

1 Miller cycle combined with EGR and post fuel injection 2 for emissions and exhaust gas temperature control of a 3 heavy-duty diesel engine

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6 Abstract

7 Miller cycle has been shown as a promising engine strategy to reduce in-cylinder nitrogen
8 oxides (NO_x) formation during the combustion process and facilitate its removal in the
9 aftertreatment systems (ATS) by increasing the exhaust gas temperature (EGT). However, the
10 level of NO_x reduction and the increase in EGT achieved by Miller cycle alone is limited.
11 Therefore, research was carried out to investigate the combined use of Miller cycle with other
12 advanced combustion control strategies in order to minimise the NO_x emissions and the total
13 cost of ownership.

14
15 In this paper, the effects of Miller cycle, exhaust gas recirculation (EGR), and post injection
16 were studied and analysed on the performance and exhaust emissions of a single cylinder
17 heavy-duty (HD) diesel engine. A cost-benefit analysis was carried out using the corrected total
18 fluid efficiency, which includes the estimated urea solution consumption in the NO_x
19 aftertreatment system as well as the fuel consumption. The experiments were performed at a
20 low load of 6 bar net indicated mean effective pressure (IMEP). The results showed that the
21 application of a Miller cycle-only strategy with a retarded IVC at -95 CAD ATDC decreased
22 NO_x emissions by 21% to 6.0 g/kWh and increased EGT by 30% to 633K when compared to

23 the baseline engine operation. This was attributed to a reduction in compressed gas temperature
24 by the lower effective compression ratio (ECR) and the in-cylinder mass trapped due to the
25 retarded IVC. These improvements, however, were accompanied by a fuel efficiency penalty
26 of 1%. A further reduction in the level of NO_x from 6.0 to 3.0 g/kWh was achieved through
27 the addition of EGR, but soot emissions were more than doubled to 0.022 g/kWh. The
28 introduction of a post injection was found to counteract this effect, resulting in simultaneous
29 low NO_x and soot emissions of 2.5 g/kWh and 0.012 g/kWh, respectively. When taking into
30 account the urea consumption, the combined use of Miller cycle, EGR and post injection
31 combustion control strategies were found to have relatively higher corrected total fluid
32 efficiency than the baseline case. Thus, the combined “Miller cycle + EGR + post injection”
33 strategy was the most effective means of achieving simultaneous low exhaust emissions, high
34 EGT, and increased corrected total fluid efficiency.

35 **Keywords**

36 Heavy-duty diesel engine, Miller cycle, EGR, post injection, exhaust gas temperatures

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45 **1. Introduction**

46 HD diesel engines are widely used in the commercial vehicle sector due to their high torque
47 output, reliability, and durability, as well as their superior thermal efficiency. However,
48 conventional diesel engines can produce significant pollutants, particularly NO_x and soot
49 emissions. This is due to the existence of locally fuel-rich and high combustion temperature
50 zones resulted from the non-premixed diffusion-controlled combustion [1]. Ever-stringent
51 emissions and fuel efficiency regulations coupled with the customers requirement to reduce the
52 vehicle operational cost, have pushed engine manufacturers to pursue highly efficient and ultra-
53 clean combustion technologies for internal combustion engines [2].

54

55 Low temperature combustion (LTC) concepts such as Homogeneous Charge Compression
56 Ignition (HCCI), Premixed Charge Compression Ignition (PCCI), and Partially Premixed
57 Charge Compression Ignition (PPCI) have been developed to produce low engine-out
58 emissions. These are generally centred on enhancing the quality of fuel-air mixing and
59 increasing the degree of pre-mixed combustion, and therefore reduced the locally fuel-rich
60 regions and peak in-cylinder gas temperature [3–5]. As a result, a simultaneous reduction in
61 NO_x and soot emissions can be obtained, but these LTC modes tend to suffer from higher
62 unburnt hydrocarbons (HC) and carbon monoxide (CO) emissions [6,7]. Additionally, the
63 aggressive use of EGR and the corresponding high boost pressure requirements limited the
64 engine operating range.

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66 In addition to the high levels of NO_x and soot formation in the conventional diesel combustion,
67 relatively lower EGT at low engine operation loads limits the efficient conversion of exhaust
68 pollutants in the ATS. The conversion efficiencies of ATS used in diesel engines such as
69 selective catalytic reduction (SCR), diesel particulate filter (DPF), and diesel oxidation catalyst

70 (DOC) are highly dependent on the EGT [8–10]. Thus, it is essential to develop effective in-
71 cylinder combustion control strategies to manage the EGT as well as the engine-out emissions,
72 decreasing the engine operational cost.

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74 Miller cycle, achieved by early or late intake valve closing (IVC) timings, has been adopted in
75 gasoline engines for reduced pumping loss at low load and suppression of knocking combustion
76 and NO_x reduction at high load operations [11–13]. However, the practical application of the
77 Miller cycle to diesel engines is more challenge. This is mainly due to the limited operation
78 range and relatively smaller benefit achieved when compared to gasoline engines [14], as diesel
79 engines have lower throttling loss and a better scavenging performance. In addition, the Miller
80 cycle would also reduce the volumetric efficiency due to the lower charging efficiency caused
81 by LIVC timing [15,16]. Furthermore, a variable valvetrain and the higher performance
82 turbocharging system to maintain power density would increase the engine's cost [11].
83 Previously, the main interest of applying Miller cycle in diesel engines is for emissions control
84 in marine applications by lowering the in-cylinder gas temperatures at the end of the
85 compression stroke [17–19]. In recent years, however, the main research focus have been on
86 the reduction of NO_x emissions from on-road diesel vehicles as the emissions regulation
87 become increasingly stringent [20–24].

88

89 Wang et al. [25] experimentally investigated the application of Miller cycle in a diesel engine
90 for NO_x emissions control, decreasing NO_x emissions by up to 17.5%. Experimental and
91 numerical studies by Rinaldini et al. [14] showed the Miller cycle strategy could led to
92 simultaneous reduction in NO_x and soot emissions by 25% and 60% respectively, but with a
93 fuel efficiency penalty of 2% in a light-duty diesel vehicle in the European Driving Cycle. The
94 NO_x reduction obtained by Miller cycle was a result of the lower charging efficiency and

95 combustion temperatures [26]. Studies also showed that the Miller cycle combined with higher
96 intake pressure can effectively minimize NO_x emissions while increasing the engine
97 performance [27,28].

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99 Other works have also show that Miller cycle can be an effective strategy for exhaust gas
100 temperature management [8,29,30]. This is mainly attributed to the lower in-cylinder mass
101 resulted from the delayed IVC timing, leading to a lower in-cylinder heat capacity. The
102 experimental and numerical study by Ratzberger et al. [31] demonstrated the potential of an
103 early exhaust valve opening (EEVO) and a late intake valve closing (LIVC) for exhaust thermal
104 management. Garg et al. [32] experimentally examined the influence of in-cylinder charge
105 control via early and late IVC events at a low engine load. A significant increase in turbine
106 outlet temperatures from 195°C to 255°C was achieved by both IVC strategies.

107

108 Despite many studies have demonstrated the capability of Miller cycle for NO_x emissions
109 control, the level of NO_x reduction achieved by Miller cycle alone is limited in diesel engines,
110 particularly at low engine loads [16]. This is primarily due to the relatively small impact on the
111 peak in-cylinder gas temperature at low load operations [26,33]. External EGR, however, is a
112 proven technology for NO_x formation control [34], because of their thermal, chemical, and
113 dilution effects [35]. Some studies [36–39] have investigated the combined use of Miller cycle
114 and EGR and have demonstrated that this strategy is effective in minimising engine-out NO_x
115 emissions.

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117 Kim et al. [16] investigated the effects of LIVC with EGR on engine performance and
118 emissions of a single cylinder diesel engine operating at low load conditions. The LIVC
119 strategy alone demonstrated a limited NO_x emissions reduction. With appropriate combination

120 of LIVC and EGR, a reduction in the NO_x emissions from more than 10 g/kWh to less than 1
121 g/kWh was obtained with slight penalty on the IMEP. Benajes et al. [40] studied the impact of
122 early intake valve closing (EIVC) timing coupled with different intake oxygen concentrations
123 (e.g. EGR rates) at low engine load. According to the analysis, EIVC showed its potential for
124 NO_x emissions control due to the lower peak in-cylinder gas temperature.

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126 However, the lower in-cylinder gas temperatures in the burned zone deteriorated the
127 combustion process and slowed down the oxidation of soot and CO emissions, increasing the
128 levels of engine-out soot and CO emissions [40,41]. Although advanced injection timing,
129 higher injection pressure, and higher boost pressure can be used to compensate this drawback,
130 they can reduce EGT and the NO_x-soot and NO_x-CO trade-offs remain. In addition, these
131 strategies would increase the engine operational cost when considering the total fluid
132 consumption including diesel fuel and the estimated urea consumption in the SCR system.

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134 In order to resolve these issues, a systematic study was carried out to investigate combined use
135 of Miller cycle with advanced in-cylinder combustion control strategies to reduce in-cylinder
136 NO_x formation and improve the EGT management while minimising the engine operational
137 cost as well as soot and CO emissions. In particular, the current work is the first attempt to
138 experimentally explore the Miller cycle combined with EGR and post injection for low load
139 emissions and EGT control. Therefore, this work includes a good novelty and originality.

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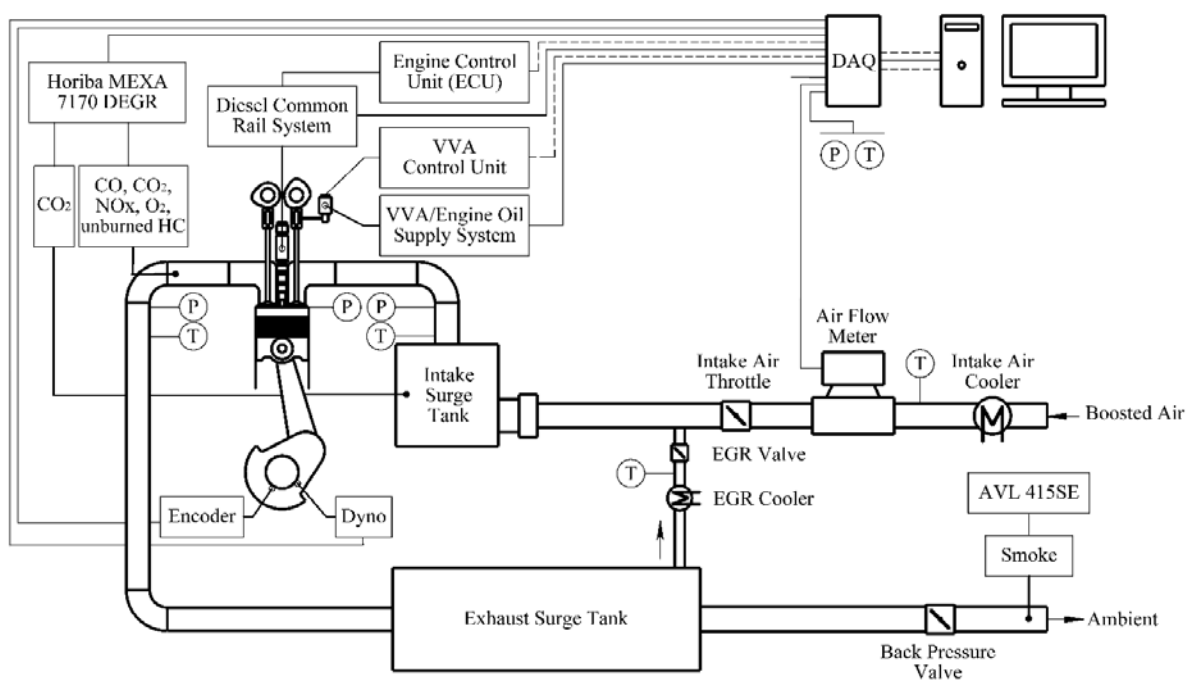
141 The experiments were conducted on a single cylinder HD diesel engine at an engine speed of
142 1250 rpm and load of 6 bar IMEP, which represents a high residency area in a typical HD drive
143 cycle and has a relatively higher weighing factor in the World Harmonized Stationary Cycle
144 (WHSC) [42]. A one-dimensional (1D) engine simulation model was used to calculate the

145 mean in-cylinder gas temperatures (T_m) and burned zone gas temperatures (T_b) based on
146 experimental pressure measurements. The effectiveness of Miller cycle was defined and
147 evaluated, followed by the analysis of a Miller cycle engine operation combined with EGR and
148 post injection. Finally, a cost-benefit and overall emissions analysis of all advanced combustion
149 control strategies were examined and compared to the baseline case.

150 2. Experimental setup

151 2.1 Overview

152 Figure 1 shows the schematic of the experimental setup, which consists of a single cylinder
153 HD diesel engine equipped with a high pressure common rail fuel injection system, an eddy
154 current dynamometer, an external boosting device, as well as measurement equipment and a
155 data acquisition system (DAQ). The key specifications of the engine are listed in Table 1. The
156 design of cylinder head with 4-valve and a stepped-lip piston bowl were based on the Yuchai
157 YC6K six-cylinder engine, while the bottom end/short block was AVL-designed with two
158 counter-rotating balance shafts.



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Figure 1. Layout of the engine experimental setup.

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Table 1. Specifications of the test engine.

Displaced Volume	2026 cm ³
Stroke	155 mm
Bore	129 mm
Connecting Rod Length	256 mm
Geometric Compression Ratio	16.8
Number of Valves	4
Piston Type	Stepped-lip bowl
Diesel Injection System	Bosch common rail
Nozzle design	8 holes, included spray angle of 150°
Maximum fuel injection pressure	2200 bar
Maximum in-cylinder pressure	180 bar

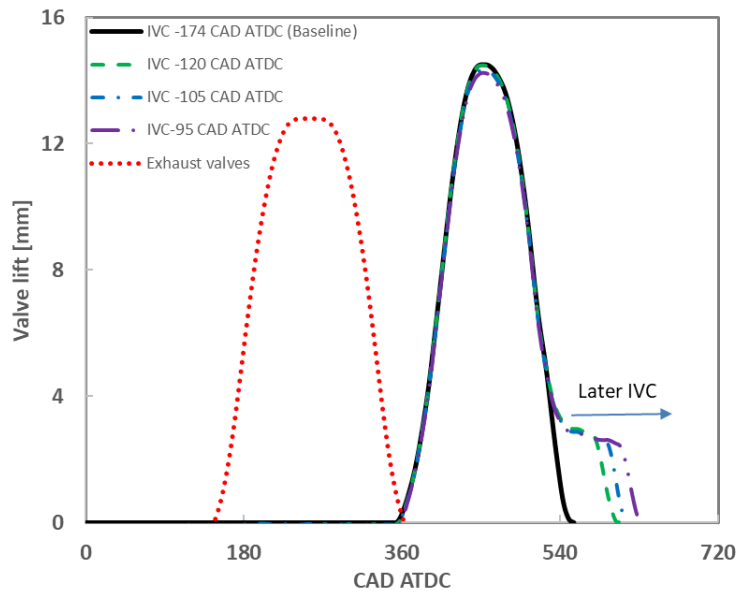
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163 2.2 Variable valve actuation system

164 The engine was equipped with a lost motion variable valve actuation (VVA) system, which
 165 incorporated a hydraulic tappet on the valve side of the rocker arm. This system allowed for
 166 Miller cycle operation via LIVC. The intake valve opening (IVO) and IVC timings of the
 167 baseline case were set at 367 and -174 crank angle degrees (CAD) after top dead centre (ATDC)
 168 respectively. The maximum intake valve lift event was 14mm. Figure 2 shows the intake and
 169 exhaust valve profiles for the baseline and Miller cycle operations. All valve events in this
 170 study were considered at 1 mm valve lift. The effective compression ratio (ECR) was calculated
 171 as

$$172 \quad ECR = \frac{V_{ivc_eff}}{V_{tdc}} \quad (1)$$

173 where V_{tdc} is the cylinder volume at TDC position, and V_{ivc_eff} is the effective cylinder
 174 volume where the in-cylinder compressed air pressure is extrapolated to be identical to the
 175 intake manifold pressure [43,44].



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Figure 2. Fixed exhaust and variable intake valve lift profiles.

178 2.3 Intake and exhaust systems

179 As depicted in Figure 1, the compressed air was supplied by an external supercharger with
 180 closed loop control. The intake mass flow rate was measured by a thermal mass flow meter.
 181 Two large surge tanks were installed in the intake and exhaust systems to damp out the pressure
 182 fluctuations in the intake and exhaust manifolds resulted from the gas exchange dynamics of
 183 the engine. Both instantaneous intake and exhaust manifold pressures were measured by piezo-
 184 resistive pressure transducers. The intake manifold pressure was fine adjusted by an intake
 185 throttle valve while the exhaust back pressure was independently controlled through a butterfly
 186 valve located downstream of the exhaust surge tank. External cooled EGR was introduced into
 187 the engine using a pulse width modulation controlled poppet valve installed upstream of the
 188 intake surge tank, and the pressure differential between the intake and exhaust manifolds.
 189 Engine coolant and lubrication oil were supplied externally and their temperatures along with
 190 the intake air and EGR flow temperatures were controlled by the water cooled heat exchangers.

191 2.4 Fuel delivery system

192 During the experiments, the diesel fuel was injected into the engine by a high-pressure solenoid
 193 injector through a high pressure pump and a common rail with a maximum fuel pressure of

194 2200 bar. A dedicated electronic control unit (ECU) was used to control fuel injection
195 parameters such as injection pressure, start of injection (SOI), and the number of injections per
196 cycle. The fuel consumption was determined by measuring the total fuel supplied to and from
197 the high pressure pump and diesel injector via two Coriolis flow meters.

198 **2.5 Exhaust emissions measurement**

199 A Horiba MEXA-7170 DEGR emission analyser was used to measure exhaust gases (NO_x,
200 HC, CO, and carbon dioxide (CO₂)). To allow for high pressure sampling and avoid
201 condensation, a high pressure sampling module and a heated line were used between the
202 exhaust sampling point and the emission analyser. The smoke number was measured
203 downstream of the exhaust back pressure valve using an AVL 415SE Smoke Meter, and
204 thereafter converted from filter smoke number (FSN) to mg/m³ [45]. All the exhaust gas
205 components were converted to net indicated specific gas emissions (in g/kWh) according to
206 [46]. In this study, the EGR rate was defined as the ratio of the measured CO₂ concentration in
207 the intake surge tank ((CO₂%)_{intake}) to the CO₂ concentration in the exhaust manifold
208 ((CO₂%)_{exhaust}) as

$$209 \quad EGR \text{ rate} = \frac{(CO_2\%)_{intake}}{(CO_2\%)_{exhaust}} * 100\% \quad (2)$$

210 **2.6 Engine test data analysis**

211 The instantaneous in-cylinder pressure was measured by a piezo-electric pressure transducer
212 with a sampling revolution of 0.5 CAD. The in-cylinder pressure data was recorded through a
213 charge amplifier and was averaged over 200 engine cycles to calculate the apparent heat release
214 rate (HRR). According to [1], the apparent HRR was calculated as

$$215 \quad HRR = \frac{\gamma}{(\gamma-1)} p \frac{dV}{d\theta} + \frac{1}{(\gamma-1)} V \frac{dp}{d\theta} \quad (3)$$

216 where γ is defined as the ratio of specific heats and assumed constant at 1.33 throughout the
217 engine cycle; V and p are the in-cylinder volume and pressure, respectively; θ is the crank
218 angle.

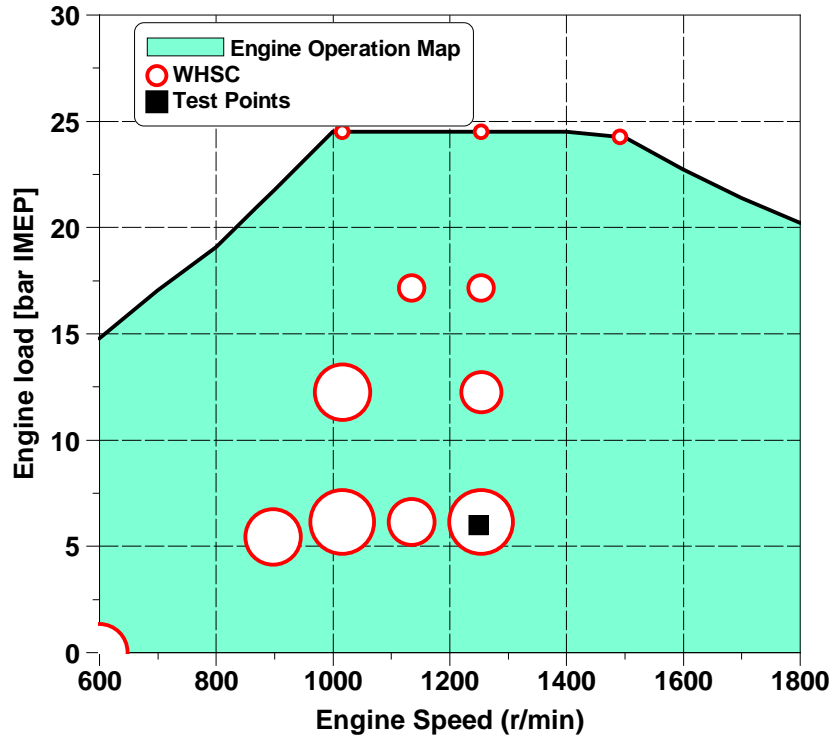
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220 In this study, the CA10, CA50 (combustion phasing) and CA90 were defined as the crank angle
221 when the fuel mass fraction burned (MFB) reached 10%, 50%, and 90%, respectively. Ignition
222 delay was defined as the period between the SOI and the start of combustion (SOC), denoted
223 as 0.3% MFB point of the average cycle. The combustion stability was measured by the
224 coefficient of variation of the IMEP (COV_{IMEP}) over the sampled cycles.

225 **3. Methodology**

226 **3.1 Engine operation**

227 The experimental work was carried out at a speed of 1250 rpm and a low engine load of 6 bar
228 IMEP. Figure 3 shows the operation points of WHSC for HD diesel engines. The size of the
229 circle represents the weighting factor. A larger circle indicates a higher relative weight of the
230 engine operation condition over the WHSC cycle. As shown in Figure 3, the test point is located
231 in a high residency area of a typical HD diesel engine drive cycle, representing an engine
232 operating condition where exhaust gas temperature is low and exhaust emissions such as NO_x
233 and soot are relatively high.



234
 235 **Figure 3. Experimental test point and WHSC operation conditions over an estimated HD diesel engine speed-**
 236 **load map.**

237 Table 2 summaries the engine operating conditions for the different combustion control
 238 strategies investigated. The effect of Miller cycle was studied with two LIVC settings and three
 239 intake pressures. The ECR was decreased from 16.8 to 13.5 and 12.5 for the Miller cycle cases.
 240 In the case of combined use of Miller cycle and EGR, the in-cylinder air flow mass was
 241 decreased noticeable, resulting in poor combustion instability and excessive smoke. Therefore,
 242 a moderate EGR rate of 15% along with an ECR of 13.5 were employed when the EGR was
 243 combined with the Miller cycle. Finally, the combined effect of Miller cycle, EGR and post
 244 injection was evaluated at two ECRs of 13.5 and 14.5 and 15% EGR.

245 **Table 2. Engine operating conditions for the different combustion control strategies. (? Add SOI valves)**

Parameter	Value			
Speed	1250 rpm			
Load	6 bar IMEP			
Injection pressure	1160 bar			
Intake temperature	303 ± 1K			
Testing modes	Baseline	Miller cycle	Miller cycle + EGR	Miller cycle + EGR + Post injection

Intake pressure	1.44 bar	1.44, 1.64, and 1.84 bar	1.44 bar	1.44 bar
Exhaust pressure	0.10 bar higher than the intake pressure			
IVC timing	-174 CAD ATDC	-95, -105 CAD ATDC	-105 CAD ATDC	-120 and -105 CAD ATDC
ECR	16.8	13.5 and 12.5	13.5	14.5 and 13.5
EGR	0%	0%	15%	15%
Post fuel injection	No	No	No	Yes

246

247 During the experiments, a small pilot injection of 3 mm³ with a constant dwell time of 1 ms
248 prior to the main injection timing was employed in order to keep the maximum pressure rise
249 rate (PRR) below 20 bar/CAD. The SOI of main injection was optimised to give the maximum
250 fuel conversion efficiency at each operating condition. The coolant and oil temperatures were
251 kept within 358 ± 2 K. Oil pressure was maintained within 4.0 ± 0.1 bar. Stable engine
252 operation was determined by controlling the COV_{IMEP} below 3%. The specifications of
253 measurement equipment can be found in Appendix A.

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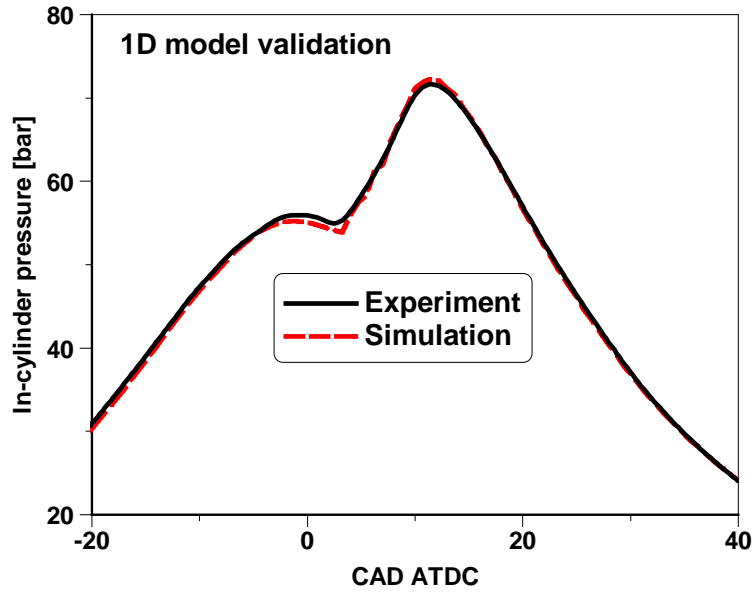
255 **3.2 Engine Modelling and Validation**

256 In order to better analyse the influence of Miller cycle, EGR, and post injection on in-cylinder
257 combustion process, a 1D thermodynamic simulation has been carried out using Ricardo Wave
258 software to estimate the mean in-cylinder gas temperatures and burned zone gas temperatures.
259 This software use finite difference method to solve the 1D unsteady compressible flow
260 equations, including conservation of mass, momentum and energy [47].

261 The geometric dimensions of intake and exhaust pipes and surge tanks were accurately
262 measured from the engine bench while the ports and valve flow coefficients were provided by
263 the engine supplier. The combustion process was simulated by using the experimentally
264 derived heat release rate profile based on the measured in-cylinder pressure. The heat transfer

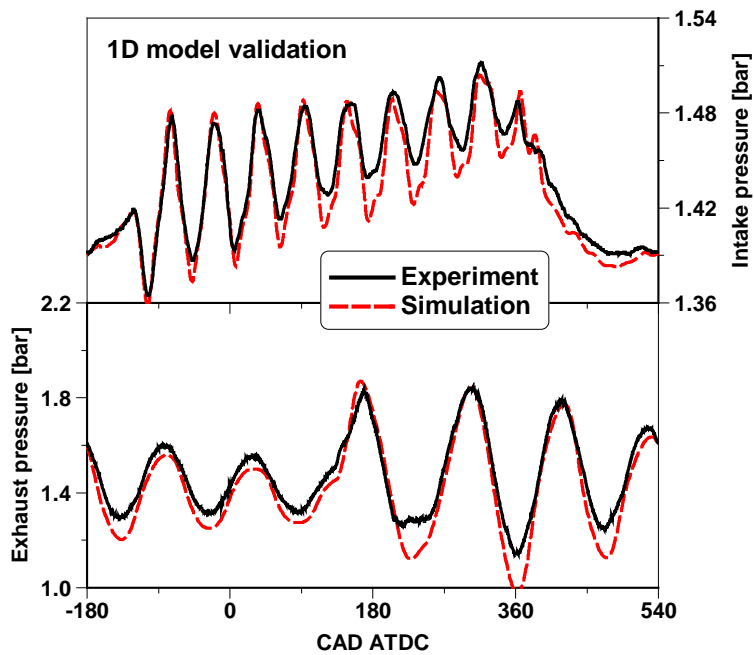
265 was calculated by the Woschni heat transfer model [48], which assumes a simple convective
266 heat transfer from a confined volume surrounded on all sides by cylinder walls. The
267 thermodynamic state of the in-cylinder gas was estimated considering the interactions among
268 the enthalpy of fluxes in and out of the chamber, heat transfer, and piston work by using a two
269 zone model, as proposed by Saegusa et al. [49].

270 The calculated intake air mass flow rate, IMEP, in-cylinder pressure, intake and exhaust
271 manifold pressures were calibrated against the experimental data. In all cases, the maximum
272 pressure is validated to within 3% of the experimental data and the modelled and experimental
273 values for average intake and exhaust manifold pressures were computed to be within 1.5%.
274 Figure 4 shows the in-cylinder pressure while Figure 5 shows the intake and exhaust pressures.
275 The pressure variations in the intake and exhaust systems are primarily due to the piston
276 velocity, valve open area variation, and the unsteady gas-flow effects that result from these
277 geometric variations. The primary frequency in both the intake and exhaust corresponds to the
278 frequency of the intake and exhaust processes [1]. Results showed that there is a good
279 agreement between the simulated and experimental results of in-cylinder pressure as well as
280 intake and exhaust pressures for the “Miller cycle + EGR” strategy. Thus, the validated 1D
281 engine model could be used to calculate T_m and T_b .



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Figure 4. Experimental and simulated in-cylinder pressures for the “Miller cycle + EGR” strategy.



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Figure 5. Experimental and simulated intake and exhaust manifold pressures for the “Miller cycle + EGR” strategy.

288 4. Results and discussion

289 The experimental results were divided into four sections. In the first section, the isolated effect
290 of Miller cycle under three intake pressures (P_{int}) is investigated. This is followed by an
291 investigation of the effect of Miller cycle combined with EGR for NO_x reduction. The third

292 section shows the potential of post injection to minimise soot and CO emissions from the
293 “Miller cycle + EGR” strategy. Finally, the effectiveness of the “Miller cycle + EGR + post
294 injection” strategy for emissions and EGT control at low engine load was evaluated by means
295 of a cost-benefit and overall exhaust emissions analysis.

296 **4.1 The effect of Miller cycle and boost pressure**

297 Figure 6 shows the in-cylinder pressure, HRR, and injector signal for the baseline engine
298 operation and Miller cycle cases (IVC at -95 CAD ATDC) with a constant main SOI at -4 CAD
299 ATDC. Figure 7 depicts T_m and T_b , which were calculated using the 1D engine thermodynamic
300 model. The intake pressure for the Miller cycle strategy was varied between 1.44 bar (e.g. same
301 P_{int} as the baseline) and 1.84 bar (e.g. same lambda as the baseline).

302

303 The Miller cycle case with a constant P_{int} of 1.44 bar (red dashed line) was characterised by a
304 significantly lower in-cylinder pressure and a higher degree of premixed combustion as
305 suggested by the appearance of the pre-mixed heat release before the main heat release process.
306 This was the result of the lower ECR, which decreased the compressed air pressure and
307 temperature and increased the ignition delay.

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309 Relatively higher in-cylinder pressure and peak HRR were observed when increasing the P_{int}
310 of the Miller cycle cases due to the higher in-cylinder charge density. The first peak heat release
311 disappeared as a result of the higher compression pressure and earlier SOC. The Miller cycle
312 strategy with higher intake pressure achieved a similar HRR profile to the baseline case when
313 operating with the same lambda. It is important to note that the in-cylinder gas pressure was
314 slightly lower than that of the baseline case. This was a result of the lower in-cylinder gas
315 temperatures achieved by this Miller cycle case with a P_{int} of 1.84 bar.

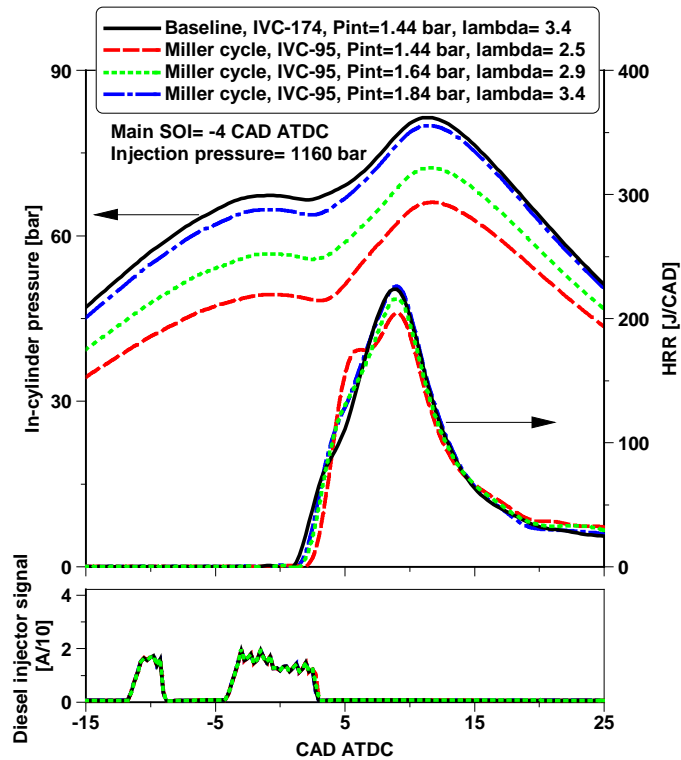
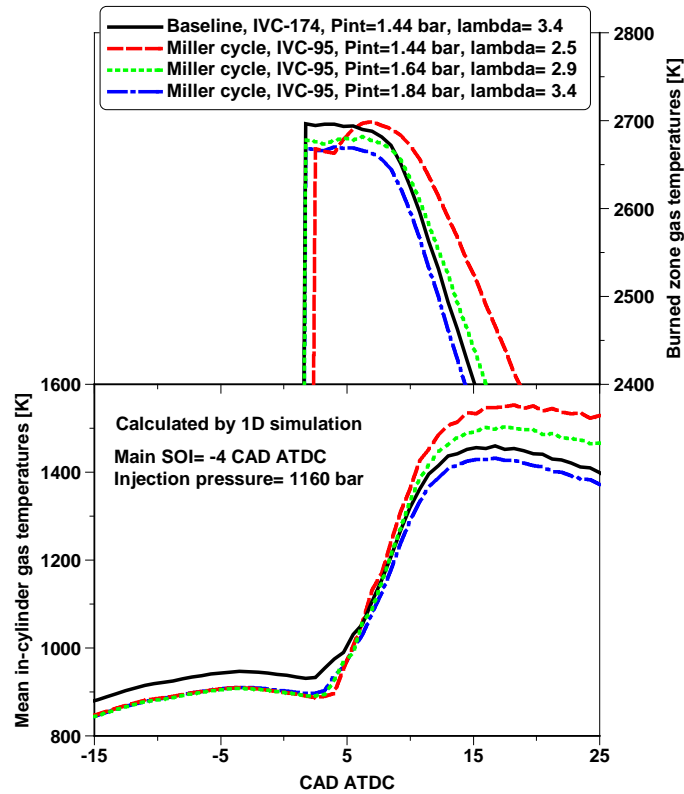


Figure 6. In-cylinder pressure, HRR, and diesel injection signal for the baseline and Miller cycle cases.

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319 The application of Miller cycle decreased the compressed air temperature by approximately 30
320 K because of the LIVC and lower ECR. However, the Miller cycle operation with constant P_{int}
321 of 1.44 bar reduced the in-cylinder fresh charge and hence the total heat capacity, leading to a
322 higher mean gas temperature, T_m , as reported in our previous work [26] and by Benajes et al.
323 [40] with EIVC. When operating with the same lambda, the peak of the mean gas temperature
324 T_m decreased with increased P_{int} and was lower than that of the baseline case. The burned gas
325 temperature, T_b , decreased slightly with the use of Miller cycle due to the lower compressed
326 gas temperatures (e.g. prior to the SOC). A relatively higher T_b after SOC was observed in the
327 Miller cycle case with a constant P_{int} of 1.44 bar due to lower in-cylinder mass, as indicated by
328 the lambda.



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Figure 7. Calculated mean in-cylinder gas temperatures and burned zone temperatures for the baseline and Miller cycle cases.

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Figure 8 shows the CA50, CA90, CA50-CA90, and combustion duration (CA10-CA90) over a sweep of main SOI. There was little difference in CA50 of the baseline and Miller cycle cases at a given main SOI. The Miller cycle with a P_{int} of 1.44 bar delayed the CA90 due to the later ignition and slowed down combustion process in the late combustion phase. This resulted in a relatively longer period time of CA50-CA90 as well as combustion duration. At the same lambda, the higher P_{int} improved the in-cylinder oxygen availability, which helped accelerate the combustion process. As a result, the CA90 was advanced and thus a shorter period time of CA50-CA90 and combustion duration. As a result, the combustion characteristics of the Miller cycle was similar to that of the baseline condition when operating with the same lambda.

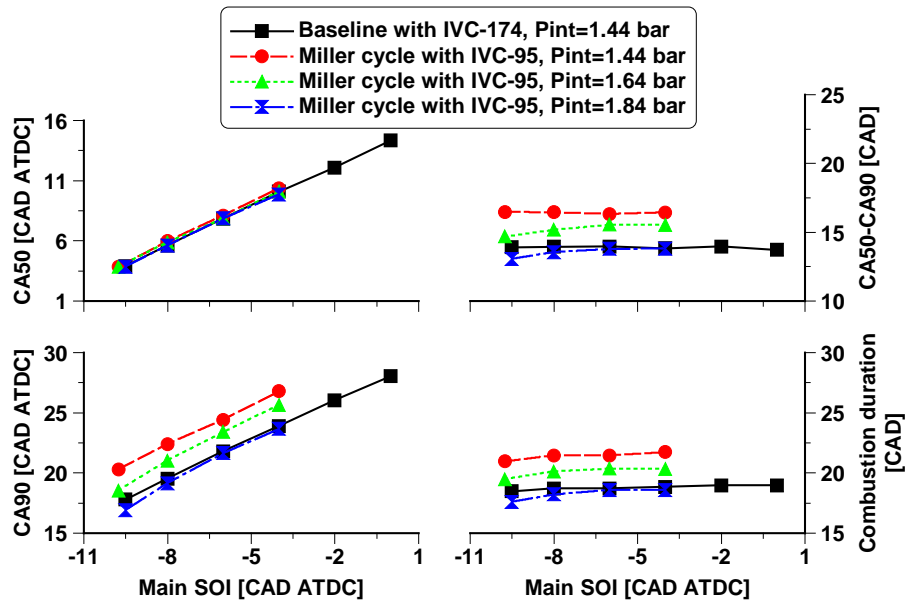


Figure 8. The effect of Miller cycle on combustion characteristics.

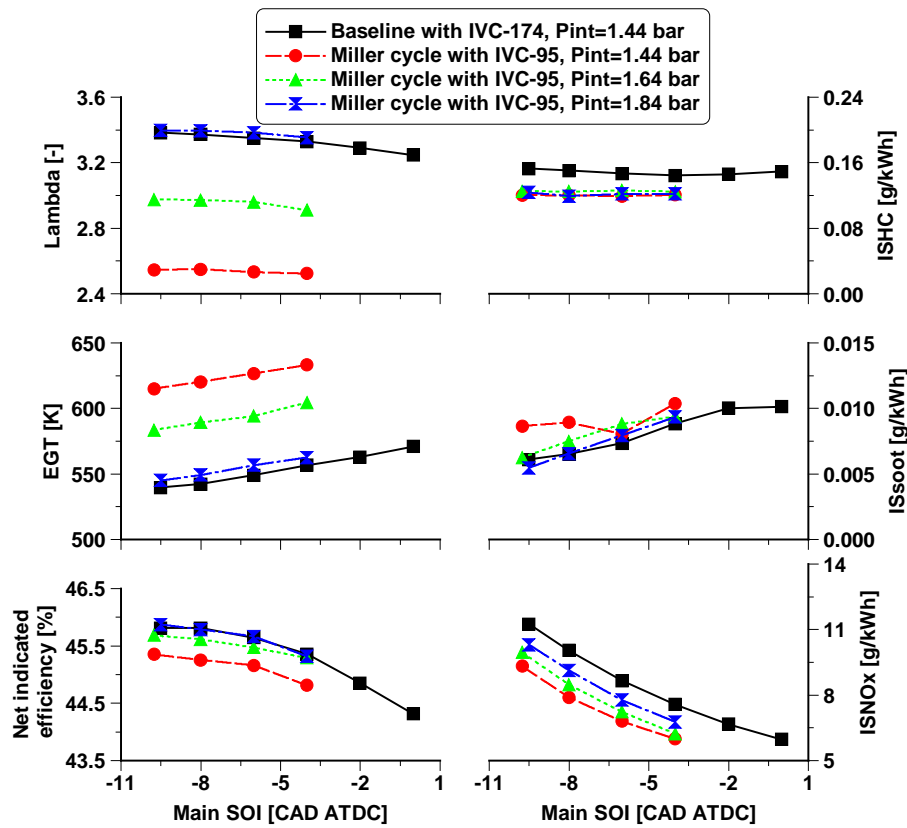
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344 Figure 9 depicts engine performance parameters and net indicated specific emissions. The use
 345 of Miller cycle with a constant P_{int} of 1.44 bar significantly reduced the lambda value due to
 346 lower in-cylinder mass trapped. This was also the reason for the 75 K increase in the EGT,
 347 which is an important achievement considering that the EGT of 539 K of the baseline case was
 348 probably insufficiently high for efficient aftertreatment operation. According to [50], the
 349 optimum performance of the ATS is achieved when the inlet temperature at ATS is maintained
 350 between 523 K and 723 K. It is also important to note that the EGT showed in this study is the
 351 engine-out temperature, which will be further reduced downstream of the turbocharger in a
 352 production multi-cylinder engine.

353

354 The net indicated efficiency of the Miller cycle at a P_{int} of 1.44 bar was decreased by
 355 approximately 1% in comparison with the baseline engine operation. This was attributed to the
 356 extended CA50-CA90 period and longer total combustion duration, and higher heat losses to
 357 cylinder walls due to higher T_m (see Figure 7). The use of higher P_{int} to maintain the in-cylinder
 358 lambda value helped increasing the net indicated efficiency due to the higher in-cylinder
 359 oxygen availability, but adversely affected the gains obtained in terms of EGT and NOx

360 emissions. Miller cycle operation with a P_{int} of 1.84 bar achieved a net indicated efficiency as
 361 high as the baseline case.



362
 363 Figure 9. The effect of Miller cycle on engine performance and exhaust emissions.
 364

365 The Miller cycle operation reduced the NOx emissions by 21% to 6 g/kWh at the expense of
 366 slightly higher soot emissions. This was primarily a result of the lower in-cylinder air mass
 367 trapped and reduced initial burned zone gas temperature. Later main SOI effectively decreased
 368 combustion temperatures and therefore NOx emissions. As the P_{int} of the Miller cycle cases
 369 was increased, soot emissions were reduced and NOx emissions increased. It can also be seen
 370 from Figure 9 that the application of Miller cycle decreased unburned HC emissions, regardless
 371 of the intake pressure used. This was possibly attributed to the relatively lower lambda and
 372 higher T_m , which can improve the oxidation rates during the expansion stroke.

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374 The above results have shown that Miller cycle was effective in the EGT management and
 375 NOx emissions control but at the expense of lower net indicated efficiency. Despite the

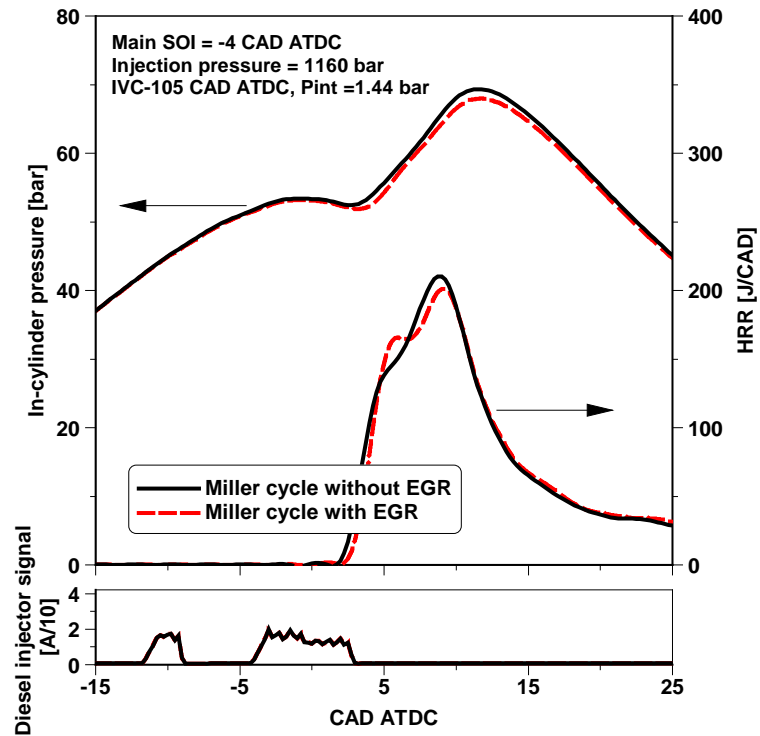
376 improvements in the net indicated efficiency obtained when increasing the P_{int} , a higher lambda
377 can impair the EGT and the NO_x emissions benefits introduced by the Miller cycle strategy.
378 In addition, it should be noted that a conventional turbocharging system is likely not able to
379 deliver the required airflow rate at such low load operations. For this reason, a more
380 sophisticated boosting system, such as a two-stage variable geometry turbocharger
381 configuration, would be needed [51,52].

382 **4.2 The combined effect of Miller cycle with EGR**

383 EGR was introduced to the Miller cycle operation in order to further reduce the engine-out
384 NO_x levels, as it is an effective and well established technology for NO_x emissions control
385 from internal combustion engines [34,53,54]. As explained previously, ECR was set at 13.5 to
386 maintain the combustion stability. Considering the fact that a higher levels of P_{int} might not be
387 achievable in a turbocharged multi-cylinder engine operating at low load condition [55], the
388 P_{int} was kept at 1.44 bar.

389

390 Figure 10 shows the in-cylinder pressure, HRR, and injector signal for the Miller cycle cases
391 with and without EGR with a constant main SOI at -4 CAD ATDC. The addition of EGR
392 delayed the SOC due to the lower in-cylinder oxygen concentration (dilution effect) and lower
393 compression temperature because of the lower specific heat values [34]. This increased the
394 degree of premixed combustion and decreased both the peak HRR and peak in-cylinder
395 pressure.



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Figure 10. In-cylinder pressure, HRR, and diesel injector signal for Miller cycle cases with and without EGR.

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Figure 11 depicts the burned zone gas temperatures and mean in-cylinder gas temperatures

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calculated using the 1D engine model. The dilution effect and higher heat capacity (thermal

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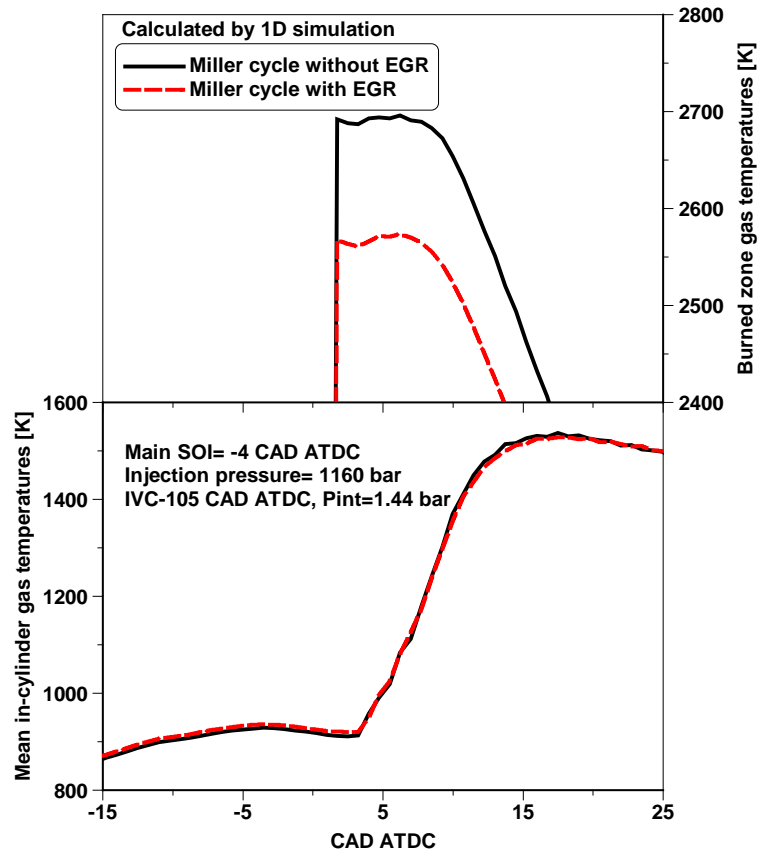
effect) introduced by EGR decreased the T_b by more than 100 K, consequently helping to curb

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NO_x formation. No significant differences in the T_m curves were observed when comparing

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the Miller cycle cases with and without EGR.



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Figure 11. Calculated mean in-cylinder gas temperatures and burned zone gas temperatures for Miller cycle cases with and without EGR.

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Figure 12 shows the resulting heat release characteristics. The addition of EGR to a Miller cycle operation had little impact on the CA50, but slowed down the late combustion process (e.g. delayed CA90). This was a result of the later SOC and the lower in-cylinder oxygen concentration. The combination of these effects with a lower peak T_b yielded a longer period of CA50-CA90, as well as increased the total combustion duration (CA10-CA90) by up to 3 CAD when compared to the Miller cycle cases without EGR.

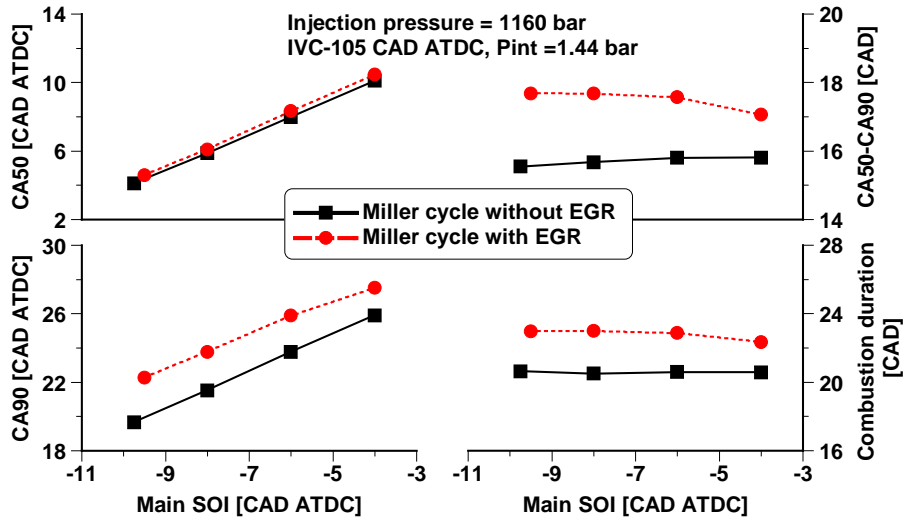


Figure 12. The effect of EGR on combustion characteristics of Miller cycle cases.

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Figure 13 depicts the engine performance parameters and net indicated specific emissions. The use of EGR decreased the lambda, which contributed to an increase of approximately 7 K in the EGT. Additionally, the lower T_b and in-cylinder oxygen concentration for the conditions with 15% EGR curbed the NO_x emissions by up to 58%. However, the later and longer combustion process of the Miller cycle cases with EGR decreased the net indicated efficiency by up to 0.8% and significantly increased soot and CO emissions. These results can limit the potential of the “Miller cycle + EGR” strategy for efficient and clean low load engine operation.

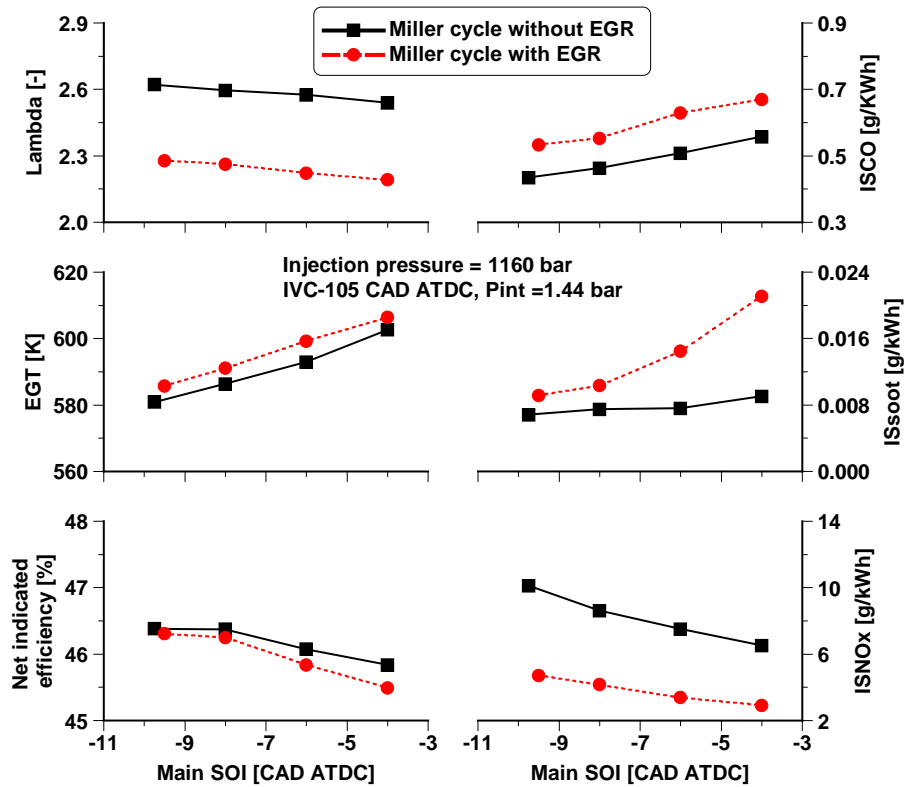


Figure 13. The effect of EGR on engine performance and exhaust emissions of Miller cycle cases.

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4.3 Exploring the potential of a post injection strategy

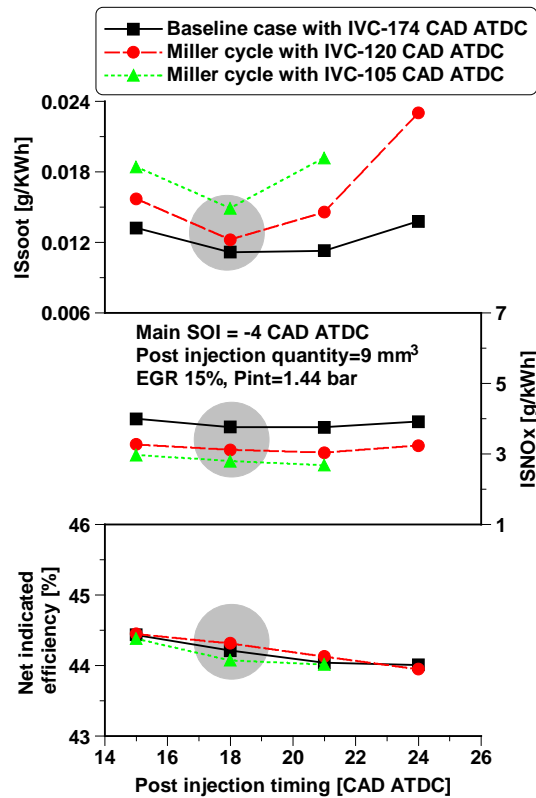
The use of post injections has been demonstrated as an effective means of mitigating soot and CO emissions [56,57]. Therefore, the post injection strategy was investigated in an attempt to improve upon the performance and emissions of the “Miller cycle + EGR” strategy.

4.3.1 The optimization of the post injection

Before analysing the potential of the “Miller cycle + EGR + post injection” strategy, the post injection timing and fuel quantity were optimised in order to achieve high efficiency and low engine-out emissions. The optimization was carried out at three IVC timings, including the baseline at -174 CAD ATDC and Miller cycle cases at -120 and -105 CAD ATDC. The main SOI was held constant at -4 CAD ATDC. At a given post injection timing, a sweep of post injection quantity of 6, 9, 12, 15 mm³ (equivalent to 7%, 11%, 15%, and 20% of the total fuel injection quantity, respectively) was performed.

439

440 Firstly, the post injection timing was varied between 15 and 24 CAD ATDC while the quantity
 441 was held constant at 9 mm³. Figure 14 shows that the soot emissions were sensitive to the post
 442 injection timing, decreasing initially and increasing significantly with the most retarded post
 443 injection timings. This was likely attributed to the trade-off effect of post injected fuel on soot
 444 formation and oxidation rates in the late combustion phase [56]. Meanwhile, changes in the
 445 post injection timing resulted in little impact on NO_x emissions. Overall, the post injection
 446 timing of 18 CAD ATDC achieved the lowest soot emissions with relatively small decrease in
 447 net indicated efficiency, and thus was selected for the post injection quantity sweep.



448
 449 Figure 14. Optimization of the post injection timing.
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451 Figure 15 depicts the optimization of the post injection quantity while maintaining the post
 452 injection timing constant at 18 CAD ATDC. An increase in the post injection quantity reduced
 453 soot and NO_x emissions at the expense of a reduction in the net indicated efficiency, regardless
 454 of the IVC timing. A post injection quantity of 12 mm³ achieved the optimum trade-off between
 455 engine performance and emissions. Therefore, the next subsection explores the potential of the

456 “Miller cycle + EGR + post injection” strategy using a post injection of 12 mm³ at 18 CAD
 457 ATDC.

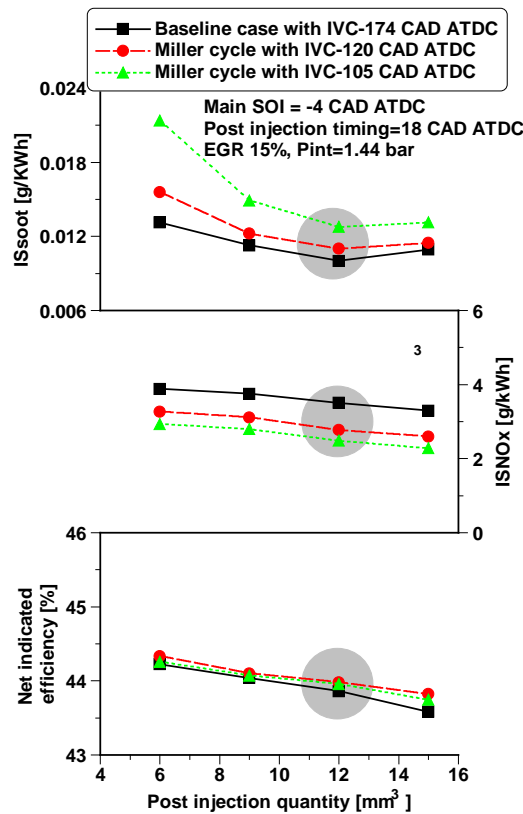
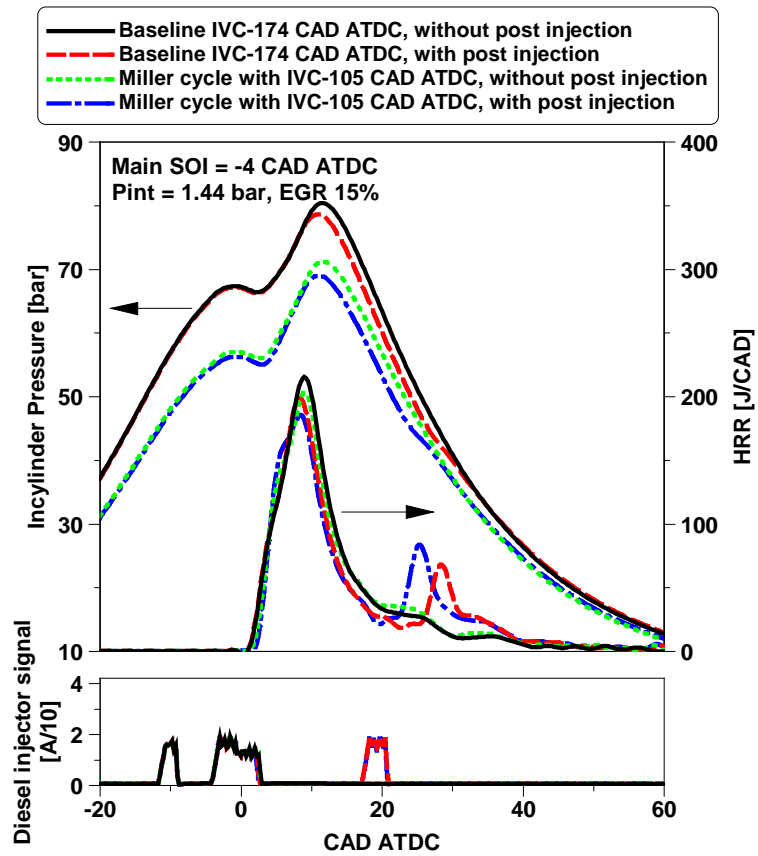


Figure 15. Optimization of the post injection quantity.

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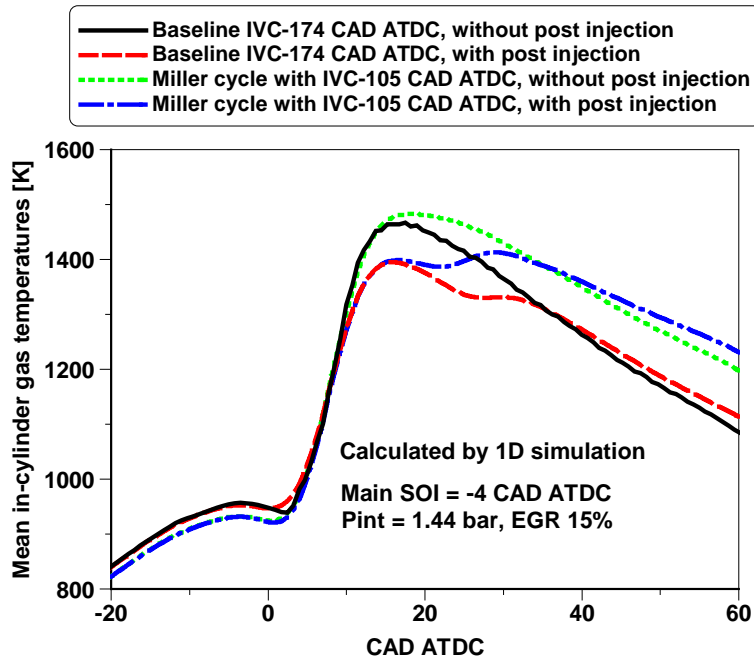
461 4.3.2 The combined effect of the post injection with Miller cycle and EGR

462 Figure 16 depicts the in-cylinder pressure, HRR, and injector signal for the baseline and Miller
 463 cycle cases with and without the optimized post injection strategy. The use of post injection
 464 yielded a second and late peak HRR, which represented the combustion of the post injected
 465 fuel. The decreased fuel mass in the main injection consequently, lowered the first peak HRR
 466 and thus the peak in-cylinder pressure when compared to the baseline cases. Figure 17 shows
 467 that the post injection strategy also decreased the peak mean in-cylinder gas temperature but
 468 increased the average temperature during the late stages of the combustion process.
 469 Additionally, it can be seen that the relatively richer mixture of the Miller cycle cases (e.g.
 470 lower lambda at a constant P_{int}) shortened the ignition delay of the post injected fuel in
 471 comparison with that of the baseline case with a post injection.



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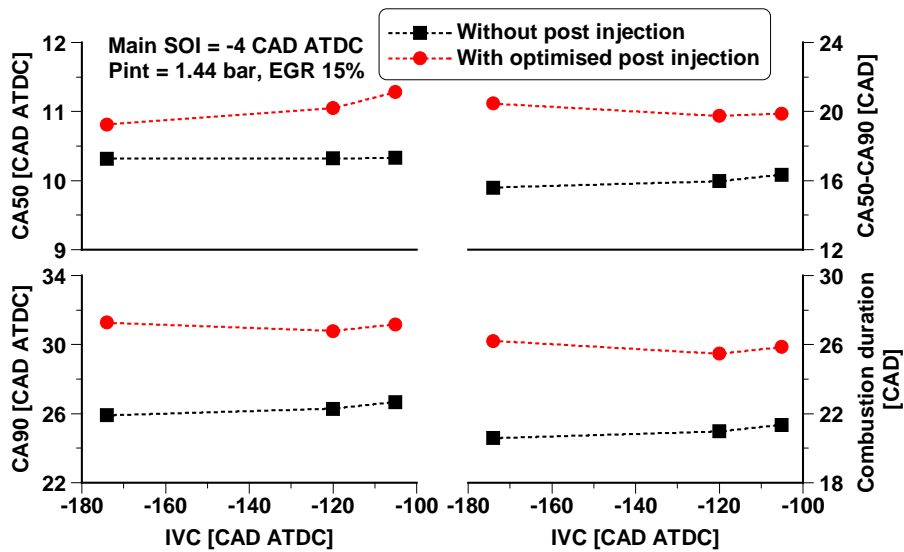
Figure 16. In-cylinder pressure, HRR, and diesel injector signal for the baseline and Miller cycle cases with and without post injection.



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Figure 17. Calculated mean in-cylinder gas temperatures for the baseline and Miller cycle cases with and without post injection.

481 Figure 18 compares the resulting heat release characteristics for different IVC timings with and
 482 without post injection. The addition of the post injection delayed the CA50 and CA90 as more
 483 fuel was burned during a relatively colder and later combustion phase. As a result, the period
 484 of CA50-CA90 was longer for the cases with post injection. These effects contributed to an
 485 increase in the combustion duration by up to 5 CAD when compared to the cases without post
 486 injection.

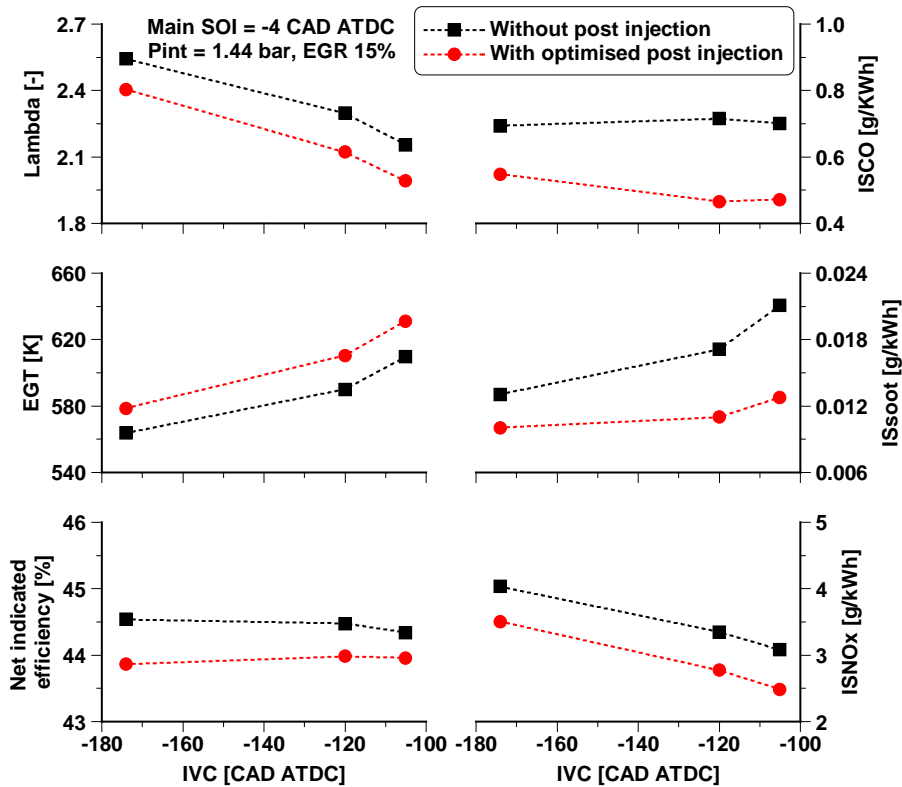


487 Figure 18. The effect of post injection on combustion characteristics.

488
 489
 490 Figure 19 shows the lambda was slightly reduced and EGT was increased with the addition of
 491 post injection. This was because part of diesel fuel was consumed later in the cycle and
 492 consequently a decrease in net indicated efficiency. Nevertheless, the difference of net
 493 indicated efficiency penalty between the cases with and without post injection was reduced
 494 with delayed IVC timing, decreasing from 1.6% at the baseline IVC to below 1% in the Miller
 495 cycle with IVC-105 CAD ATDC. This was probably due to the relatively faster heat release of
 496 the post injected fuel in the Miller cycle cases.

497
 498 The emission results showed that soot emissions were significantly reduced by the post
 499 injection. The primary reasons for this improvement are likely the shorter injection durations,
 500 the enhanced fuel-air mixing, and the increased late combustion temperature achieved with the

501 multiple injections [57]. Meanwhile, the use of a post injection led to a reduction in NOx
 502 emissions of 20% on average. This was due to the lower lambda (e.g. less oxygen availability)
 503 and most likely the lower burned zone gas temperatures achieved with this injection strategy.
 504 The levels of CO showed a similar trend to the soot emissions, and were also reduced with the
 505 post injection strategy attributed to the increased late combustion temperatures.



506 Figure 19. The effect of post injection on engine performance and exhaust emissions.
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509 4.4 Cost-benefit and overall analysis

510 In this section, the total fluid consumption (\dot{m}_{total}) was calculated by summing the measured
 511 diesel flow rate (\dot{m}_{diesel}) and aqueous urea solution consumption in an SCR system (\dot{m}_{urea}),
 512 which allowed for a cost-benefit and overall emissions analysis of the different combustion
 513 control strategies.

$$514 \quad \dot{m}_{total} = \dot{m}_{urea} + \dot{m}_{diesel} \quad (4)$$

515

516 Since the urea consumption in the SCR system depends on the operating conditions as well as
517 engine-out NO_x emissions, reductions in the levels of engine-out NO_x can help minimise the
518 urea flow rate. According to [58,59], the urea consumption in the SCR system can be estimated
519 as 1% of the diesel equivalent fuel flow per g/kWh of NO_x reduction necessary to meet the
520 Euro VI limit (NO_{x_{EuroVI}}) of 0.4 g/kWh.

521

$$522 \quad \dot{m}_{urea} = 0.01 (\text{ISNO}_x - \text{NO}_{x_{EuroVI}}) \dot{m}_{diesel} \quad (5)$$

523

524 which is then used in the calculation of the corrected total fluid efficiency

525

$$526 \quad \text{Corrected total fluid efficiency} = \frac{P_i}{\dot{m}_{total} \text{LHV}_{diesel}} \quad (6)$$

527

528 where P_i is the net indicated power and LHV_{diesel} is the diesel lower heating value of 42.9
529 MJ/kg.

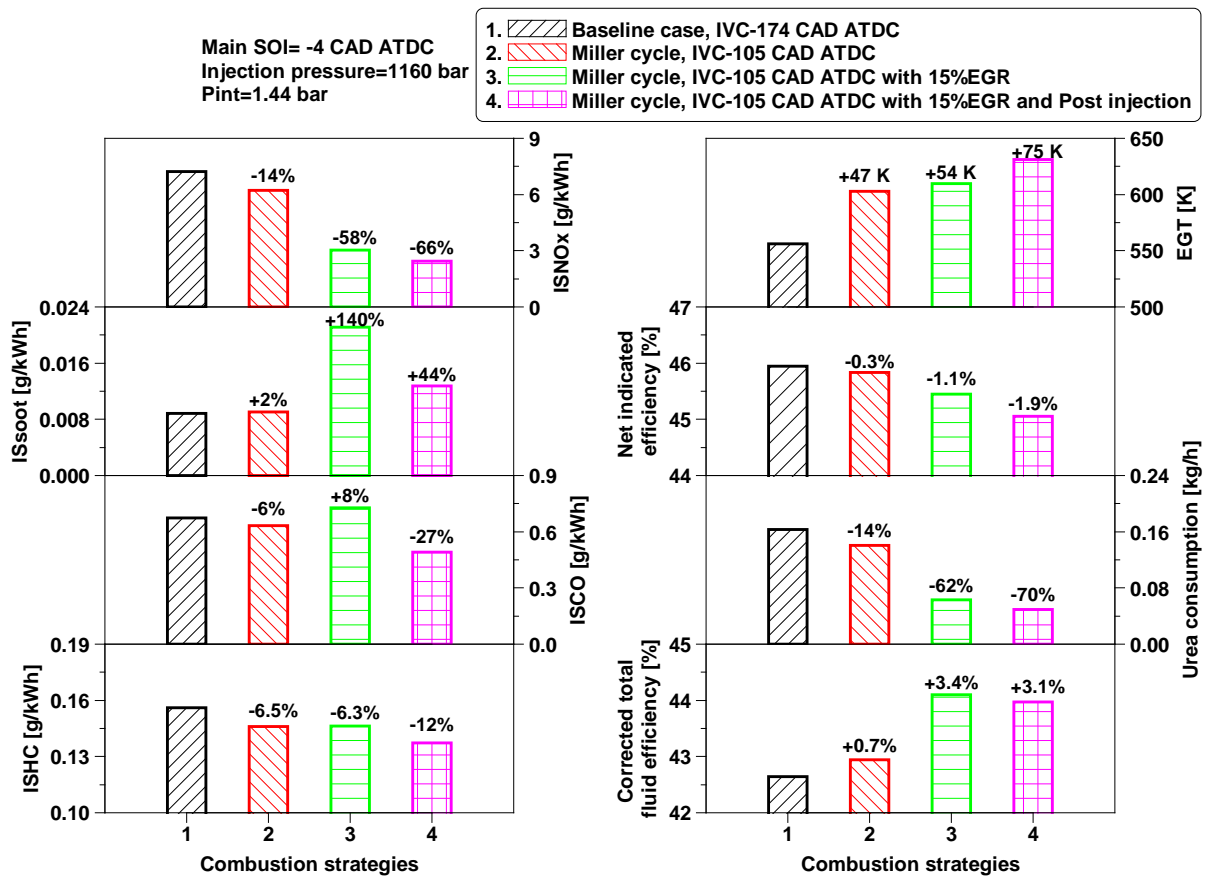
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531 Figure 20 provides an overall assessment of the potential of Miller cycle with EGR and post
532 injection to improve upon the EGT, exhaust emissions, and engine operational cost of a diesel
533 engine operating at low load. The results of various advanced combustion control strategies are
534 compared to the baseline case using a constant main SOI of -4 CAD ATDC and P_{int} of 1.44 bar.
535 Additionally, all Miller cycle cases were operated with a same IVC at -105 CAD ATDC in
536 order to allow a fair comparison.

537

538 Compared to the baseline case, a “Miller cycle-only” strategy decreased NO_x emissions by 14%
539 accompanied with an increase of 47 K in EGT. This was achieved at the expense of a reduction
540 of 0.3% in the net indicated fuel conversion efficiency. The combination of a Miller cycle

541 strategy with EGR achieved better improvement in NO_x emissions and EGT, decreasing NO_x
 542 emissions by 58% and increasing EGT by 54 K in comparison with the baseline case. This
 543 significant improvement was obtained with a reduction of 1.1% in the net indicated fuel
 544 conversion efficiency. However, the “Miller cycle + EGR” strategy increased soot emissions
 545 significantly from 0.009 g/kWh in the baseline case to 0.022 g/kWh and also increased CO
 546 emissions by 8%. The introduction of a post injection helped control soot and CO emissions,
 547 allowed for further reductions in NO_x emissions and increased the EGT by 75 K. It can be also
 548 seen that all advanced combustion control strategies achieved relatively lower levels of HC
 549 emissions than that of the baseline case. As discussed in Section 4.3, the later and longer
 550 combustion process of the “Miller cycle + EGR + post injection” strategy adversely affected
 551 net indicated fuel conversion efficiency, decreasing it by 1.9% when compared with the
 552 baseline engine operation.



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Figure 20. Cost-benefit and overall emissions for various combustion control strategies.

556 Figure 20 also revealed that the advanced combustion control strategies helped decrease the
557 urea consumption via lower engine-out NO_x emissions. This minimised the total fluid
558 consumption and therefore increased the corrected total fluid efficiency, despite a lower net
559 indicated efficiency. The highest improvement in corrected total fluid efficiency of 3.4% was
560 achieved by the “Miller cycle + EGR” strategy as a result of the high NO_x reduction capability
561 combined with a relatively lower net indicated efficiency penalty. Alternatively, the “Miller
562 cycle + EGR + post injection” strategy increased the corrected total fluid efficiency by 3.1%
563 while improving upon EGT as well as soot and CO emissions. Therefore, the combined strategy
564 of “Miller cycle + EGR + post injection” was identified as the most effective means to achieve
565 low engine operational cost and enable exhaust emissions and EGT control.

566 **5. Conclusions**

567 This study investigated the effect of a Miller cycle-only strategy as well as Miller cycle
568 operations combined with EGR and post injection on the combustion process, exhaust
569 emissions, and performance of a HD diesel engine. The aim of the research was to identify an
570 effective combustion control strategy for exhaust emissions and exhaust gas temperatures
571 control, in order to minimise the combined consumption of fuel and urea solution at a low
572 engine load of 6 bar IMEP. Miller cycle, EGR, and multiple injections were achieved by means
573 of a VVA system, a high-pressure loop cooled EGR, and a common rail fuel injection system,
574 respectively. A 1D simulation was used for the calculation of the mean in-cylinder gas
575 temperatures and burned zone gas temperatures, improving the analysis and discussion. The
576 main findings can be summarized as follows:

577

- 578 1. A Miller cycle-only strategy with IVC at -95 CAD ATDC increased the EGT by 75 K and
579 reduced NO_x emissions by 21%, but with a penalty of 1% in the net indicated fuel

580 conversion efficiency when operating the engine with the same intake pressure as the
581 baseline case. This was due to the reduced in-cylinder charge mass and lower peak
582 combustion temperature.

583 2. A higher intake pressure to keep the in-cylinder air/fuel ratio constant allowed for
584 improvements in the net indicated fuel conversion efficiency and soot emissions of a Miller
585 cycle operation at the expense of a reduction in the NO_x and EGT benefits.

586 3. The “Miller cycle + EGR” strategy enabled lower burned zone gas temperatures,
587 significantly reducing NO_x emissions. However, the lower combustion temperatures and
588 lambda (e.g. in-cylinder oxygen availability) adversely affected soot and CO emissions.

589 4. The addition of a post injection was effective in minimising soot and CO emissions of the
590 “Miller cycle + EGR” strategy while achieving lower NO_x emissions and higher EGT. The
591 improvements, however, were achieved at the expense of a decrease in net indicated fuel
592 conversion efficiency.

593 5. Overall, the “Miller cycle + EGR + post injection” strategy was identified as the most
594 effective means for exhaust emissions and EGT control among the various approaches
595 examined in this study. This combination of combustion control strategies reduced engine-
596 out NO_x emissions by 66% and increased the EGT by 75K, which in turn resulted in an
597 increased in the corrected total fluid efficiency by 3.1%. Furthermore, the use of this
598 combined strategy with a 12 mm³ post injection at 18 CAD ATDC helped attain
599 simultaneous low levels of soot, CO and HC emissions.

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611 **Acknowledgments**

612
613 The first author, Mr W Guan, would like to acknowledge the Guangxi Yuchai Machinery Company
614 for supporting his PhD study supervised by Prof. Hua Zhao at Brunel University London.

615 **Declaration of conflicting interests**

616 The author(s) declared no potential conflicts of interest with respect to the research, authorship,
617 and/or publication of this article.

618 **Funding**

619 The author(s) disclosed receipt of the following financial support for the research, authorship,
620 and/or publication of this article: Funding for this project was provided by Guangxi Yuchai Machinery
621 Company.

622 **Definitions/Abbreviations**

ATDC	After Firing Top Dead Center.
ATS	Aftertreatment System.
CA10	Crank Angle of 10% Cumulative Heat Release.
CA50	Crank Angle of 50% Cumulative Heat Release.
CA90	Crank Angle of 90% Cumulative Heat Release.
CAD	Crank Angle Degree.
CO	Carbon Monoxide.
CO₂	Carbon Dioxide.
COV_{IMEP}	Coefficient of Variation of IMEP.
(CO₂%)_{intake}	CO ₂ concentration in the intake manifold.
(CO₂%)_{exhaust}	CO ₂ concentration in the exhaust manifold.
DOC	Diesel Oxidation Catalyst.
DPF	Diesel Particulate Filter.
ECR	Effective Compression Ratio.
ECU	Electronic Control Unit.

EGR	Exhaust Gas Recirculation.
EGT	Exhaust Gas Temperatures.
EEVO	Early Exhaust Valve Opening.
FSN	Filter Smoke Number.
HCCI	Homogenous Charge Compression Ignition.
HRR	Heat Release Rate.
HC	Hydrocarbons.
HD	Heavy-duty.
IMEP	Indicated Mean Effective Pressure.
IVO	Intake Valve Opening.
IVC	Intake Valve Closing
ISFC	Net Indicated Specific Fuel Consumption.
ISSoot	Net Indicated Specific Emissions of Soot.
ISNOx	Net Indicated Specific Emissions of NOx.
ISCO	Net Indicated Specific Emissions of CO.
ISHC	Net Indicated Specific Emissions of Unburned HC.
LTC	Low Temperature Combustion.
LIVC	Late Intake Valve Closing.
\dot{m}_{urea}	Aqueous Urea Solution Consumption
\dot{m}_{total}	Total Fluid Consumption
\dot{m}_{diesel}	Diesel Flow Rate
NOx	Nitrogen Oxides.
1D	One Dimensional
PCCI	Premixed Charge Compression Ignition.
P_{int}	Intake Pressure.
PPCI	Partially Premixed Charge Compression Ignition.
PRR	Pressure Rise Rate.
SCR	Selective Catalytic Reduction.
SOI	Start of Injection.
SOC	Start of Combustion.
TDC	Firing Top Dead Centre.
T_{m}	Mean In-cylinder Gas Temperatures.
T_{b}	Burned Zone Gas Temperatures.

VVA	Variable Valve Actuation.
WHSC	World Harmonized Stationary Cycle

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Appendix A. Test cell measurement devices

Variable	Device	Manufacturer	Measurement range	Linearity/Accuracy
Speed	AG 150 Dynamometer	Froude Hofmann	0-8000 rpm	± 1 rpm
Torque	AG 150 Dynamometer	Froude Hofmann	0-500 Nm	$\pm 0.25\%$ of FS
Diesel flow rate (supply)	Proline promass 83A DN01	Endress+Hauser	0-20 kg/h	$\pm 0.10\%$ of reading
Diesel flow rate (return)	Proline promass 83A DN02	Endress+Hauser	0-100 kg/h	$\pm 0.10\%$ of reading
Intake air mass flow rate	Proline t-mass 65F	Endress+Hauser	0-910 kg/h	$\pm 1.5\%$ of reading
In-cylinder pressure	Piezoelectric pressure sensor Type 6125C	Kistler	0-300 bar	$\leq \pm 0.4\%$ of FS
Intake and exhaust pressures	Piezoresistive pressure sensor Type 4049A	Kistler	0-10 bar	$\leq \pm 0.5\%$ of FS
Oil pressure	Pressure transducer UNIK 5000	GE	0-10 bar	$< \pm 0.2\%$ FS
Temperature	Thermocouple K Type	RS	233-1473K	$\leq \pm 2.5$ K
Intake valve lift	S-DVRT-24 Displacement Sensor	LORD MicroStrain	0-24 mm	$\pm 1.0\%$ of reading using straight line
Smoke number	415SE	AVL	0-10 FSN	-
Fuel injector current signal	Current Probe PR30	LEM	0-20A	± 2 mA

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797 **Paper Correction**

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799 **Dear Organizers and Reviewers,**

800 Thank you for your kind comments and suggestions to the manuscript. We have modified the
801 manuscript accordingly, and detailed corrections are listed below point by point. The paragraphs in
802 black are the reviewers' comments, while our responses are listed in blue. All the modifications in the
803 manuscript are highlighted in red.

804 We look forward to hearing from you.

805 Sincerely,

806 Wei Guan

807 Brunel University London

808

809 **Reviewer #1**

810 1) The abstract should be reformulated. The abstract is an extremely important and powerful
811 representation of the article. The authors should describe better their research, the main contributions
812 and results, and the conclusions. They have to be direct and highlight the novelty of this study. First
813 paragraph statement is not right and not relevant with the performed work; can be removed.

814 **Thanks, the first paragraph in Abstract has been removed.**

815 2) Introduction:

816 i) Please avoid lumping references. Instead of lumping references, please describe in one sentence the
817 most spectacular results of each cited study.

818 **The cited references have been reformulated accordingly in Introduction.**

819 ii) Review should be more critical, rather than a description of previous studies. Following questions
820 must be answered in this literature review:

821 • How does EGR affect emissions, exhaust temp and volumetric efficiency?

822 • How does IVC affect emissions, exhaust temp and volumetric efficiency?

823 • What's the difference between LPEGR and HPEGR? Which strategy is preferred and when?

824 • Why to combine EGR (LP or HP?) and Miller?

825 After reading the introduction, the reader must have a clear understanding of the problem.

826 Thanks for the kind suggestions, the reasons for how do EGR and IVC affect emissions, exhaust temp
827 and volumetric efficiency have been explained accordingly. The type of EGR used is only HP EGR
828 and the difference between LPEGR and HPEGR is not the focus in this work. The reason to combine
829 EGR and Miller cycle is because EGR can effectively curb NO_x formation while Miller cycle is a
830 more effective means for exhaust temperature management but with a limited capability for NO_x
831 control. Therefore, the combined use of Miller cycle with EGR can achieve high NO_x reduction and
832 EGT increase at such low load operation.

833 iii) Last but not least, what is the research gap that this study bridges? Please highlight the novelty.

834 There is no such a work investigating the effects of the combination of Miller cycle with EGR and
835 post injection on low load emissions and EGT control. Therefore, this work includes a good novelty
836 and originality. This has been highlighted at the end of Introduction.

837 iv) A more critical reformulation of the introduction could significantly improve the quality of this
838 study and to get cited more in the future.

839 Thanks very much for your kind suggestion.

840 3) Experimental Set-up: Nice presentation of the experimental set up. Few minor comments:

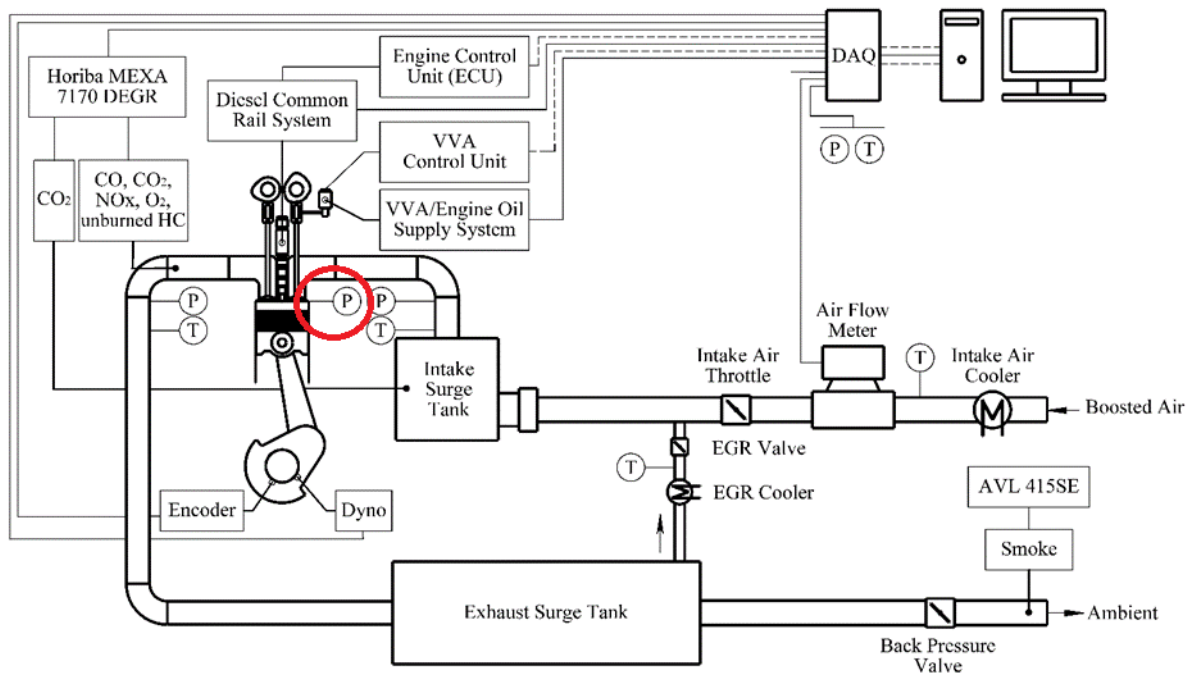
841 i) Please add error analysis of your data. What's the RSO of the utilized sensors?

842 We ensure that the error of all experimental data is controlled below 5% during the study.

843 Please see attached the Appendix A for the resolution of the primary utilized sensors.

844 ii) If in-cylinder pressure is monitored, please add that discussion on the document and update figure
845 1.

846 Yes, the in-cylinder pressure is monitored and has been shown in Figure 1, which can be seen in the
847 Figure below. The discussion can be seen in Section 2.6.



848

849 iii) Pg 10, ln 177: please add the full description of ECR in the document and the ECR in parenthesis.

850 [It has been modified accordingly.](#)

851 iv) Pg 12, ln 222: please provide the value of the specific heat of the polyonymic equation.

852 [Thanks. It has been added accordingly.](#)

853 4) Methodology: Figure 4: please zoom from 20 to 80 bar range. It seems that the model cannot
 854 predict accurately the in-cylinder pressure rise as well as the peak cylinder pressure. Small differences
 855 in the pressure trace result much higher error in the predicted heat release rate. What is the error
 856 between peak cylinder pressure between sim and exp? Why the authors didn't use a two zone heat
 857 release model, presented in many other studies in literature, to calculate burned and unburned
 858 temperatures, so as to avoid errors between simulation and experiments? Also error curve vs Cad at
 859 figure 5 is important to be added.

860 [In all cases, the maximum pressure is validated to within 3% of the experimental data. The model
 861 predicts slightly lower in-cylinder pressures near to the end of compression stroke. This is likely to be
 862 a result of over-predicted heat transfer and the fact the modelled in-cylinder pressure is insensitive to
 863 the simulated diesel injection pressure.](#)

864 [The model used is a two zone heat release model as presented in Section 3.2. In Figure 5, the
 865 modelled and experimental values for average intake and exhaust manifold pressures were computed
 866 to be within 1.5%, which has been added in the text.](#)

867 5) Results:

868 i) Please express exp data with dots and simulation data with curves when comparison between sim
869 and exp is presented

870 There is not comparison between simulation and experiment in the Results section. The 1D model
871 was used only to calculate the mean in-cylinder gas temperatures and burned zone gas temperatures
872 once it was validated with the experimental data.

873 ii) Calculation of temperatures include an error as a result of the deviation between estimated pressure
874 and measured. Please comment that in the text and refer to this error? How much do you expect is
875 that?

876 Thanks for the kind suggestion. It has been added accordingly in the text in Section 3.2.

877 iii) The critical point of Miller is boost pressure. Can the results be realistic when conventional
878 turbocharging is applied instead of the external compressor? How does Miller and EGR affect
879 turbocharger operation? What turbocharger design would be required? Please add comments in the
880 text.

881 Thanks for this good question. These have been added in the text in page 20.

882 The use of Miller cycle results in less total mixture mass trapped while the introduction of a HP EGR
883 decreased the exhaust flow to pass turbocharger, both which could result in lower turbocharger
884 efficiency. As the Miller degree increases the boost pressure has to accordingly increase in order to
885 maintain the fresh mixture mass trapped. Thus, a conventional supercharging system is not able to
886 deliver the required airflow rate, particularly at low load operations. For this reason, a two-stage
887 variable geometry turbocharger configuration may be very promising but it largely increases the
888 engine complexity. On the other hand, a highly efficient intake intercooler is always needed to cool
889 down the intake air after boosted by a turbocharger.

890 iv) At session 4.3.2: It would be better to apply only EGR at the baseline scenario and then Miller +
891 EGR to see the effect of Miller. In the way results are presented, the effect of EGR is well known,
892 NO_x are reduced because burned temperature is increased and soot is increased. This is what happens
893 at any diesel engine. Please reformulate this chapter and use more data if available.

894 Thanks for the kind suggestions, the main objective of session 4.3.2 is to estimate the effect of post
895 injection at various IVC timings. Therefore, the EGR rate was maintained constant at 15% for all
896 cases.

897 v) English and grammar could be reviewed at the results session.

898 Thanks. The manuscript has been revised and proofread.

899 vi) Ln 538, pg 32: Could you please add in the discussion the EU limits for any standard off-highway
900 cycle to understand how much higher than the limits this value of 0.022g/kWh is?

901 This research study was focused on the development of an engine for on-road heavy-duty
902 applications. The Euro VI emission regulation for such vehicles limits soot emissions to less than 0.01
903 g/kWh. Current soot limits for EU off-highway applications is 0.015 g/kWh, as can be seen in Stage
904 V emission standards for non-road engines.

905 6) In the conclusions, in addition to summarising the actions taken and results, please highlight the
906 novelty of this work.

907 The novelty of this work has been highlighted in the conclusions.

908 **Reviewer #2**

909 There are a few suggestions for the authors to consider.

910 1) To include how the '15% EGR' on page 24 was determined.

911 As mentioned in the Methodology, the in-cylinder air flow mass can be decreased noticeable in the
912 combined use of Miller cycle and EGR. This will results in poor combustion instability and excessive
913 smoke. Therefore, a moderate EGR rate of 15% was used for all cases in order to avoid the
914 abovementioned issues as well as to achieve a fair comparison for all strategies investigated with
915 EGR.

916 2) When reading the post injection strategy, the absolute value of the post injection quantity is of
917 course important. However, the readers may like to know how much in percentage of the overall
918 injection quantity it is.

919 Thanks very much for your kind suggestions. The percentage of the post injection quantity to the
920 overall injection quantity has been added in Section 4.3.1.

921 3) The reference No. 53 and 55 were repeated.

922 Thanks. It has been corrected accordingly.