

Exploring the potential of Miller cycle with and without EGR for maximum efficiency and minimum exhaust emissions in a heavy-duty diesel engine

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Abstract

In order to improve the fuel conversion efficiency and meet more stringent exhaust emissions regulations, Miller cycle and exhaust gas recirculation (EGR) have been researched as separate means to reduce carbon dioxide (CO₂) and pollutant emissions from the internal combustion engines. In this paper, an experimental work was carried out to explore the potential benefits of Miller cycle operation via late intake valve closing (LIVC) with and without EGR in a single cylinder heavy-duty (HD) diesel engine equipped with a variable valve actuation (VVA) system. The overall engine-out emissions, fuel conversion efficiency, and estimated urea consumption in the selective catalyst reduction (SCR) aftertreatment were analysed and compared over the World Harmonized Stationary Cycle (WHSC) for different combustion control strategies. Additionally, the potential of Miller cycle with and without EGR based on the “SCR-only” and “SCR + EGR” technical routes to meet the Euro VI nitrogen oxides (NO_x) limit of 0.4 g/kWh was assessed at different NO_x aftertreatment efficiencies.

When considering the urea consumption in the SCR, the results showed that the introduction of EGR allowed for an engine operation with higher corrected net indicated efficiency (NIE_{corr.}) or lower specific total fluid consumption than the baseline cases without EGR due to the relatively lower engine-out NO_x emissions. However, the use of EGR adversely affected soot and CO emissions when operating with constant intake pressure (P_{int}) of the baseline case. The application of Miller cycle with and without EGR strategies decreased the net indicated efficiency (NIE) and NIE_{corr.} when operating with the same P_{int} of the baseline operation. The use of higher P_{int} helped to improve upon the NIE and NIE_{corr.} of the Miller cycle cases. The WHSC cycle-averaged analysis showed that different combustion and engine control technologies can be adopted with and without EGR to meet Euro VI NO_x limit. A conventional baseline engine operation without EGR would require a high SCR efficiency of 96% in order to curb a cycle-averaged NO_x emissions level of 10 g/kWh. Miller cycle operation without EGR achieved the optimum NIE_{corr.} at the cycle-averaged NO_x level of 8 g/kWh. When

32 increasing the P_{int} , this strategy enabled an increase of 2.6% in the NIE_{corr} , and reduced the
33 required SCR efficiency to 93.5%, but with a penalty on the NIE of 3.3% when controlling the
34 cycle-averaged NOx level at 6.5 g/kWh. Alternatively, Miller cycle operation with EGR and
35 higher P_{int} allowed for a cycle-averaged NOx levels of 4.0 g/kWh, decreased the total fluid
36 consumption by 8% and minimised the required SCR efficiency to 90%. Therefore, this study
37 has presented promising cost-effective emission control and fuel efficiency technologies that
38 could be suitable for the “SCR-only” and “SCR + EGR” technical routes for the future HD
39 diesel engines.

40 **Keywords**

41 Heavy-duty diesel engine, Miller cycle, EGR, total fluid consumption, exhaust gas temperature,
42 exhaust emissions.

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55 **1. Introduction**

56 Growing concerns over the greenhouse gas (GHG) such as CO₂ and increasing stringent
57 emission regulations are driving the development of advanced combustion and engine control
58 technologies. Despite occupy only 4% of the total number of on-road vehicles, the heavy-duty
59 (HD) vehicles account for 18% of the fuel consumption and CO₂ emissions within the
60 transportation sector [1].

61 HD vehicles, which typically powered by diesel engines, are facing another important concern
62 over high levels of NO_x and soot emissions. This is due to the fact that conventional diesel
63 combustion produced a wide range of local in-cylinder gas temperatures and equivalence ratios,
64 which are the two main conditions dominating the formation of NO_x and soot emissions [2,3].
65 These emissions are limited to 0.4 g/kWh and 0.01 g/kWh accordingly in the Euro VI
66 legislation [4]. To meet these low emission targets, significant reduction in these pollutants by
67 in-cylinder combustion technologies and emission control aftertreatment systems is required
68 [5].

69 Some technical routes have been adopted to meet the emission legislation for HD applications
70 [6]. The first solution that some companies have adopted is the “EGR-only” concept, which
71 relies heavily on EGR utilization requiring up to 40% EGR at full engine load [7,8]. This
72 requires a high performance turbocharging system in order to maintain the in-cylinder air/fuel
73 ratio [9]. The high amount of exhaust gas recirculated back to the engine also increases the
74 demand on the engine cooling systems. Furthermore, advanced high pressure common rail fuel
75 injection system is required in order to avoid excessive soot emissions and poor fuel efficiency.
76 All these required improvements on the original systems would increase the initial cost and the
77 complexity of the engine.

78 The second widely adopted technical route by manufactures is the “SCR-only” strategy, which
79 can relieve the burden on the engine design while achieving relatively higher fuel conversion
80 efficiency and lower soot emissions. However, the high engine-out NO_x emissions requires a
81 very complicated SCR control strategy combined with a high NO_x conversion efficiency of
82 more than 95% to comply with the Euro VI NO_x limits. The high urea consumption in the SCR
83 system also raises the risk of NH₃ slip, which is limited to less than 10 ppm in the Euro VI
84 regulation [10,11]. Additionally, high SCR conversion efficiency is difficult to maintain at the

85 low engine load operations where the exhaust gas temperature (EGT) is relatively low and
86 probably insufficient to support an efficient SCR system [12].

87 In comparison with the two aforementioned technical routes, another approach is the combined
88 use of EGR with the SCR strategy. This can simplified the SCR control strategy and lower the
89 EGR requirement, varied from 15% at full load to 30% at lower loads [13,14]. With regard to
90 the engine operational cost, the three technical routes are facing the challenge of the trade-off
91 between exhaust emissions and fuel efficiency. For example, an increase in the fuel conversion
92 efficiency by 1% would lead to higher engine-out NO_x emission, increasing from 10 g/kWh to
93 14 g/kWh [6]. This can increase the total engine operational cost (EOC) due to higher urea
94 consumption in the SCR system [6,15,16]. Likewise, the lower engine-out NO_x emissions can
95 be achieved at the expense of a lower fuel efficiency, which can adversely affect the total EOC.
96 Therefore, it is necessary to research alternative in-cylinder combustion and engine control
97 technologies coupled with aftertreatment control strategy in order to achieve fuel efficient and
98 cost-effective solutions [17,18].

99 Recently, Miller cycle has been shown as an effective engine control technology to reduce in-
100 cylinder NO_x formation during the combustion process and is being considered as a
101 mainstream technology to be adopted to HD diesel engines [19]. Firstly, Miller cycle achieved
102 via early or late intake valve closing (IVC) timings can be an effective means for NO_x control
103 and therefore potentially minimize the requirement on the EGR rate used. This is because the
104 lower effective compression ratio (ECR) decreases the in-cylinder gas temperatures at the end
105 of the compression stroke and thus lower peak combustion temperature [20–24]. In addition,
106 Miller cycle can effectively decrease the mechanical and thermal loads of a modern
107 turbocharged HD diesel engine by reducing the peak in-cylinder pressure and temperature via
108 a later initiation of compression process. This allows for the application of advanced
109 combustion technologies such as advanced diesel injection timing, higher fuel injection
110 pressure, and higher boost pressure to improve the engine efficiency [25–28]. Moreover, Miller
111 cycle is one of the effective strategies of improving upon exhaust temperature management to
112 facilitate NO_x removal by SCR, particularly at light engine load operation. This is primarily
113 attributed to the reduced in-cylinder mass trapped and the delayed combustion process [29–32].

114 This study aims to explore Miller cycle based cost-effective emission control and fuel
115 efficiency technologies incorporated in the “SCR-only” and “SCR + EGR” technical routes for
116 future HD diesel engines by combining with and/or without EGR. In particular, the current

117 work is the first attempt to experimentally investigate the influence of Miller cycle operating
118 with and without EGR using constant boost pressure and constant lambda of the baseline cases
119 respectively from low to full engine loads, as well as their potential to meet the Euro VI
120 emission regulations over the WHSC test cycle.

121 The experimental work was carried out on a single-cylinder HD diesel engine. The
122 investigation was conducted at test points within the WHSC test cycle. The results of the
123 different emission control and fuel efficiency technologies were compared with the maximum
124 $NIE_{corr.}$ at a steady-state engine speed of 1250 rpm and engine loads between 25% and 100%
125 of full load. In addition, the cycle-averaged results calculated over the WHSC test cycle were
126 analysed and discussed. Finally, the potential of Miller cycle with EGR and without EGR based
127 on the “SCR + EGR” and “SCR-only” technical routes to meet the Euro VI emission regulation
128 was assessed at different NO_x aftertreatment efficiency requirements.

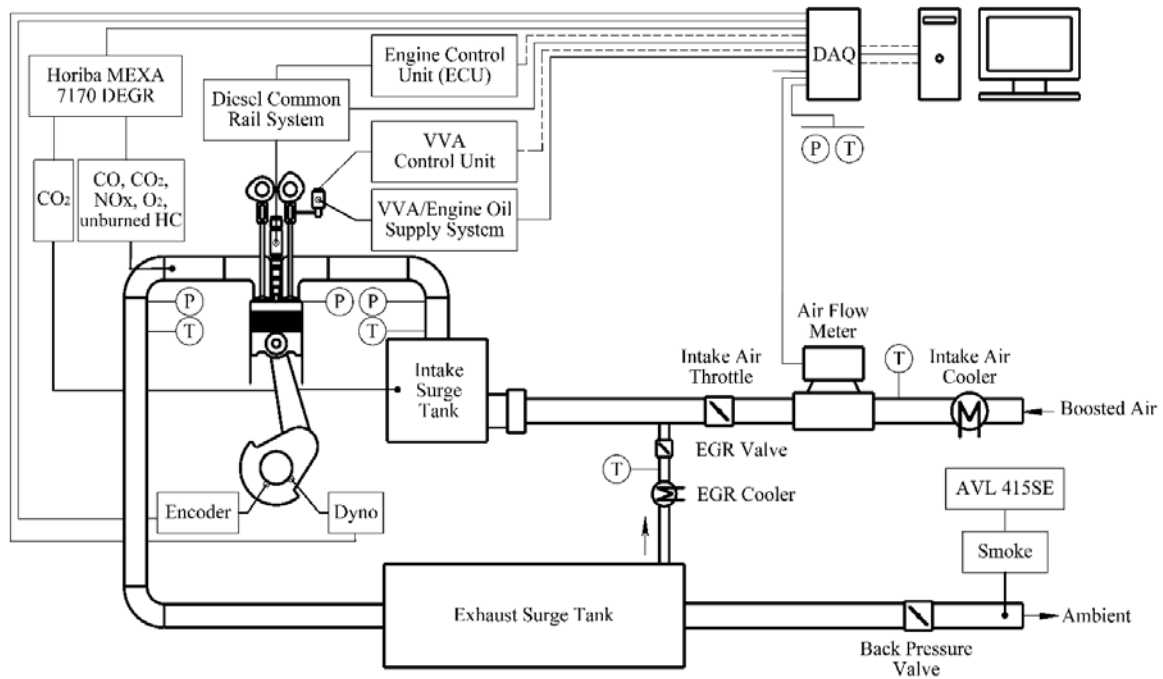
129 **2. Experimental setup**

130 **2.1 Test engine**

131 Experiments were carried out on a single cylinder HD diesel engine equipped with a high
132 pressure common rail fuel injection system. The engine was coupled to a Froude Hofmann
133 AG150 eddy current dynamometer in order to absorb the power output. Figure 1 shows the
134 schematic diagram of the experimental setup and Table 1 outlines the specifications of the test
135 engine. The design of the cylinder head with 4-valve and a stepped-lip piston bowl were based
136 on the Yuchai YC6K 6-cylinder engine, while the bottom end/short block was AVL-designed
137 with two counter-rotating balance shafts.

138 An AVL 515 sliding vanes supercharger was used to provide the boosted intake air with closed
139 loop control. The intake and exhaust surge tanks were installed to damp out the strong pressure
140 fluctuations. Two piezo-resistive pressure transducers were used to measure the instantaneous
141 intake and exhaust manifold pressures. The intake manifold pressure was fine adjusted by an
142 intake throttle valve located upstream of the intake surge tank while the exhaust back pressure
143 was independently controlled through a butterfly valve located downstream of the exhaust
144 surge tank. The intake mass flow rate was measured by an Endress+Hauser Proline t-mass 65F
145 thermal mass flow meter. High-pressure loop cooled external EGR was introduced into the

146 engine upstream of the intake surge tank using a pulse width modulation-controlled EGR valve
 147 and the pressure differential between the intake and exhaust manifolds.



148
 149 Figure 1. Layout of the engine experimental setup.

150 Table 1. Specifications of the test engine.

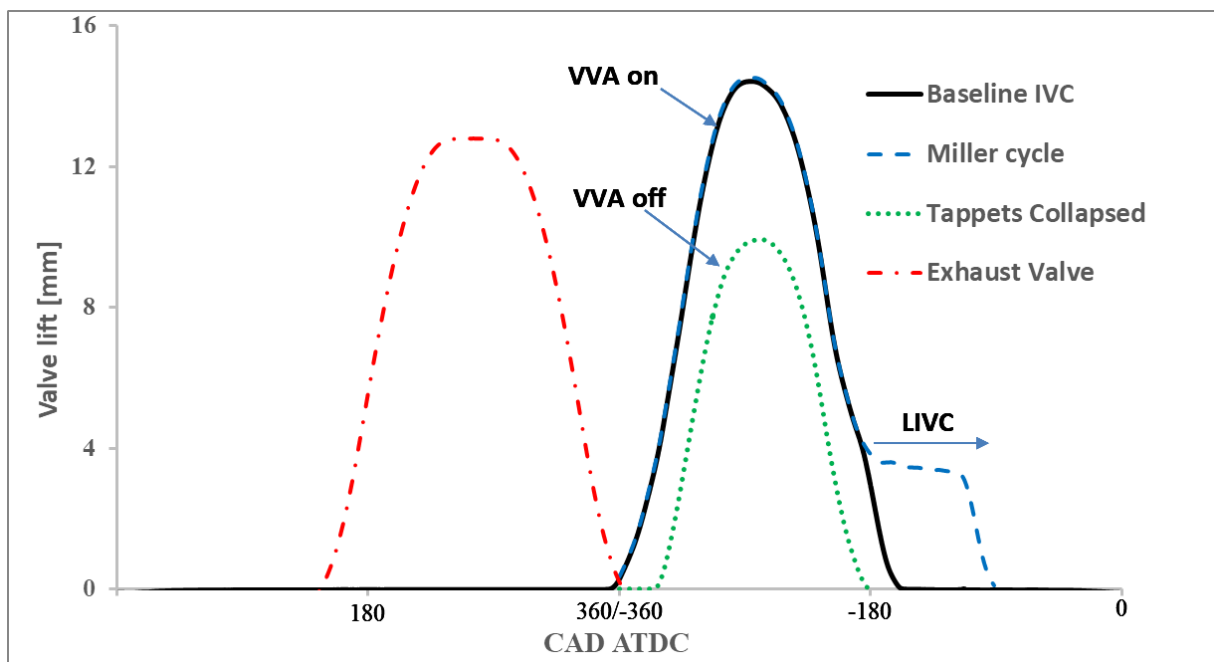
Displaced Volume	2026 cm ³
Stroke	155 mm
Bore	129 mm
Connecting Rod Length	256 mm
Geometric Compression Ratio	16.8
Number of Valves	4
Piston Type	Stepped-lip bowl
Diesel Injection System	Bosch common rail
Nozzle design	8 holes, 0.176 mm hole diameter, included spray angle of 150°
Maximum fuel injection pressure	2200 bar
Maximum in-cylinder pressure	180 bar

151 Water cooled heat exchangers were used to control the temperatures of the boosted intake air
 152 and external EGR as well as engine coolant and lubrication oil, which were supplied externally
 153 by separate electric motors.

154 A dedicated engine control unit (ECU) was used to control the fuel injection parameters such
155 as the injection pressure, start of injection (SOI), and the number of injection pulses (up to three
156 injections per cycle). During the experiments, the diesel fuel was injected into the engine by a
157 high-pressure solenoid injector through a high pressure pump and a common rail with a
158 maximum fuel pressure of 2200 bar. Two Endress+Hauser Promass 83A Coriolis flow meters
159 were used to measure the fuel consumption, which was determined by measuring the total fuel
160 supplied to and from the high pressure pump and diesel injector. The specifications of
161 measurement equipment can be found in Appendix A.

162 2.2 Variable valve actuation system

163 A prototype hydraulic lost-motion VVA system was installed on the intake camshaft. On the
164 valve side of the rocker arm, a collapsing tappet was incorporated to allow for the adjustment
165 of the IVC timing and therefore enable Miller cycle operation [33]. The intake valve opening
166 (IVO) and IVC timings of the baseline case were set at 367 and -178 crank angle degrees (CAD)
167 after top dead centre (ATDC), respectively. The maximum intake valve lift event was 14mm.
168 All valve events in this study were considered at 1 mm valve lift. Figure 2 shows the intake
169 and exhaust valve profiles for the baseline as well as the late IVC (LIVC) strategy.



170 Figure 2. Engine fixed exhaust and variable intake valve lift profiles.
171

172 **2.3 Exhaust emissions measurement**

173 Gaseous emissions such as NO_x, carbon monoxide (CO), CO₂, unburnt hydrocarbon (HC), and
174 oxygen (O₂) were measured by a Horiba MEXA-7170 DEGR emission analyser. A high-
175 pressure sampling module and a heated line were used between the exhaust sampling point and
176 the emission analyser to allow for high-pressure sampling and avoid water condensation. In
177 this study, the EGR rate was defined as the ratio of the measured CO₂ concentration in the
178 intake surge tank ((CO₂%)_{intake}) to the CO₂ concentration in the exhaust manifold ((CO₂%)_{exhaust})
179 as

$$180 \quad EGR \text{ rate} = \frac{(CO_2\%)_{intake}}{(CO_2\%)_{exhaust}} * 100\% \quad (1)$$

181 An AVL 415SE Smoke Meter was used to measure the concentration of the black carbon
182 containing soot downstream of the exhaust back pressure valve. The measurement was taken
183 in filter smoke number (FSN) basis, and thereafter was converted to mg/m³ [34]. All measured
184 exhaust gas components were converted from parts per million (ppm) to net indicated specific
185 gas emissions (in g/kWh) according to the methodology described in the agreement of
186 Regulation number 49 for the European Union [35].

187 **2.4 Data acquisition and analysis**

188 The instantaneous in-cylinder pressure was measured by a Kistler 6125C piezo-electric
189 pressure transducer with a sampling revolution of 0.5 CAD. The measured in-cylinder pressure
190 data was recorded through an AVL FI Piezo charge amplifier. Two National Instruments data
191 acquisition (DAQ) cards were used to acquire the signals from the measurement device. The
192 high speed DAQ card captured high frequency signals such as crank angle resolved data while
193 the low speed DAQ card acquired the low frequency signals of the engine operation. An in-
194 house developed transient combustion analysis software was used to display the captured data
195 from DAQ as well as the processed important engine parameters in real-time.

196 The crank angle based in-cylinder pressure traces were averaged over 200 consecutive engine
197 cycles for each test point and used to calculate the IMEP and apparent Heat Release Rate (HRR).
198 According to [36], the apparent HRR was calculated as

$$199 \quad HRR = \frac{\gamma}{(\gamma-1)} p \frac{dV}{d\theta} + \frac{1}{(\gamma-1)} V \frac{dp}{d\theta} \quad (2)$$

200 where γ is defined as the ratio of specific heats, which was assumed constant at 1.33 throughout
201 the engine cycle [37]; V and p are the in-cylinder volume and pressure, respectively; θ is the
202 crank angle.

203 The pressure rise rate (PRR) was determined by the average value of the maximum pressure
204 variations of one-hundred in-cylinder pressure cycles. The in-cylinder combustion stability was
205 monitored by the coefficient of variation of the IMEP (COV_{IMEP}) over the sampled cycles.
206 The mass fraction burnt (MFB) was defined by the ratio of the integral of the HRR and the
207 maximum cumulative heat release. Combustion phasing (CA50) was determined by the crank
208 angle of 50% MFB. Combustion duration was represented by the period of time between the
209 crank angles of 10% (CA10) and 90% (CA90) MFB. Ignition delay was defined as the period
210 of time between the main SOI and the start of combustion (SOC), denoted as 0.3% MFB point
211 of the average cycle.

212 **2.5 The definition of the corrected net indicated efficiency**

213 As the urea consumption in the SCR system depends on the operating conditions as well as
214 engine-out NO_x emissions, reductions in the levels of engine-out NO_x can help minimise the
215 urea flow rate. Therefore, the corrected net indicated efficiency (NIE_{corr.}) is used in this study
216 in order to take into account both the measured diesel flow rate (\dot{m}_{diesel}) and the aqueous urea
217 solution consumption in an SCR system (\dot{m}_{urea}). This allowed for a cost-benefit analysis of
218 the total cost of the different combustion control strategies. The prices of diesel fuel and urea
219 vary in countries and regions and for simplicity they are assumed to be the same in this study
220 [17,38]. According to [17,39,40], the urea consumption in the SCR system can be estimated as
221 1% of the diesel equivalent fuel flow per g/kWh of NO_x reduction necessary to meet the Euro
222 VI limit (NO_{xEuroVI}) of 0.4 g/kWh.

$$223 \quad \dot{m}_{urea} = 0.01 (\text{NO}_{x\text{Engine-out}} - \text{NO}_{x\text{EuroVI}}) \dot{m}_{diesel} \quad (3)$$

224 The total fluid consumption (\dot{m}_{total}) can then be calculated by summing the measured diesel
225 flow rate and the estimated urea flow rate,

$$226 \quad \dot{m}_{total} = \dot{m}_{urea} + \dot{m}_{diesel} \quad (4)$$

227 from which the NIE_{corr.} is determined by

228
$$\text{NIE}_{corr.} = \frac{P_i}{\dot{m}_{total} \text{LHV}_{diesel}} \quad (5)$$

229 where P_i is the net indicated power and the LHV_{diesel} is the diesel lower heating value of 42.9
230 MJ/kg. It is noted that in the calculation of the $\text{NIE}_{corr.}$ the power used to inject urea is not
231 included and could be included if such data can be available.

232 **3. Methodology**

233 **3.1 Test conditions**

234 In this study, the effects of Miller cycle combined with different combustion control
235 technologies on engine performance, exhaust emissions, and total engine operational cost were
236 evaluated over the WHSC test cycle, which is a legislated test cycle adopted in the Euro VI
237 emission standard.

238 Figure 3 shows the location of the WHSC tests points over a HD diesel engine operation map.
239 There are 13 modes in the WHSC test cycle (red circles), which consist of five speeds (25%,
240 35%, 45%, 55%, and 75%, which are abbreviated as A, B, C, D, and E, respectively) and four
241 engine loads (25%, 50%, 70%, and 100%), as well as two idle modes at the beginning and the
242 end of test cycle. The size of the circle represents the weighting factor. A bigger size indicates
243 a higher relative weighting of the engine operation conditions over the WHSC cycle.

244 Table 2 summarises the engine operating conditions for the different combustion control
245 strategies investigated between 25% and 100% of full engine loads. The intake pressure (P_{int})
246 set point of the baseline operation was taken from a Euro V compliant multi-cylinder HD diesel
247 engine of the same cylinder design as the single cylinder engine. The boosted air was supplied
248 by an external boosting system instead of a turbocharger. The exhaust pressures were adjusted
249 to provide a constant pressure differential of 0.10 bar above the intake pressure, in order to
250 realize the required EGR rate and to achieve a fair comparison with equivalent pumping work.
251 Constant lambda and constant P_{int} analyses were carried out on the Miller cycle operation
252 (using the respective set points from the baseline engine operation) to help evaluate the
253 potential of the technology.

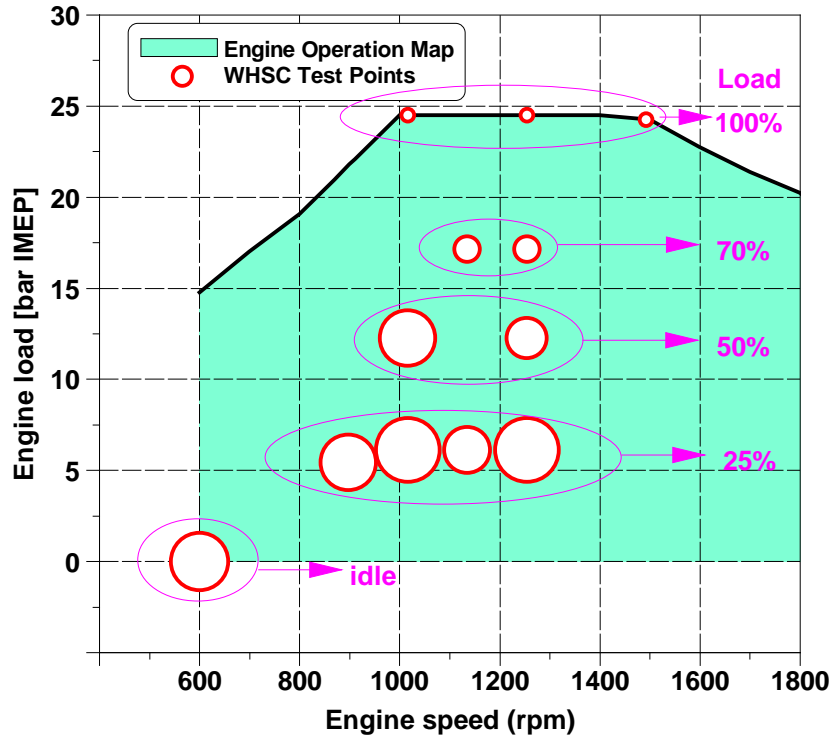


Figure 3. The WHSC operation conditions over an estimated HD diesel engine speed-load map.

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256 With regard to the operations with EGR, the EGR rate was kept constant at a given load,
 257 regardless of the combustion control strategy used. An EGR rate of 15% was used between 25%
 258 and 70% engine loads, and was decreased to 8% at 100% engine load. The moderate EGR rates
 259 were used in this study to avoid the combustion instability, excessive smoke, poor fuel
 260 efficiency, as well as to minimise the demand on the boosting system when operating the
 261 engine with Miller cycle and EGR. These were also the reasons for the relatively earlier IVC
 262 timings used in the cases with EGR when compared to a Miller cycle operation without EGR.
 263 The IVC timings of EGR cases were varied from -100 to -130 CAD ATDC and they were from
 264 -92 to -120 CAD ATDC for cases without EGR as the engine load increased.

265 Table 2. Test conditions for baseline and Miller cycle with and without EGR from low to full engine loads.

Engine speed	Engine load	Rail Pressure	EGR	Exhaust pressure	Baseline operation		Miller cycle operation			
					Intake pressure	IVC	Intake pressure		IVC	
							Constant P_{int}	Constant lambda	Without EGR	With EGR
rpm	% of full load	bar	%	bar	bar	CAD ATDC	bar	bar	CAD ATDC	CAD ATDC

897	25	900	0 and 15	0.10 bar higher than the intake pressure	1.20	-178	1.20	1.63	-92	-100
1016	25	1000	0 and 15		1.25	-178	1.25	1.74	-92	-100
	50	1300	0 and 15		1.48	-178	1.48	2.08	-100	-110
	100	1800	0 and 8		2.80	-178	2.80	2.90	-120	-130
1135	25	1000	0 and 15		1.33	-178	1.33	1.95	-92	-100
	70	1400	0 and 15		2.10	-178	2.10	2.32	-110	-120
1250	25	1150	0 and 15		1.44	-178	1.44	2.10	-92	-100
	50	1400	0 and 15		1.74	-178	1.74	2.35	-100	-110
	70	1500	0 and 15		2.30	-178	2.30	2.70	-110	-120
	100	1800	0 and 8		2.80	-178	2.80	3.00	-120	-130
1492	100	1800	0 and 8		2.90	-178	2.90	3.10	-120	-130

266 During the experiments, a small pilot injection of 3 mm³ with a constant dwell time of 1 ms
267 prior to the main injection timing was employed for both 25% and 50% engine load conditions
268 in order to keep the maximum PRR below 30 bar/CAD. The coolant and oil temperatures were
269 kept within 85 ± 2°C. Oil pressure was maintained within 4.0 ± 0.1 bar. The maximum in-
270 cylinder pressure was limited to 180 bar. Stable engine operation was determined by
271 controlling the COV_IMEP below 3%.

272 **3.2 The calculation of the pressure-based ECR**

273 In this study, Miller cycle was realised via a LIVC strategy, where the period of intake valves
274 opening was longer than that of the baseline operation. This caused backflow from cylinder
275 into the intake manifold due to the upward piston motion before the IVC, which decreased the
276 actual in-cylinder mass trapped.

277 The ECR is a commonly used parameter for indicating the extent of compression and
278 quantifying the effect of LIVC, which was reduced by the delayed initiation of the compression
279 process. One of commonly used ECR definitions is based on the volumetric ratio, which is
280 geometrically calculated as the ratio of the cylinder volume at the specified IVC to the cylinder
281 clearance volume at TDC. The actual compression process, however, does not occur exactly
282 when the intake valve close [41]. Figure 4 shows the illustration of the definition of volume-
283 based and pressure-based ECR at a speed of 1250 rpm and a 25% of full load. The cylinder
284 charge has already been partially compressed before the IVC. This is primarily due to the flow
285 resistance across the intake valves and the gas-momentum-induced compression [42]. As a
286 result, the volume-based geometric ECR is generally lower than the pressure-based ECR, as
287 shown in Figure 5. Therefore, the volume-based geometric ECR is inadequate to explain the
288 experimental results.

289 In this work, a pressure-based method was applied in order to more accurately characterize the
290 actual in-cylinder compression process. The effective IVC volume was taken as the volume
291 corresponding to the intersection point by extrapolating the polytropic compression process
292 curve linearly down to intersect the average intake manifold air pressure (MAP). The pressure-
293 based ECR is then defined as the ratio of the volume at effective IVC to the clearance volume
294 at TDC [43,44].

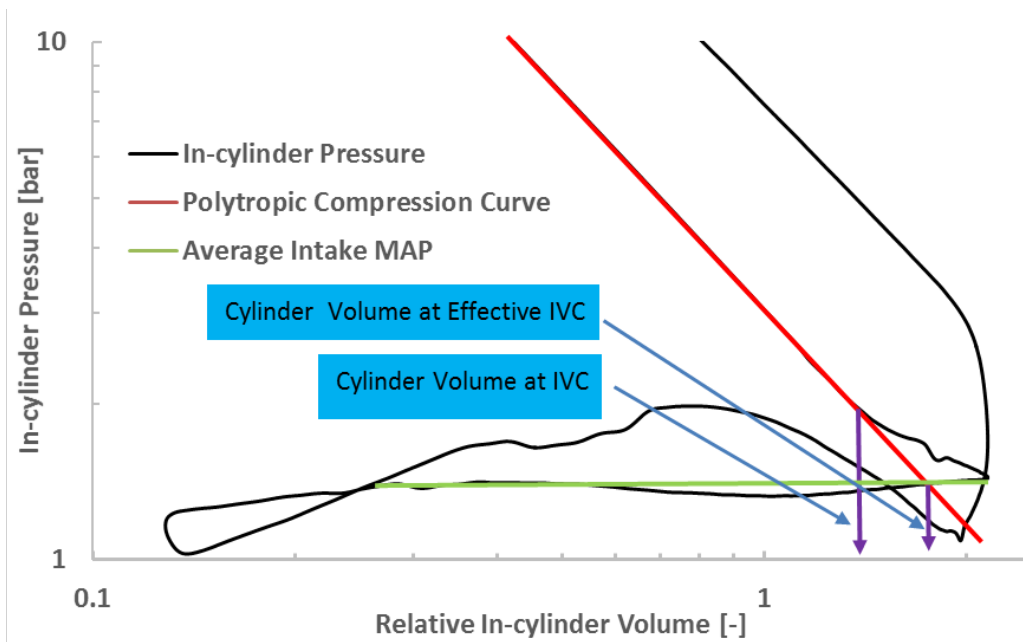


Figure 4. The illustration of volume-based and pressure-based methods for ECR calculation at 1250 rpm and 25% of full load.

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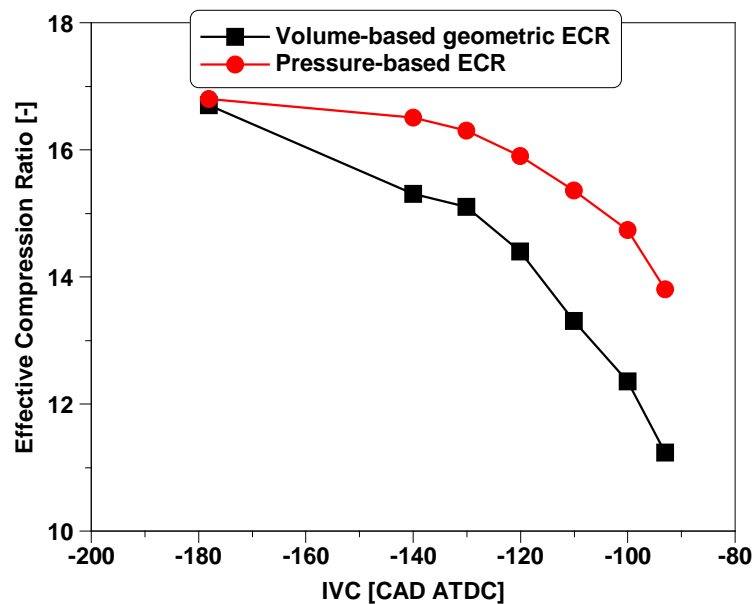


Figure 5. The volume-based and pressure-based ECR as a function of the IVC at 1250 rpm and 25% of full load.

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302 4. Results and discussions

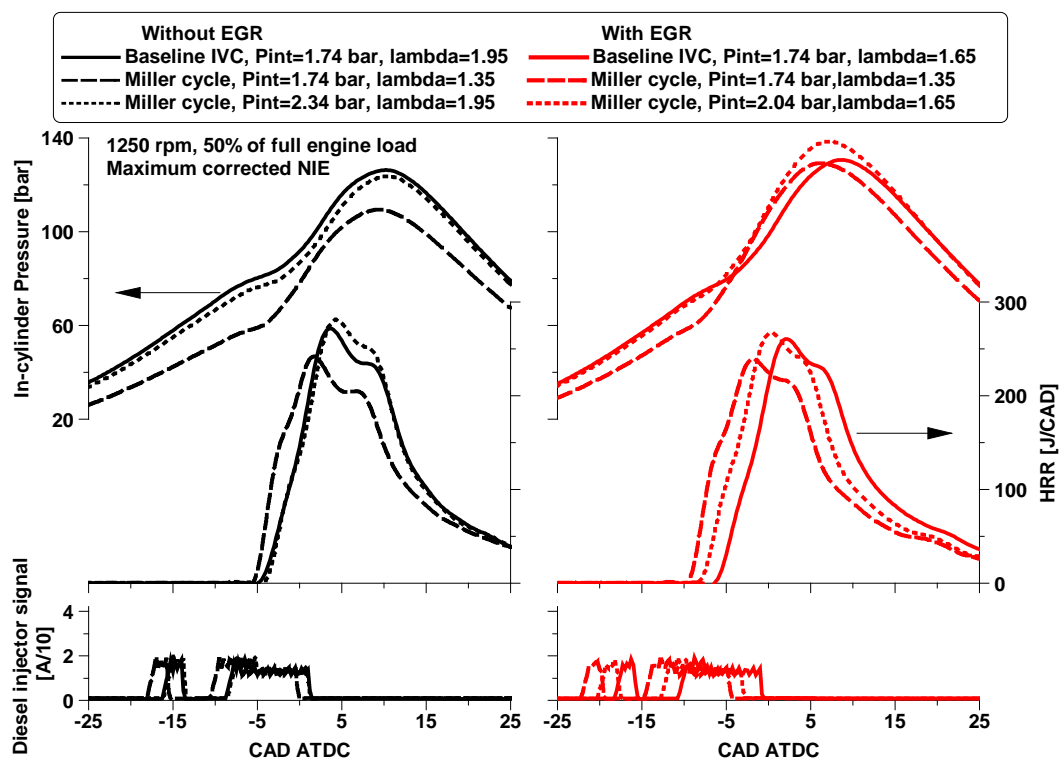
303 4.1 Miller cycle operation with and without EGR

304 In this subsection, Miller cycle operation with and without EGR using constant P_{int} and a
 305 constant lambda as those of the baseline cases were investigated in order to analyse the effect
 306 of Miller cycle on engine performance, exhaust emissions, and total engine operational cost
 307 over the engine loads. The results were analysed and comparisons were made with the baseline

308 operations at 1250 rpm over a range of loads from 25% to 100% of full engine load. Diesel
 309 injections were optimised for the maximum $NIE_{corr.}$.

310 4.1.1 Analysis of the in-cylinder pressure and heat release rate

311 Figure 6 shows the in-cylinder pressure, HRR, and diesel injector signal for cases attained the
 312 maximum $NIE_{corr.}$ at 50% of full engine load. The Miller cycle operation without EGR and
 313 with a constant P_{int} of 1.74 bar as the baseline operation was characterised by lower in-cylinder
 314 pressure and peak HRR. This was due to the later initiation of the compression process by the
 315 LIVC and hence lower in-cylinder compressed pressure and temperature. The use of EGR
 316 allowed for a more advanced diesel SOI to achieve the maximum $NIE_{corr.}$ when compared to
 317 the cases without EGR. This enabled an earlier SOC, consequently increasing the peak in-
 318 cylinder pressures (P_{max}) of the optimum cases. When operating with constant lambda by
 319 increasing the intake pressure, Miller cycle with and without EGR achieved relatively higher
 320 peak HRR, although a later main SOI than that with constant P_{int} .

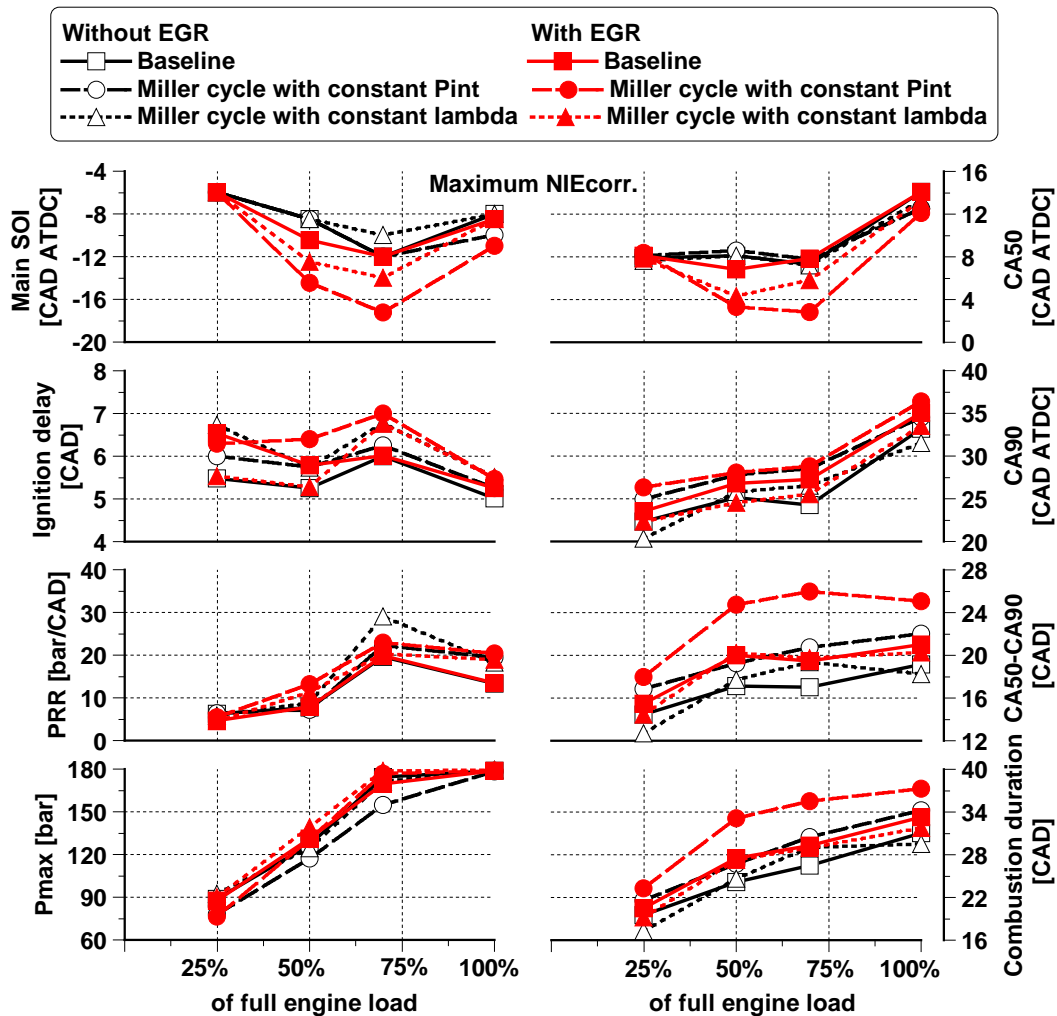


321
 322 Figure 6. In-cylinder pressure, HRR, and diesel injector signal for the baseline and Miller cycle cases with the
 323 maximum $NIE_{corr.}$

324 4.1.2 Combustion characteristics

325 Figure 7 shows the main SOI and the resulting combustion characteristics for the baseline and
 326 Miller cycle cases that attained the maximum $NIE_{corr.}$. The Miller cycle with constant P_{int} of

327 the baseline cases allowed for more advanced main SOI, particularly in the cases with EGR.
 328 At 100% of full engine load, the optimisation of SOI was constrained by the P_{max} of 180 bar.
 329 The Miller cycle and EGR strategies increased the ignition delay in most cases. This was
 330 attributed to the lower compression pressures and in-cylinder oxygen availability, as well as
 331 the advanced SOI. The PRR was kept below 30 bar/CAD for all cases.



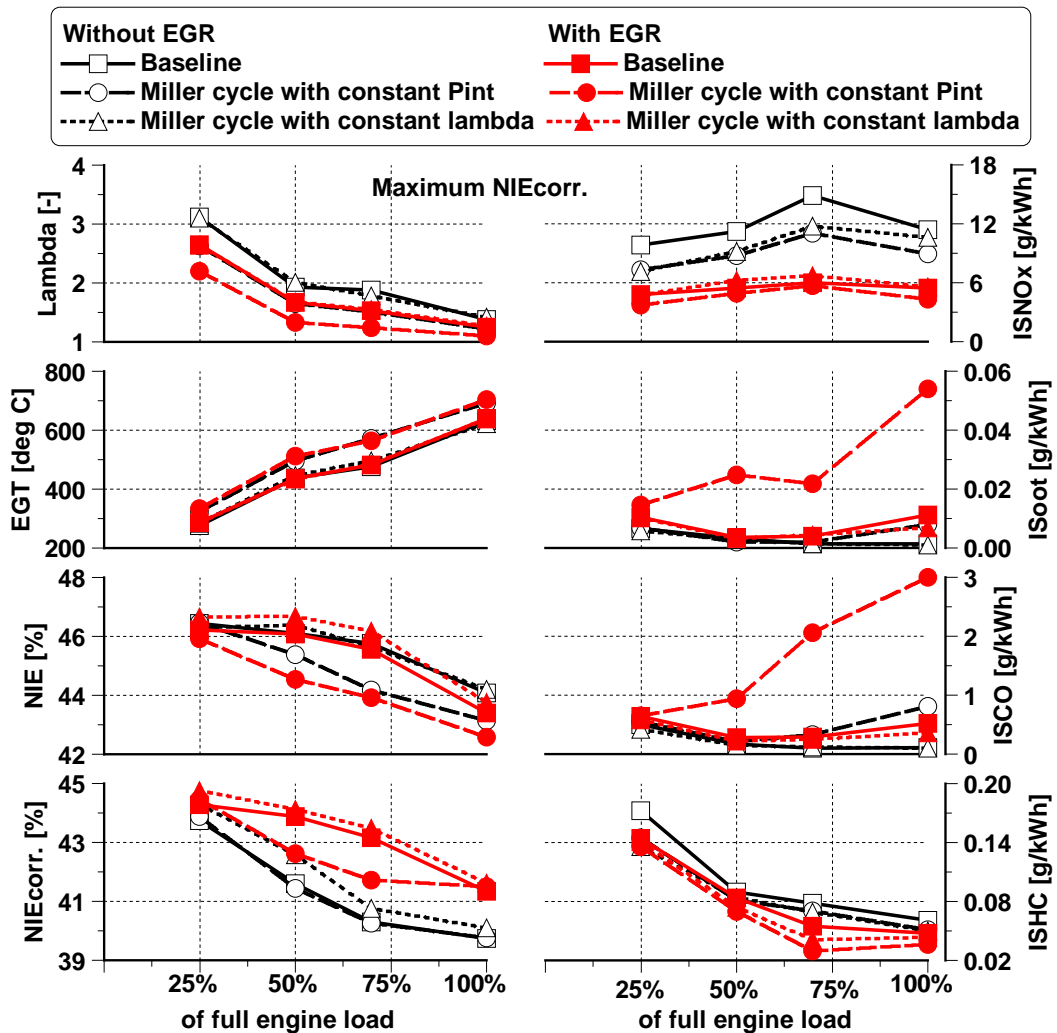
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 333 Figure 7. Injection timing and combustion characteristics of the baseline and Miller cycle cases with maximum
 334 NIE_{corr} .

335 The optimum CA50 was initially advanced from low to medium loads likely due to the longer
 336 period CA10-CA90 and lower heat transfer losses, and then was delayed at high engine loads
 337 attributed to the limit of peak in-cylinder pressure [16]. This trend became more obvious for
 338 the cases with EGR. The reduced in-cylinder oxygen availability via the use of Miller cycle
 339 and EGR slowed down the rate of heat release, leading to a later CA90 and longer period of
 340 CA10-CA90 (e.g. combustion duration). A higher P_{int} improved the in-cylinder oxygen
 341 availability and accelerated the combustion process of Miller cycle with and without EGR.

342 This advanced the CA90 and yielded shorter CA50-CA90 period and combustion duration
 343 when operating with a constant lambda.

344 **4.1.3 Engine performance, exhaust emissions, and fuel efficiency**

345 Figure 8 shows the engine performance parameters and net indicated specific emissions for the
 346 baseline and Miller cycle cases with and without EGR. Miller cycle with a constant P_{int}
 347 decreased the lambda due to a reduction in the in-cylinder mass trapped, particularly with EGR.
 348 This was the primary reason for an increase in EGT. However, the “EGR-only” strategy
 349 showed little impact on the EGT due to the replacement of air with the recirculated exhaust gas
 350 and lower combustion temperatures [24].



351
 352 Figure 8. Engine performance, emissions, and fuel efficiency of the baseline and Miller cycle cases with the
 353 maximum $NIE_{corr.}$.

354 Compared to the baseline engine operation, Miller cycle with and without EGR decreased the
 355 NIE at a constant P_{int} . This was a result of the slower combustion process and longer
 356 combustion duration. The reduction in NIE is also associated with an increase in heat loss due

357 to the higher mean in-cylinder gas temperature calculated by a 1-D engine model, as reported
358 in [24,45]. By using a higher P_{int} to improve the in-cylinder oxygen availability, however, the
359 NIE of Miller cycle was improved and was slightly higher than that of the baseline operation
360 when operating with a constant lambda.

361 When considering the urea consumption in the SCR system, the total engine operational cost
362 is related to the $NIE_{corr.}$, which will be determined by the engine-out NOx emissions and the
363 fuel conversion efficiency. As shown in the bottom left of Figure 8, the engine operations with
364 EGR achieved higher maximum $NIE_{corr.}$ than those cases without EGR over the sweep of
365 engine loads as a result of the lower NOx emissions. When operating without EGR, the
366 maximum $NIE_{corr.}$ of Miller cycle with constant P_{int} was comparable to that of the baseline.
367 When operating with EGR, Miller cycle strategy significantly reduced the maximum $NIE_{corr.}$.
368 However, by increasing P_{int} and keeping the lambda constant helped to improve the trade-off
369 between NOx emissions and NIE, increasing the $NIE_{corr.}$ of the Miller cycle strategy.

370 It can be also seen from Figure 8 the engine-out emissions of NOx, soot, unburned HC, and
371 CO. The NOx emissions were increased between 25% and 70% of full engine load but then
372 decreased at 100% of full engine load. This was likely due to the increased combustion
373 temperature from 25% to 70% of full engine load. The NOx reduction at full engine load was
374 associated with the significantly lower lambda value and the relatively retarded CA50. The use
375 of EGR was more effective in minimizing engine-out NOx emissions than Miller cycle in all
376 cases. The “Miller cycle + EGR” strategy with constant P_{int} achieved low NOx levels but
377 yielded excessive smoke, particularly at high engine loads. Soot emissions were reduced when
378 operating with the same lambda of the baseline cases and were maintained below the Euro VI
379 limit of 0.01 g/kWh at 50% and 70% of full engine load. Therefore, the use of high boost
380 pressures is a key enabler for achieving simultaneous low NOx and soot emissions when
381 operating Miller cycle and EGR, especially at high engine loads.

382 The observed trend of CO emissions was similar to that of soot emissions. This was mainly
383 because the in-cylinder oxygen concentration played an important role on the CO and soot
384 formations. Miller cycle and EGR strategies showed little impact on CO emissions except when
385 they were combined. The use of a higher P_{int} between medium and full engine loads helped to
386 curb CO emissions from the Miller cycle operating with EGR. Finally, Figure 8 revealed that
387 all advanced combustion control strategies decreased HC emissions compared to the baseline
388 cases. Overall, the lowest levels of HC emissions were achieved by the “Miller cycle + EGR”

389 strategy with constant P_{int} . This could be linked to the use of more advanced diesel injections
390 and higher average in-cylinder gas temperature at lower lambda [28].

391 **4.2 Analysis of Miller cycle operation with and without EGR over the WHSC** 392 **test cycle**

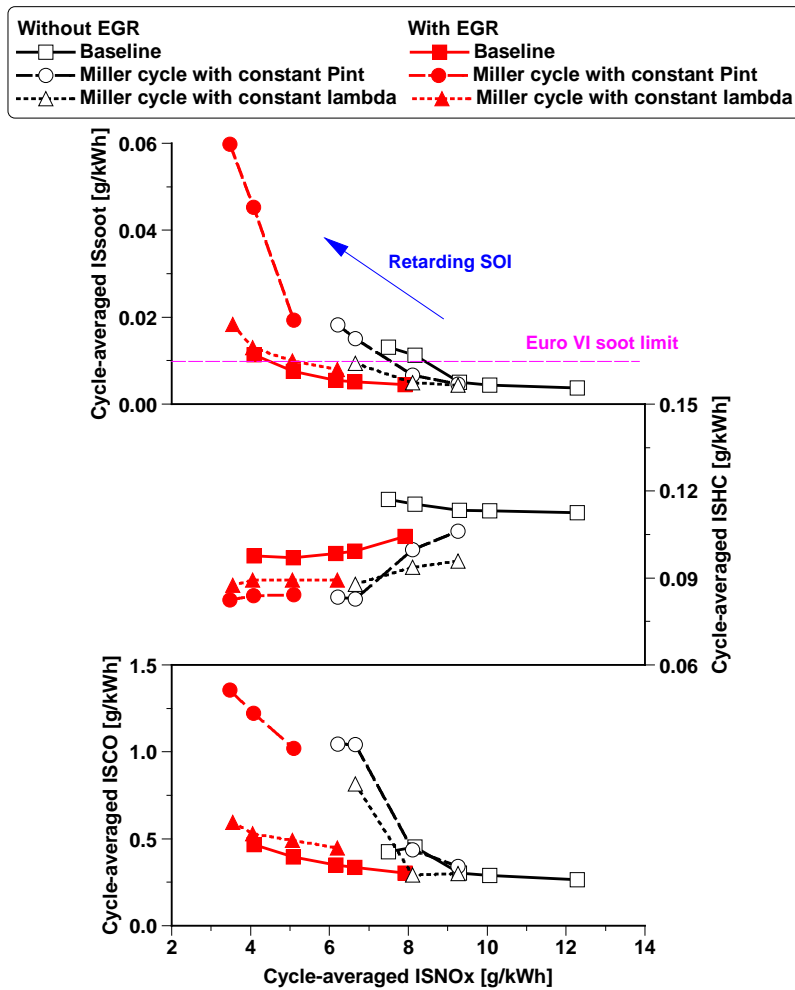
393 The purpose of this subsection is to analyse and compare the cycle-averaged performance,
394 emissions, and total engine operational cost of different combustion control strategies over the
395 WHSC test cycle. The diesel SOI was delayed at each combustion control strategy in order to
396 obtain the different cycle-averaged NOx emissions.

397 **4.2.1 Cycle-averaged engine exhaust emissions as a function of engine-out NOx emissions**

398 Figure 9 shows the exhaust emissions as a function of different cycle-averaged NOx emissions
399 over the WHSC test cycle for the baseline and Miller cycle operations with and without EGR.
400 The diesel SOI was varied to achieve different engine-out NOx emissions in order to obtain
401 the required cycle-average NOx levels for all strategies investigated.

402 The cycle-averaged soot emissions of all combustion control strategies increased with the
403 reduction in cycle-averaged NOx emissions. The baseline operation without EGR showed little
404 impact on soot emissions when the NOx emissions were controlled at approximately 10 g/kWh
405 or higher. Further reduction in the NOx emissions by delaying the SOI resulted in higher soot
406 emissions. Miller cycle operation without EGR produced less soot emissions than the baseline
407 cases at the NOx level of 8 g/kWh. This was a result of a more advanced diesel injection, which
408 helped improve the air-fuel mixing. However, the soot emissions were increased linearly as the
409 cycle-averaged NOx emissions decreased to below 6.5 g/kWh.

410 EGR is required to achieve a cycle-averaged NOx emission lower than 6 g/kWh. The results
411 indicated that the use of EGR was effective in reducing NOx emissions while producing
412 slightly higher soot emissions. However, the combination of Miller cycle and EGR at a constant
413 P_{int} produced excessive soot emissions due to the lower lambda at such conditions. This was
414 overcome by increasing intake pressure and was maintained below Euro VI soot limits, as
415 denoted with a red dash line in Figure 9. As a result, simultaneous low levels of soot and NOx
416 emissions were obtained when the engine was operated with the same lambda as the baseline
417 case with EGR.



418
419

Figure 9. Cycle-averaged exhaust emissions for baseline and Miller cycle cases over the WHSC test cycle.

420 The data in Figure 9 also shows the cycle-averaged CO emissions, which increased with the
 421 reduction in cycle-averaged NOx emissions. Similar to the impact on soot emissions, the SOI
 422 in the baseline operations had little impact on the CO emissions. However, the use of late SOI
 423 in the Miller cycle operation with a constant P_{int} increased the CO emissions, particularly when
 424 using EGR. This drawback was overcome when operating with a higher P_{int} . Different from
 425 the soot and CO emissions, HC emissions decreased with a reduction in NOx emissions for all
 426 advanced combustion control strategies. These improvements were relatively higher in the
 427 Miller cycle cases. Overall, the cycle-averaged CO and HC emissions of all cases were
 428 maintained within the Euro VI limits. It should be also noted that CO and HC emissions are
 429 not the primary issues in conventional diesel engines. This is due to the fact that HD diesel
 430 engines are equipped with a diesel oxidation catalyst (DOC) which can effectively reduce CO
 431 and unburnt HC emissions from the engine exhaust gases when the exhaust temperatures were
 432 held between 200 and 450°C [46].

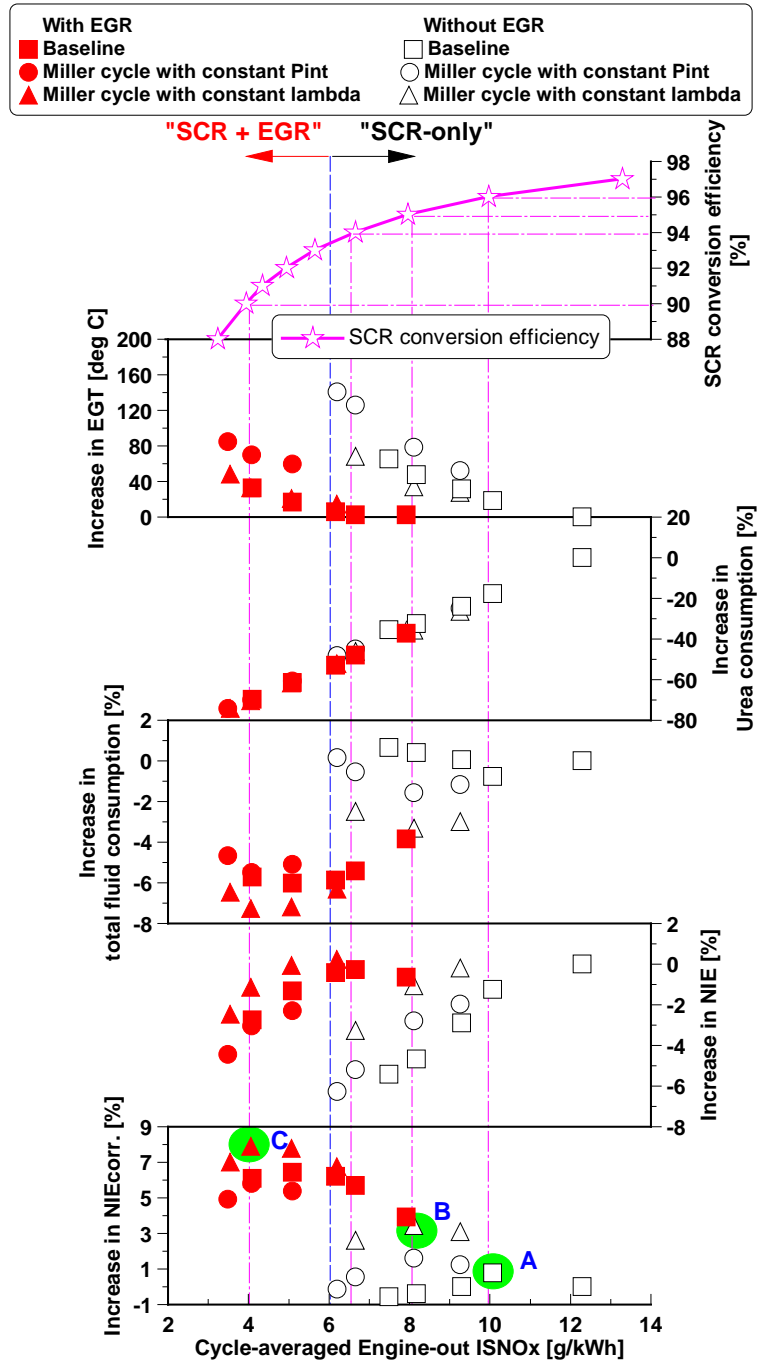
433 4.2.2 Analysis of cycle-averaged performance and potential of different technical routes

434 This section is focused on the analysis and estimation of the different technical routes in terms
435 of the total engine operational cost and the economic effect by taking into account both fuel
436 consumption and the usage of urea. Figure 10 depicts the cycle-averaged performance as a
437 function of various cycle-averaged NO_x emissions for all combustion control strategies. The
438 EGT increased with the reduction in the cycle-averaged NO_x emissions. This was due to the
439 use of Miller cycle, EGR, and late SOI for NO_x control, which decreased the in-cylinder air
440 mass trapped and/or delayed the combustion process. The highest increase in EGT of
441 approximately 140°C was achieved by the “Miller cycle-only” strategy. The addition of EGR
442 in the Miller cycle operation had little impact on the EGT, despite the lower lambda.

443 In addition, all combustion control strategies yielded lower NIE when reducing the cycle-
444 averaged NO_x emissions. The baseline operations achieved lower NO_x emissions via a
445 retarded SOI, reducing the NO_x emissions by 39% with a penalty of 6% in NIE. Compared to
446 the baseline without EGR, the “Miller cycle-only” strategy achieved a reduction in NO_x
447 emissions of 49% with the same NIE penalty of 6%. The NIE penalty could be reduced to 3%
448 while preserving the NO_x reduction benefit by increasing the P_{int} and maintaining the same
449 lambda of the baseline engine operation. The “EGR-only” strategy reduced the NO_x emissions
450 by 67% with a NIE penalty of 2.7%. Alternatively, the “Miller cycle + EGR” strategy with the
451 same lambda as the baseline cases with EGR reduced NO_x emissions by up to 71%. This was
452 achieved with the same NIE penalty of 2.7% measured in the “EGR-only” strategy.

453 Furthermore, the SCR conversion efficiency was used to calculate the urea consumption and
454 the $NIE_{corr.}$ of the engine operation, in order to fully evaluate the potential of these different
455 combustion control strategies to meet different engine-out NO_x levels over the WHSC test
456 cycle. It can be seen from Figure 10 the cycle-averaged urea consumption in the SCR system
457 as a function of cycle-averaged NO_x emissions. The advanced combustion control strategies
458 helped decrease the urea consumption via lower engine-out NO_x emissions. This could
459 minimise the cycle-averaged total fluid consumption and therefore increased the $NIE_{corr.}$. The
460 “Miller cycle + EGR” strategy with a constant lambda attained the highest $NIE_{corr.}$ over the
461 WHSC, increasing the cycle-averaged engine efficiency by up to 8% in comparison with the
462 baseline without EGR.

463



465 Figure 10. Potential of the proposed “SCR-only” and “SCR + EGR” technical routes to meet the Euro VI NO_x
 466 limit (0.4 g/kWh) over the WHSC test cycle. A) Optimum baseline engine operation without EGR; B) Optimum
 467 “Miller cycle-only” strategy with constant lambda; C) Optimum “Miller cycle + EGR” strategy with constant
 468 lambda.
 469

470 Finally, Figure 10 demonstrates the potential of different combustion control strategies to meet
 471 the Euro VI emission regulations. It should be noted that the proposed “SCR + EGR” and
 472 “SCR-only” technical routes represent the combustion control strategy with and without using
 473 EGR, respectively. The results demonstrated that the baseline case without EGR was adequate

474 for the cycle-averaged NO_x emissions of 10 g/kWh and would require an overall SCR
475 conversion efficiency of 96% at the expense of 1.5% reduction in NIE. In addition, the optimum
476 NIE_{corr.} of the baseline engine operation was achieved at this cycle-averaged NO_x level of 10
477 g/kWh, as denoted by point “A” in the bottom of Figure 10. When the cycle-averaged NO_x
478 emissions were required to be lower than 10 g/kWh, the Miller cycle without EGR strategy
479 was preferable when operating with a constant lambda of the baseline case without EGR. This
480 strategy enabled a relatively lower penalty of 3.3% on the NIE and an increase of 2.6% in the
481 NIE_{corr.} at the NO_x level of 6.5 g/kWh. Besides, the EGT was increased by 68°C and the
482 minimum required SCR conversion efficiency was reduced to 93.5%. It should be also noted
483 that the optimum NIE_{corr.} of “Miller cycle-only” strategy was improved up to 3.5% when
484 controlling the NO_x level at 8 g/kWh, as denoted by point “B” in Figure 10. Therefore, the
485 “Miller cycle-only” strategy with a constant lambda of the baseline case without EGR was
486 determined as the most effective means for the “SCR-only” technical route, achieving lower
487 engine operational cost, higher EGT, and lower soot emissions when the engine-out NO_x
488 emissions were kept between 6 and 10 g/kWh.

489 When the cycle-averaged NO_x emissions were controlled to below 6 g/kWh, however, the
490 “Miller cycle-only” strategy resulted in a penalty of 6.3% on NIE and adversely affected the
491 NIE_{corr.}. Therefore, the introduction of EGR was necessary owing to its high NO_x reduction
492 capability and relatively lower penalty on the fuel conversion efficiency. Particularly, the
493 “Miller cycle +EGR” strategy with constant P_{int} could decrease the cycle-averaged NO_x
494 emissions from 12.3 g/kWh in the baseline operation without EGR to 3.4 g/kWh while
495 increasing upon the NIE_{corr.} by 4.9%, despite a reduction in NIE of approximately 4.6%.
496 Meanwhile, this lower cycle-averaged NO_x emissions allowed for a reduction in the minimum
497 required SCR conversion efficiency down to 88%, which can significantly minimize the
498 catalyst volumes and thus can simplify the SCR system [47]. However, this strategy increased
499 the cycle-averaged soot emissions to up to 0.06 g/kWh as a result of the lower lambda, which
500 was significantly higher than the Euro VI limit of 0.01 g/kWh (seen in Figure 9). These results
501 can limit the potential of the “Miller cycle + EGR” strategy for efficient and clean engine
502 operation.

503 Preferably, a more efficient Miller cycle operation with EGR was achieved by keeping the
504 lambda as much as the baseline operation with EGR via a higher P_{int}. The cycle-averaged soot
505 emissions were significant reduced and were controlled within the Euro VI limit at most cases,

506 except for the cases with cycle-averaged NO_x emissions below 4 g/kWh, as shown in Figure
507 9. Meanwhile, the penalty on the NIE was notably minimized and the NIE_{corr.} was significantly
508 increased while maintaining low levels of NO_x emissions. Overall, the “Miller cycle + EGR”
509 strategy with a constant lambda achieved the highest improvement in the NIE_{corr.} of 8% when
510 controlling the cycle-averaged NO_x emissions at 4 g/kWh, as denoted by point “C” in Figure
511 10. This was accompanied with an increase in EGT by 40°C and a lower minimum required
512 SCR conversion efficiency of 90% when compared to the baseline engine operation without
513 EGR. Therefore, the “Miller cycle + EGR” strategy with a higher P_{int} was identified as the most
514 effective means for the “SCR + EGR” technical route, achieving low engine operational cost
515 while simultaneously enabling low cycle-averaged exhaust emissions.

516 **5. Conclusions**

517 This study investigated the effect of Miller cycle with and without EGR on the combustion
518 characteristics, exhaust emissions, and performance of a HD diesel engine over the WHSC
519 tests cycle. Miller cycle, EGR, and multiple injections were achieved by means of a VVA
520 system, a high-pressure loop cooled EGR, and a common rail fuel injection system,
521 respectively. A comparison was performed between the baseline and Miller cycle operations
522 with and without EGR attained the maximum NIE_{corr.} from low to full engine loads.
523 Experimental analysis of cycle-averaged results of the different combustion control strategies
524 were carried out in order to demonstrate their potential to meet the Euro VI NO_x limit. The aim
525 of the research was to explore a cost-effective emission control and fuel efficiency technology
526 based on the “SCR-only” and “SCR + EGR” technical routes for the future HD diesel engines.
527 The primary findings can be summarised as follows:

- 528 1. At the optimum efficiency and without taking into account the urea consumption in the
529 SCR system, Miller cycle with and without EGR decreased the NIE at the same intake
530 pressure as the baseline operation. This was due to the decreased cylinder mass trapped,
531 resulting in slower combustion process and a longer combustion duration. By increasing
532 the boost pressure to keep the same lambda as the baseline, the NIE of Miller cycle
533 operation was improved and comparable to that of the baseline operation at all engine loads.
- 534 2. When considering the total fluid consumption (e.g. fuel and urea), the baseline engine
535 operation with EGR produced higher NIE_{corr.} attributed to the reduction of urea
536 consumption in SCR systems. This reduced the total fluid consumption but at the expense
537 of higher soot and CO emissions. Particularly, the “Miller cycle + EGR” strategy with

538 constant lambda achieved the highest NIE and NIE_{corr.} while achieving simultaneous low
539 levels of exhaust emissions from low to full engine loads.

540 3. The cycle-averaged results over the WHSC test cycle showed that all advanced combustion
541 control strategies enabled a reduction in the cycle-averaged NO_x emissions and higher
542 EGT. These improvements were attained at the expense of lower NIE and higher soot and
543 CO emissions. However, a higher NIE_{corr.} could be achieved via lower urea consumption
544 in the SCR system, especially when operating with a higher intake pressure to improve the
545 combustion process.

546 4. The “Miller cycle-only” strategy with constant intake pressure achieved a reduction in the
547 cycle-averaged NO_x emissions by 49% at the expense of a penalty on NIE of 6%. When
548 increasing the intake pressure to maintain the same lambda of the baseline operation
549 without EGR, the “Miller cycle-only” strategy enabled a relatively lower penalty of 3.3%
550 on the NIE and an increase in the NIE_{corr.} of 2.6% at the cycle-averaged NO_x level of 6.5
551 g/kWh. In the meantime, the cycle-averaged EGT was increased by 68°C, which was
552 associated with the decrease in the cylinder mass and thus a lower total heat capacity.

553 5. The “Miller cycle + EGR” strategy with constant intake pressure could decreased the
554 cycled-average NO_x emissions from 12.3 g/kWh in the baseline operation without EGR
555 to 3.4 g/kWh, but adversely affected soot emissions and NIE. The use of a higher intake
556 pressure to maintain the same lambda of the baseline operation with EGR enabled a “Miller
557 cycle + EGR” strategy achieving the highest improvement of approximately 8% in the
558 NIE_{corr.} while elevating the EGT by 40°C and controlling the NO_x levels at 4 g/kWh.
559 Meanwhile, the highly boosted strategy helped increase the NIE and curb soot emissions
560 due to improved oxygen availability and thus the combustion process.

561 6. Overall, the “Miller cycle-only” and “Miller cycle+ EGR” strategies with the same lambda
562 as the baseline operations were identified as the most effective emission control and fuel
563 efficiency technologies for the “SCR-only” and “SCR + EGR” technical routes
564 accordingly. Moreover, the minimum required SCR conversion efficiency was reduced
565 from 96% in the baseline engine operation to 93.5% in the “Miller cycle-only” strategy
566 and 90% in the “Miller cycle+ EGR” strategy when they were combined with a highly
567 boosted strategy.

568 **6. Future work**

569 From the results presented in this study, it can be seen that the application of Miller cycle
570 required a higher boost pressure to recover the cylinder charge air in order to avoid lower
571 engine efficiency. This increases the requirements on the boosting system, a conventional
572 turbocharging system is likely not able to deliver the required airflow rate. Thus, a more
573 sophisticated boosting system such as a two-stage VGT configuration will be needed to deliver
574 the desired boost pressure, but it largely increases the engine complexity and requires
575 additional cost and may increase the total engine operational cost. Additionally, a highly
576 efficient intake intercooler will be essential to cool down the boosted intake air.

577 Moreover, the impact of applying Miller cycle on turbocharger and pumping work as well as
578 aftertreatment systems needs to be investigated on a multi-cylinder engine, so that further
579 improvement and better understanding of such technology in real world applications can be
580 assessed.

581 **Definitions/Abbreviations**

CA90	Crank Angle of 90% Cumulative Heat Release.
CA50	Crank Angle of 50% Cumulative Heat Release.
CA10	Crank Angle of 10% Cumulative Heat Release.
ATDC	After Firing Top Dead Center.
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.
ECR	Effective Compression Ratio.
ECU	Electronic Control Unit.
EGR	Exhaust Gas Recirculation.
EGT	Exhaust Gas Temperature.
EOC	Engine Operational Cost
FSN	Filter Smoke Number.

GHG	Greenhouse Gas
HRR	Heat Release Rate.
HC	Hydrocarbons.
HD	Heavy Duty.
IMEP	Indicated Mean Effective Pressure.
IVO	Intake Valve Opening.
IVC	Intake Valve Closing
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.
LIVC	Late Intake Valve Closing.
MFB	Mass Fraction Burnt
MAP	Manifold Air Pressure
NOx	Nitrogen Oxides.
NIE	Net Indicated Efficiency
O₂	Oxygen
P_{int}	Intake Pressure.
P_{max}	Peak In-cylinder Pressure
PRR	Pressure Rise Rate
SCR	Selective Catalytic Reduction.
SOI	Start of Injection.
SOC	Start of Combustion.
TDC	Firing Top Dead Centre.
VVA	Variable Valve Actuation.
WHSC	World Harmonized Stationary Cycle.

582 **References**

- 583 1. Gravel, R., “Freight Mobility and SuperTruck,” 2016.
- 584 2. Dec, J.E., “A Conceptual Model of DI Diesel Combustion Based on Laser-Sheet Imaging*,”
585 1997, doi:10.4271/970873.
- 586 3. Rissman, B.J. and March, H.K., “Advanced Diesel Internal Combustion Engines,” (March):1–
587 14, 2013.

- 588 4. Nations, U., "UNECE Regulation 49," (March 1958), 2013.
- 589 5. Johnson, T. V., "Review of Vehicular Emissions Trends," *SAE Int. J. Engines* 8(3), 2015,
590 doi:10.4271/2015-01-0993.
- 591 6. Görsmann, C., "Improving air quality while reducing the emission of greenhouse gases,"
592 *Johnson Matthey Technol. Rev.* 59(2):139–151, 2015, doi:10.1595/205651315X687524.
- 593 7. Cloudt, R., Baert, R., Willems, F., and Vergouwe, M., "SCR-only concept for heavy-duty Euro-
594 VI applications," *MTZ, Mot. Zeitschrift* 70(9):58–63, 2009, doi:10.1007/BF03226980.
- 595 8. Yao, M., Zhang, Q., Liu, H., Zheng, Z., Zhang, P., Lin, Z., Lin, T., and Shen, J., "Diesel Engine
596 Combustion Control: Medium or Heavy EGR?," *SAE Tech. Pap.* 1125–2010, 2010,
597 doi:10.4271/2010-01-1125.
- 598 9. Liu, J., Wang, H., Zheng, Z., Zou, Z., and Yao, M., "Effects of Different Turbocharging Systems
599 on Performance in a HD Diesel Engine with Different Emission Control Technical Routes," *SAE
600 Tech. Pap.*, 2016, doi:10.4271/2016-01-2185.
- 601 10. Vressner, A., Gabrielsson, P., Gekas, I., and Senar, E., "Meeting the EURO VI NO_x Emission
602 Legislation using a EURO IV Base Engine and a SCR / ASC / DOC / DPF Configuration in the
603 World Harmonized Transient Cycle," *Sae Pap. 2010-01-1216*, 2010, doi:10.4271/2010-01-1216.
- 604 11. Cloudt, R., Willems, F., and Heijden, P. van der, "Cost and Fuel Efficient SCR-only Solution
605 for Post-2010 HD Emission Standards," *SAE Int. J. Fuels Lubr.* 2(1):399–406, 2009,
606 doi:10.4271/2009-01-0915.
- 607 12. Cavina, N., Mancini, G., Corti, E., Moro, D., Cesare, M. De, and Stola, F., "Thermal
608 Management Strategies for SCR After Treatment Systems," *SAE Int.*, 2013, doi:10.4271/2013-
609 24-0153.
- 610 13. Baert, R.S.G., Beckman, D.E., and Veen, a, "SAE TECHNICAL Efficient EGR Technology
611 for Future HD Diesel Engine Emission Targets," *SAE Tech. Pap.* 1(0837):1–13, 1999,
612 doi:10.4271/1999-01-0837.
- 613 14. Morgan, R., Banks, A., Auld, A., and Heikal, M., "The Benefits of High Injection Pressure on
614 Future Heavy Duty Engine Performance," (x), 2015, doi:10.4271/2015-24-2441.Copyright.
- 615 15. Buckendale, L.R., Stanton, D.W., and Stanton, D.W., "Systematic Development of Highly
616 Efficient and Clean Engines to Meet Future Commercial Vehicle Greenhouse Gas Regulations,"
617 *SAE Int.* 2013-01–2421, 2013, doi:10.4271/2013-01-2421.
- 618 16. Pedrozo, V.B., May, I., Guan, W., and Zhao, H., "High efficiency ethanol-diesel dual-fuel
619 combustion: A comparison against conventional diesel combustion from low to full engine load,"
620 *Fuel* 230(February):440–451, 2018, doi:10.1016/j.fuel.2018.05.034.
- 621 17. Charlton, S., Dollmeyer, T., and Grana, T., "Meeting the US Heavy-Duty EPA 2010 Standards
622 and Providing Increased Value for the Customer," *SAE Int. J. Commer. Veh.* 3(1):101–110, 2010,
623 doi:10.4271/2010-01-1934.
- 624 18. Dallmann, T. and Menon, A., "Technology Pathways for Diesel Engines Used in Non-Road
625 Vehicles and Equipment," *ICCT White Pap.* (September), 2016.
- 626 19. Bruce Morey, "IAV brings variable valvetrains to heavy duty," *SAE Int. United States*, 2017.
- 627 20. Imperato, M., Antila, E., Sarjoavaara, T., Kaario, O., Larmi, M., Kallio, I., and Isaksson, S., "NO_x
628 Reduction in a Medium-Speed Single-Cylinder Diesel Engine using Miller Cycle with Very
629 Advanced Valve Timing," *SAE Tech. Pap.* 4970, 2009, doi:10.4271/2009-24-0112.
- 630 21. Rinaldini, C.A., Mattarelli, E., and Golovitchev, V.I., "Potential of the Miller cycle on a HSDI
631 diesel automotive engine," *Appl. Energy* 112(x):102–119, 2013,

- 632 doi:10.1016/j.apenergy.2013.05.056.
- 633 22. Gonca, G., Sahin, B., Parlak, A., Ayhan, V., Cesur, I., and Koksall, S., "Application of the Miller
634 cycle and turbo charging into a diesel engine to improve performance and decrease NO
635 emissions," *Energy* 93:795–800, 2015, doi:10.1016/j.energy.2015.08.032.
- 636 23. Benajes, J., Serrano, J.R., Molina, S., and Novella, R., "Potential of Atkinson cycle combined
637 with EGR for pollutant control in a HD diesel engine," *Energy Convers. Manag.* 50(1):174–183,
638 2009, doi:10.1016/j.enconman.2008.08.034.
- 639 24. Guan, W., Pedrozo, V., Zhao, H., Ban, Z., and Lin, T., "Investigation of EGR and Miller Cycle
640 for NOx Emissions and Exhaust Temperature Control of a Heavy-Duty Diesel Engine," *SAE*
641 *Tech. Pap.*, 2017, doi:10.4271/2017-01-2227.
- 642 25. Kovács, D. and Eilts, P., "Potentials of the Miller Cycle on HD Diesel Engines Regarding
643 Performance Increase and Reduction of Emissions," *SAE Tech. Pap. 2015-24-2440 (X)*, 2015,
644 doi:10.4271/2015-24-2440.Copyright.
- 645 26. Zhang, Y., Wang, Z., Bai, H., Guo, C., and Li, Y., "The Reduction of Mechanical and Thermal
646 Loads in a High-Speed HD Diesel Engine Using Miller Cycle with Late Intake Valve Closing,"
647 *SAE Tech. Pap.*, 2017, doi:10.4271/2017-01-0637.
- 648 27. Benajes, J., Molina, S., Martín, J., and Novella, R., "Effect of advancing the closing angle of the
649 intake valves on diffusion-controlled combustion in a HD diesel engine," *Appl. Therm. Eng.*
650 29(10):1947–1954, 2009, doi:10.1016/j.applthermaleng.2008.09.014.
- 651 28. Guan, W., Pedrozo, V., Zhao, H., Ban, Z., and Lin, T., "Exploring the NOx Reduction Potential
652 of Miller Cycle and EGR on a HD Diesel Engine Operating at Full Load," *SAE Tech. Pap.* 2018-
653 April:1–12, 2018, doi:10.4271/2018-01-0243.
- 654 29. Gehrke, S., Kovács, D., Eilts, P., Rempel, A., and Eckert, P., "Investigation of VVA-Based
655 Exhaust Management Strategies by Means of a HD Single Cylinder Research Engine and Rapid
656 Prototyping Systems," *SAE Tech. Pap.* 01(0587):47–61, 2013, doi:10.4271/2013-01-0587.
- 657 30. Ratzberger, R., Kraxner, T., Pramhas, J., Hadl, K., Eichlseder, H., and Buegler, L., "Evaluation
658 of Valve Train Variability in Diesel Engines," *SAE Int. J. Engines* 8(5):2015-24–2532, 2015,
659 doi:10.4271/2015-24-2532.
- 660 31. Garg, A., Magee, M., Ding, C., Roberts, L., Shaver, G., Koeberlein, E., Shute, R., Koeberlein,
661 D., McCarthy, J., and Nielsen, D., "Fuel-efficient exhaust thermal management using cylinder
662 throttling via intake valve closing timing modulation," *Proc. Inst. Mech. Eng. Part D J. Automob.*
663 *Eng.* 230(4):470–478, 2016, doi:10.1177/0954407015586896.
- 664 32. Guan, W., Zhao, H., Ban, Z., and Lin, T., "Exploring alternative combustion control strategies
665 for low-load exhaust gas temperature management of a heavy-duty diesel engine," *Int. J. Engine*
666 *Res.* 146808741875558, 2018, doi:10.1177/1468087418755586.
- 667 33. Schwoerer, J., Kumar, K., Ruggiero, B., and Swanbon, B., "Lost-Motion VVA Systems for
668 Enabling Next Generation Diesel Engine Efficiency and After-Treatment Optimization," *SAE*
669 *Tech. Pap.* 01(1189), 2010, doi:10.4271/2010-01-1189.
- 670 34. AVL, "AVL 415SE Smoke Meter," *Prod. Guid. Graz, Austria*, 2013.
- 671 35. Regulation No 49 – uniform provisions concerning the measures to be taken against the emission
672 of gaseous and particulate pollutants from compression-ignition engines and positive ignition
673 engines for use in vehicles. Off J Eur Union, 2013.
- 674 36. Heywood J.B, "Internal Combustion Engine Fundamentals," ISBN 007028637X, 1988.
- 675 37. Zhao, H., "HCCI and CAI engines for the automotive industry," ISBN 9781855737426, 2007.

- 676 38. Hanson, R., Ickes, A., and Wallner, T., "Comparison of RCCI Operation with and without EGR
677 over the Full Operating Map of a Heavy-Duty Diesel Engine," *SAE Tech. Pap.* (x), 2016,
678 doi:10.4271/2016-01-0794.
- 679 39. Johnson, T. V., "Diesel Emissions in Review," *SAE Int. J. Engines* 4(1):143–157, 2011,
680 doi:10.4271/2011-01-0304.
- 681 40. Pedrozo, V.B., May, I., and Zhao, H., "Exploring the mid-load potential of ethanol-diesel dual-
682 fuel combustion with and without EGR," *Appl. Energy* 193:263–275, 2017,
683 doi:10.1016/j.apenergy.2017.02.043.
- 684 41. Ickes, A., Hanson, R., and Wallner, T., "Impact of Effective Compression Ratio on Gasoline-
685 Diesel Dual-Fuel Combustion in a Heavy-Duty Engine Using Variable Valve Actuation," 2015.
- 686 42. Modiyani, R., Kocher, L., Alstine, D.G. Van, Koeberlein, E., Stricker, K., Meckl, P., and Shaver,
687 G., "Effect of intake valve closure modulation on effective compression ratio and gas exchange
688 in turbocharged multi-cylinder engines utilizing EGR," *Int. J. Engine Res.* 12(6):617–631, 2011,
689 doi:10.1177/1468087411415180.
- 690 43. Pedrozo, V.B. and Zhao, H., "Improvement in high load ethanol-diesel dual-fuel combustion by
691 Miller cycle and charge air cooling," *Appl. Energy* 210(March 2017):138–151, 2018,
692 doi:10.1016/j.apenergy.2017.10.092.
- 693 44. Stricker, K., Kocher, L., Koeberlein, E., Alstine, D. Van, and Shaver, G.M., "Estimation of
694 effective compression ratio for engines utilizing flexible intake valve actuation," *Proc. Inst.*
695 *Mech. Eng. Part D J. Automob. Eng.* 226(8):1001–1015, 2012, doi:10.1177/0954407012438024.
- 696 45. Kawano, D., Suzuki, H., Ishii, H., Goto, Y., Odaka, M., Murata, Y., Kusaka, J., and Daisho, Y.,
697 "Ignition and Combustion Control of Diesel HCCI," *Sae Tech. Pap. Ser.*, 2005,
698 doi:10.4271/2005-01-2132.
- 699 46. Gehrke, S., Kovács, D., Eilts, P., Rempel, A., and Eckert, P., "Investigation of VVA-Based
700 Exhaust Management Strategies by Means of a HD Single Cylinder Research Engine and Rapid
701 Prototyping Systems," *SAE Tech. Pap.* 01(0587):47–61, 2013, doi:10.4271/2013-01-0587.
- 702 47. Chi, J.N., "Control Challenges for Optimal NOx Conversion Efficiency from SCR
703 Aftertreatment Systems," *SAE Tech. Pap.*, 2009, doi:10.4271/2009-01-0905.

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712 **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	± 0.25% of FS
Diesel flow rate (supply)	Proline promass 83A DN01	Endress+Hauser	0-20 kg/h	± 0.10% of reading
Diesel flow rate (return)	Proline promass 83A DN02	Endress+Hauser	0-100 kg/h	± 0.10% of reading
Intake air mass flow rate	Proline t-mass 65F	Endress+Hauser	0-910 kg/h	± 1.5% of reading
In-cylinder pressure	Piezoelectric pressure sensor Type 6125C	Kistler	0-300 bar	≤ ± 0.4% of FS
Intake and exhaust pressures	Piezoresistive pressure sensor Type 4049A	Kistler	0-10 bar	≤ ± 0.5% of FS
Oil pressure	Pressure transducer UNIK 5000	GE	0-10 bar	< ± 0.2% FS
Temperature	Thermocouple K Type	RS	233-1473K	≤ ± 2.5 K
Intake valve lift	S-DVRT-24 Displacement Sensor	LORD MicroStrain	0-24 mm	± 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