circuiting rate was reduced.
- 530 • At 800 rpm and 1500rpm the heat release process was dominated by spark ignited
531 flame propagation combustion. At higher engine speeds, CAI combustion took place
532 and the spark timing had no effect whatsoever.
- 533 • The ISFC was primarily determined by the combustion efficiency, which was directly
534 related to the in-cylinder air/fuel ratio. The relative air/fuel ratios of the in-cylinder
535 mixture could be increased by optimisation of the valve timings for maximum
536 scavenging efficiency.
- 537 • The CO emissions were directly affected by the in-cylinder lambda. At 800rpm,
538 negligible CO emission was measured with optimised valve timings.
- 539 • Compared to CO emissions, uHC emissions and exhaust smoke levels were found
540 to be more affected by the fuel impingement and localised over-rich fuel mixtures in
541 the cylinder.
- 542 • NOx emissions were found to be very low at higher engine speeds when there was
543 high residual gas concentration and CAI combustion.

544 The above results have demonstrated that the scavenging process and fuel
545 preparation are the two most important issues affecting the two-stroke poppet valve
546 engine's performance and emissions. The scavenging process can be further optimised by
547 additional experiments with different valve opening duration and timings on the engine. To

548 improve the fuel preparation process, it would be necessary to increase the injection
549 pressures and employ split injections. A more robust stratified charge combustion system
550 design, such as a centrally mounted fast DI injector, would be also desirable.

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553 technological development (CNPq - Brasil) for supporting his PhD at Brunel University
554 London.

555 **Abbreviations**

556 ATDC: after top dead centre

557 CA: crank angle

558 CAI: controlled auto-ignition

559 CO: carbon monoxide

560 COV_{IMEP} : covariance of the indicated mean effective pressure

561 $dP/d\theta$: rate of pressure rise

562 ECR: effective compression ratio

563 EER: effective expansion ratio

564 EGR: exhaust gas recycling

565 EVC: exhaust valve closing

566 EVO: exhaust valve opening

567 GDI: gasoline direct injection

568 uHC: unburned hydrocarbon

569 IMEP: indicated mean effective pressure

570 ISCO: indicated specific carbon monoxide

571 ISFC: indicated specific fuel consumption

572 ISuHC: indicated specific unburned hydrocarbon
573 ISNOx: indicated specific oxides of nitrogen
574 IVC: intake valve closing
575 IVO: intake valve opening
576 LHV: lower heating value
577 KLS: knock limited spark advance
578 MBT: minimum spark advance for maximum break torque
579 NOx: oxides of nitrogen, rpm: revolutions per minute
580 SACI: spark assisted compression ignition
581 SI: spark ignition

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