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F.Z. Aklouche, Khaled Loubar, A. Bentebbiche, Sary Awad, Mohand Tazerout. Experimental investigation of the equivalence ratio influence on combustion, performance and exhaust emissions of a dual fuel diesel engine operating on synthetic biogas fuel. *Energy Conversion and Management*, 2017, 152, pp.291-299. 10.1016/j.enconman.2017.09.050 . hal-01731269

HAL Id: hal-01731269

<https://hal.science/hal-01731269>

Submitted on 20 Nov 2018

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1 **Experimental investigation of the equivalence ratio influence on**
2 **combustion, performance and exhaust emissions of a dual fuel**
3 **diesel engine operating on biogas fuel**

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8

9 **ABSTRACT**

10 In this work, an experimental investigation was conducted to analyze dual fuel engine (DF)
11 operation using biogas fuel under high load at constant percentage energy substitution rate
12 (PES). The performance, the ignition delay and other combustion characteristics of engine
13 operating in dual fuel mode (biogas/diesel) are compared to the conventional mode. The
14 biogas, composed of 60% methane and 40% carbon dioxide, is the primary fuel which is
15 blended with air in the engine inlet manifold, whereas the pilot fuel is diesel. The equivalence
16 ratio (ϕ) was varied by changing air flow rate while the energy introduced into the engine
17 remained constant for all the examined cases. Combustion analysis showed that with
18 increasing ϕ , the ignition delay tends to become longer and the peak of heat release rate was
19 increased. Furthermore, as the ϕ increased from 0.35 to 0.7, THC and CO emissions were
20 reduced by 77% and 58% respectively. The NO_x emissions decreased at 60% PES by 24%
21 while the BTE was improved by 13%.

22

23 **Keywords:** Biogas; dual fuel engine; combustion; pollutant emissions; equivalence ratio.

24

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27 **1. Introduction**

28 The development of internal combustion engines and the process of reducing pollutant
29 produced by fuel combustion have become mandatory requirements for propulsion systems
30 designers. One of the main objectives is to find the best way to minimize the emissions
31 without considerably modify the engine's mechanical structure while maintaining the same
32 performance. One of the proposed solutions is the method of injection of diesel fuel and the
33 use of diverse gaseous fuels like biogas, natural gas, etc.

34 As a result of the gradual depletion of fossil oil and the environmental degradation, the
35 utilization of gaseous fuel from biomass attracts the attention of the researchers. Thus, the use
36 of some alternative gaseous fuels in the diesel engine increases worldwide. Biogas is an
37 alternative fuel produced through a biological process. In the absence of oxygen (anaerobic
38 condition), the methanization reaction starts spontaneously when organic matter is in a tepid
39 environment, where it decomposes to form a gas mixture called biogas. Raw materials like
40 industrialized waste, agricultural and biomass present an important potential for the
41 generation of biogas but are underutilized [1-3]. However, the biogas composition varies
42 according to parameters such as humidity, gas pockets and variations of the waste location
43 [4].

44 Because of the increasing demand to reduce the pollutant emissions of existing technologies
45 of internal combustion engine (ICE), several researches have been carried out investigating
46 new approaches to achieve this goal. However, amongst several studies which were realized,
47 the conversion of the diesel engine into a dual fuel (DF) engine, so as to use the alternative
48 fuel, showed interesting results. In this mode, gaseous fuel is injected into the inlet manifold
49 where it is mixed with air. After that, the gaseous mixture is entered into the combustion
50 chamber where the pilot fuel, which has an important cetane number, is directly injected, and
51 then ignites to initiate the combustion [6,7]. It is noted that biogas is a recommended fuel for

52 engines which have a big compression ratio taking advantage of its important octane number
53 allowing higher thermal efficiency. It was also reported that with minor modifications of the
54 diesel engine, the biogas could be used [8]. In fact, a compression ignition engine running in
55 conventional mode can be just altered into DF mode by branching a gas blender to its intake
56 manifold. Moreover, a fuel test system must be placed to adjust the flow rate of the injected
57 diesel fuel [9]. Also, it was suggested that the engine operating in DF mode running in biogas
58 can be used to reduce the quantity of the diesel injected and at the same time diminish the
59 compromise between NO_x and smoke [9-13].

60 In diesel engines, nearest the injection of the diesel, the air temperature is around 553K.
61 However, in dual fuel mode, under these conditions, the biogas cannot ignite without the
62 presence of a little quantity of pilot fuel to raise the temperature up to a temperature of about
63 1087K [14]. Furthermore, it has also been observed that, the dual fuel engine requires an
64 extended combustion duration and at the same time an over long ignition delay [15-17]. In
65 addition, it was noticed that the ignition delay can be shortened with the use of a fuel
66 possessing a great stage of dissolved oxygen, and a high cetane number. Also, a reduction of
67 the emission of exhaust can be observed [18].

68 Various gaseous fuels (methane, propane, CNG, LPG, hydrogen) can be used to power the
69 diesel engine operating in DF mode while maintaining engine efficiency, and reducing
70 exhaust emissions [19, 20]. Furthermore, the reduction of some exhausts emissions and the
71 improvement of the engine efficiency can be achieved by the optimizing of certain parameters
72 influencing the engine, such as, substitution ratio, injection timing, engine loads [21], and
73 inlet air temperature [22]. Moreover, it was noticed by Makareviciene et al. [23] that the
74 combustion performance was better with variation of injection timing.

75 Srinivasan et al [24] found that DF (natural gas/diesel) engine could participate to minimize
76 carbon dioxide emission by reason of the smallest carbon-hydrogen ratio. The natural gas may

77 decrease greatly both NO_x emission and soot [25, 26], which is complicated to attain in
78 conventional mode. In addition, for various advantages of natural gas, it may be used like an
79 additional fuel to operating in DF mode [26]. That is why former studies showed that the use
80 of the biomethane as a fuel for vehicles is preferable for the energy production [27].

81 Additionally, dual fuel mode using biodiesel as pilot fuels [28-31] shows a lowering of
82 particulate emissions with the increasing of load [30]. This was explained by the important
83 combustion temperature and enabling combustion amelioration. Luijten and Kerkhof [32]
84 have replaced the diesel by the Jatropha oil to ignite the DF engine operating with biogas. A
85 reduction of about 10% of the brake thermal efficiency was observed. Similar observation
86 was reported by other researchers using different pilot fuels (karanja methyl) [33].
87 Papagiannakis et al [34] obtained the effects of the amount of natural gas on NO_x emissions.
88 The results showed that the augmentation of natural gas mass ratio has a good effect to
89 minimize NO_x emissions.

90 On the other hand, the effect of the biogas composition in DF mode was studied by Daouk et
91 al [35]. They noted that the biogas containing 70% methane (by volume) and about 30%
92 carbon dioxide has advantages in terms of emissions compared to the conventional mode.
93 Also, the study of the performance of dual fuel (biogas-diesel) was established by Lounici et
94 al [20], where a good knock resistance was observed from measured torque at the knock
95 onset.

96 It was reported by several researchers that the high levels of THC emissions produced by DF
97 combustion is considered as a disadvantage of this mode. In fact, it was achieved in other
98 works that the hydrocarbon emission in normal diesel mode was lower than that in dual fuel
99 operation, with about 6000 ppm [36, 37]. In spite of the fact that, in DF mode, the THC
100 emissions were decreased as engine load increases, but the value of hydrocarbon emission

101 was even more important by about 2000 ppm [38] and 2.5 times [39] in DF mode with natural
102 gas. Thus, the need to get a method to diminish the THC emissions on a DF mode emerges.

103 Various methods were used in DF engine (NG/diesel) to reduce the THC without lowering the
104 thermal efficiency, such as using injection timing [40], gas recirculation burning [41], raising
105 the quantity of pilot fuel [42], etc.

106 Previous studies focused mainly on the use of natural gas for engines running in DF mode
107 with respect to biogas. There are fewer studies about the biogas as a high potential alternative
108 source of energy for compression ignition engines. Moreover, these studies stated that the use
109 of biogas can be limited by the high levels of pollutant emissions, especially unburned
110 hydrocarbon. Thus, the main objective of the present study is to investigate the use of biogas
111 in CI engine and to propose a technique to reduce the pollutant emissions without any
112 drawbacks on the performance of engine at high load. The proposed technique is based on
113 varying the total equivalence ratio using a throttle valve to adjust the air mass flow rate.

114

115 **2. Materials and procedure**

116 *2.1. Biogas used in engine test*

117 In general, the composition of biogas fuel is ranging from 50% to 70 % (by volume) for
118 methane and 30% to 50% for carbon dioxide. In this study, CH₄ and CO₂, stored separately in
119 pressurized bottles, were used in the engine test with a percentage of 60% and 40%
120 respectively. The biogas used was prepared using a computer controlled mixer.

121 In the beginning, the engine is operated only on diesel fuel to prepare the engine (warm up
122 and stabilize). After that, the engine is switched to dual fuel mode. The synthesized gaseous
123 fuel passes through a line to the intake manifold after mixing. Close to the compression
124 process ending, the flow rate of the pilot fuel injected into the combustion chamber is
125 maintained constant to cover ten percent of the power output. After that, by increasing only

126 the gaseous fuel flow (biogas), the power output is more rises. The compositions of primary
127 fuel and diesel fuel used for this study are shown in table 1.

128 **Table 1**
129 Diesel fuel and biogas properties.

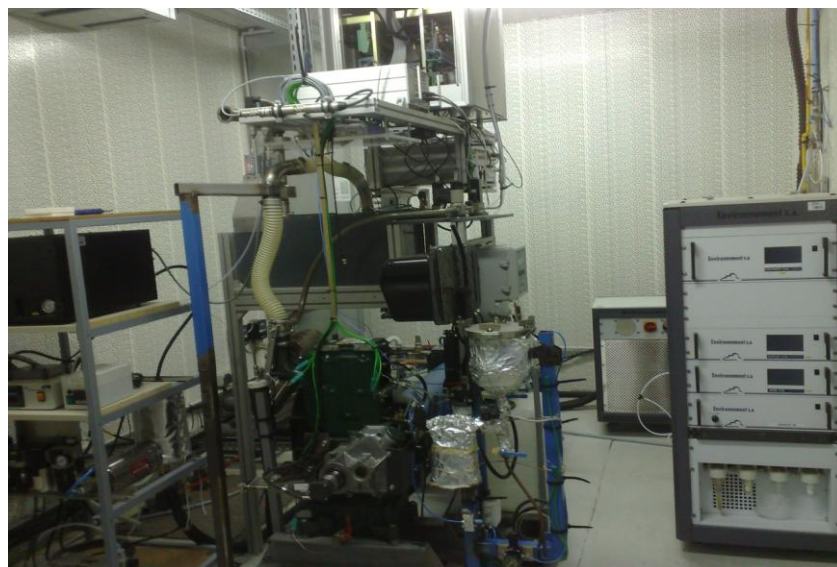
Component	Primary fuel (Biogas)	Pilot fuel (diesel)
Chemical Composition	60% CH ₄ , 40% CO ₂ (V/V (%))	C ₁₂ H ₂₆
Cetane number	-	49
Density (kg/m ³)	1.33	840
Stoichiometric air fuel ratio	6.04	14.60
LHV (MJ/kg)	17.65	42

130

131 2.2. Engine test and procedure

132 In this study, the engine tests were carried out in the laboratory of Energy of Ecole des Mines
133 de Nantes. It consists of a single-cylinder engine LISTER-PETTER, which is made to
134 function in a speed range of 0 to 2500 rpm, capable of generating a power up to 7.4kW. It
135 includes a dynamometer, a particle analyzer, an exhaust gas analyzer and a gas supply system
136 to feed the engine when it is operating in dual fuel mode. The fig.1 shows the engine test
137 scheme.

138



139

140

Fig.1. Experimental setup scheme.

141

142 Using AVL software, the data was registered and analyzed from 100 cycles. A sampling
143 interval of 0.2 crank angle (CA) is considered to obtain the combustion parameters. More
144 details about the acquisition system; sensors precision and uncertainty calculation could be
145 found in our previous works [25, 30].

146 Through the tests, the energy input under dual fuel operation (diesel pilot fuel+biogas) was
147 kept constant; this energy is allows to produce about 80% of the full load under conventional
148 mode. The percentage energy substitution rate (PES) of biogas was kept constant (60%) for
149 the different total equivalence ratios studied. The total equivalence ratio and the biogas
150 percentage energy substitution rate are determined as following:

$$151 \quad PES = (\dot{m}_B LHV_B) / (\dot{m}_B LHV_B + \dot{m}_D LHV_D) \quad (1)$$

$$152 \quad \phi = (\dot{m}_B AFR_{B-th} + \dot{m}_D AFR_{D-th}) / (\dot{m}_{air}) \quad (2)$$

153 Where \dot{m}_B , \dot{m}_D and \dot{m}_{air} (kg/h) are the mass flow rates of biogas, diesel and air respectively.
154 AFR_{B-th} and AFR_{D-th} (MJ/kg) are the theoretical air fuel ratios of biogas and diesel
155 respectively. LHV_B and LHV_D are the lower heating values of biogas and diesel respectively.

156 The injection timing of diesel fuel is fixed at 13°CA before top dead center (TDC). During
157 dual fuel operation, ϕ was varied by adjusting the air flow rate using a throttle valve.
158 However, for conventional diesel mode, the engine tests were conducted under wide open
159 throttle (WOT) condition.

160 The calibration of the engine test bench was done according to the instructions given in the
161 manufacturer catalog. The brake torque is measured from the dynamometer control system for
162 a given operating condition. Parameters, such as the pressure cylinder, the flow rates of diesel,
163 biogas and air are registered to analyze the engine efficiency.

164

165

166

167 3. Analysis combustion model

168 In order to examine the combustion process in both modes, the heat release rate (HRR) must
169 be determined. The HRR can supply valuable information for understanding the engine
170 operating. It is calculated analytically by applying and the first law of thermodynamics and
171 the law of ideal gas. Using the variation of the cylinder volume and the recorded value of the
172 cylinder pressure, the following expression is used to obtain the HRR [43].

$$173 \frac{dQ_{net}}{d\theta} = \frac{dQ_c}{d\theta} - \frac{dQ_w}{d\theta} = \frac{\gamma}{\gamma-1} P \left[\frac{dV}{d\theta} \right] + \frac{1}{\gamma-1} V \left[\frac{dP}{d\theta} \right] \quad (3)$$

174 Where $\frac{dQ_{net}}{d\theta}$ is the net HRR, $\frac{dQ_c}{d\theta}$ is the rate of heat released by the fuel combustion and
175 $\frac{dQ_w}{d\theta}$ is the heat transfer rate through the cylinder wall obtained from the Woschni's
176 correlation [43].

177 P is the cylinder pressure. γ is the ratio of specific heats. V represents the combustion chamber
178 volume; it depends on the crank angle (θ) and the geometric parameters of the engine. The
179 cylinder volume V is obtained by the following equation:

$$180 V(\theta) = V_d \left[\frac{Cr}{Cr-1} - \frac{1-\cos\theta}{2} + \frac{1}{2} \sqrt{\left(2\frac{L}{C}\right)^2 - \sin^2\theta} \right] \quad (4)$$

181 where V_d , Cr , L and C are respectively the displacement volume, the compression ratio, the
182 connecting rod length and the stroke.

183

184 4. Results and discussion

185 4.1 Combustion characteristics

186 The results shown below are discussed for a percentage energy substitution rate (PES) of
187 60%. The effect of the variation of total equivalence ratio was studied for the engine dual fuel

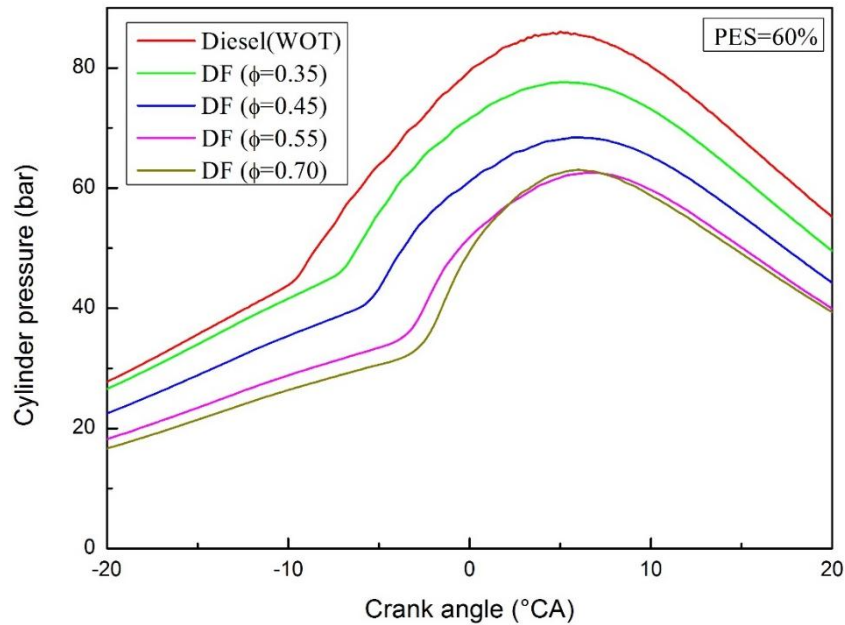
188 (biogas/diesel) at high load. To diagnose combustion behavior, it is important to analyze the
189 cylinder pressure history, which is obtained by the cylinder pressure data. The cylinder
190 pressure data is used to plot the curves of cylinder pressure and the evolution of heat release
191 rate (HRR), for the different total equivalence ratios with respect to crank angle. The in-
192 cylinder pressure, ignition delay and net HRR for the conventional diesel and dual fuel (DF)
193 operating modes at a constant engine speed of 1500 rpm are shown below for 80% of load. A
194 comparison was done between the results of the conventional and DF engine mode.

195

196 *4.1.1 Cylinder pressure*

197 Fig.2 shows the evolution of the cylinder pressure versus crank angle at various total
198 equivalence ratios. As shown in fig.2, the highest peak of cylinder pressure, of 86 bars, is
199 noticed for the conventional diesel mode, which occurs 5.4 after TDC. However, as the total
200 equivalence ratio increases, the evolution, as well as the peak, of cylinder pressure of dual
201 fuel operation decreases. A minimum peak of cylinder pressure of 63 bars is observed at
202 $\phi=0.7$; it represents a reduction of about 26% in comparison to conventional mode. In
203 addition, it can be shown that the evolution of cylinder pressure also decreases even during
204 the compression stage. Indeed, the reduction of the intake air flow, reduces the volumetric
205 efficiency which reduces the in cylinder pressure.

206

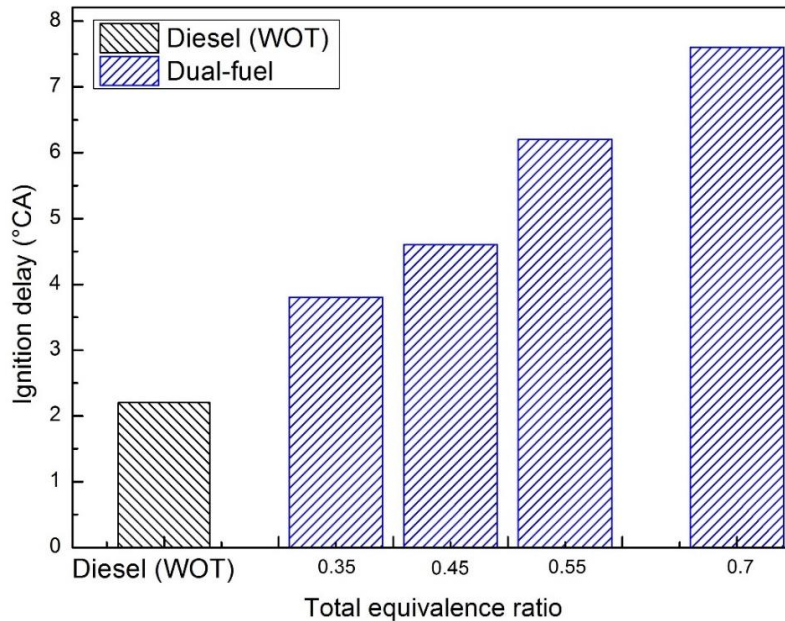


207
 208 Fig. 2. Cylinder pressure variation with crank angle for both modes, for different equivalence
 209 ratios (ϕ).

210 4.1.2 Ignition delay

211 The ignition delay (ID) can be defined as the crank angle (CA) interval between two points.
 212 The first point is the start of injection, and the second is the beginning of combustion. The
 213 combustion of biogas in DF mode undertakes a much complex process compared to that of
 214 diesel. Fig.3 shows the plot of the ID evolution versus ϕ for DF mode (PES 60%) compared to
 215 conventional mode. It can be seen that the ID of DF is greater than the conventional mode for
 216 all studied cases. In addition, the ID increases as ϕ increases. The value of the ignition delay
 217 varies between 3.8 and 7.6 for ϕ ranging from 0.35 to 0.7. The reasons of the prolongation of
 218 ID are the reduction in the oxygen concentration, as the ϕ increases, and the diminution of the
 219 charge temperature [43] due to lower pressure as discussed previously (fig.2). On the other
 220 hand, this extension can also be caused by the specific heat capacity of biogas which is high.
 221 Furthermore, as reported by Wei Lijiang et al [44], the addition of biogas into the cylinder can
 222 delay the auto-ignition of pilot fuel because of the coupling between free radicals. In fact, the
 223 gaseous mixture (air/biogas) goes through a pre-ignition reaction over the compression phase.

224 This chemical reaction affects the ignition of the pilot fuel, due to the formation of energetic
225 radicals. Hence, the prolongation of ID can also be referred to chemical factors.
226



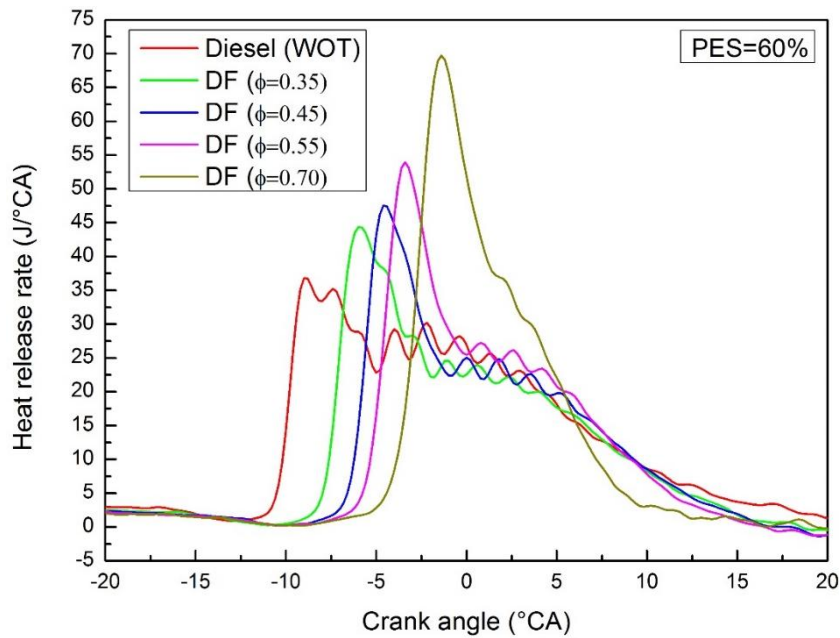
227
228 Fig.3. Ignition delay variation with ϕ .

229

230 4.1.3. Heat release

231 Fig.4 shows the evolution of the heat release rate (HRR) versus the crank angle (CA) for
232 conventional and dual-fuel (DF) modes at different total equivalence ratios. As shown in
233 fig.3, the combustion process, is divided into three phases namely: ignition delay, premixed
234 phase (rapid combustion), and the diffusion combustion. At the beginning, one can visualize a
235 negative heat release over the first phase which is explained by the vaporization of diesel fuel
236 accumulated. As the combustion starts, the HRR takes positive value and increases until the
237 peak of the second phase. This is followed by the diffusion combustion which is controlled by
238 the (fuel- air) gaseous mixture. From fig.4, the peak of HRR increases with the increasing of
239 ϕ . Furthermore, the DF combustion start later with a few degrees of CA compared to the
240 conventional mode due to the extension of ID as explained previously. The maximum and the
241 minimum values of premixed peak of HRR for DF mode are 70 J/deg CA and 43 J/deg CA

242 corresponding to ϕ of 0.7 and 0.35, respectively. In addition, the minimum of HRR of 36.7
243 J/deg CA is recorded in the case of conventional mode, which takes place at 8.8 °CA before
244 TDC. Furthermore, at 0.35, 0.45, 0.55 ϕ , the diffusion combustion of DF mode is similar to
245 that in conventional mode. This can be interpreted by the augmentation of ϕ of the gaseous
246 mixture, owing to the rise of combustion temperature. Also, as ϕ increases the combustion
247 process enhances and the second phase is reinforced as result of the good mixture
248 concentration (fig.5), which gives a highest flame speed of gaseous mixture (biogas/air).
249



250
251

Fig.4. Heat release rate (HRR) with crank angle for different ϕ .

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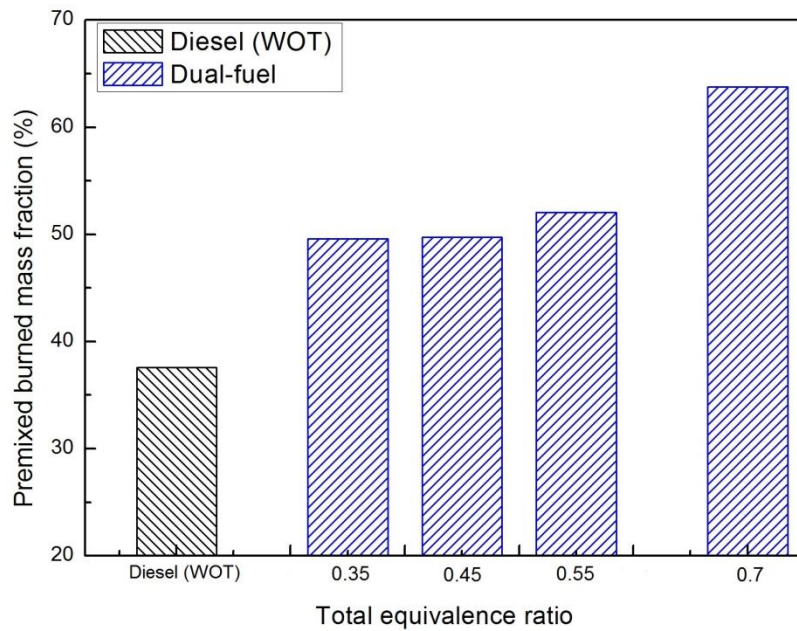


Fig.5. Evolution of the premixed burned mass fraction versus ϕ .

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254

255 4.1.4. Combustion duration

256 Fig.6 presents the evolution of the combustion duration with respect to total equivalence
 257 ratio. The combustion duration is presented in terms of crank angle. It is defined as the
 258 duration between the instance where the HRR takes positive value and the instance at which
 259 90% of the net heat is released [43]. The combustion duration decreases as ϕ rises. It varies
 260 from 20.5 to 12.2 CA when ϕ varies from 0.35 to 0.7 respectively. As known, at high ϕ the
 261 burning velocity of biogas is higher than that at low ϕ [43], which consequently reducing the
 262 combustion duration, the same observation was achieved by Debabrata Barik et al [45].

263

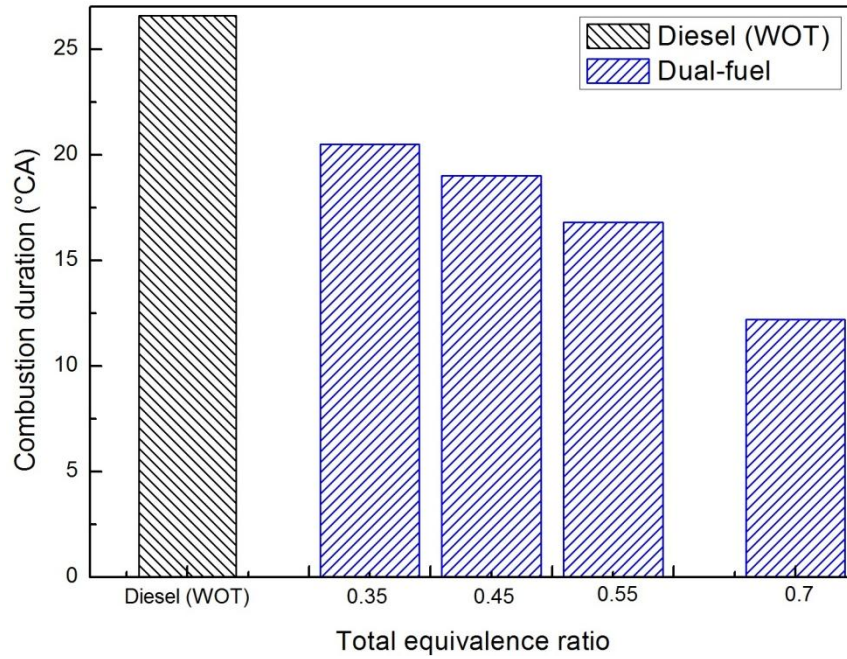


Fig.6. Variation of combustion duration versus ϕ .

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266

267 4.2 Engine performance

268 The evolution of the brake thermal efficiency (BTE) for DF operation and conventional diesel
269 engine with ϕ are shown in fig.7. It can be calculated by the following expression:

$$270 \quad \eta_{eff} = P_{eff} / (\dot{m}_B LHV_B + \dot{m}_D LHV_D) \quad (7)$$

271 Where P_{eff} (kW) is the brake power. The BTE in dual fuel (DF) mode increases with
272 increasing equivalence ratio (fig.7). Else, the BTE for the DF mode is higher than that of
273 conventional mode for all the cases studied. For instance, the BTE of the engine operating in
274 DF at $\phi=0.7$ is better than that of diesel mode by about 19%. Indeed, as known, at high load,
275 the BTE is improved when more biogas is provided to the cylinder. Regarding DF mode, the
276 BTE increases with about 13% as ϕ increases. In fact, at high ϕ gaseous mixture (biogas/air)
277 became more homogeneous. This is beneficial factor to have a complete combustion and
278 limits misfires cases, which reduce THC emissions as it is shown in the next section and
279 improve the BTE.

280

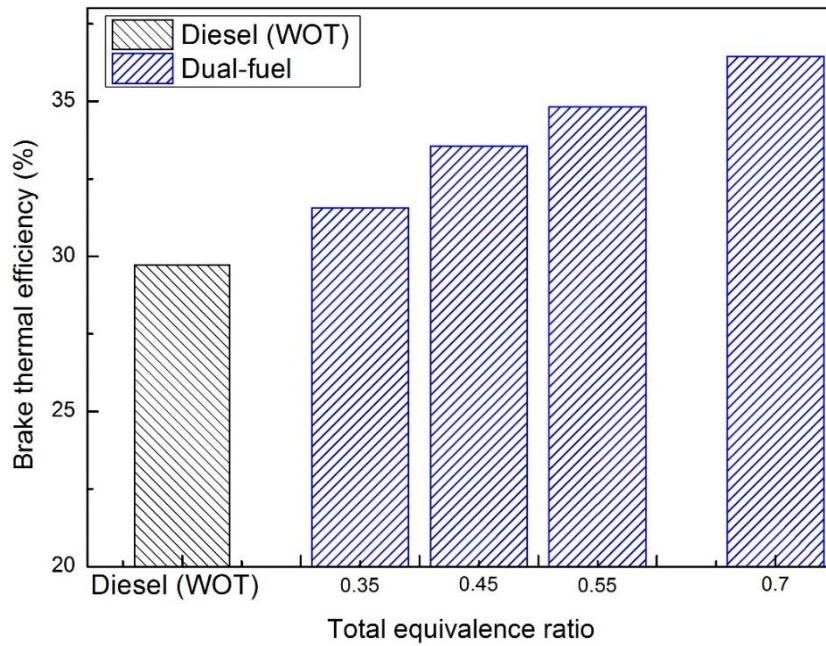


Fig.7. Evolution of the BTE with ϕ .

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283

284 The evolution of the brake specific consumption (BSFC) for both modes is depicted in fig.8.

285 It can be observed that the BSFC in DF mode takes much important values than that recorded

286 for conventional mode. This result is explained by the lower energy content of biogas fuel. On

287 the other hand, one can notice a reduction of BSFC as the ϕ increases from 0.35 to 0.7 for DF

288 mode. This can be assigned to the amelioration of combustion efficiency, leading to better

289 BTE and consequently reducing BSFC.

290

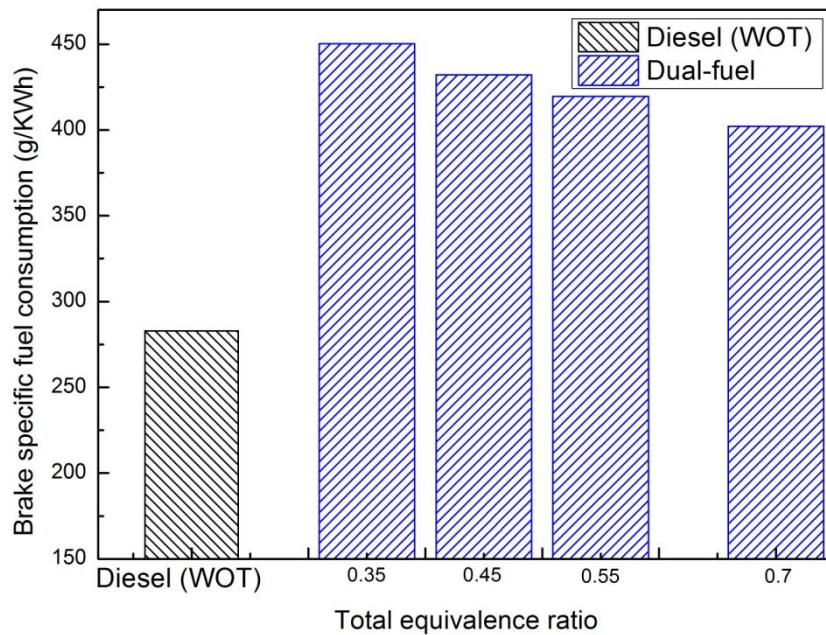


Fig.8. Evolution of the BSFC with ϕ .

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294 4.3 Exhaust emissions

295 In this part, the evolution of pollutant emissions was analyzed for different ϕ at 60% PES. In
296 DF mode, the concentration of soot in the exhaust gas is very low at high load; a similar
297 observation was reported by Lyes Tarabet et al [30]. Therefore, the evolution of soot is not
298 presented in this study.

299 The variation of total unburned hydrocarbon (THC) related to the variation of ϕ is plotted in
300 fig.9. As shown in fig.9, the concentration of THC emissions under DF mode is much
301 important than that in conventional mode. On the other hand, it can be observed that THC
302 emissions in DF mode decrease significantly with the augmentation of ϕ . Indeed, changing ϕ
303 from 0.35 to 0.7 results in a THC emissions decrease of about 77%. This reduction can be
304 particularly explained by the decrease of dilution of air/biogas mixture, which provides a
305 good condition of fast flame propagation of biogas-air mixture. On the other hand, as stated
306 previously the rich mixture leads to a better combustion process due to higher temperatures in
307 the combustion chamber and limits misfires cases.

308

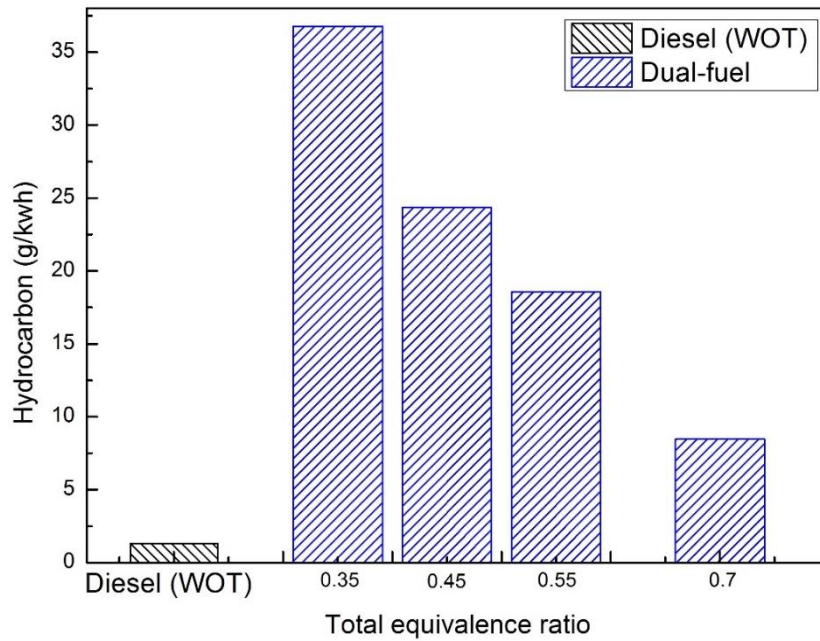


Fig.9. Variation of THC emissions versus ϕ .

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312 Fig.10 indicates the evolution of NO_x emissions versus ϕ . From this figure, a decrease of 24%
 313 of the NO_x emissions was observed for DF mode with the rising of ϕ . The conditions for
 314 formation of NO_x emissions according to zeldovich mechanism can be summarized in three
 315 terms: high peak flame temperature, O₂-rich and the time of burned gas during the highest
 316 temperature. However, NO_x emissions are essentially constituted by two gases, oxygen and
 317 nitrogen. The absence of one of these components has a negative effect for the NO_x
 318 formation. Indeed, the introduction of biogas in the cylinder and the lowering of the intake air
 319 reduce the concentration of oxygen present in the cylinder, and consequently affect the NO_x
 320 emission formation. On the other hand, a comparison with conventional mode shows that a
 321 reduction of NO_x emissions of about 42% was achieved when $\phi=0.7$.

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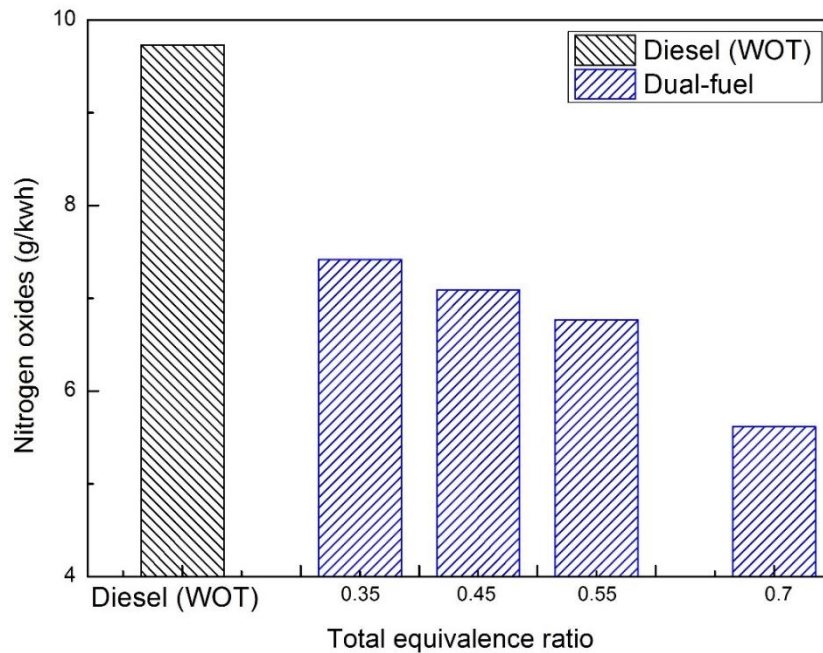


Fig.10. Variation of NOx emissions with different ϕ .

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324

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326 Fig.11 shows the evolution of the carbon monoxide emissions (CO) with ϕ . It may be noticed
 327 that as the ϕ increases the concentration of the CO emissions decreases about 58%. This is
 328 again the result of the better flame propagation leading to a more complete combustion due to
 329 the rich mixture. However, it can be also observed that, although the concentration of CO
 330 emission decreases with increasing ϕ , the concentration of these pollutants is still twice that of
 331 the conventional mode.

332

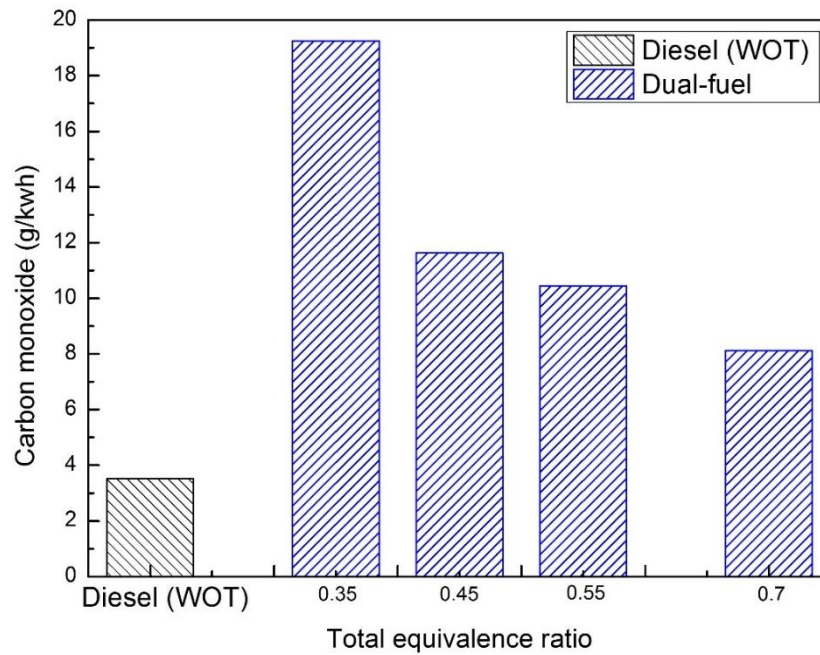


Fig.11. Evolution of CO emissions with various ϕ .

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336 Fig.12 shows the evolution of the carbon dioxide with increasing ϕ . It can be observed that at
 337 high ϕ , the difference between the concentrations of CO_2 for both modes is about 7%. The
 338 specific emission of CO_2 (g/kWh) decreases about 14% with the augmentation of ϕ in the case
 339 of DF mode. Indeed, as discussed previously, the increase of ϕ induces an improvement of
 340 combustion process resulting in higher CO_2 concentrations (%Vol.) in the exhaust gas.
 341 However, in the same time, the power output increases since the BET is becoming higher and
 342 thus resulting in lower specific emission (g/kWh) when increasing ϕ .

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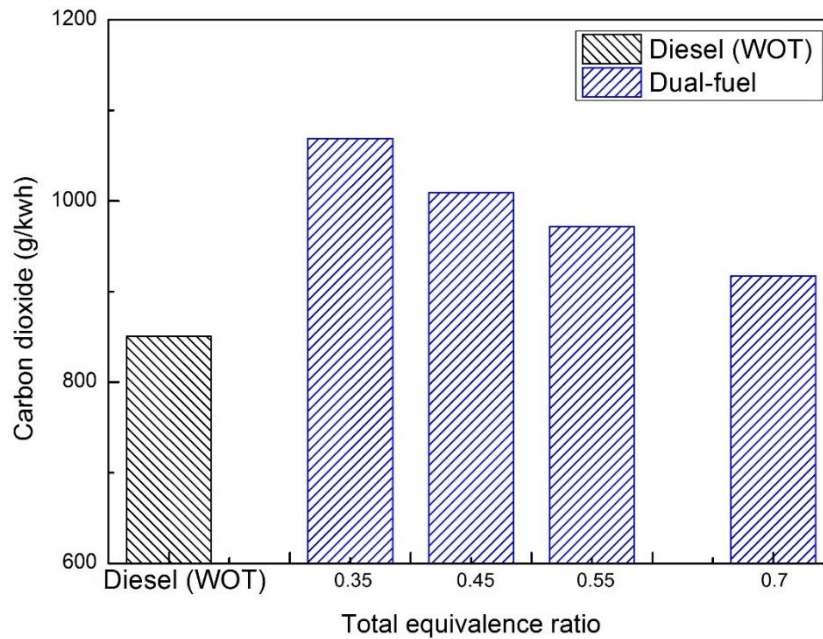


Fig.12. Variation in CO₂ emissions with different ϕ .

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5. Conclusion

349 In the present work, an experimental investigation was conducted to study the effects of
350 equivalence ratio (ϕ) variations on the combustion characteristics, performance and exhaust
351 emissions of a dual fuel (DF) diesel engine operating on biogas fuel at high load (80% of full
352 load). The experiments were performed with a fixed energy input to the engine while the
353 percentage energy substitution (PES) was kept constant (60%). ϕ was varied from 0.35 to 0.7.
354 The results led to the following conclusions:

- 355 - The peak of in-cylinder pressure under DF mode was lower than that observed for
356 conventional mode for all the cases studied. Also, the peak of in-cylinder pressure decreased
357 as ϕ increases for DF operation.
- 358 - Under DF mode, as ϕ increases, the ignition delay became longer and the heat release rate
359 gave higher peak for the premixed phase.
- 360 - Concentrations of THC, CO, CO₂ and NO_x were reduced with about 77%, 58%, 14% and
361 24% respectively when ϕ was varied from 0.35 to 0.7.
- 362 - The BTE was improves by about 13% with increasing ϕ under DF mode. Moreover, at
363 $\phi=0.7$, the BTE is higher than that of conventional mode by about 19%.

364 **6. Acknowledgments**

365 The authors are very thankful to all the technicians in the laboratory of Ecole des Mines de
366 Nantes for their help and for kind collaboration in this study.

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