

# Performance, emission and combustion characteristics of an indirect injection (IDI) multi-cylinder compression ignition (CI) engine operating on neat jatropha and karanj oils preheated by jacket water

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## ABSTRACT

Renewable non-edible plant oils such as jatropha and karanj have potential to substitute fossil diesel fuels in CI engines. A multi-cylinder water cooled IDI type CI engine has been tested with jatropha and karanj oils and comparisons made against fossil diesel. The physical and chemical properties of the three fuels were measured to investigate the suitability of jatropha and karanj oils as fuels for CI engines. The engine cooling water circuit and fuel supply systems were modified such that hot jacket water preheated the neat plant oil prior to injection. Between jatropha and karanj there was little difference in the performance, emission and combustion results. Compared to fossil diesel, the brake specific fuel consumption on volume basis was around 3% higher for the plant oils and the brake thermal efficiency was almost similar. Jatropha and karanj operation resulted in higher CO<sub>2</sub> and NO<sub>x</sub> emissions by 7% and 8% respectively, as compared to diesel. The cylinder gas pressure diagram showed stable engine operation with both plant oils. At full load, the plant oils gave around 3% higher peak cylinder pressure than fossil diesel. With the plant oils, cumulative heat release was smaller at low load and almost similar at full load, compared to diesel. At full load, the plant oils exhibited 5% shorter combustion duration. The study concludes that the IDI type CI engine can be efficiently operated with neat jatropha (or karanj) oil preheated by jacket water, after small modifications of the engine cooling and fuel supply circuits.

**Keywords:** Non-edible plant oil, CI engine, Performance, Combustion, Emissions, Biofuel

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## 1. Introduction

The total estimated world greenhouse gas (GHG) emissions in the year 2005 amounted to 44 billion tonnes of CO<sub>2</sub> equivalent, of which 66.5% were associated with energy services. Of the total emissions, the share from transportation, electricity and heat was 39.2% – corresponding to 59% of the emissions related to energy services [1]. Internal combustion engines, gas turbines and steam turbines are widely used in the provision of these services. Use of internal combustion engines, and especially CI engines, is rapidly growing for small-to-medium scale decentralised power, irrigation, combined heat and power, and tri-generation applications due to a number of advantages e.g. robustness, high efficiency, fuel flexibility. Therefore a substantial reduction in GHG emissions is possible by replacing fossil diesel in CI engines with biomass derived renewable fuels. Different types of liquid biofuels have been discussed in the literature [2-5]. One of these is non-edible or edible plant oil, either neat or blended with fossil diesel [6-12]. Plant oils can also be transesterified to produce biodiesel [13-15]. However, neat (unprocessed) plant oils offer considerable life cycle energy and GHG emission savings compared to both biodiesel and fossil diesel [16,17]. Use of neat oils also offers economic advantages due to the fewer requirements for machinery, chemicals and processes.

Plant oil, when blended with fossil diesel or transesterified into biodiesel, does not generally require any engine modification [18,19]. Modification is recommended, however, if the plant oil is used

57 directly in neat form, mainly due to higher viscosity of the plant oils compared to fossil diesel which  
58 leads to incomplete combustion and coke formation inside the cylinder and in fuel supply systems  
59 including the injectors [16,20,21]. For example, in one study, Bari [21] reported that, after 500 hours  
60 of running with crude palm oil, maximum power decreased by 20% and minimum brake specific fuel  
61 consumption increased by 26%. Heavy carbon deposits in the combustion chamber and wear in the  
62 injector pump plunger and delivery valve were also observed [21]. Plant oils are usually preheated to  
63 reduce the viscosity before injection for better engine performance and reduced emissions. Typically  
64 engine exhaust gas [22-24] or electricity [7,8,10,25,26] has been used for preheating. Use of engine  
65 exhaust gas is better as it utilises waste heat and therefore involves no additional running costs;  
66 however, it requires close monitoring and control of the exhaust gas flow and temperature of the  
67 preheated oil. Overheating due to either malfunction or sudden overloading of the engine could cause  
68 the oil to flash. Preheating by exhaust gas therefore requires automation in order to meet health and  
69 safety requirements, which can be expensive. An alternative is to use the engine jacket water for  
70 preheating. In one recent study, for example, Basinger et al. [20] designed and installed a V-shaped  
71 plug type oil preheater by modifying the Change Over Valve (COV) of the Lister engine, which  
72 collected heat from the cooling water jacket. The heat transfer modelling and identification of the  
73 channel geometry of the preheater enabled the authors to maintain the oil temperature before the  
74 injector at about 90°C across all engine loads [20].

75  
76 To avoid direct competition with food crops, non-edible plant oils are an attractive substitute for fossil  
77 diesel fuel. Of the non-edible plant oils, jatropha (*Jatropha curcas*) and karanj (*Pongamia pinnata*)  
78 have received considerable attention [27-31] mainly due to their availability and ease of cultivation.  
79 The jatropha tree is notable for its tolerance of poor soils and dry climates. Karanj is a forest tree that  
80 is especially interesting as a nitrogen fixing species. Their oils have usually been evaluated separately  
81 in different engines and under different test conditions [22-24, 28, 32-35]. Moreover, most studies  
82 were conducted in only single cylinder engines of mainly direct injection (DI) type with compression  
83 ratios in the range of 16.5–18.5 [22-24, 28, 32-35]. Only a few studies investigated engine combustion  
84 characteristics [23, 32]. Chauhan et al. [22] tested preheated and unheated jatropha oil in a 5.9 kW  
85 single cylinder DI engine; jatropha oil was preheated to 100°C and they reported 3% decrease in  
86 thermal efficiency when compared to fossil diesel operation. These authors reported that NO<sub>x</sub>  
87 emission was lower than fossil diesel in the case of unheated jatropha operation, but increased by 45%  
88 when preheated jatropha was used [22]. Preheated neat karanj oil gave around 40% increase in Brake  
89 Specific Fuel Consumption (BSFC) and 25% decrease in brake thermal efficiencies as compared to  
90 fossil diesel operation [23]. Khatri et al. [24] investigated the effect of injection timing on engine  
91 performance when operated on preheated karanj oil-diesel blends using a single cylinder 3.7 kW  
92 Kirloskar DI engine with compression ratio of 16.5 and injection timing of 23° BTDC. The thermal  
93 efficiency of the 40:60 karanj-diesel blend was found to be similar with fossil diesel operation at  
94 optimum injection timing of 19° BTDC. In another similar study, 23% higher BSFC, 14% lower  
95 brake thermal efficiency and 15% higher CO<sub>2</sub> emission were reported with preheated neat jatropha oil  
96 used instead of fossil diesel in a single cylinder DI engine with compression ratio of 17.5 [35].

97  
98 As interest in jatropha and karanj is growing, it is important to investigate these oils in CI engines of  
99 types not prominent in the current literature. Furthermore, for valid comparison, the oils need to be  
100 tested in the same engine and under the same conditions. Given the high viscosity of plant oils, use of  
101 an IDI type engine may be advantageous over DI, because mixing of the plant oil with air should be  
102 better due to the turbulence created by the partial burning in the secondary combustion chamber (pre-  
103 chamber) of the IDI engine. Consequently, it is expected that, the combustion of high viscous plant  
104 oils will be better in IDI type engine, and the coke formation in the combustion chamber will be  
105 minimum. Furthermore, NO<sub>x</sub> emission may be lower in the IDI than in the DI engine, due to the lack  
106 of air in the secondary combustion chamber and lower temperature in the main combustion chamber  
107 [36]. With the aim of verifying these anticipated benefits, an experimental study of the performance,  
108 exhaust emissions and combustion parameters of a multi-cylinder IDI type CI engine running on neat  
109 plant oils is presented in this study. Jatropha and karanj will be compared against each other and  
110 against fossil diesel operation for reference. The engine cooling circuit will be modified so that the hot

111 jacket water (which is normally wasted) preheats the plant oil. The physical and chemical  
112 characteristics of the fuels will first be measured for use in the performance calculations and  
113 evaluation of the results. The study includes combustion analysis by means of cylinder pressure  
114 measurements, and a preliminary investigation of the coke formation inside the cylinder, piston and  
115 injectors.

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## 118 **2. Materials and methods**

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### 121 **2.1. Fuel Characterisation**

122

123 Jatropha oil produced in Ghana was obtained from a UK supplier (Matrix Biofuels Ltd.), whereas  
124 karanj oil produced in India was obtained from an Indian supplier (A-Mark Procon Ltd.). The oils  
125 were cleaned in the laboratory using 10, 5 and 1  $\mu\text{m}$  sock filters. A Parr 6100 bomb calorimeter was  
126 used to measure the higher heating values (HHV). Canon Fenski u-tube viscometers and thermostatic  
127 water bath ( $\pm 0.1^\circ\text{C}$ ) were used to measure the kinematic viscosities at different temperatures  
128 according to ISO 3104 having an accuracy of  $\pm 0.22\%$ . The pH values were measured using a HACH  
129 HQ40d meter and densities using a standard hydrometer. Flash point was measured using a Setaflash  
130 series 3 plus closed cup flash point tester (model 33000-0). A Shatox OPLCM instrument was used to  
131 measure the cetane number, pour point temperature and filterability temperatures. Elemental analysis  
132 to investigate the carbon, hydrogen, nitrogen, oxygen and sulphur content (% wt) was performed by  
133 an accredited laboratory using a Thermo EA1108 series elemental analyser. The lower heating value  
134 (LHV) was calculated from the HHV and hydrogen content.

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### 137 **2.2. Experimental set-up**

138

139 The specifications of the IDI water cooled CI engine selected for the study are given in Table 1. The  
140 experimental set-up, including modifications carried out to the engine fuel supply and cooling water  
141 systems, is shown in Fig. 1.

142

#### 143 *2.2.1. Fuel supply and preheat*

144

145 A dual-fuel supply system was designed so that the engine can be started and warmed up with fossil  
146 diesel before switching to plant oil after about 10 minutes (Fig. 1). At the end of each test run  
147 reversion to diesel operation flushed the remaining plant oil from the fuel supply and engine injection  
148 system. This avoided clogging of the fuel filters and injector holes due to thickening of the oil once  
149 the system cools. The fuel supply lift pump supplied with the engine was disconnected, and the spill  
150 from the injector was connected to the main fuel supply line using a T-connector. Additional filters  
151 were connected into the fuel supply system. The fuel supply tanks were placed at 3 m height for  
152 reliable fuel flow and to overcome the pressure losses in the additional fuel filters and heat exchanger  
153 (HX). Note that Bari et al. [37] have already reported the advantages as regards power output of  
154 connecting additional fuel filters and raising the fuel supply tank in plant-oil fuelled engine operation.  
155 Two 12V DC fuel transfer pumps were installed to transfer the fuels from the ground tank to the  
156 overhead tanks (not shown). Stainless steel piping and valves were used to avoid any corrosion or  
157 erosion.

158

159 At full load, engine jacket water temperature is around  $100^\circ\text{C}$  with a flow of 33 l/min (Table 1); this  
160 means that a considerable amount of the heat energy is available in the hot jacket water. To preheat  
161 the plant oil with this water, a plate type HX was connected to the engine jacket water cooling system  
162 via two manually operated valves that were opened when using plant oils but kept closed for diesel.  
163 Another manually operated valve controlled the flow of the hot jacket water to the plate HX and  
164 hence the temperature of the preheated plant oils (Fig. 1).

165 In a standard engine, hot jacket water is cooled by a radiator and fan and then returned to the cooling  
166 circuit. To facilitate testing within a standard test cell, however, the radiator was by-passed by a  
167 header tank type HX (Bowman UK) supplied with tap water. Although the radiator water pipes were  
168 disconnected from the engine, the radiator itself and the fan were not disconnected to enable  
169 comparisons of the engine performance with the rated figures provided by the manufacturer.  
170 Moreover the radiator fan was needed to cool the engine accessories.

### 171 172 *2.2.2. Instrumentation*

173  
174 An eddy current dynamometer (Froude Hofmann AG80HS) was used to measure and adjust the  
175 engine load and speed (Fig. 1). The torque and speed accuracies of the dynamometer were  $\pm 0.4$  Nm  
176 and  $\pm 1$  rpm respectively. A five-gas emission analyser (Bosch BEA 850) and smoke opacity meter  
177 (Bosch RTM 430) were used to analyse the exhaust gas components and to measure the smoke  
178 intensity respectively. A graduated cylinder and stop watch were used to measure the fuel  
179 consumption rate (Fig. 1). K-type thermocouples were used to measure the temperatures of the  
180 exhaust gas, fuel inlet, lub oil and engine jacket water. Fuel temperature was measured immediately  
181 after the plate HX and after the fuel injection pumps ie. just before the injector, thus approximating as  
182 closely as possible the temperature of the injected fuel (Fig. 2). A LabVIEW data acquisition system  
183 was used to log the temperatures at the various locations. The dynamometer was calibrated and the  
184 calibration curve (not shown) was used to obtain the brake torque applied at the engine shaft. The  
185 engine was operated at different loads at a constant speed of 1500 rpm. Brake thermal efficiency was  
186 calculated based on LHV.

187  
188 A pressure sensor (Kistler 6125C11) and charge amplifier (Kistler 5064B11) were used to measure  
189 pressure in the cylinder nearest the radiator end of the engine. A suitable location in the main  
190 combustion chamber was identified and a mounting sleeve with locking mechanism was designed and  
191 fabricated for this installation. Machining of the cylinder head through the water jacket was performed  
192 accordingly (Fig. 2). The sensor was not installed flush to the combustion chamber surface; rather it  
193 was recessed by 2 mm to avoid damage to the diaphragm and to aid maintenance. Another pressure  
194 sensor (Kistler 4065A500A0) and amplifier (Kistler 4618A0) were used to measure the fuel line  
195 injection pressure (Fig. 2). This sensor was installed on the fuel delivery line of the same cylinder. An  
196 optical encoder (Kistler 2614A) was installed for detection of the crank angle position. The amplifiers  
197 and the encoder electronics were connected to the 'KiBox' (Kistler, model 2893AK8) for data  
198 logging, which was connected to a PC through the ethernet port (Fig. 1). KiBoxCockpit software was  
199 used to measure and analyse the various engine combustion parameters.

## 200 201 202 **3. Results and discussion**

### 203 204 205 **3.1. Fuel properties**

206  
207 With regard to LHV, density and cetane number, Table 2 shows only very small differences between  
208 karanj and jatropha and minor differences of both plant oils compared to fossil diesel. The LHVs of  
209 the plant oils were lower by about 11% and their densities higher by about 12%. Their cetane numbers  
210 were almost 10% lower than for fossil diesel. In contrast, their viscosities and flash point temperatures  
211 were considerably higher than those of diesel. Viscosity of the plant oils is temperature dependent and  
212 can be reduced by 80-90% through preheating to 90°C (Fig. 3). Note that high flash point is  
213 advantageous as regards safety in storage, handling and preheating.

214  
215 The carbon content in the fossil diesel was 7–8% higher than in the plant oils. Nitrogen content in  
216 jatropha oil was much higher than in karanj oil and in fossil diesel (Table 2). Hydrogen content was  
217 comparable to fossil diesel but oxygen content was 6 to 7 times higher in the plant oils. For all three  
218 fuels sulphur was at trace levels. The properties of any plant oil depend on the plant variety and

219 conditions of cultivation; however, the properties of the oils measured in our study are in line with  
220 those found in the literature [16].

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### 223 **3.2. Engine performance and exhaust emissions**

224

225 Preheating raised the temperature of the oils to 58–75°C, the higher temperatures being achieved at  
226 higher brake power outputs due to increasing jacket water temperature (Fig. 4). Although the HX was  
227 installed close to injection pump, the temperature of the fuel dropped by 12–15°C by the time it  
228 reached the injectors (Fig. 4). The level of preheating achieved by using the jacket water reduced the  
229 viscosity of the oil considerably by 60–70% (Fig. 3).

230

231 At full load, BSFC was 16% and 3% higher on weight and volume bases respectively, when the  
232 engine was fuelled with preheated neat jatropha (or karanj) oil, as compared to unheated fossil diesel  
233 (Fig. 5a, 5b). Because the calorific value of the jatropha or karanj oil is lower than that of fossil diesel,  
234 more fuel is needed for the same engine output. In addition, although the plant oils were preheated to  
235 60–75°C, they remained 10 times more viscous than diesel leading to less efficient mixing of fuel and  
236 air. These factors explain the somewhat higher fuel consumptions. Comparison on a volume rather  
237 than weight basis showed smaller differences in BSFC due to the higher densities of the plant oils  
238 (Table 2). Between jatropha and karanj, BSFC results were very close on both weight and volume  
239 bases (Fig. 5a, 5b). Brake thermal efficiencies from both plant oils were close to those from diesel at  
240 high load, but 10% lower than from diesel at low load condition (Fig. 5c).

241

242 At low loads, CO and CO<sub>2</sub> emissions were almost the same for all fuels (Fig. 6a, 6b). The amount of  
243 intake air (i.e. oxygen content) was constant throughout the engine test – so, at higher loads the fuel to  
244 air ratio decreased (lower lambda values) and hence CO emission increased sharply due to the lower  
245 oxygen content in the mixture. In the case of jatropha and karanj oil operation and at higher load  
246 conditions, CO and CO<sub>2</sub> emissions were higher than with fossil diesel as higher amounts of the plant  
247 oils were burned for the same engine output (Fig. 6a, 6b). In addition, it is probable that the higher  
248 oxygen content in the plant oils helped to combust the jatropha (or karanj) oil and caused higher CO<sub>2</sub>  
249 emission than with fossil diesel. At full load condition, CO<sub>2</sub> emission was 7% higher for plant oil  
250 operation (Fig. 6b). It was also observed that at higher load condition, CO emission was slightly  
251 higher with jatropha than karanj. This may be explained by the lower oxygen content in the jatropha  
252 oil.

253

254 Higher combustion temperatures in CI engines generally give higher NO<sub>x</sub> emission. For jatropha and  
255 karanj operation, NO<sub>x</sub> emissions were 8% higher than fossil diesel in these tests (Fig. 6c). This is  
256 consistent with the higher peak cylinder pressures observed with the plant oils (section 3.3), as higher  
257 pressures are normally associated with higher combustion temperatures. Oxygen emissions were  
258 almost similar under low load conditions for all fuels; but at high loads O<sub>2</sub> emissions were lower for  
259 jatropha (or karanj) oil operation (Fig. 6d). Higher level of fuel-borne oxygen in the jatropha (or  
260 karanj) oil probably caused this difference.

261

262 Exhaust temperature is important for co-generation applications (CHP and tri-generation). Little  
263 difference was observed in exhaust temperatures among the three fuels tested (Fig. 7a). At high load  
264 condition, the smoke opacity values were slightly higher than fossil diesel when the engine was  
265 operated either on jatropha or karanj oil (Fig. 7b). Lambda values were found to be mostly similar for  
266 all fuels (Fig. 7c). However, at higher loads, lower lambda values were observed for plant oil  
267 operation corresponding to the higher BSFC.

268

269 At low load, an additional means to preheat the oil is desirable, as the jacket water temperature is low.  
270 An off-the-shelf glow plug type electrical heater could be used for this purpose. This type of heater is  
271 powered by the engine alternator and can be switched on during low load operation. Overall, the  
272 performance and exhaust emission results obtained in the current study compare favourably to those  
273 typically reported in the literature from studies on DI engines [23, 32-35]. Use of an IDI type engine,

274 with additional fuel filters and raised fuel tank, caused this better engine performance. Previous  
275 studies were mostly conducted in single cylinder DI engines with compression ratios in the range of  
276 16.5–18.5, whereas the compression ratio of the engine used in this study was 22. Although the  
277 engine was not tested at other compression ratios, it is believed that this high compression ratio could  
278 be another reason for better engine performance and emission.

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### 281 **3.3. Combustion characteristics**

282

283 Stable engine operation was obtained regardless of fuel and load, as evident from the smooth  
284 pressure-crank angle diagram (Fig. 8). No significant differences in peak cylinder pressure were noted  
285 between jatropha and karanj oils over a range of loads. Compared to fossil diesel, however, peak  
286 cylinder pressures were about 3% higher (Fig. 8). The cylinder pressure profiles before the peak  
287 pressure point were almost similar for all three fuels, and only minor variations were observed after  
288 the peak cylinder pressure point. For all fuels and load conditions, the peak cylinder pressure occurred  
289 between 10° and 12.5° crank angle (CA) after top dead centre (ATDC) position. In general, jatropha  
290 and karanj oil operation resulted in peak cylinder pressure occurring close to the top dead centre  
291 position, earlier than with diesel (Fig. 8).

292

293 At low loads, integral heat release from fossil diesel combustion was higher (Fig. 9). This can be  
294 explained by the lower preheating temperature and higher viscosities of the plant oils at lower loads  
295 (Fig. 3, Fig. 4) leading to incomplete combustion. At higher loads, the heat release behaviour was  
296 similar to fossil diesel (Fig. 9). For any specific load condition, peak integral heat release values were  
297 almost similar between jatropha and karanj. In general, the fuel line injection peak pressure values  
298 were higher in the case of the plant oil operation, due to the higher viscosity values of the plant oil  
299 even after preheating (Fig. 10); at full load it was almost 9% higher than for fossil diesel.

300

301 Total combustion duration is defined as the duration of the crank angle between 5% and 90%  
302 combustion. Ignition delay is another important combustion parameter, which is related to the ignition  
303 quality (ie. cetane number) of the fuel. Engine parameters affecting the ignition delay period are:  
304 compression ratio, engine speed, cylinder gas pressure, temperature of the air intake, and quality of  
305 fuel spray [34]. In the case of plant oil operation, the start of combustion was delayed compared to  
306 fossil diesel operation which is consistent with the lower cetane number (Table 2). Total combustion  
307 duration increased with engine load for all fuels. The duration was shorter for the plant oils than for  
308 diesel (Fig. 11a, 11b). At full load, combustion duration of the jatropha (or karanj) operation was  
309 almost 5% lower than for fossil diesel; no significant differences were observed between jatropha and  
310 karanj in this respect.

311

312 It was observed that in the case of plant oils, the maximum heat release rates were advanced by about  
313 1 to 2.5°CA (figure not shown) relative to fossil diesel. The ignition delay was longer, but the  
314 combustion duration was shorter than fossil diesel (Fig. 11). Quick burning in a short period of crank  
315 angle duration (i.e. in smaller cylinder volume) caused higher peak cylinder pressures in the case of  
316 plant oils. Higher peak cylinder pressures in the case of plant oils have also been reported in the  
317 literature [20,21,26,37].

318

319 In case of plant oils, the start of combustion was delayed; but once combustion started, the oils burnt  
320 quickly resulting in shorter combustion duration than fossil diesel. It is likely that the presence of fuel-  
321 borne oxygen in the plant oils helped complete combustion, and together with the high ignition delay,  
322 this caused shorter combustion duration. Furthermore, it was observed that the peak cylinder pressures  
323 and maximum heat release rates were advanced in case of jatropha (or karanj) oil relative to fossil  
324 diesel, due to the shorter combustion duration. The higher ignition delay period and shorter  
325 combustion duration in the case of plant oils (or blends) resulted in higher peak cylinder pressures  
326 than with fossil diesel [38]. The combustion duration or ignition of fuel is a complex system and may  
327 depends on many variables, for example: engine type (IDI/DI) and speed, compression ratio,  
328 geometry of combustion chamber, injection parameters (IOP, injection timing), injector geometry,

329 temperature of air intake and fuel properties. Generally, plant oils or biofuels have lower cetane  
330 number than fossil diesel; therefore, longer ignition delays were observed [38-39]. In contrast to this,  
331 shorter ignition delay period has also been reported in the literature [23, 26, 37]. Some studies found  
332 that fuels with high cetane number have shorter ignition delay and longer combustion duration [40-  
333 41]. However, both longer [23, 26] and shorter [42-43] combustion duration were reported for plant  
334 oils and other biofuels. This is an area where further research work is recommended.

335

336

### 337 **3.4. Engine durability: coke formation**

338

339 Coke deposition inside the engine cylinder and in the fuel supply systems, piston ring sticking,  
340 leakage of the plant oils into the crank case, clogging of fuel filters and injectors, engine wear and  
341 other engine durability problems were reported in the literature when neat plant oils were used in  
342 standard CI engines [16,21,37,44-46]. Although a complete endurance study was not performed  
343 during this study, a visual observation of the piston, cylinder, fuel supply system, and injector  
344 conditions was carried out after 50 hours of engine operation with preheated non-edible plant oils. A  
345 very small amount of coke formation occurred on to the piston surfaces and cylinder heads (Fig. 12).  
346 Injectors were inspected and no corrosion or erosion of the injector was observed. The fuel filters  
347 were not clogged. Neither engine knocking nor any instability in the engine operation was noticed.  
348 Adoption of a preventive maintenance schedule and some changes in the routine maintenance strategy  
349 should be conducive to longer term engine operation when the engine is fuelled with these oils.

350

351

## 352 **4. Conclusions**

353

354 A three-cylinder indirect-injection compression ignition engine, with nominal output 9.9 kW and  
355 compression ratio 22, has been tested with non-edible plant oils from jatropha and karanj trees.  
356 Performance, emissions and combustion characteristics were compared against fossil diesel operation.  
357 The engine was modified with a heat exchanger such that the plant oils were preheated by the engine  
358 jacket water to temperatures of 46–60°C prior to injection. The three fuels were characterised and  
359 found to have properties similar to those reported in the literature.

360

361 Between the jatropha and karanj oils, there were few differences in the results from the engine tests.  
362 Compared to fossil diesel, however, there were a number of small but significant differences when  
363 using the plant oils:

- 364 i. At full load the BSFC was increased by 3% on a volume basis and 16% on a weight basis,  
365 while brake thermal efficiency was similar.
- 366 ii. CO<sub>2</sub> and NO<sub>x</sub> emissions were higher by 7% and 8% respectively; CO emissions were  
367 similar to fossil diesel operation at low load but at higher loads the plant oils gave slightly  
368 higher CO emissions; the O<sub>2</sub> emissions were either similar to or less than those from  
369 fossil diesel.
- 370 iii. Peak cylinder pressures were slightly higher (~3%).
- 371 iv. Peak integral heat release values were smaller at low load, translating into a lower  
372 efficiency than with fossil diesel, but at full load they were almost the same.
- 373 v. At full load, fuel line injection pressure was around 9% higher.
- 374 vi. Total combustion duration was lower under all load conditions.

375

376 The cylinder gas pressure diagrams with jatropha and karanj oils indicated stable engine operation  
377 with these alternative fuels. This study concludes that, an IDI type CI engine used in the current study  
378 combusted the plant oils more efficiently and produced better engine performance, emission and  
379 combustion characteristics, compared to results typically reported in the literature using jatropha (or  
380 karanj) oil in DI engines with lower compression ratio. This preliminary study reports that neat non-  
381 edible jatropha or karanj oils can be used effectively (without blending with fossil diesel or  
382 conversion to biodiesel) in a suitably selected CI engine, provided minor modifications to the standard

383 engine are made. However, full endurance tests are required to assess the engine durability issues and  
384 are recommended as further work.

385  
386

### 387 **Acknowledgements**

388  
389 This work was supported by the Research Councils (UK) and DST (Government of India) under the  
390 Science Bridges programme (grant number EP/G039992/1). The authors would like to thank Mr  
391 David Smith for providing technical assistance in various stages of the study, and Mr Asrar Razzaq  
392 (project student, Aston University) and Mr Dhruvatar Mitra (SRI, Ranchi, India) for their help during  
393 the experiments.

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Table 1

## Specification of the experimental engine

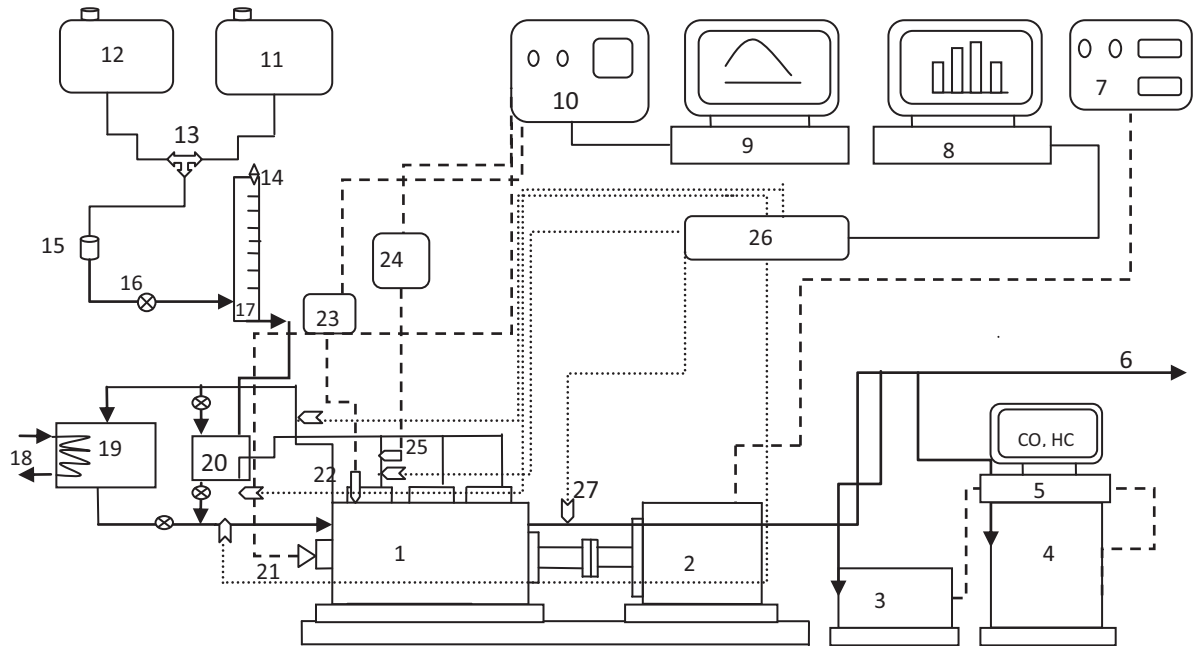
Manufacturer	Lister Petter (UK)
Model/type	LPWS Bio3 water cooled
No. of cylinders	3
Bore/stroke	86/80 mm
Rated speed	1500 rpm
Continuous power at rated speed	9.9 kW
Overload power at rated speed	10.9 kW
Type of fuel injection	Indirect injection. Self-vent fuel system with individual fuel-injection pumps
Fuel pump injection timing	20° BTDC
Aspiration	Natural
Cylinder capacity (litre)	1.395
Compression ratio	1:22
Firing order	1-2-3
Minimum full load speed (rpm)	1500
Maximum permissible exhaust back pressure (mbar)	75
Continuous power fuel consumption at 1500 rpm	3.19 litre/hr (fossil diesel)
Glow plug	Combustion-chamber glow plugs
Exhaust gas flow	41.4 litres/sec at full loads at 1500 rpm
Jacket water flow at full load	33 litres/min (at 1500 rpm)
Maximum engine jacket water temperature	99 - 102°C

Table 2

Measured properties of jatropha, karanj and fossil diesel

Physical and chemical properties	Jatropha oil	Karanj oil	Fossil diesel
Kinematic viscosity (cSt) at 25°C	56.01	79.21	3.37
Pour point temperature (°C)	-6.05	-7.65	-6.17
Cetane number	44.6	45.6	50.2
Filterability temperature (°C)	-7.2	-	-8.03
Flash point temperature (°C)	245	257	70
pH value @ 22°C	6.21	5.82	6.85
Density (kg/m <sup>3</sup> ) @ 18°C	916	934	834
Higher heating value (HHV) (MJ/kg)	39.87	39.10	44.67
Lower heating value (LHV) (MJ/kg)	37.25	36.61	41.87
Carbon (% wt)	77.88	78.10	84.73
Hydrogen (% wt)	12.38	11.74	13.20
Nitrogen (% wt)	1.19	<0.10	<0.10
Sulphur (% wt)	<0.10	<0.10	<0.10
Oxygen (% wt)	8.75	10.30	1.40

Figure 1



1: Engine; 2: Dynamometer; 3: Smoke meter; 4: Exhaust analyser; 5: Exhaust data acquisition; 6: Exhaust gas discharge; 7: Dynamometer controller; 8: NI data acquisition for temperature; 9, 10: Kistler combustion analyser; 11: Jatropha/Karanj tank; 12: Diesel tank; 13: 3-way valve; 14: Vent screw; 15: Additional fuel filter; 16: Valve; 17: Fuel measurement; 18: Cold water flow to HX; 19: HX to cool jacket water; 20: Plant oil preheater; 21: Crank angle encoder; 22: Cylinder pressure transducer; 23, 24: Amplifier; 25: Injection pressure sensor; 26: NI DAQ; 27: Thermocouple

Fig. 1. Schematic diagram of the engine modifications and experimental rig

Figure 2

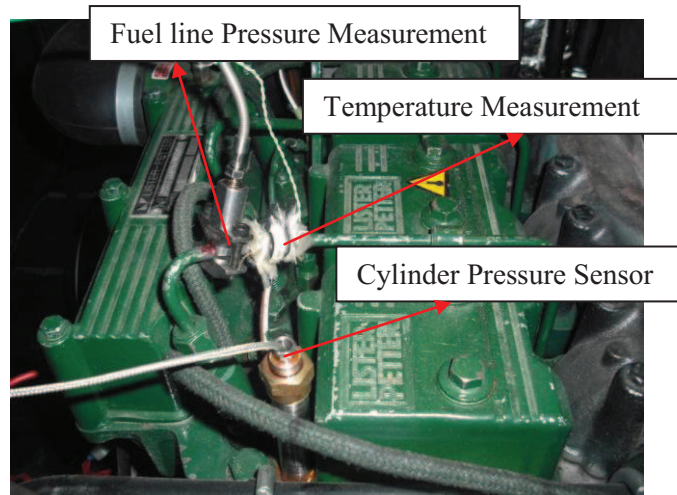


Fig. 2. Measurement of cylinder pressure, and fuel line injection temperature and pressure

Figure 3

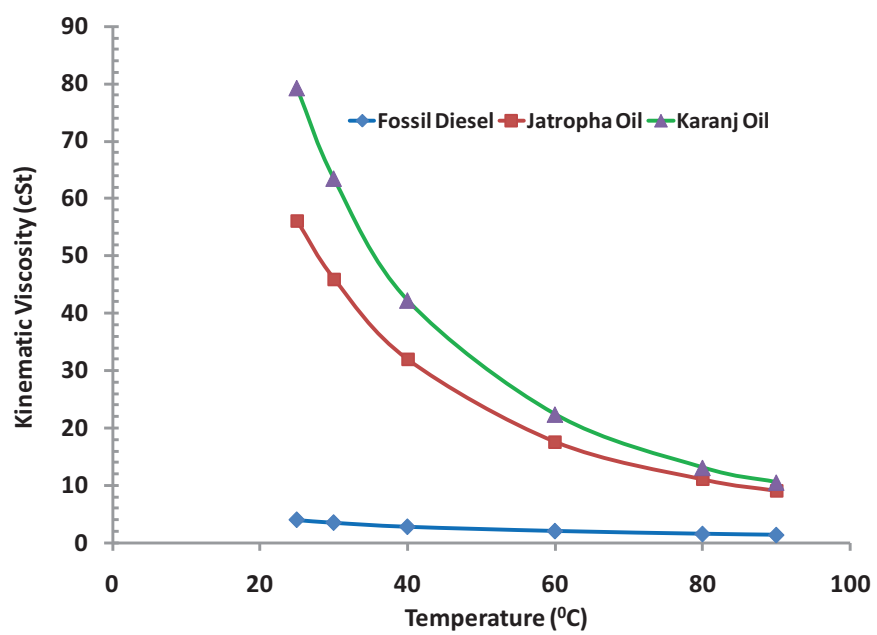


Fig. 3. Viscosity vs. temperature of jatropha, karanj and fossil diesel

Figure 4

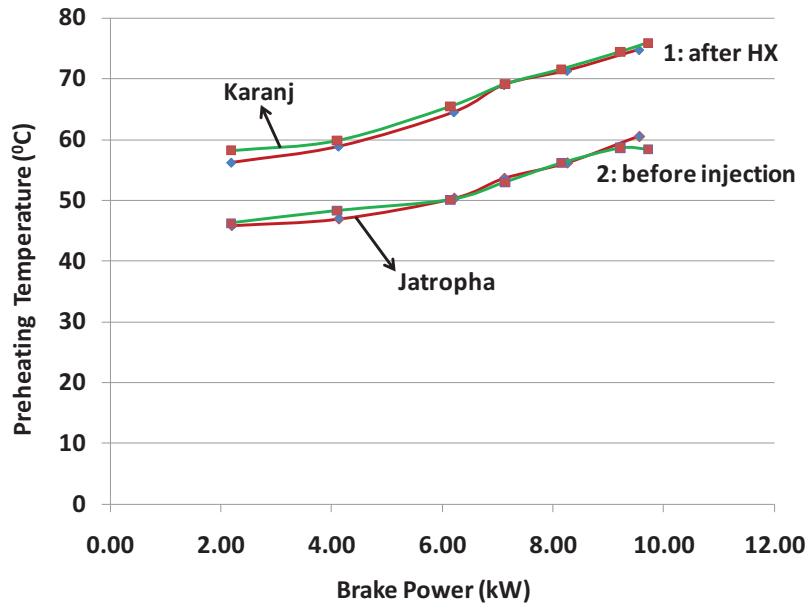


Fig. 4. Preheating temperatures of neat jatropha and karanj oils at different engine load conditions

Figure 5

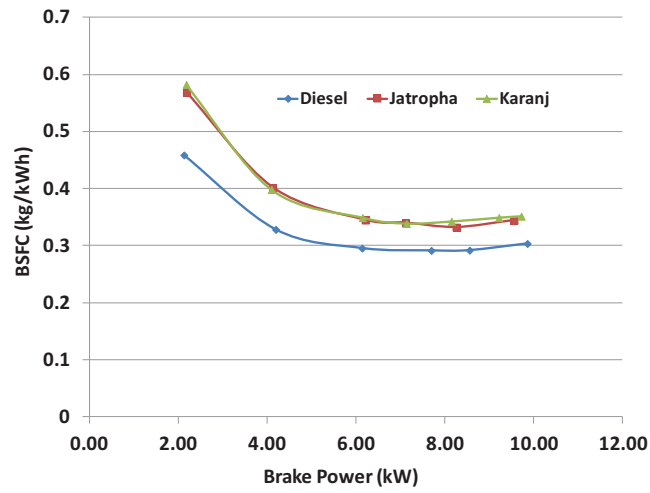


Fig. 5a - BSFC (kg/kWh) vs. load

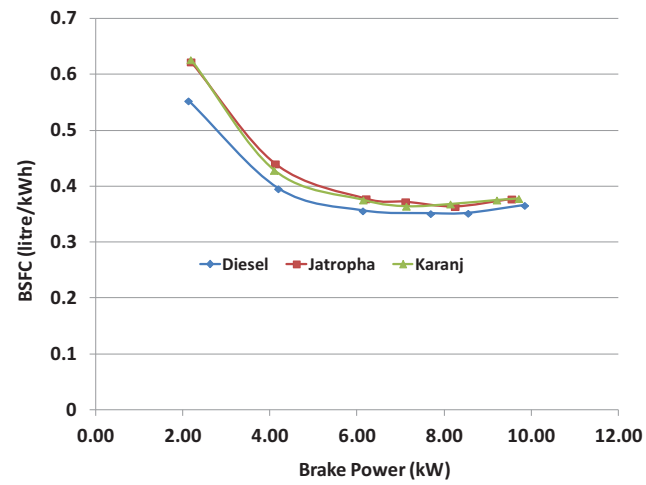


Fig. 5b - BSFC (litre/kWh) vs. load

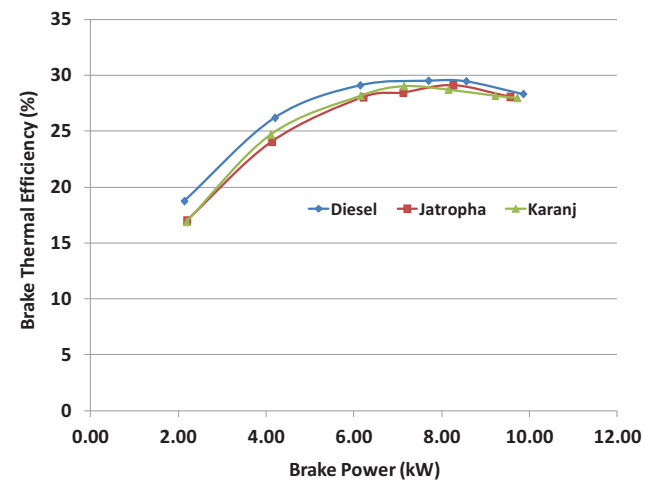


Fig. 5c - Brake thermal efficiency vs. load

Fig. 5. Brake specific fuel consumption and brake thermal efficiency vs. engine load



Figure 6

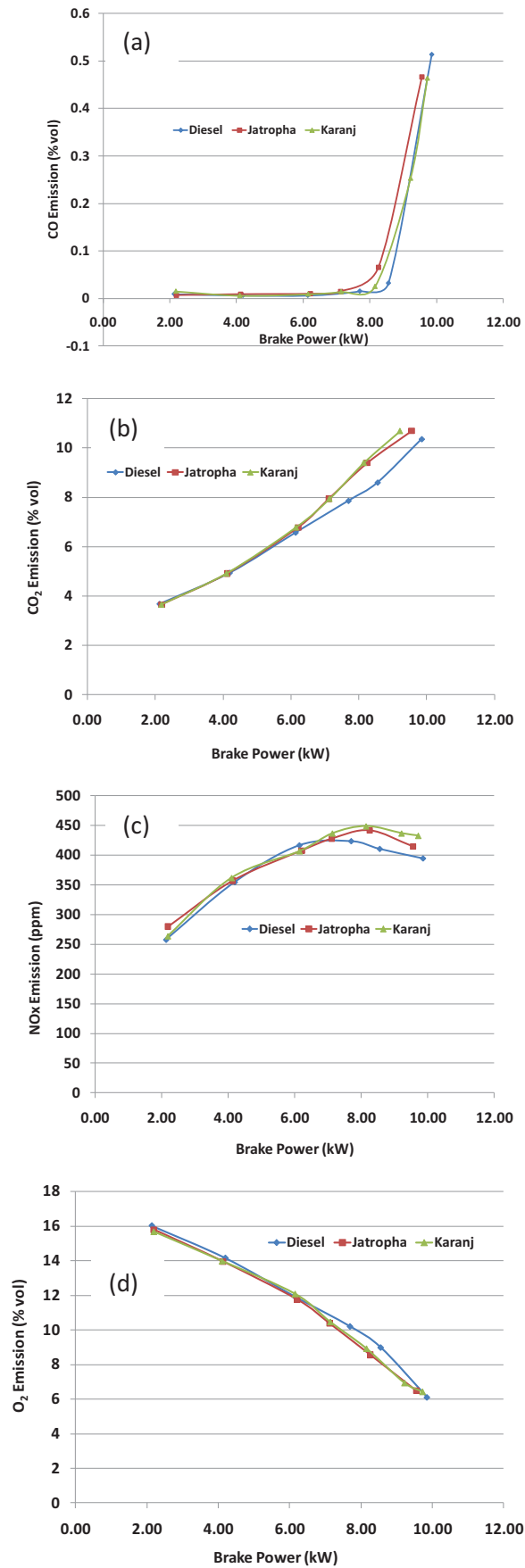
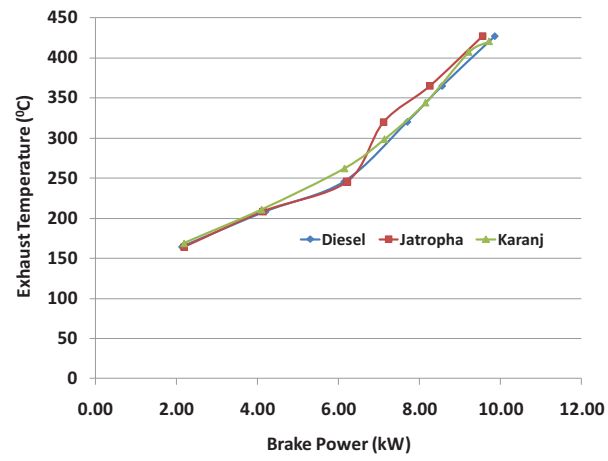
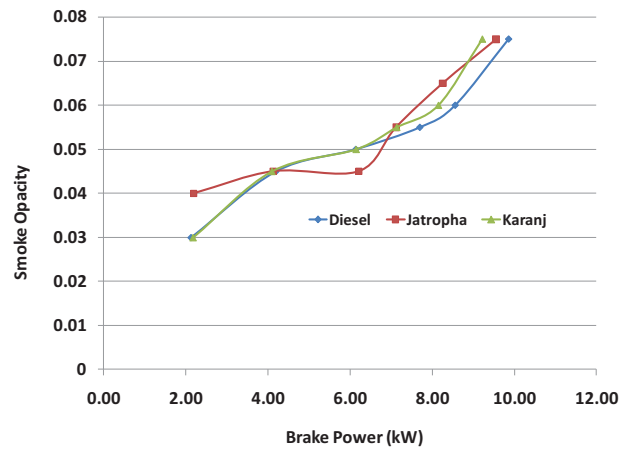


Fig. 6. Comparison of exhaust emissions at different engine load conditions: (a) CO emission, (b) CO<sub>2</sub> emission, (c) NO<sub>x</sub> emission, and (d) O<sub>2</sub> emission

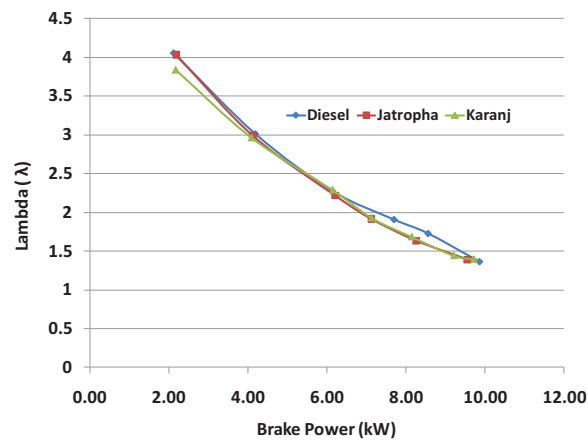
Figure 7



7a – Exhaust temp vs. load



7b – Smoke opacity vs. load



7c – Lambda values vs. load

Fig. 7. Exhaust gas temperature, smoke opacity and lambda values vs. engine load

Figure 8

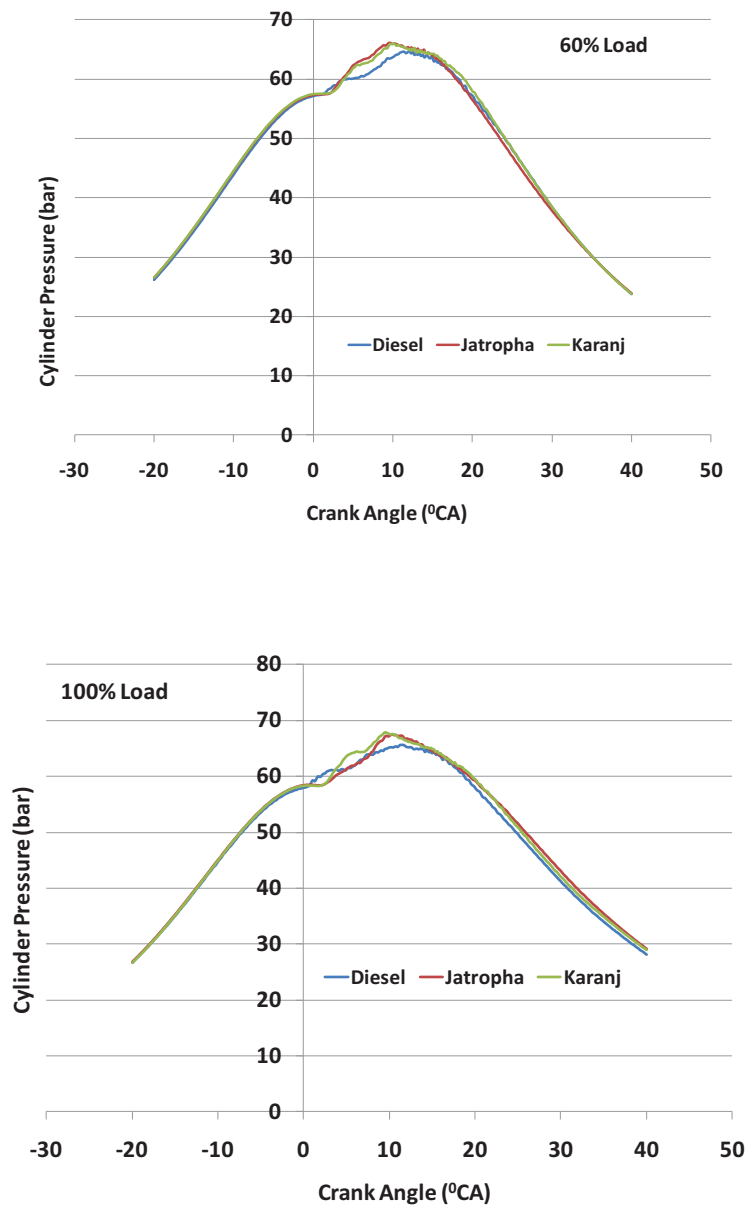


Fig. 8. Cylinder pressure vs. crank angle at different engine loads

Figure 9

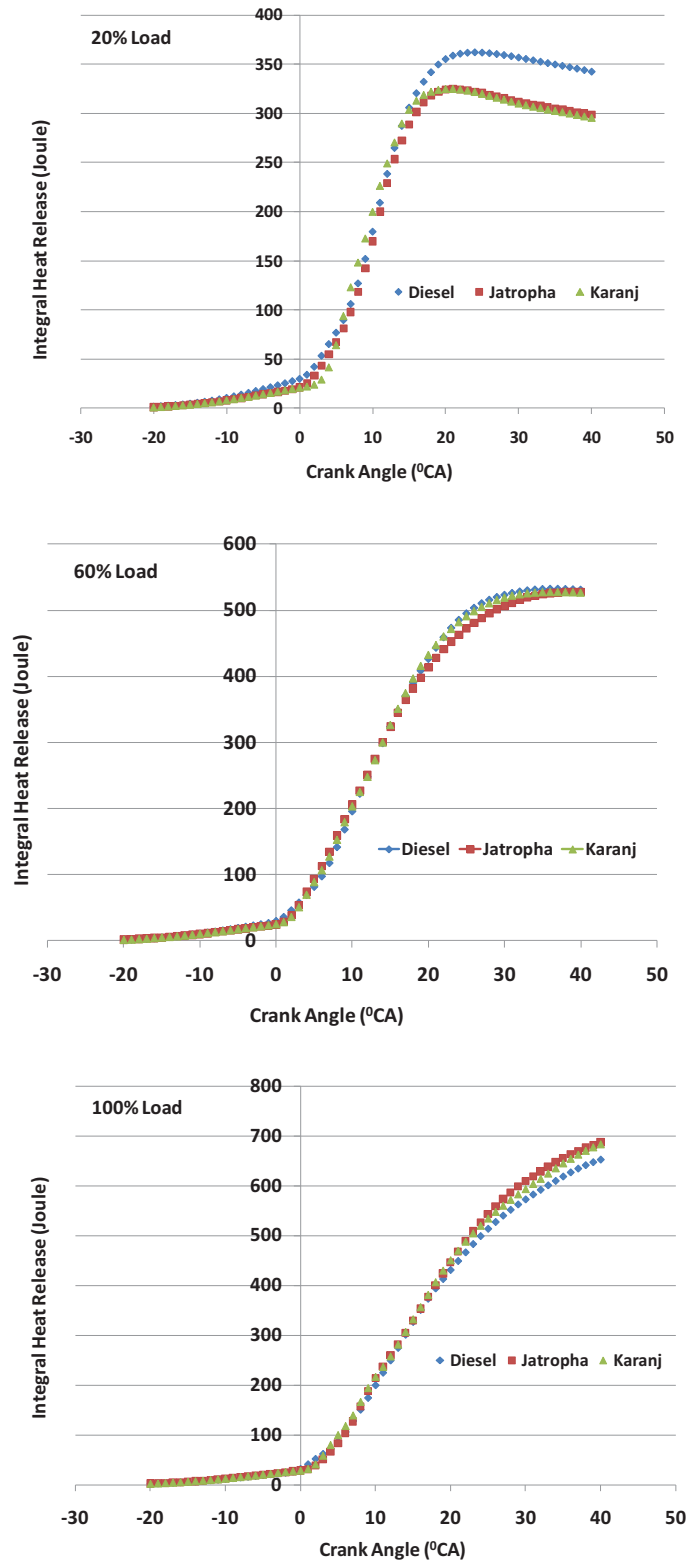


Fig. 9. Integral heat release vs. crank angle for different fuels and at different loads

Figure 10

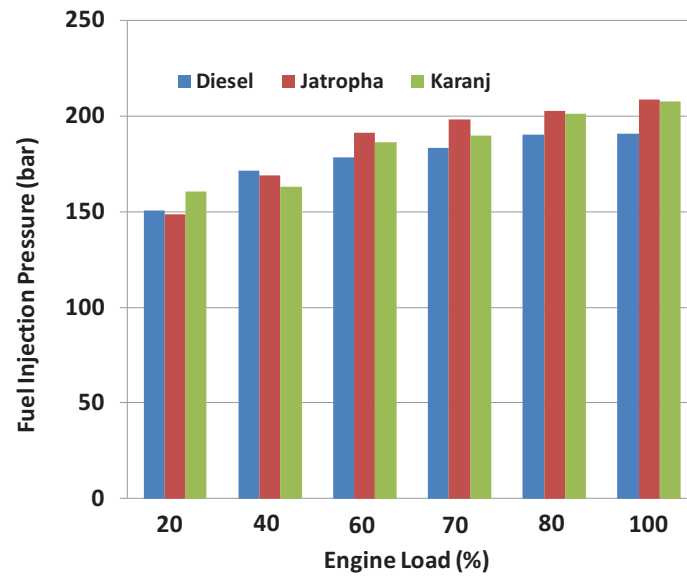
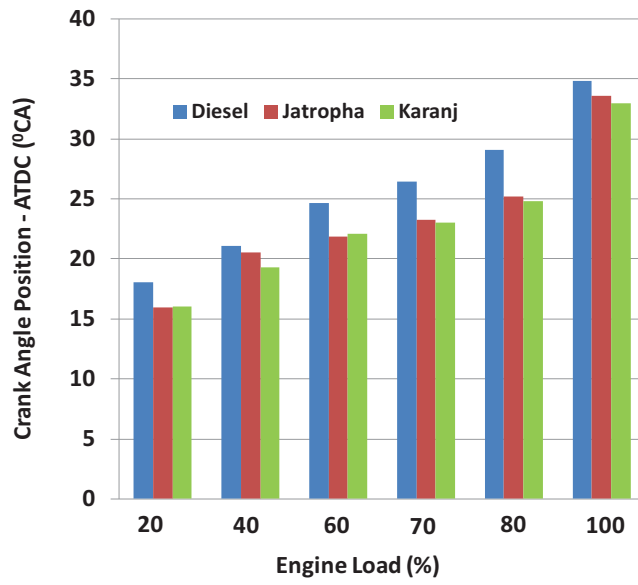
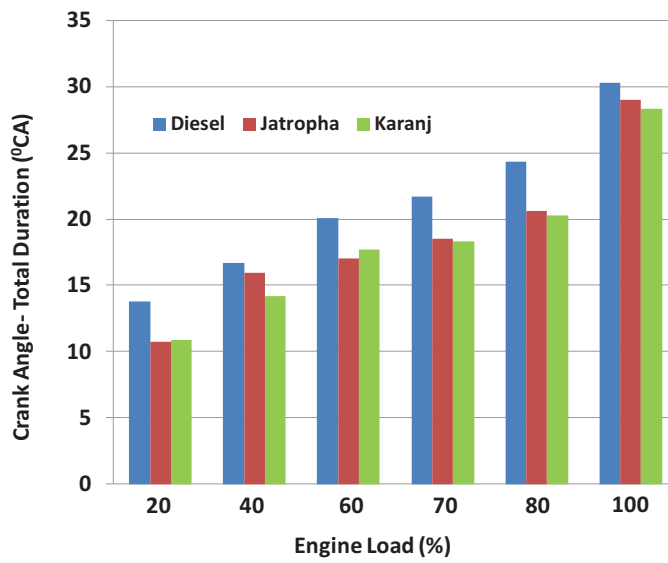


Fig. 10. Maximum fuel injection pressure vs. engine load



(a) 90% combustion



(b) Total combustion duration

Fig. 11. Combustion of fuels at various engine load conditions

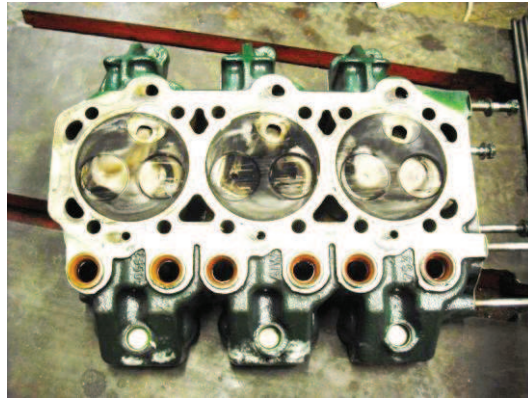


Fig. 12. Coke deposits in the cylinder head and intake/exhaust valves