Proceedings of the 2nd World Congress on Momentum, Heat and Mass Transfer (MHMT'17)
Barcelona, Spain – April 6 – 8, 2017
Paper No. CSP 109
ISSN: 2371-5316
DOI: 10.11159/csp17.109

Effect of Pilot Pre-Injection on Methane-Diesel Dual Fuel Engine Combustion

Abdelrahman Hegab[1, 2], Paul Shayler[1]
[1]Department of Mechanical, Materials, and Manufacturing Engineering, the University of Nottingham
University Park, Nottingham NG7 2RD, United Kingdom
paul.shayler@nottingham.ac.uk
[2]Department of Mechanical Engineering, Al-Azhar University
Nasr City, Cairo 11371, Egypt
abdelrahman.hegab@nottingham.ac.uk

Abstract **-** Dual fuelling of diesel engines with natural gas (NG), mainly composed of methane, is considered an attractive approach to reduce the dependence on diesel fuel and provide a more eco-friendly alternative to conventional diesel engines. In this fuelling strategy, most of the engine power output is provided by the NG, while a pilot amount of diesel fuel is used as an ignition source to ignite the NG-air mixture. However, this engine configuration suffers from lower thermal efficiency, slower burning rate, increased engine instability, and higher CO and HC emissions; particularly at part loads. The present work aims at investigating the effect of using pilot pre-injection (i.e. splitting the diesel pilot into a main injection preceded by a pre-injection) on dual fuel engine combustion and performance at part load conditions. The experiments were conducted on a modern automotive direct injection (DI) diesel engine; properly modified to suite dual fuel operation with methane and diesel. The tests were performed at different engine speeds ranging from 1400 to 2000 rpm at 25% of the engine load at the particular speed, with different substitution ratios of methane for diesel fuel (from 20 to 80% on energy basis). The results demonstrated that optimized pilot pre-injection could bring about a change in the combustion mode from the conventional diesel dual fuel (DDF) combustion to a novel two-stage combustion mode; referred to as partially premixed dual fuel (PPDF) combustion. With this strategy, combustion process becomes more intensified yet smoother, with around 17% lower rate of pressure rise (ROPR) and up to 5% improvement in the brake thermal efficiency. PPDF strategy also maintains the engine stability at acceptable levels (even with high substitution ratios) and allows leaner operation at all conditions; indicating lower levels of harmful emissions.

Keywords**:** Dual fuel engine, diesel fuel, methane, pilot pre-injection, partially-premixed, combustion, emissions

1. Introduction

The use of natural gas (NG) as a partial substitute for diesel fuel in compression ignition engines is recognized as a potent approach towards a more sustainable fuel market, to keep diesel fuel supply-and-demand in balance and mitigate the harmful effects of diesel engines emissions [1]. In recent years, there have been continuous research efforts with regard to NG-diesel dual fuel engines combustion, performance, and emissions. Various engine parameters have been assessed, among which pilot injection strategy (size, spacing, timing, and pressure) is most influential and should be optimized in accordance with the operating conditions [1]. Diesel dual fuel (DDF) engines at part load suffer from retarded performance, reduced engine stability and partial misfire, on top of increased HC and CO emissions. At high load, the major drawbacks are the elevated rates of pressure rise and the increased tendency to knock [2].

The development of advanced techniques of diesel engines brings many novel ideas to dual fuel engines, where the continued evolution in engine electronic control (EEC) and fuel injection equipment (FIE) promotes the progress in DDF engine to the next level. The latest generation of high pressure common rail (HPCR) injection systems incorporates piezo-electric injectors that can perform a very rapid switching. As a result, shorter injection durations, smaller injection quantities, and multiple injections per cycle all become possible. With accurate control of pilot injection parameters, novel combustion modes in DDF engines, such as homogeneous charge compression ignition (HCCI) and also partially premixed compression ignition (PPCI), become achievable [3]. The key to combustion mode change is the pilot injection strategy. Conventional DFF operates with a single late injection of the pilot, HCCI is achieved through a very early injection of the pilot to form a homogeneous mixture, while PPCI uses two injections of the pilot; an early pre-injection to promote the ignition, and a late main injection to initiate the combustion [3]. With two split injections, pilot ignition mode changes from the classical auto-ignition of diesel to the novel two-stage ignition mode that can bring several benefits to engine performance, combustion, and emissions [4].

The present work aims at investigating the viability of employing pilot pre-injection to approach partially premixed dual fuel (PPDF) combustion; with different substitution ratios of methane for diesel fuel. Cylinder pressure, rate of pressure rise (ROPR), coefficient of variation of the gross IMEP ($COV_{IMEPgross}$), heat release rate (HRR), combustion phasing (CA50), brake thermal efficiency, total equivalence ratio, and total diesel fuel injected are studied and compared with the data obtained from conventional DDF operation with single injection; at the same operating conditions.

2. Experimental Facility and Test Conditions

2.1. Engine and test bed

The experimental work in the present study was carried out on a Ford Puma (Duratorq ZSD 422 Range) four cylinder, 2.2 liters, turbocharged intercooled direct injection (DI) diesel engine; typical of that used in commercial light duty vehicles. The engine develops a peak power of 92 kW (125 hp) at 3700 rpm and a peak torque of 320 Nm at 2000 rpm. This engine employs a high-pressure common-rail (HPCR) fuel injection system producing injection pressure as high as 1800 bar. Figure 1 shows a general view of the engine on the test bed. The engine was directly coupled to a David McClure DC electric motor regenerator dynamometer has a rated power of 90 kW and a maximum speed of 4000 rpm. The dynamometer was controlled through a DC drive unit that can maintain steady speed motoring/absorbing operation against varying torque.

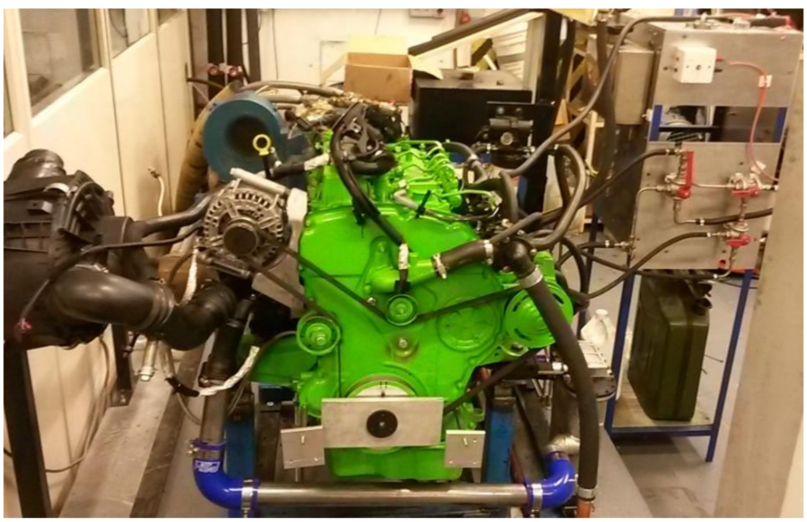

Fig. 1: General view of the Ford Puma (ZSD 422) HPCR diesel engine used in the present work; mounted on the test bed.

The engine intake manifold was properly modified to suite dual fuel operation with port injection of methane. The modification involves attaching a special gas injection system comprising a gaseous fuel rail and a mixing section, where the gaseous fuel is injected and mixed with the intake stream before being admitted to the engine. Chemically pure methane (99.95% concentration) was supplied from a compressed gas bottle, through a set of pressure regulators to bring the pressure down from the bottle pressure of 200 bar to the injectors working pressure of 7 bar.

2.2. Data acquisition and instrumentations

All experimental data were acquired to a desktop PC via an integral data acquisition system and control programs. The setup involves the use of an advanced NIRA engine management system in the place of the original engine control unit (ECU). This system is capable of controlling both the diesel as well as the gaseous fuel injection strategies. It comprises a NIRA-i7r engine control unit complete with NIRA-rk application tool for loading software program, tuning engine data, monitoring, etc. NIRA-i7r ECU is connected through a special dongle to the PC, where NIRA-rk application tool is installed. Engine variables, including diesel fuel injection parameters, were measured and controlled by this system. Gaseous fuel injection was also controlled by the NIRA system, while its consumption was measured through a thermal mass flow meter, model F-113AI by Bronkhorst, and then imported into a dedicated software. The in-cylinder pressure was measured using Kistler 6055C piezoelectric non-cooled combustion pressure sensors fitted into the cylinder head glow plug holes. The output signal of each pressure transducer was fed to a dedicated Kistler 5011 charge amplifier. Crankshaft position was monitored using a Hohner W2D11R incremental optical shaft encoder, with an accuracy of 0.5 degree. In-cylinder pressure and crank position signals were fed into an X-Series NI USB-6351 high-speed (1MHz) data acquisition module then to the PC, to obtain the crank angle-synchronized in-cylinder pressure data in the LabVIEW environment.

2.3. Test conditions and procedure

The present work targets the operation of methane-diesel dual fuel engines at partial load conditions that prevail in modern city traffic. Accordingly, tests were conducted at different engine speeds ranging from 1400 to 2000 rpm at 25% of the engine load at the particular speed, with different substitution ratios of methane for diesel fuel ranging from 20 to 80%. The engine was first run under the mono operation of diesel fuel, then methane was introduced on the account of diesel to achieve the required substitution ratio. The percentage of methane substitution is defined on energy basis as:

$$\%\mathrm{CH}_4 = \frac{\dot{m}_{CH4} \times LHV_{CH4}}{\dot{m}_D \times LHV_D \ + \ \dot{m}_{CH4} \times LHV_{CH4}} \tag{1}$$

where $(\dot{m}_{CH4})$ and $(\dot{m}_D)$ are the mass flow rates of methane and diesel, respectively, and (LHV_{CH4}) and (LHV_D) are the lower heating values of the two fuels. As tests involve the use of two fuels, the total equivalence ratio is calculated as:

$$\emptyset_{tot} = \frac{AFR_{CH4}^{stoic} \times \ \dot{m}_{CH4} + \ AFR_D^{stoic} \times \ \dot{m}_D}{\dot{m}_{air}} \tag{2}$$

where (AFR_{CH4}^{stoic}) and (AFR_D^{stoic}) are the stoichiometric air-fuel ratios of methane and diesel, respectively, and $(\dot{m}_{air})$ is the mass flow rate of the intake air.

For each test case examined, the variation of cylinder pressure with crank angle position was recorded for 50 consecutive cycles, where the ensemble average of these 50 combustion cycles was then filtered to remove any noise spikes, then processed to calculate HRR, ROPR, CA50, IMEP, and $\mathrm{COV}_{\mathrm{IMEPgross}}$.

The net heat release rate (HRR) is calculated by the traditional first law equation [5]:

$$\frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \tag{3}$$

where (θ) is the crank angle, (p) is the in-cylinder pressure at a given crank angle, (V) is the cylinder volume at that point, and (γ) is the specific heat ratio (C_p/C_v). Integrating the HRR as a function of crank angle provides a representation of the total energy released up to a specified angle (aka cumulative HR). The crank angle at which 50% of heat release occurs (CA50) is used to present combustion phasing [6].

The ROPR is obtained from the pressure traces and used as an indication of engine combustion noise [7], while the coefficient of variation of the gross indicated mean effective pressure ($\mathrm{COV}_{\mathrm{IMEPgross}}$) is used to define the cyclic variability as a measure of engine stability, and calculated as [5]:

$$COV_{IMEPgross} = \frac{\sigma_{IMEPg}}{\overline{IMEPg}} \times 100\% \tag{4}$$

where (σ_{IMEPg}) is the standard deviation and ($\overline{IMEPg}$) is the mean value of the gross IMEP over the recorded cycles.

For tests that involve the use of pre-injection, the size of pre-injection was quantified on mass basis; as a percentage of the total diesel fuel used:

$$\% \text{ PRE} = \frac{\dot{m}_{Pre}}{\dot{m}_D} \quad (5)$$

where $(\dot{m}_{Pre})$ is the mass flow rate of pre-injection part of the pilot, and $(\dot{m}_D)$ is the total mass flow of diesel fuel (supplied to the engine; inclusive of the pre-injection.

3. Theory and method

To approach PPDF combustion mode, the pre-injection part of the pilot should be optimized for timing and quantity. With regard to timing, it should be noted that too early injection will cause spray impingement and poor atomization of fuel, while too late injection will lead to the incidence of classical diesel ignition [4]. To assess the optimum, a sweep of different pre-injection timings (ranging from 20° to 80° BTDC) was investigated under conventional diesel operation at 1600 rpm and 25% load, while the main injection was kept at 10° BTDC. Any significant changes in the diesel combustion mode would essentially be reflected on the subsequent combustion of the gaseous fuel-air mixture [4]. To avoid any possible high rates of pressure rise with early injection timings, the pilot pre-injection quantity (%PRE) was kept at 10%. The resulting HRR curves with pre-injection were compared with those obtained from conventional single injection of the diesel fuel, to reveal the injection strategy that influences the combustion mode. Figure 2 shows the resulting HRR curves for some selective test cases examined (the whole set of results could not be presented due to space limitation).

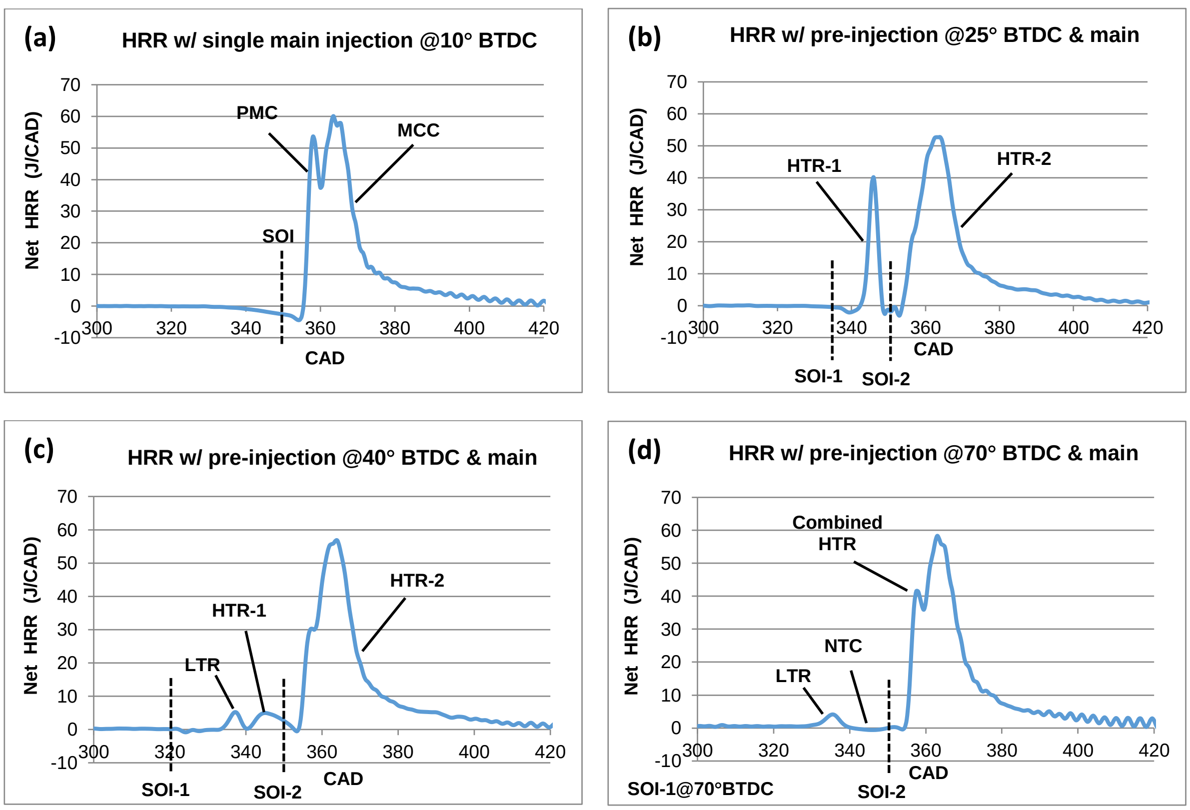

Fig. 2: The influence of diesel injection strategy on the net HRR and combustion mode; (a) with single main injection of diesel @10° BTDC; (b), (c), (d) with two split injections with the main injection fixed @10° BTDC and the pre-injected set to 25°, 40°, and 70° BTDC; respectively. Data given is for conventional diesel operation at 1600 rpm and 25% load.

It can be seen from Fig.2 (b) that the relatively late injection timing @ 25°BTDC of the pre-injection will result in the characteristic CI spray combustion mode (i.e. that distinguishes diesel engines) of that pilot. This is attributed to the elevated high pressure and temperature inside the engine cylinder upon injection, and hence as soon as the ignition delay (ID) of that part of injection ends, it burns immediately. Nevertheless, in contrast with the classical diesel combustion that is characterized by the presence of two combustion phases; premixed combustion (PMC) and mixing-controlled combustion (MCC) as shown by Fig.2 (a) and described in [5], the combustion of the pre-injection part of the pilot will take place over only one stage; due to its limited quantity. This stage (referred to as HTR-1) is primarily a high temperature reaction that increases the cylinder temperature significantly. Accordingly, the main part of the fuel subsequently injected will experience a very short ID, after which it will burn in a second HTR (referred to as HTR-2) with much less premixed part and lower noise levels.

When pre-injection timing is advanced, its ignition mode changes from the classical diesel combustion mode into a two-stage combustion mode. As shown in Fig.2 (c) and (d), this mode is characterized by the presence of a low temperature reaction (LTR), or a cool flame zone, that has a high degree of reactivity due to a high level of radicals [8]. In an ideal HCCI two-stage combustion, a negative temperature coefficient (NTC) zone separates the LTR and the HTR regimes without any other heat release processes [8]. However, with average advance of the pre-injection timing, e.g. 40°BTDC as shown in Fig. 2 (c), where the cylinder conditions (pressure and temperature) are relatively high, the LTR could be followed by a HTR of some part of the pre-injection (referred to as HTR-1), and the NTC is not realised. Any unburned fractions of the pre-injection will burn in the main combustion event (HTR-2) that occurs following the auto ignition of the main injection. With further pre-injection advance, e.g. 70°BTDC as shown in Fig. 2 (d), the NTC zone becomes recognizable after the LTR zone, and the HTR-1 no longer exists. Instead, a combined HTR regime for the pre-injection along with the main injection occurs, with larger premixed part compared with retarded per-injection timings, and at higher rates of heat release. But in both cases, as long as the two-stage ignition mode is attained, the LTR zone always occurs at a specific position; about 20-25°BTDC [8].

In view of that, it was concluded that an optimum pre-injection timing for the current setup would be around 50°BTDC with main injection timing set to 10°BTDC. This would bring about a two-stage ignition mode of the pilot that promotes the combustion of the main fuel, while it provides a stable combustion at acceptable rates of heat release.

As far as the quantity of pre-injection part of the pilot is considered, too large pre-injections may cause high rates of pressure rise beyond the acceptable levels, while too small quantities may not be capable of producing the desired cool flame reactions and active radicals. The limitation to the minimum amount of fuel that could be injected through a given FIE should also be considered. To set a base of comparison, the value of %PRE was kept constant for all tests with pre-injection. As the current work involves CH_4 substitution ratios as high as 80% (on the account of diesel fuel), a value of 20% was found to be the optimum %PRE that could be realized at all CH_4 substitution ratios examined, without compromising the cool flame effect or ROPR. The effect of %PRE on conventional diesel combustion was examined to ensure that the proposed value satisfies the required criteria. Figure 3 shows the HRR curves for 10% and 20% PRE, for the optimum pre-injection timing of 50°BTDC (as determined earlier) and main injection timing of 10°BTDC; at 1600 rpm and 25% load.

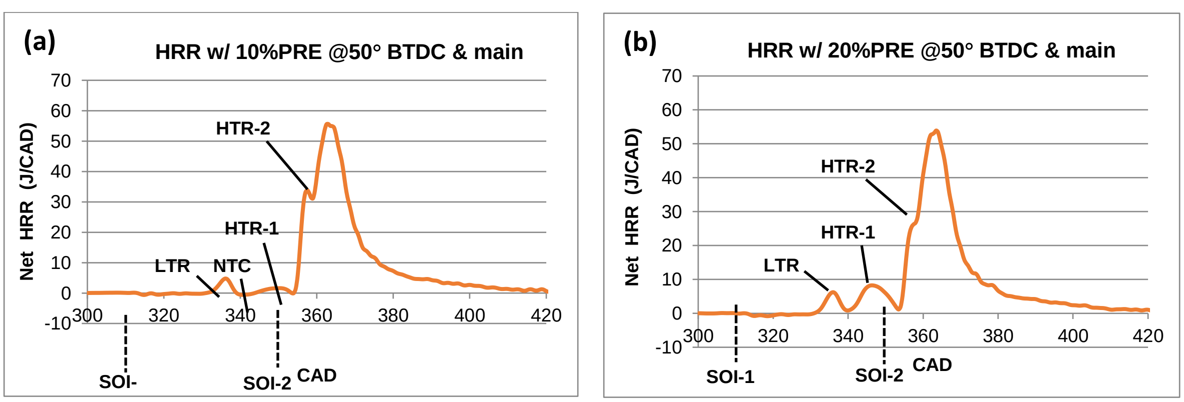

Fig. 3: The influence of pre-injection quantity (%PRE) on the net HRR and combustion mode at optimum injection timing of 50° BTDC; (a) 10%PRE, and (b) 20%PRE. The quantity of main injection varies accordingly, but its timing is always @10° BTDC. Data given is for conventional diesel operation at 1600 rpm and 25% load.

First, it could be seen from Fig. 3 that the quantity of pre-injection does not affect the position of the LTR zone, but it may slightly affect its magnitude; due to the availability of a large amount of fuel to go through the cool flame reactions. The relative magnitudes of the HTR zones experience the more apparent change, where the larger %PRE leads to the release of a larger amount of active radicals that promotes the combustion of some of the pre-injection part; owing to the relatively elevated cylinder temperature at that conditions. Accordingly, the HTR-1 zone becomes clearly distinguished in Fig. 3 (b), while HTR-2 exhibits smaller premixed part; owing to the reduced contribution of the pre-injection from one side, in addition to the reduced ID of the main injection itself from the other side.

4. Results and Discussion

In this section, only test results at 2000 rpm and 25% of the engine load are presented for both DDF and PPDF modes; results at other speeds could not be presented due to space limitation. Figure 4 shows the in-cylinder pressure traces and lists the associated numeric values of maximum ROPR at each combustion mode; at different CH4 substitution ratios. From the figure, it can be seen that PPDF combustion exhibits higher in-cylinder pressure values compared with conventional DDF combustion; particularly with high %CH4. The reason behind this is the earlier start of combustion (SOC) associated with the use of pre-injection in PPDF, where the presence of active radicals from the LTR zone promote the combustion process. The earlier start of that intensified combustion yields higher cylinder pressure values [9]. The effect is more voluminous at high %CH4, as conventional DDF combustion at these conditions exhibits prolonged combustion duration (as demonstrated in Fig. 5) that results in lower cylinder pressure values. Above and beyond, the use of pre-injection in PPDF means that the combustion process will start with an auto-ignition of a less amount of the diesel fuel, and hence the maximum ROPR values are much less; the average reduction in maximum ROPR is as high as 17%. This is advantageous to engine combustion noise, fuel economy, and NOx and PM emissions [7].

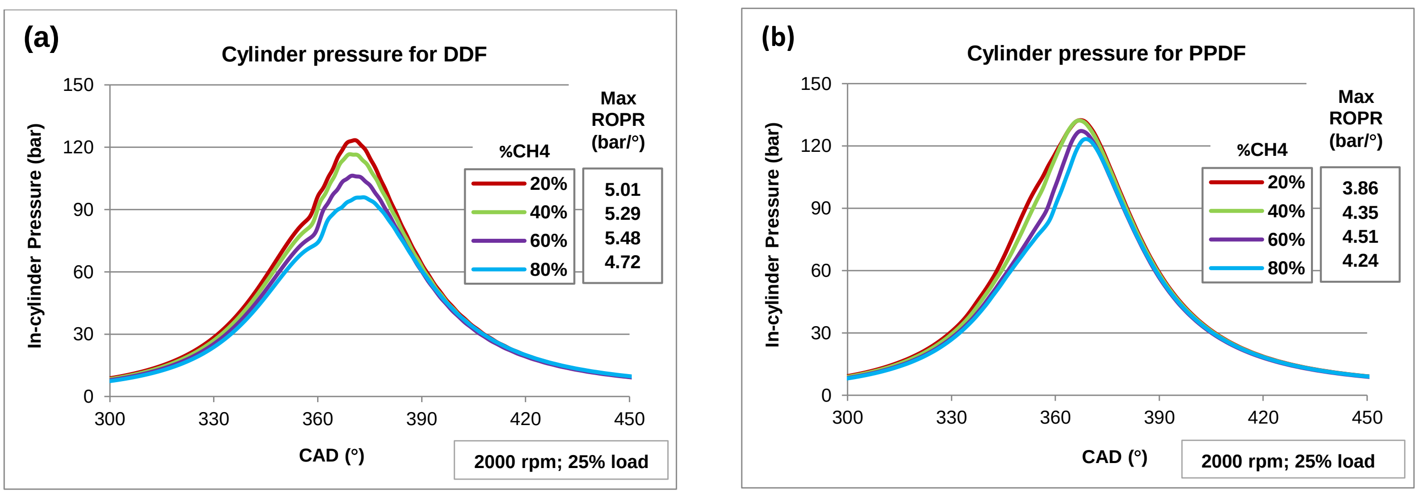

Fig. 4: In-cylinder pressure and the associated maximum ROPR, for (a) conventional DDF operation with single pilot injection @10° BTDC, and (b) PPDF operation with 20% PRE @50° BTDC and main injection @10° BTDC.

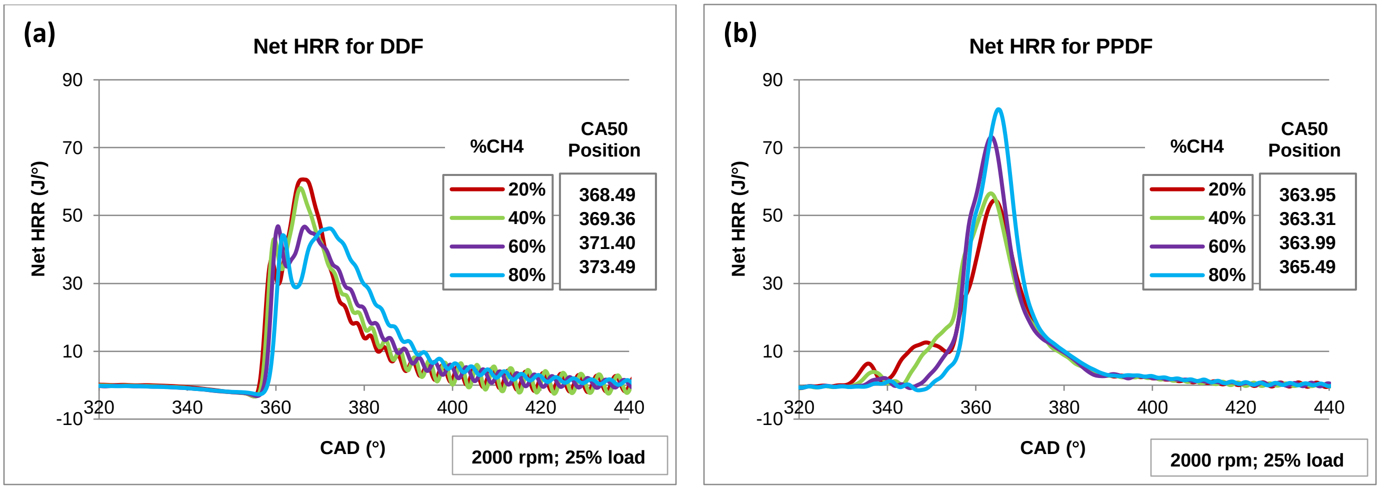

Fig. 5: Net HRR data along with the corresponding positions of the occurrence of CA50, for (a) conventional DDF operation with single injection @10° BTDC, and (b) PPDF operation with 20% PRE @50° BTDC and main injection @10° BTDC.

Figure 5 shows the net HRR, along with the corresponding positions of the occurrence of CA50; for both DDF and PPDF modes. The LTR zone could be observed with PPDF combustion as already discussed, although its magnitude notably decreases with the increase of % CH_4. This is the natural consequence of the decrease in total diesel fuel injected; according to Eq. (5) above. Yet, it is believed that the small pre-injection is still capable of increasing the degree of reactivity inside the cylinder and promoting combustion. This is evident by the enhanced burning rates and reduced combustion duration at all % CH_4 ratios; in contrast with DDF. Combustion phasing, as represented by CA50, is much improved in PPDF, where the major part of the heat release takes place closer to the TDC; producing a more useful work [6]. The late stages of heat release with PPDF are also free of any ringing visible (unlike DDF); demonstrating the reduction of combustion noise.

The effect of pre-injection on PPDF engine stability and performance is shown in Fig. 6. It could be seen from Fig. 6 (a) that the use of pre-injection in PPDF mode maintains the engine cyclic variability (as represented by the $COV_{IMEPgross}$) at acceptable levels (2-3%) throughout the entire range of % CH_4. This is attributed to the intensified combustion in PPDF mode that provides better engine stability [9]. With DDF, in contrast, the use of high % CH_4 is always associated with high cyclic variability, particularly at part load, due to flame propagation problems [10]. In addition, Fig. 6 (b) shows that more than 5% improvement in the engine performance, as represented by the brake thermal efficiency, could be achieved when the engine is switched from DDF to PPDF operation. This improved performance is the result of the combination of increased in-cylinder pressure, reduced ROPR, enhanced combustion intensity, faster burning rate, better combustion phasing, and increased engine stability.

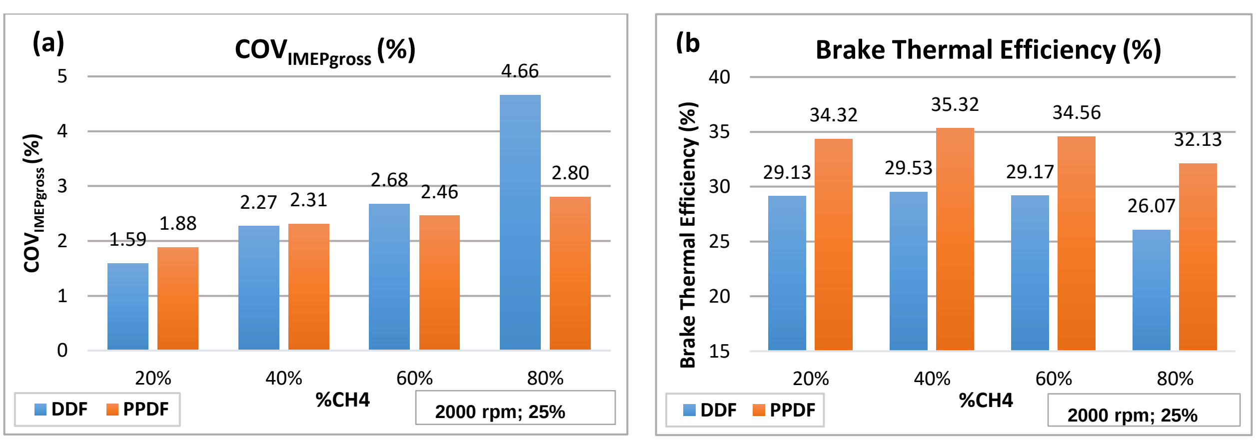

Fig. 6: (a) Coefficient of variation ($COV_{IMEPgross}$), and (b) brake thermal efficiency values; for both DDF and PPDF modes.

The improved fuel economy with PPDF mode means that the engine will work at lower total equivalence ratio (ϕ_{tot}) compared with DDF; as shown in Fig. 7 (a). The reduced (ϕ_{tot}) involves the reduction of both fuels; ($\dot{m}_D$) and ($\dot{m}_{CH4}$), as implied by Eq. (2). That is, the lean operating limit of the engine could be extended. Moreover, the reduced ($\dot{m}_D$) with PPDF, as shown by Fig. 7 (b), implies a corresponding reduction in the PM emissions originated from the diesel pilot. The effect is more voluminous at low % CH_4 where the diesel pilot fuel is more significant, and hence the reduction in the PM emissions becomes more sensible.

While no emissions results are presented in the current work, it is believed that PPDF mode practically produces lower CO, HC, and NOx emissions than conventional DDF. This is attributed to the effect of splitting the diesel injection into an early pre-injection followed by the main injection. The pre-injection part is expected to homogeneously mix with the gaseous fuel; increasing the mixture strength and reducing CO and HC emissions to some extent [3]. At the same time, the size of the main injection becomes smaller, which means that the rich mixture zone in the spray cone is reduced in some measure; reducing NOx emissions [5].

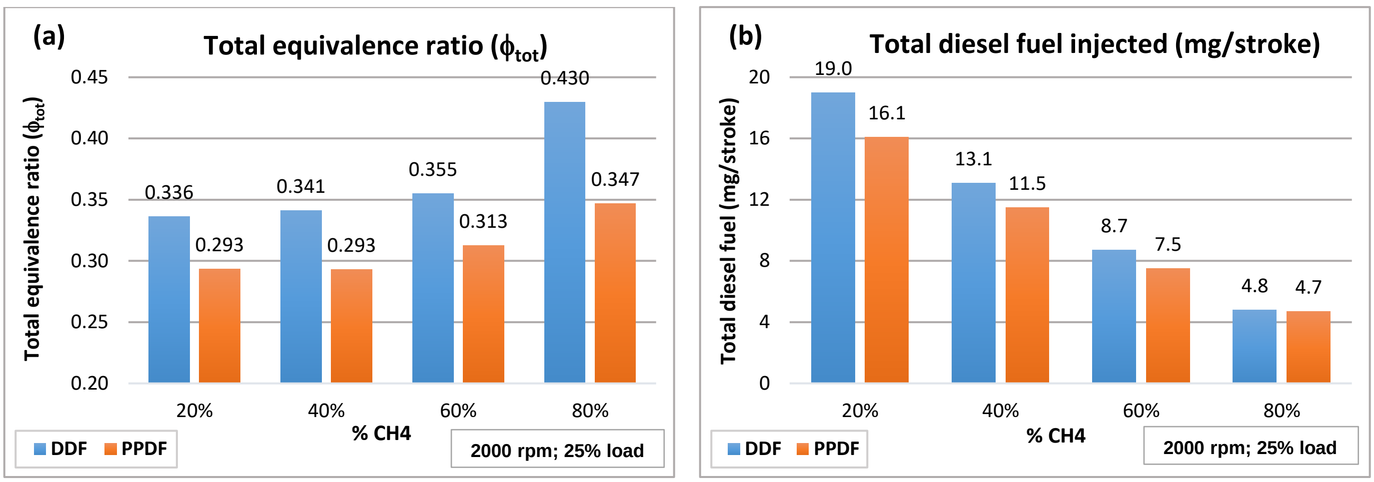

Fig. 7: (a) Total equivalence ratio (ϕ_{tot}), and (b) total diesel fuel injected (mg/stroke); for both DDF and PPDF modes.

5. Conclusion

In this work, the use of pilot pre-injection in methane-diesel dual fuel engine was investigated over a wide range of CH_4 substitution ratios (20% – 80%). It was found that this technique could change the ignition mode from the conventional DDF combustion into a two-stage PPDF combustion. To attain PPDF, the pre-injection part of the pilot should be optimized for timing and quantity, where the practical range of timing was found to be 40°– 80°BTDC and that of quantity was 10% – 20% of the total diesel fuel used. The high level of active radicals with PPDF brings about a more intensified yet smoother combustion with better phasing; compared with conventional DDF. Around 17% reduction in the maximum ROPR and 5% increase in the brake thermal efficiency could be obtained with PPDF while marinating the engine stability at acceptable levels; even with high % CH_4 substitution levels. The total equivalence ratio with PPDF is less than that with DDF, meaning that the engine lean operating limit could be extended. While emissions results are not presented in the current work, it is believed that PPDF exhibits lower CO and HC emissions due to the increased mixture strength with pre-injection, and lower NOx and PM emissions due to the reduced size of the main injection.

References

[1] A. Hegab, A. La Rocca, P. Shayler, "Towards keeping diesel fuel supply and demand in balance: Dual-fuelling of diesel engines with natural gas," *Renewable Sustainable Energy Rev.*, [online]. Available: http://dx.doi.org/10.1016/j.rser.2016.11.249

[2] M. M. Abdelaal, B. A. Rabee, A. H. Hegab, "Effect of adding oxygen to the intake air on a dual-fuel engine performance, emissions, and knock tendency," *Energy*, vol. 61, pp. 612-620, 2013.

[3] F. Königsson, “On combustion in the CNG-diesel dual fuel engine,” Ph.D. thesis, Dept. Mach. Des., Royal Institute of Technology (KTH), Sweden, 2014.

[4] Z. Wang, Z. Zhao, D. Wang, M. Tan, Y.Han, Z. Liu, H. Dou, “Impact of pilot diesel ignition mode on combustion and emissions characteristics of a diesel/natural gas dual fuel heavy-duty engine,” *Fuel*, vol. 167, pp. 248-256, 2016.

[5] J. B. Heywood. *Internal Combustion Engine Fundamentals*. New York, USA, McGraw-Hill, 1988.

[6] L. de O. Carvalho, T. C. C. de Melo, R. M. de A. C. Neto, “Investigation on the fuel and engine parameters that affect the half mass fraction burned (CA50) optimum crank angle,” *SAE Tech. Pap.*, 2012-36-0498, 2012.

[7] R. Stone. *Introduction to Internal Combustion Engines*. 3rd ed., Oxford, UK: Macmillan; 1999.

[8] M. Yao, Z. Zheng, and H. Liu, “Progress and recent trends in homogeneous charge compression ignition (HCCI) engines,” *Prog. Energy Combus.t Sci.*, vol. 35, pp. 398-437, 2009.

[9] M. Xu, W. Cheng, H. Zhang, T. An, S. Zhang, “Effect of diesel pre-injection timing on combustion and emission characteristics of compression ignited natural gas engine,” *Energy Convers. Manage.*, vol. 117, pp. 86-94, 2016.

[10] L. Sun, Y. Liu, K. Zeng, R. Yang, Z. Hang, “Combustion performance and stability of a dual-fuel diesel–natural-gas engine,” *Proc. Instn. Mech. Engr.s, Part D: J. Automob. Eng.*, vol. 229, no. 2, pp. 235-246, 2014.