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Effect of natural gas enrichment with hydrogen on combustion characteristics of a dual fuel diesel engine

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Abstract

Environmental benefits are one of the main motivations encouraging the use of natural gas (NG) as fuel for internal combustion engines. In addition to the better impact on pollution, natural gas is available in many areas. In this context, investigation has been carried on the improvement of diesel engine operating under dual fuel mode. This study mainly focuses on the influence of natural gas enrichment with different hydrogen fractions (10%, 20% and 30%) volume basis on the combustion parameters and performance of a dual fuel engine. All tests were performed at constant engine speed of 1500 rpm and supplied with different natural gas – hydrogen mixtures to vary the engine load, namely 30%, 40%, 50% and 70% of full load. It has been observed that the hydrogen addition increases the peak of in-cylinder pressure, reaching an increase of 5.1% with 10% H_2 blend compared to natural gas dual fuel operation. Maximum of cumulated heat release was obtained with 10% H_2 due to combined effect of charge homogeneity, ignitability and burning velocity improvement. Moreover, hydrogen addition to natural gas resulted in better combustion stability while no major effect on combustion noise was noticed. Concerning engine performance, a maximum brake thermal efficiency is achieved with 10% hydrogen enrichment at 70% engine load. Almost the same

29 brake thermal efficiency was reached with pure diesel (32.53%). Overall, an improvement is
30 achieved with natural gas enrichment with hydrogen compared to pure natural gas.

31

32 **Keywords:** CI engine; dual fuel; natural gas; hydrogen; combustion characteristics.

33

34 1. Introduction

35 The rapid increase in energy demand with the possible depletion of fossil fuels has
36 created an alternative fuel requirement. On the other hand, the stringent norms to reduce
37 harmful emissions promote the utilization of low carbon or carbon-neutral alternative fuels. It
38 is recommended that engine designers study the viability of these fuels [1]. Natural gas (NG)
39 is one of the various alternative fuels that offers considerable potential and interest for the
40 automotive sector [2]. The dual fuel (DF) mode is a promising technique that allows utilizing
41 NG in internal combustion engines [3, 4].

42 Natural gas is introduced into the intake manifold in most dual fuel applications to
43 provide uniform mixing with air. Next, the NG-air mixture is added to the combustion
44 chamber. The NG-air mixture is ignited by a liquid diesel pilot fuel using the original diesel
45 fuel injection system [4]. In general, a constant amount of the liquid fuel is injected near the
46 end of the compression stroke to act as an ignition source of the gaseous fuel-air mixture.
47 Major part of the engine power output is provided and controlled by the gaseous fuel. The
48 high octane number of NG allows using higher compression ratios, which can significantly
49 increase thermal efficiency. Several studies [5-7] showed that potential benefits of dual fuel
50 mode using NG as primary fuel are both environmental and economic. However, a number of
51 critical issues remain under dual fuel operation and previous studies affirmed that emissions
52 of unburned hydrocarbons (HC) and carbon monoxide (CO) are higher than those under
53 conventional diesel at low loads condition [6-9]. In addition, engine performance decreases as
54 well.

55 To overcome those obstacles, several solutions were proposed for improving the dual
56 fuel operation. This improvement could be achieved by adjusting some critical parameters of
57 the engine related to the combustion process such as pilot quantity, injection timing [10, 11],
58 compression ratio, air inlet preheating [12, 13] and hot exhaust gas re-circulation [14]. The
59 effect of pilot quantity and intake charge temperature at part load in dual fuel engine were
60 analyzed by Srinivasan et al. [12]. Authors reported a better combustion stability and lower
61 emissions with an increase in intake charge temperature at low engine load. It was also
62 reported that advanced pilot injection timing, accompanied with a larger proportion of pilot
63 fuel, allows obtaining better engine efficiency associated with a reduction of some emissions,
64 except NO_x [15].

65 Among the various methods, the combination of NG with a gaseous fuel which
66 possesses fast burning velocity and wide flammability limit can allow improvement in
67 combustion characteristics, especially lean-burn limit and combustion velocity. Regarding
68 spark ignition (SI) engines, there is a considerable available literature related to the use NG
69 enriched with hydrogen as a fuel [16-18]. Indeed, the use of NG-hydrogen mixture as fuel for
70 SI engines proved to be a promising way to eliminate some of the disadvantages of SI natural
71 gas engines, including the lower lean burn limit, which reduces the engine's efficiency.
72 However, the benefits from burning gaseous fuels in spark ignition engines is limited due to
73 their lower compression ratios resulting in lower thermal efficiency.

74 Recently, many studies have examined the positive features of NG enriched with
75 hydrogen in compression ignition engines and the outcomes on performance and pollutant
76 emissions are very promising. However, very few studies have investigated the combustion
77 behavior of a HNG dual fuel engine.

78 Lounici et al. [19] evaluated the performance and emission characteristics of a CI
79 engine. They reported that the addition of hydrogen to NG is a promising method for

80 improving the dual fuel mode, especially at low engine loads. The emission results showed
81 that THC and CO emissions are in general reduced by H₂ enrichment as a result of the
82 improvement of gaseous fuel utilization. They also found that the total brake specific fuel
83 consumption (BSFC) is considerably reduced. An important benefit in terms of BTE, reaching
84 to increase a 12% with the 10%H₂ blend compared to the pure NG case, is achieved.

85 Zhou et al. [20] have analyzed the effect of using hydrogen, CH₄ and their mixture as
86 primary fuel on combustion characteristics and pollutant emissions of a dual fuel diesel
87 engine. Five steady engine loads were performed at 1800 rev/min. The addition of higher
88 hydrogen fraction in methane has improved the engine performance at 30% load and lowered
89 the CO and HC emission trends. Tri-fuel operation with 30% hydrogen addition in methane
90 was observed to be the best fuel in reducing particulate and NO_x emissions at 70 and 90%
91 loads. Talibi et al. [21] reported the effect of methane-hydrogen mixtures on a direct injection
92 CI engine, with diesel as pilot fuel. The engine was supplied with different methane-hydrogen
93 mixtures to vary the engine load. It is reported that increasing the proportion of H₂ in the CH₄-
94 H₂ mixture shorter the ignition delay period and increased peak heat release rates, but only at
95 high engine loads. Also the fuel properties are not having the significant effect on the ignition
96 and the reactivity of the fuels is affected by the mixture concentrations. The emissions levels
97 of CO₂ and particulate are reduced for all engine loads. While, the emissions of CO and
98 unburned THC increased for methane-hydrogen mixtures at all engine loads when compared
99 to diesel mode.

100 Some of the research works has been reported on the effect of NG enrichment with various
101 H₂ blends for improving biodiesel use in DI diesel engine under dual fuel mode. Tarabet et al.
102 [22] have analyzed the combustion behavior of hydrogen under NG dual fuel mode with
103 eucalyptus biodiesel as pilot fuel. For all examined engine loads, the pilot biodiesel quantity
104 was kept constant to supply around 10% of the engine nominal power while the engine load is

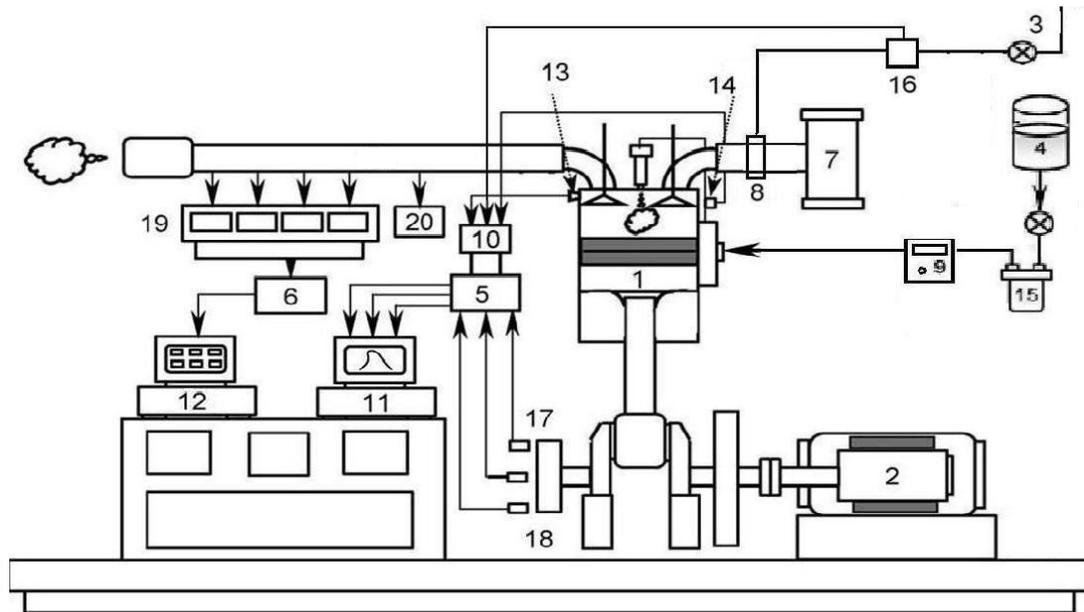
105 further increased using only the gaseous fuel (NG-H₂). They concluded that NG enrichment
106 with H₂ leads to a significant increase in HRR and pressure peak as a consequence of the
107 short ignition delay and the rapid combustion during the premixed combustion phase, as
108 compared to pure NG. At high engine loads, the highest pressure peak corresponding to
109 EB/75NG-25H₂ is 10.2 bars higher and 2.2 °CA earlier than that of conventional diesel mode.
110 In another study Kalsi et al.[23], carried out tests on pilot biodiesel dual fuel RCCI
111 (Reactivity Controlled Compression Ignition) engine. Results indicate that hydrogen addition
112 led to higher heat release rate. While, a lower ignition delay (pilot fuel) and combustion
113 duration was reported as compared to CNG engine. Moreover, a significant improvement was
114 noted in the brake thermal efficiency of the engine with blends (biodiesel-HCNG) containing
115 up to 64% of HCNG energy share.

116 From the literature, it has been identified that NG enrichment with hydrogen in CI engine
117 has very promising on performance and pollutant emissions. However, few researches focus
118 on the effect of hydrogen addition on combustion behavior of a dual fuel diesel engine.
119 Hence, this work aims to gain a better comprehension and especially examining the effect of
120 hydrogen addition. This can contribute improving this very promising engine operating
121 technique. In this study, various test blends were examined with varying H₂ percentages and
122 different engine loads (30, 40, 50 and 70% load) is therefore investigated. Results are
123 compared with conventional diesel and pure natural gas dual fuel operation.

124 **2. Experimental setup and analysis model**

125 *2.1. Engine test*

126 Engine tests were carried out using a single cylinder Lister Petter (TS1) diesel air-
127 cooled engine. The output power of this engine is 4.5 kW at 1500 rpm. Its basic data and the
128 experimental methodology are provided in previous author's works [7, 19]. The experimental
129 setup scheme is presented in Fig.1.



130

- | | |
|--------------------------|----------------------------------|
| 1. Test engine | 11. Fast data acquisition system |
| 2. Dynamometer | 12. Slow data acquisition system |
| 3. Gaseous fuel supply | 13. Cylinder pressure sensor |
| 4. Diesel fuel tank | 14. Injection pressure sensor |
| 5. A/D card for pressure | 15. Liquid fuel filter |
| 6. A/D card for Analyzer | 16. Gaseous fuel flow meter |
| 7. Air tank | 17. TDC encoder |
| 8. Air-gas Mixer | 18. Speed sensor |
| 9. Diesel flow meter | 19. Exhaust gas analyzer |
| 10. Charge Amplifier | 20. Smoke meter |

131

Fig. 1. Experimental setup scheme.

132 2.2. Model for heat release evaluation

133 The heat release calculation is an attempt to obtain information on the combustion
 134 process in the engine. These include calculations that determine the heat required to be added
 135 to the cylinder content to generate the observed in-cylinder pressure trends [24]. A single-
 136 zone heat release rate model, based on the first law of thermodynamic, is used.

137 The space trapped in the cylinder is considered as a control volume when the valves
 138 are closed (a constant mass is considered). The first law of thermodynamics can be applied as
 139 following:

$$140 \quad \delta Q_{net} = \delta Q_{gross} - \delta Q_w = \delta U + \delta W \quad (1)$$

141 Where δQ_{gross} , δQ_w are the heat released by combustion and the heat lost through combustion
 142 chamber walls, respectively. δW is the mechanical work done by the system. δQ_{net} is the net
 143 heat release representing the difference between the gross heat release and the heat loss to the
 144 walls.

145 Cylinder contents is considered as a mixture of ideal gases ($C_p/C_v = \gamma$ and $C_p - C_v = R$).
 146 Equation (1) can be written as:

$$147 \quad \frac{dQ_{net}}{d\theta} = \left[\frac{\gamma}{\gamma - 1} \right] P \frac{dV}{d\theta} + \left[\frac{1}{\gamma - 1} \right] V \frac{dP}{d\theta} \quad (2)$$

148 Where P (Pa) is the in-cylinder pressure at a given crank angle; V (m³) is the volume of
 149 combustion chamber. γ is the specific heat ratio being considered independent of cylinder
 150 temperature as a result of the negligible effect of γ variation within a typical range on the
 151 phasing of $[dQ_{net}/d\theta]$. A constant value of ($\gamma=1.35$) is commonly adopted [7].

152 3. Results and discussions

153 3.1. Combustion characteristics

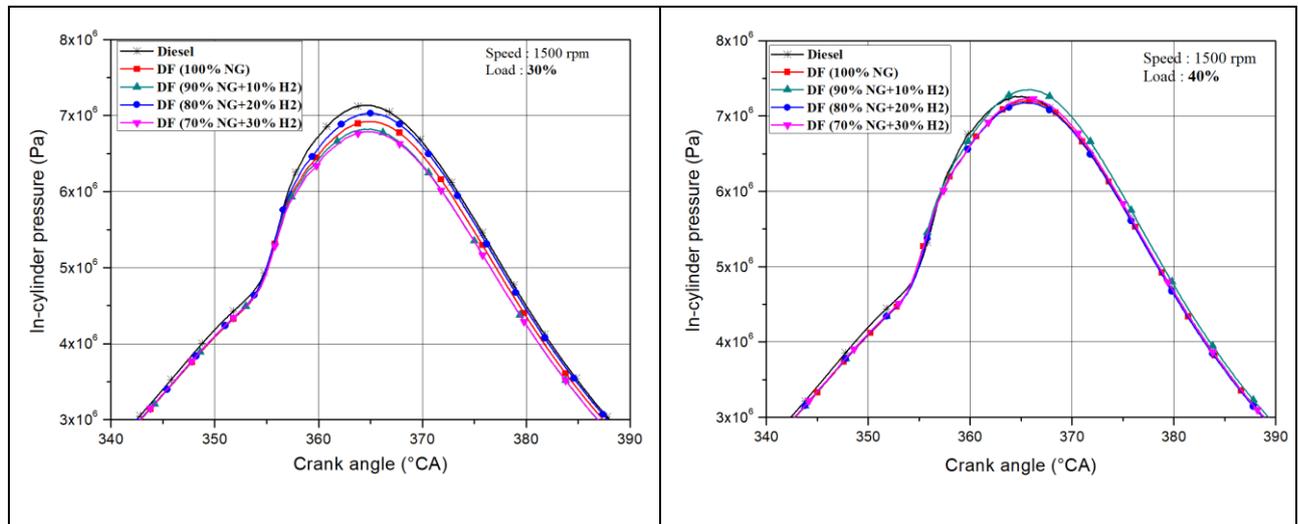
154 In this section, combustion characteristics such as heat release rate, combustion
 155 duration, combustion noise, and combustion stability, are examined for conventional diesel,
 156 NG and HNG dual fuel modes at different engine load conditions.

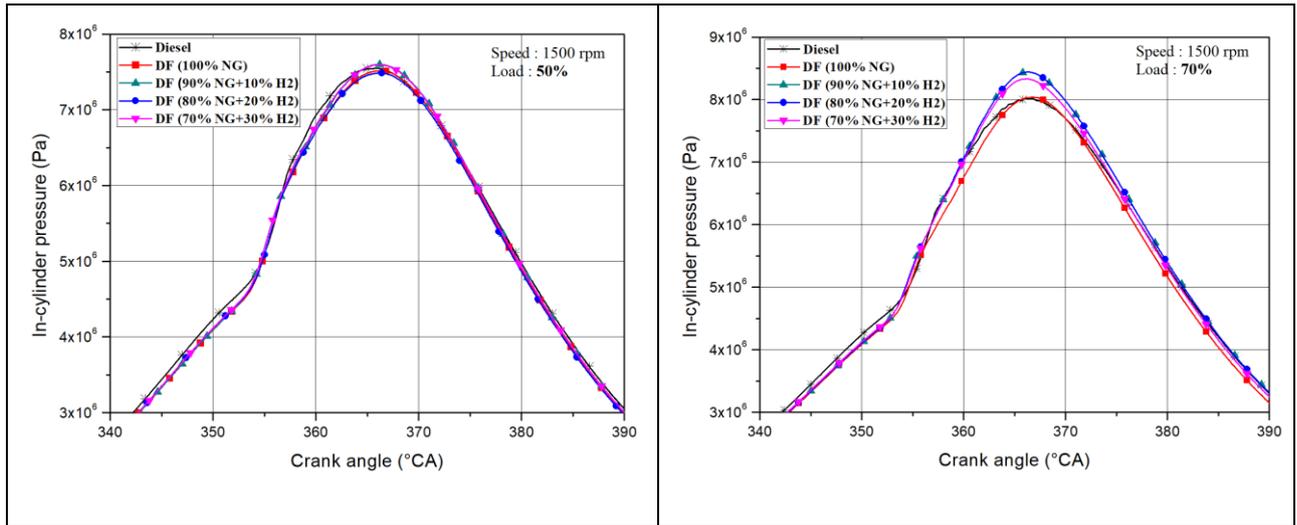
157 3.1.1. In-cylinder Pressure

158 The variation of in-cylinder pressure for conventional diesel, NG and HNG dual fuel
 159 modes is presented in Fig.2. In-cylinder pressure curves are presented for different engine
 160 loads (30, 40, 50 and 70%), which are representative of low, medium and relatively high
 161 engine loads. Obviously, a larger amount of fuel is inducted at high engine loads leading to
 162 higher pressure peak for all examined test blends. Under dual fuel operation, a slightly lower
 163 in-cylinder pressure is observed during the compression stroke, which is probably due to the

164 higher specific heat capacity of the NG-air mixture and therefore a lower in-cylinder
 165 temperature [2]. While, the low combustion rate of natural gas is the main factor causing the
 166 low pressure observed at initial stages of combustion in dual fuel mode. However, the
 167 addition of H₂ to NG results in a higher in-cylinder pressure compared to pure NG dual fuel
 168 mode. When operating at 50% load, under HNG mode, the maximum in-cylinder pressure
 169 gradually increases with the increase in H₂ fraction and exceed that of diesel operation. This is
 170 due to the improved gaseous fuels combustion.

171 At 70% engine load, the dual fuel mode generates a slightly higher in-cylinder
 172 pressure, which is governed by higher in-cylinder gas temperature, revealing an improvement
 173 in gaseous fuel combustion. At this condition, the peak in-cylinder pressure was measured as
 174 80.3 bar for pure NG dual fuel operation and 80.1 bar for conventional diesel operation. With
 175 hydrogen addition (10%, 20%, and 30% of hydrogen), an increase of 5.1%, 4.5%, and 3.7%,
 176 is obtained respectively, compared to NG dual fuel mode. Thus, the maximum peak pressure
 177 is observed for 10% hydrogen blending case.

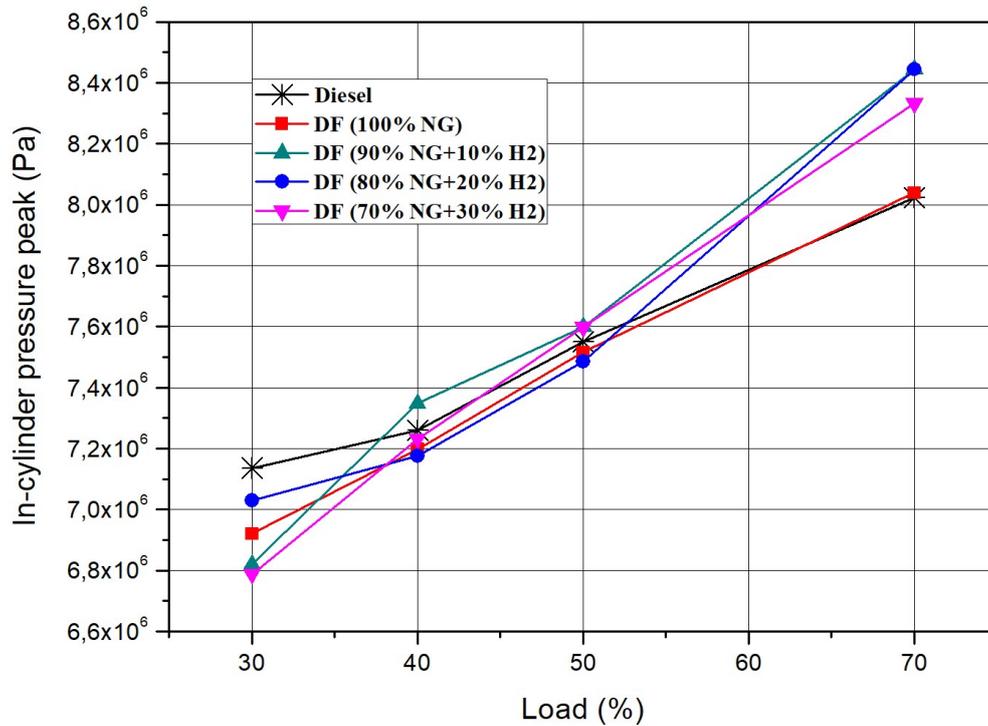




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179

Fig. 2. In-cylinder pressure versus crank angle at different engine loads.



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Fig.3. In-cylinder pressure peak variation with engine loads.

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183 3.1.2. Heat release rate

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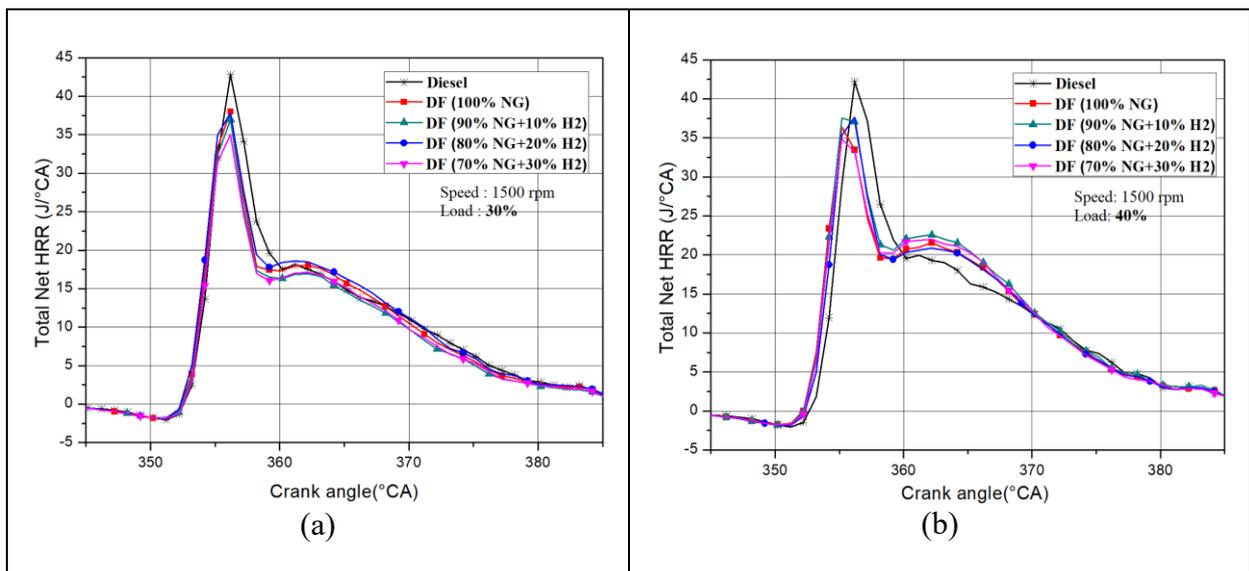
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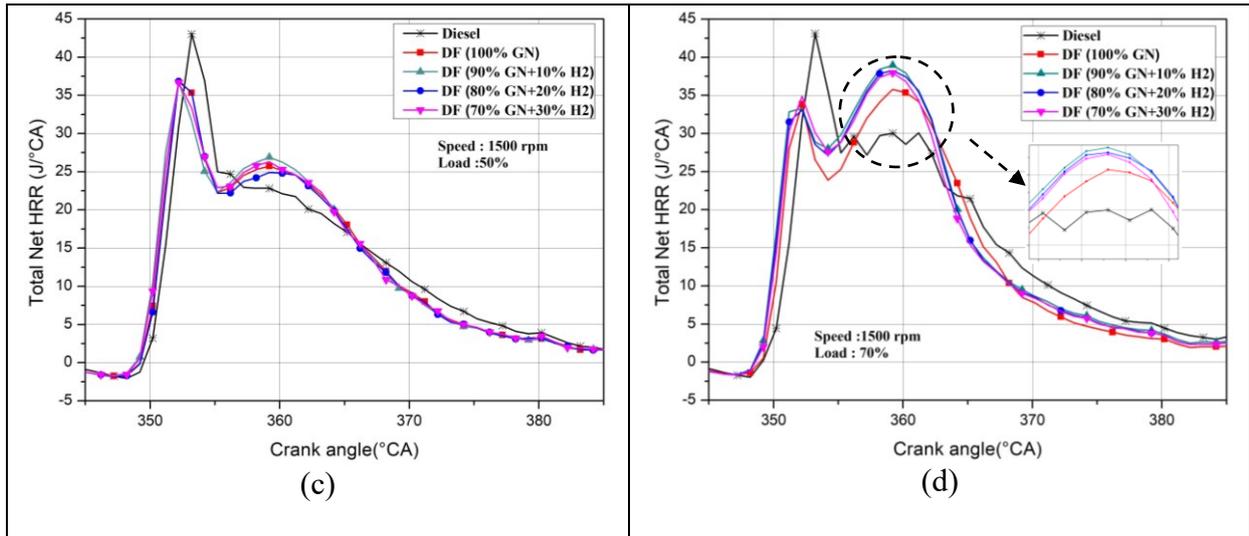
The heat release rate (HRR) calculation is an attempt to get information concerning the combustion process in the engine. HRR allows knowing how heat is released by

186 combustion. The variation in heat release rate for conventional diesel, NG and HNG dual fuel
 187 modes is presented in Fig.4.

188 As expected, a slightly higher HRR peak was obtained under dual fuel mode during
 189 the second phase of combustion than that of conventional diesel, announcing a late
 190 combustion of the gaseous fuel. Regarding the effect of hydrogen addition to NG, it can be
 191 noticed that, at medium to high loads, the maximum HRR increased which is mainly due to
 192 the improvement of the gaseous fuel combustion process.

193 The flame speed enhancement with hydrogen addition, leads to a domination of the
 194 premixed combustion phase (second peak) [23] and therefore a higher heat release rate during
 195 this phase (Fig.4.d). Indeed, hydrogen has a high diffusivity ($0.63 \text{ cm}^2/\text{s}$) in the air, which
 196 leads to formation of more homogeneous gaseous charge (air+HNG) [25]. As a result,
 197 increasing charge homogeneity with HNG allows a better premixed charge combustion
 198 characteristics, which corresponds to a significant HRR. The maximum HRR was obtained
 199 for 10% of H_2 addition. This can be related to the combined effect of charge homogeneity,
 200 ignitability and burning velocity improvement. In addition, volumetric heating value (VHV)
 201 decreases as hydrogen concentration increases.





202

203

Fig.4. Net heat release rate versus crank angle at different engine loads.

204

205 3.1.3. Cumulative heat release

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The cumulative heat release (CHR) represents the amount of heat that is released during the combustion process. Results showed that the effect of hydrogen enrichment at low loads (Fig 5.a) is not significant since the gaseous fuel energy share is not important. However, for higher loads, improvement in gaseous fuel combustion leads to higher CHR with hydrogen blending (Fig 5.b). Moreover, the previous results obtained with 10% H₂ addition for HRR are confirmed with those concerning CHR.

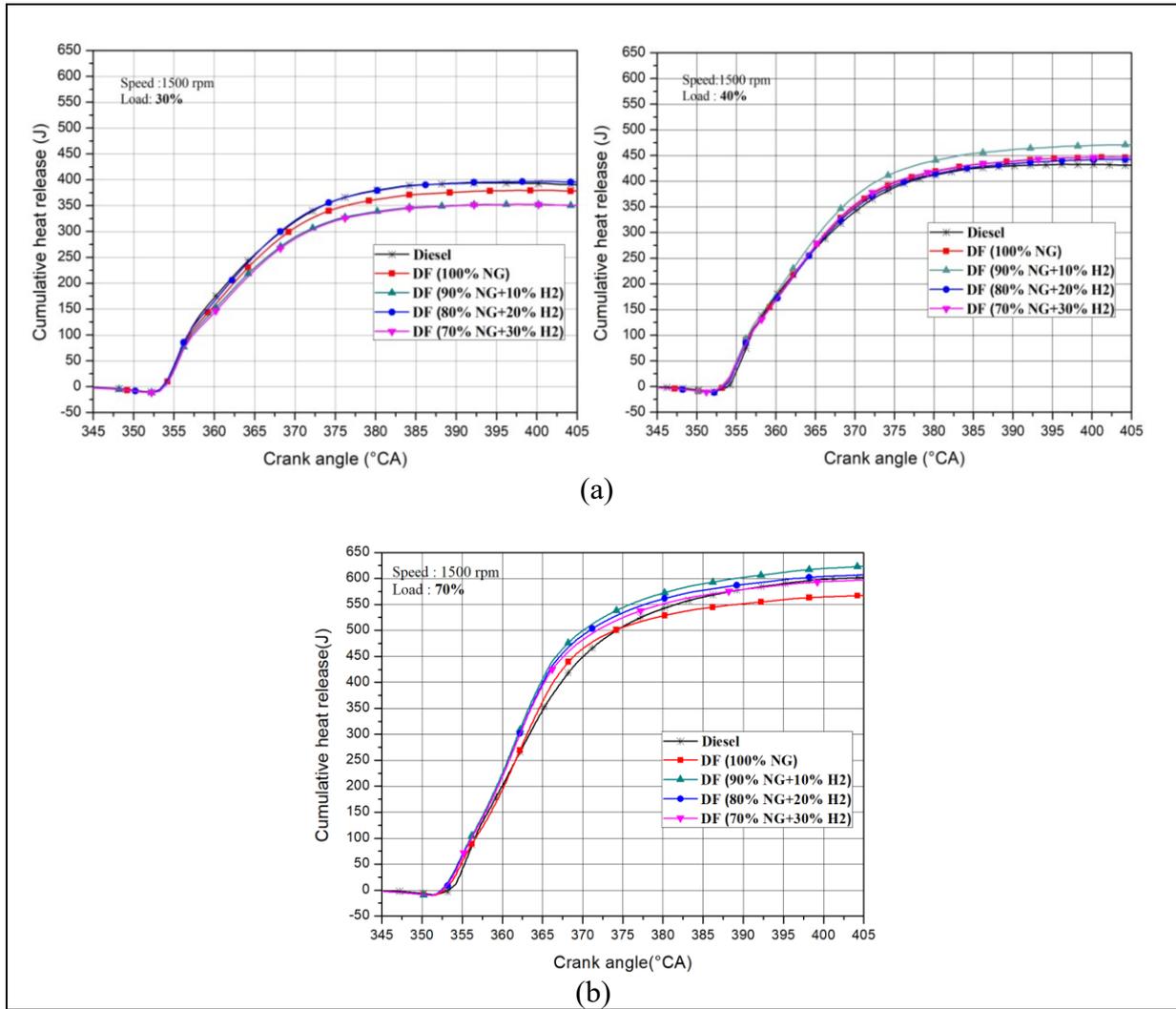


Fig.5. Cumulative heat release for 30%, 40% (a) and 70% (b) engine loads.

213

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215 3.1.4. Combustion noise

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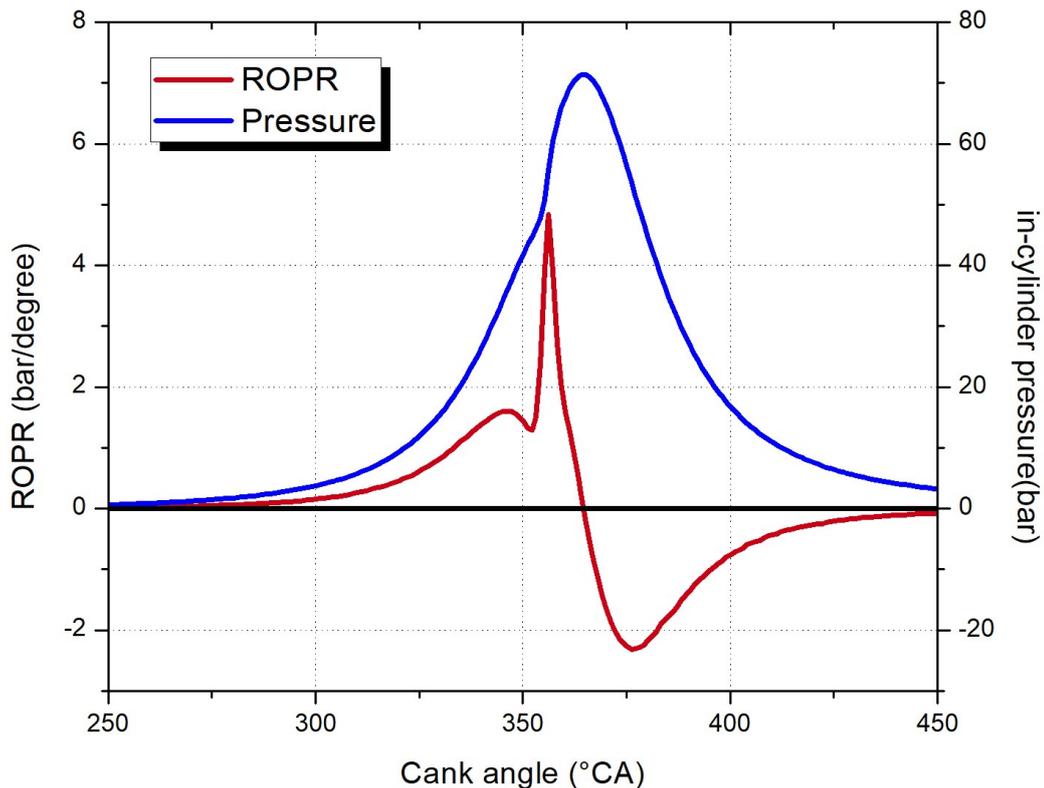
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A typical rate of pressure rise (ROPR) curve against the crank angle is illustrated in Fig. 6 together with corresponding in-cylinder pressure trace. Fig. 6 shows that ROPR increases during the compression and early combustion phases to its maximum value at a certain crank angle, followed by a decrease, while the pressure continues to increase until it reaches its maximum point. The ROPR becomes negative during the expansion stroke.



221

222

Fig.6. Typical ROPR for a given pressure data.

223

224

225

226

The rate of pressure rise indicates implicitly the increase and decrease in combustion noise due to the fact that combustion noise is related to the rise in rate of in-cylinder pressure [26]. Several authors [14, 27] represent the combustion noise by the maximum value of the rate of pressure rise at the considered conditions.

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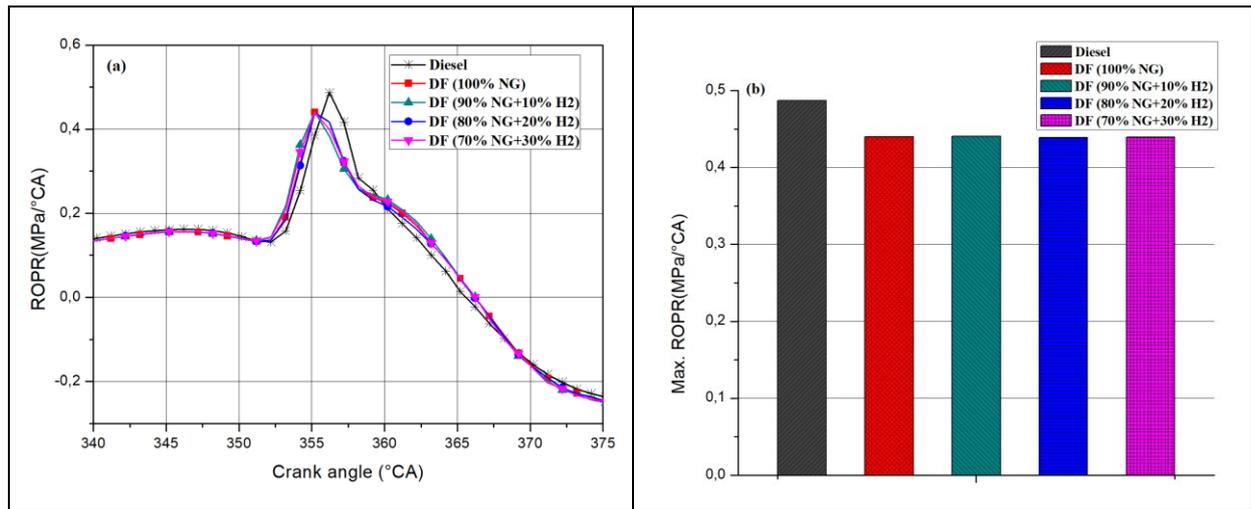
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231

232

233

Figs. 7 shows the rate of pressure rise variations and its maximum for various engine operating modes at 50% load. A lower combustion noise is noticed under dual fuel mode than that of diesel mode at all engine loads. On the other hand, this reduction in combustion noise is assumed to be caused by the reduction in the maximum rate of heat release since conventional diesel mode involves an intense combustion process in the premixing phase (Fig. 4), which is responsible of the rapid pressure rise rates. Therefore, the combustion noise of conventional diesel mode is always higher compared to dual fuel mode.



234

235 Fig. 7. The rate of pressure rise versus crank angle (a) and corresponding combustion noise
 236 (b) at 50% engine load.

237

238 Concerning hydrogen addition, no major effect was observed. Indeed, both NG and HNG
 239 dual fuel operations have almost similar combustion noise levels. The maximum values of
 240 ROPR and their positions in terms of crank angle are given in Table 2.

241

242

Table 2. Maximum ROPR and °CA positions						
Engine load	Unit	Diesel	DF (100%GN)	10HNG	20HNG	30HNG
30%	°CA	357	357	357	357	357
	MPa/°CA	0.483	0.438	0.429	0.438	0.410
40%	°CA	357	356	356	357	356
	MPa/°CA	0.477	0.435	0.445	0.436	0.421
50%	°CA	357	356	356	356	356
	MPa/°CA	0.486	0.440	0.440	0.438	0.439
70%	°CA	357	356	355	356	356
	MPa/°CA	0.488	0.413	0.412	0.410	0.425

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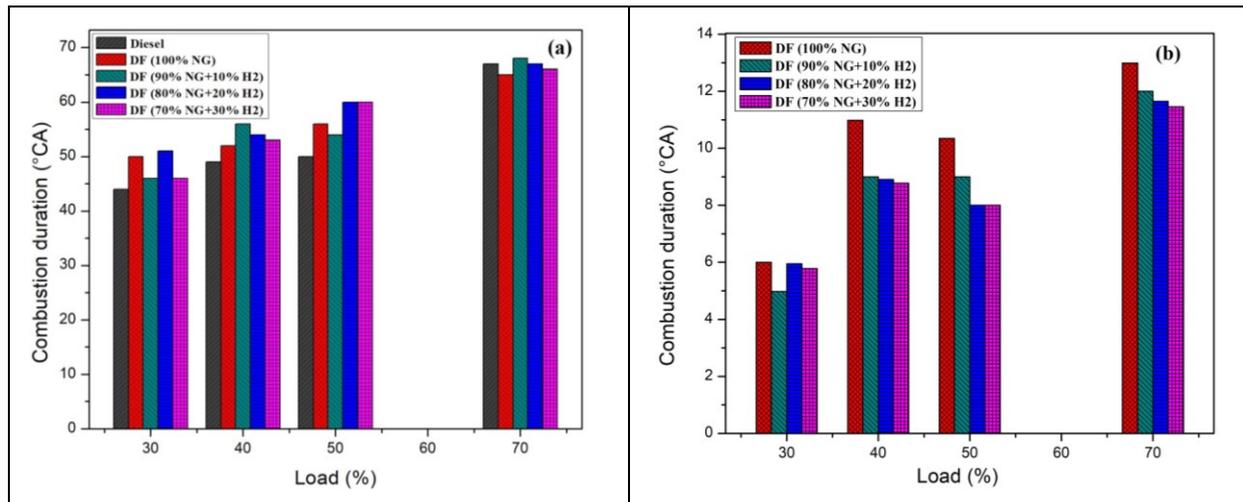
246 *3.1.5. Combustion duration*

247 Combustion duration (CD) is the angle between start of combustion (SOC) and end of
248 combustion (EOC). In literature, several methods for identifying the SOC have been used,
249 based either on the in-cylinder pressure signal analysis or on the heat release analysis.
250 According to [28, 29], the start of combustion was defined as the local minimum of the ROPR
251 curve. Meanwhile, according to [30, 31], the start of combustion is taken as the maximum of
252 ROPR in terms of crank angle.

253 Fig. 8 (a) presents the combustion durations in terms of crank angle for different
254 engine load. The CD was defined as the interval between the local minimum of ROPR curve
255 to the maximum cumulative heat release (CHR). It can be seen that CD increased as engine
256 load increases due to largest quantity of fuel involved. On the other hand, at a given engine
257 load, higher combustion durations are noticed under dual fuel operation than that of
258 conventional diesel, especially at low and medium loads. This difference decreased with
259 higher engine load (70%).

260 Concerning hydrogen addition to NG, the combustion duration of gaseous fuel
261 combustion phase is reduced in all cases as shown in Fig. 8 (b). However, it's not the case for
262 the overall combustion duration (Fig. 8 (a)) and we cannot see a clear effect as it is affected
263 by other combustion phases.

264 According to Gharehghani et al. [32], a shorter combustion duration leads to lower
265 heat loss through cylinder walls. This decrease in the heat loss contributes to the thermal
266 efficiency improvement.



267

268

Fig.8. Overall combustion duration (a) and gaseous fuel combustion duration (b)

269

at different engine loads.

270 3.1.6. Ignition delay

271

Ignition delay (ID) is a significant parameter in determining the combustion process. The

272

ID is presented in terms of °CA and defined as duration between the start of injection (SOI) of

273

pilot fuel and the start of combustion of dual fuel charge [33]. The SOI was determined from

274

fuel pressure diagram, it is taken as the time when the injector needle lifts off its seat, whereas

275

start of combustion was identified from the rate of in-cylinder pressure rise variation.

276

Fig. 9 shows the ignition delay periods for conventional diesel and DF modes at different

277

engine loads. It can be observed that the ID tends to decrease as engine load increases. The

278

value of ID varies between 3.8 and 5.2 for NG dual fuel operation. At low engine loads, a

279

slight difference has been noticed in ignition delays between NG dual fuel and conventional

280

diesel. This might be due to the poor gaseous fuel contribution in NG-air mixture. At high

281

engine load (70%), the ignition delay is greater with NG dual fuel mode compared to

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conventional diesel. This is due to the fact that a portion of air was replaced by gaseous fuel,

283

which reduced the oxygen concentration in the charge. The extension of ignition delay can

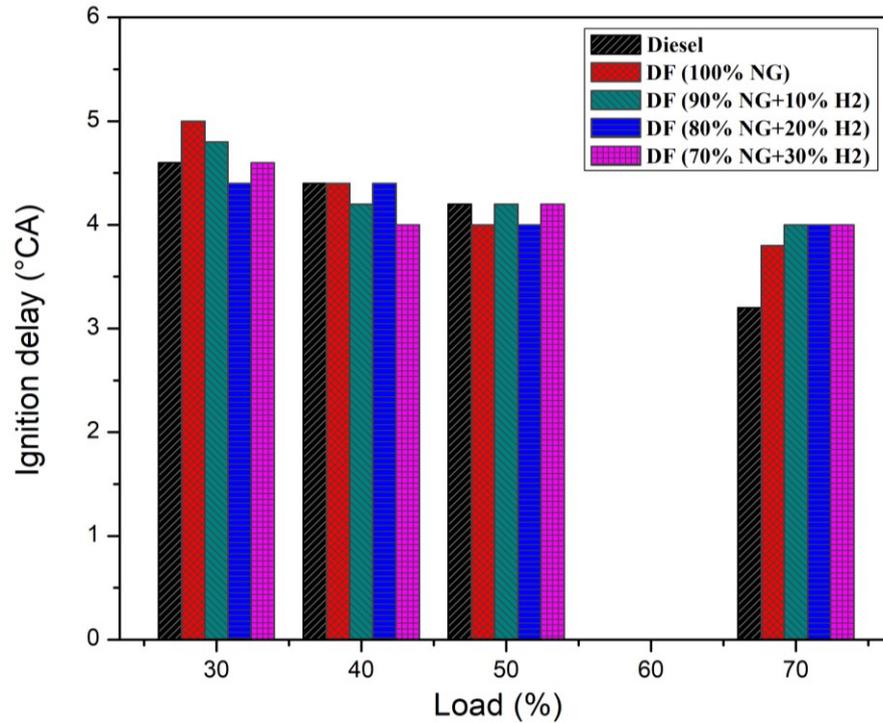
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also be referred to the higher specific heat capacity of gaseous fuel-air mixture than the pure

285

air. However, addition of H₂ to NG did not affect significantly the ignition delay period. In

286 fact, no noteworthy effect in the ignition delay period can be revealed with all HNG test
 287 blends compared to NG dual fuel mode.



288

289

Fig.9. Ignition delay at different engine loads.

290 3.1.7. Cyclic variability

291 The Cyclic variability, defined as a combustion stability criterion, is frequently
 292 represented by the coefficient of variation (COV) of the indicated mean effective pressure
 293 (IMEP) and the maximum in-cylinder pressure. The results of 100 successive engine cycles
 294 were analyzed to determine the cyclic variability at different test conditions. The coefficient
 295 of variation was calculated using the following equations and is usually expressed in percent.

$$296 \quad COV(x) = \frac{\sigma}{\bar{x}} \times 100 \quad (3)$$

297 Where (σ) is the standard deviation and \bar{x} the mean value given by:

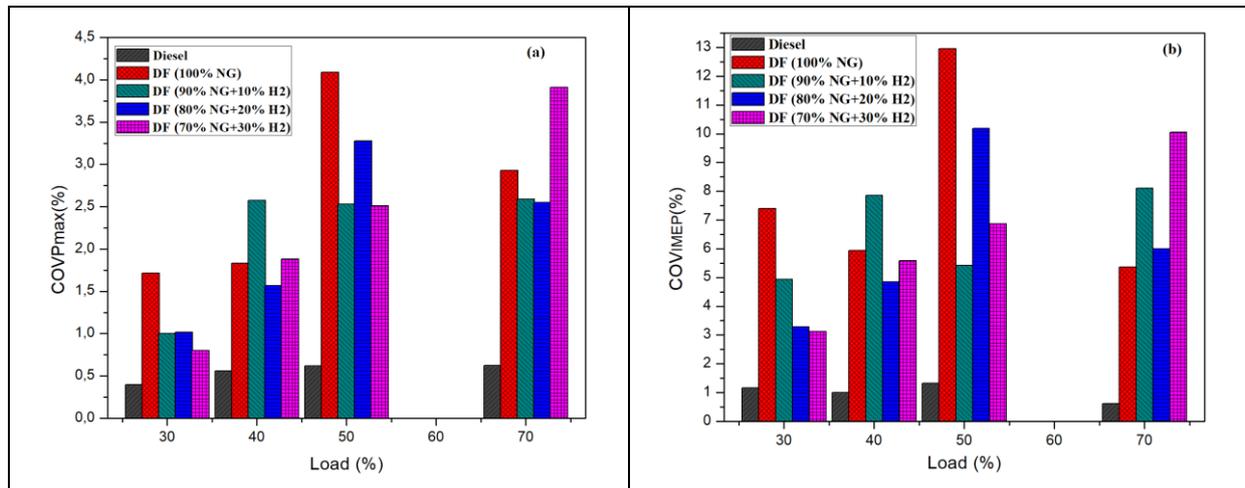
$$298 \quad \sigma = \sqrt{\frac{\sum_{i=1}^n (\bar{x} - x_i)^2}{n-1}} \quad (4)$$

$$\bar{x} = \frac{1}{n} \sum_{i=1}^n x_i \quad (5)$$

299

300 n is the number of sampled cycles

301 Fig. 10 (a) and (b) shows the coefficients of variation for maximum pressure
 302 ($COV_{P_{max}}$) and for IMEP (COV_{IMEP}) respectively. It can be seen that COV values for dual
 303 fuel mode, with either NG or HNG, are higher compared to those of conventional diesel
 304 mode. Moreover, the COV values for conventional diesel are relatively constant and do not
 305 exceed 0.6% for $COV_{P_{max}}$, for instance. Regarding COV_{IMEP} , the highest value was about
 306 13%, with dual fuel mode, which is much higher than conventional diesel case.



307

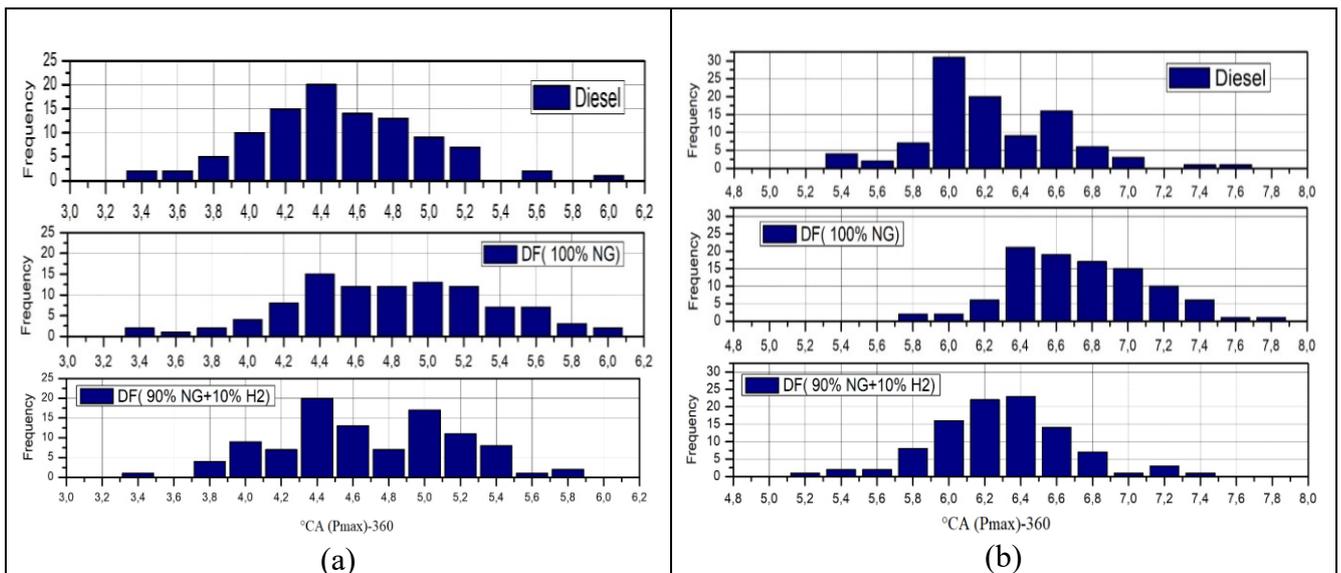
308 Fig. 10. $COV_{P_{max}}$ (a) and COV_{IMEP} (b) at different engine loads.

309

310 The most frequently used parameter is the COV_{IMEP} . It determines the cyclical
 311 variability of the indicated work per cycle. A higher value of COV_{IMEP} indicates more
 312 instability in combustion and it has been found that drivability problems of vehicle appears
 313 generally when COV_{IMEP} exceeds about 10 percent in conventional combustion engines [33].
 314 It can be noticed from Fig. 10 (b) that for all hydrogen test blends, the COV_{IMEP} satisfy this
 315 condition. In fact, the addition of hydrogen has a noticeable effect on reducing the cyclic
 316 variability of the indicated mean effective pressure. As mentioned above, hydrogen addition

317 to NG is effective in combustion quality improvement, thus resulting in better combustion
 318 stability.

319 On the other hand, another important combustion parameter is the crank angle
 320 corresponding to the maximum pressure in the combustion chamber. Variability from one
 321 cycle to another influences the crank angle position of maximum pressure. The frequency
 322 distribution of P_{\max} position in term of crank angle, for all operating conditions is shown in
 323 Fig. 11. The P_{\max} positions are observed to be, in most of the cases after the top dead center
 324 (TDC). For instance, the P_{\max} positions are in the range of 3.4-6.6 °CA after TDC for low
 325 loads and 5–7.6°CA for high loads, respectively. The maximum repeatability of the same
 326 crank angle for P_{\max} position is up to 30%.

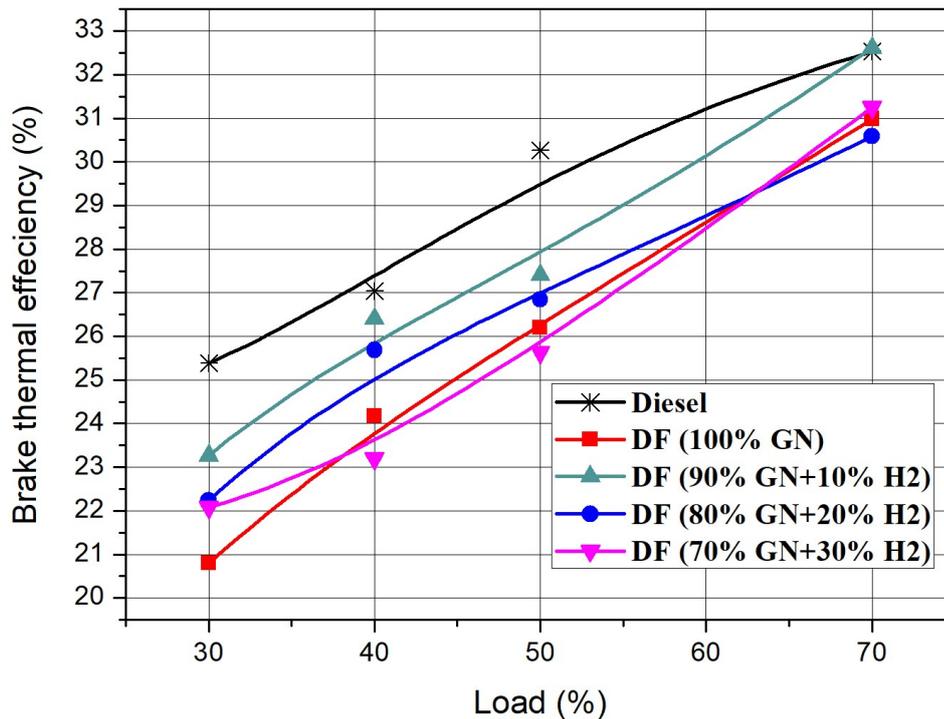


327
 328 Fig. 11. Frequency distribution of crank angle corresponding to P_{\max} at 30% (a) and 70%
 329 (b) engine load.

330 3.2. Engine performance

331 Combustion quality has a direct effect on engine performance such as brake thermal
 332 efficiency (BTE). The brake thermal efficiency evolution (Fig. 12) is investigated to examine
 333 the influence of NG enrichment with hydrogen on dual fuel engine performance. It can be
 334 seen that the brake thermal efficiency is lower for dual fuel mode, with either NG or HNG in

335 comparison to conventional diesel case at all engine loads. This reduced thermal efficiency in
 336 dual fuel mode can be explained by the poor lean-burn capability, slow-burning velocity of
 337 the gaseous fuel and low in-cylinder gas temperature. For instance, at 30% engine load, the
 338 BTE decreased from 25.39% with conventional diesel mode to 20.8% with NG dual fuel
 339 mode. However, this difference in the BTE is reduced with H₂ addition and the results show
 340 an enhancement under HNG dual fuel mode with almost all hydrogen concentrations
 341 compared to NG dual fuel operation. This is the consequence of improved gaseous fuel
 342 combustion and the enhanced combustion rate, as shown before which is due to combustion
 343 characteristics of hydrogen. As observed from Fig.12, the maximum brake thermal efficiency
 344 is achieved with 10% hydrogen enrichment at 70% engine load. Almost the same BTE was
 345 reached with pure diesel (32.53%).



346
 347 Fig.12. Brake thermal efficiency at different engine loads.

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4. Conclusion

350 In our present study, combustion analysis has been carried out for examining the effect of
 351 natural gas enrichment with various hydrogen blends. In addition, a comparison was

352 performed for various combustion characteristics of CI engine, running under conventional
353 diesel mode and dual fuel with both NG and HNG. Main conclusions are summarized in
354 the following points:

- 355 • The hydrogen addition to NG leads to a significant increase in the in-cylinder pressure
356 peak and maximum HRR at high engine load, as a result of improvement in gaseous
357 fuel combustion behavior. In fact, this increase is due to the fast burning velocity and
358 better ignitability of hydrogen. Higher CHR is obtained at 10% H₂ addition.
- 359 • The combustion noise is always higher for conventional diesel mode compared to dual
360 fuel mode. Both NG and HNG dual fuel operations have almost similar combustion
361 noise.
- 362 • Hydrogen addition to NG has a noticeable effect on reducing the cyclic variability,
363 thus resulting in better combustion stability.
- 364 • Combustion duration of gaseous fuel combustion phase is considerably reduced by
365 hydrogen addition to NG. However, this does not necessary result in reduced overall
366 combustion duration.
- 367 • At high engine load, the ignition delay is greater with NG dual fuel mode compared to
368 conventional diesel. However, addition of hydrogen to NG did not affect significantly
369 the ignition delay period.
- 370 • The BTE was lower for DF mode compared to conventional diesel operation.
371 Regarding hydrogen addition, a significant improvement was obtained with almost all
372 hydrogen blends as a result of improved gaseous fuel combustion.
- 373 • Maximum BTE was attained with 10% hydrogen enrichment at 70% engine load and
374 was similar to that obtained with pure diesel.

375 Overall, enriching NG with H₂ in dual fuel mode is an interesting technique to improve
376 engine combustion characteristics and engine performance.

377 **Notations**

- 378 m mass flow rate, kg/s
 379 N engine rotational speed, rpm
 380 P instantaneous in-cylinder gas pressure, bar

381

382 **Abbreviations**

383

- 384 BTE brake thermal efficiency
 385 CA crank angle
 386 CI compression ignition
 387 CHR cumulative heat release
 388 COV coefficient of variation
 389 DI direct injection
 390 DF dual fuel
 391 EOC end of combustion
 392 HNG hydrogen natural gas
 393 HRR heat released rate
 394 ID ignition delay
 395 NO_x nitrogen oxide
 396 ROPR rate of pressure rise
 397 RCCI reactivity controlled compression ignition
 398 SOC start of combustion
 399 SOI start of injection
 400 VHV volumetric heating value

401

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