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Numerical Investigation into Natural Gas–Diesel Dual-Fuel Engine Configuration

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Abstract

Natural gas is used as an additive to a Diesel engine in a dual-fuel configuration. The natural gas is port injected with the charge, while the Diesel is directly injected. The effect of increasing the energy content contributed by the natural gas (to 11% and 22%) is tested for two engine speeds (2000 and 2500 rpm) using ANSYS Forte with *n*-heptane as the surrogate fuel for Diesel and methane as the surrogate fuel for natural gas. It was found that the main combustion phase was retarded by additional amounts of natural gas, with 22% methane approaching misfire at 2500 rpm. The thermal efficiency for the other methane cases was higher than Diesel at the corresponding engine speeds, with 11% methane the highest at both engine speeds. Increasing the amount of natural gas significantly reduced the emissions of CO and NO_x. Because the natural gas is separately regulated, in practice the amount can be dynamically controlled by the ECU to satisfy the desired performance/emissions levels. The current results suggest that small amounts of natural gas can have a significant positive influence on the emissions profile.

1 Introduction

The transportation sector in Australia produced 93 Mt CO_2-e in the year to June 2016, which accounted for 18% of the nation's greenhouse gas emissions [1]. Of this figure, 37% is due to trucks, busses and light commercial vehicles [2]. Improvements in Diesel engine efficiency can contribute significantly to the improvement of these figures, reduce operating costs and extend fossil fuel reserves.

A promising alternative is to use natural gas with Diesel in dual-fuel operation because this can reduce NO_x and soot, but could result in increased CO and unburned CH_4. The conventional configuration is to maintain direct injection of the Diesel, with the natural gas port injected to premix with the charge. Various studies have investigated this concept. One study used CONVERGE CFD to study the effect of Exhaust Gas Recirculation (EGR) for a light-duty VW TDi engine [3]. They found that increasing the natural gas content retarded the main combustion phase so that once the energy content of the natural gas exceeded a certain level, no reactions occurred. The NO_x emissions also decreased, which can be partially attributed to the decrease in power generation, while the unburned hydrocarbon (UHC) and CO levels were approximately constant until the misfire point. Advancing the injection timing substantially reduced the NO_x, with corresponding significant increases in UHC and CO. Finally, splitting the Diesel injection into separate pulses increased the NO_x and decreased the UHC and CO the more mass there was injected into the second pulse.

Another study used experiments and a two-zone model (where the expanding "burning zone" is conical during fuel injection and spherical during combustion) to study the performance of a single-cylinder research engine [4]. The Diesel injection timing was close to TDC (5°, 10° and 15°BTDC) and the consistent ignition delay time resulted in all conditions being close to misfire. Increasing the energy content supplied by the natural gas mitigated the misfire conditions. Conditions closer to misfire had decreased NO and soot production but increased CO and specific energy consumption.

A modified single-cylinder Caterpillar 3400-series heavy-duty engine was studied at 25% load both experimentally and using the AVL FIRE v2014 software coupled with CHEMKIN [5]. A fixed energy content of 75% natural gas was used with different injection timings in the range 10–50°BTDC. At 10°BTDC, the unburned methane and CO were highest, but NO_x was low. At 30°BTDC, the NO_x was at its maximum and beyond this injection timing, unburned methane and CO emissions were approximately the same.

The current study investigates the effect of engine speed and energy content of natural gas for a single-cylinder engine at full load.

2 Test conditions

ANSYS Forte was used to simulate the engine with test conditions listed in Table 1. The RNG k-e model was used for the turbulence; a 35-species 74-reaction *n*-heptane chemical kinetics scheme was used. Pure Diesel (modelled as *n*-heptane) was considered along with natural gas (modelled as methane) contributing 10% and 20% of the mass of fuel (which equates to 11% and 22% of the energy based on the LHV of the fuels).

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Bore	8.25 cm
Stroke	11.6 cm
Compression Ratio (CR)	17
Injection start–end	30°BTDC–7°ATDC
Equivalence ratio λ	3.31
Engine speeds	2000, 2500 rpm
Load	Full

Table 1: Engine parameters for Results in Sect. 4.

3 Validation

A qualitative validation was performed against Diesel experiments [6]. In analysing the effect of engine speed (Figs. 1 and 2), similar effects can be observed in that increasing engine speed retards the location and magnitude of the peak pressure, eventually leading to the onset of misfire when there are effectively two local maxima of pressure. The heat release rate shows the two separate instances of heat release, with the first release always occurring at essentially the same timing, with the major heat release becoming retarded and weaker with increasing engine speed. At the highest engine speed tested, the major heat release has two distinct increases, so the final heat release is curtailed as misfire begins at higher engine speeds.

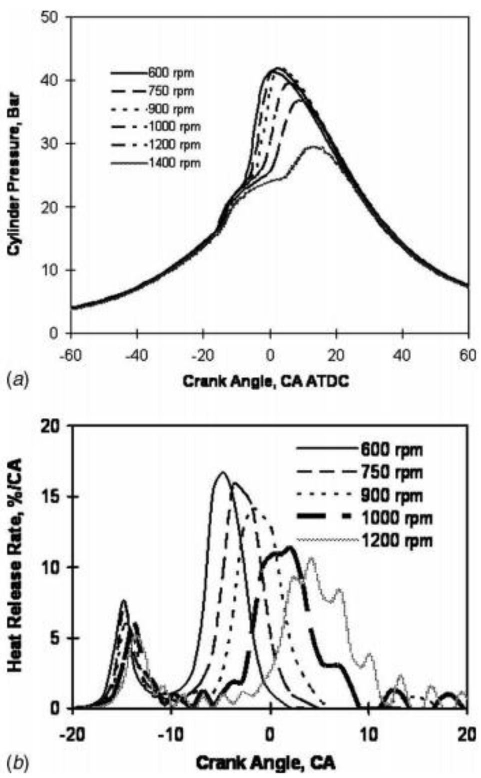

Figure 1: Effect of engine speed on (a) cylinder pressure and (b) heat release rate at CR = 10, AFR = 50, T_{in} = 40°C [6].

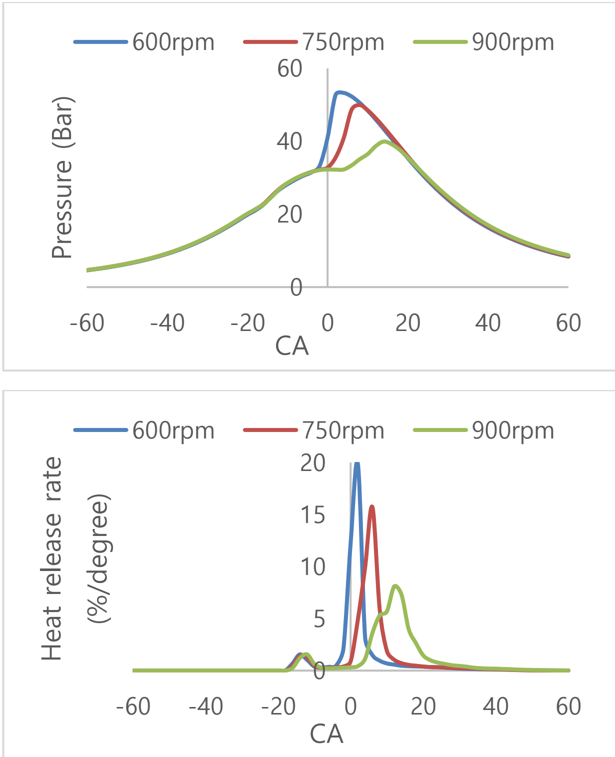

Figure 2: Simulation results corresponding to Fig. 1.

4 Results

4.1 2000 rpm

The pressure trace for the different fuels is shown in Fig. 3. In line with other studies [3,4], the main combustion phase is retarded by the addition of the methane. The fundamental reason for this is that the autoignition temperature of methane is not reached by TDC for the motored pressure: it requires the main combustion phase of the Diesel to reach the autoignition temperature. Therefore the port-injected methane acts as a diluent during the ignition delay of the Diesel. For the methane to autoignite before TDC requires extraordinary levels of charge pressure and temperature.

The corresponding heat release rate is shown in Fig. 4. The initial combustion phase is unaffected by the addition of methane, but the main combustion phase is slightly delayed, takes longer and reaches a lower peak heat release rate with increasing amounts of methane, with a corresponding decrease in accumulated heat (Table 2).

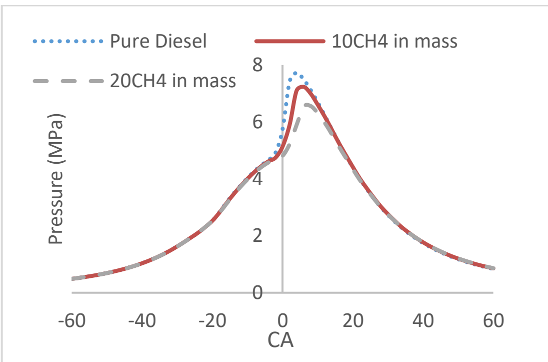

Figure 3: Pressure trace for 2000 rpm.

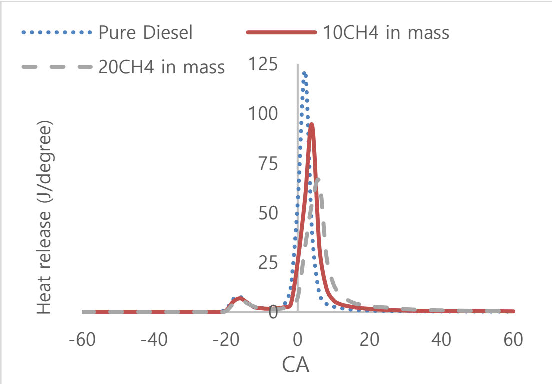

Figure 4: Heat release rate for 2000 rpm.

Engine speed	Diesel	10% CH_4	20% CH_4
2000 rpm	567 J	558 J	540 J
2500 rpm	586 J	546 J	498 J

Table 2: Accumulated heat for different cases tested.

4.2 2500 rpm

The pressure trace for the different fuels is shown in Fig. 5. Because of the faster engine speed, the duration shown in Fig. 5 is only 80% of the duration in Fig. 3, therefore it is unsurprising that the main combustion phase is significantly retarded, with the 20% methane approaching misfire. This outcome is reflected in the heat release rate (Fig. 6), where the initial combustion phase is unaffected, but the main combustion phase has two stages, which are separated in the case of 20% methane. Because the main combustion commences after TDC, the peak heat release rate is reduced by more than the 20% implied by the increased engine speed. The total energy released actually slightly increases for Diesel, but decreases for the methane cases, with a marked loss of power for 20% (Table 2).

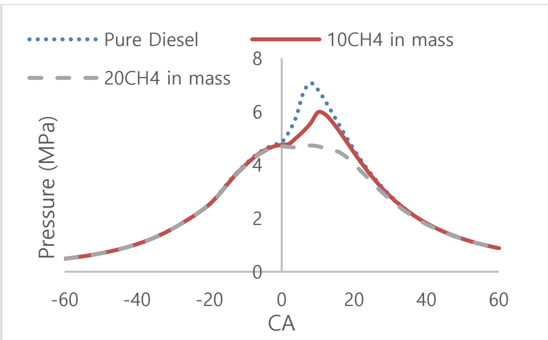

Figure 5: Pressure trace for 2500 rpm.

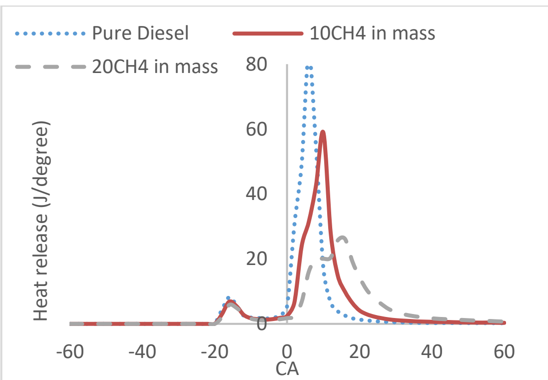

Figure 6: Heat release rate for 2500 rpm.

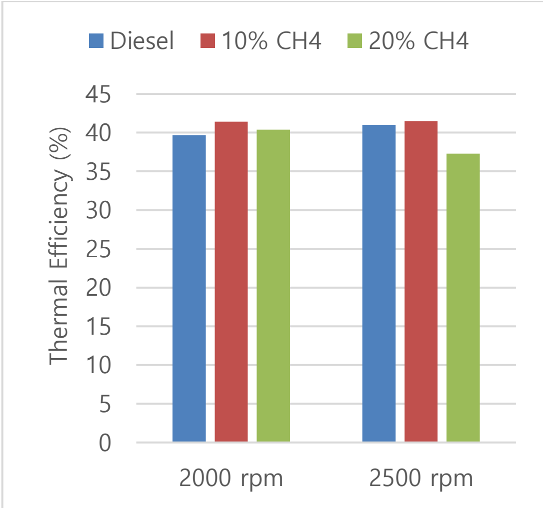

Figure 7: Thermal efficiency for all cases.

4.3 Performance and emissions

The thermal efficiency for the various cases is reported in Fig. 7. The 10% methane case had the highest value at both engine speeds, with 20% methane higher at 2000 rpm, but significantly lower at 2500 rpm due to its closeness to misfire. The CO and NO_x levels (Figs. 8 and 9 respectively) both reduced with increasing methane content. During the initial combustion phase, small amounts of methane are produced by the Diesel, so the additional methane aids the combustion process approach completion. The reduced cylinder pressure (and corresponding temperature)—due to the methane dilution retarding the moment of peak heat release—results in conditions that are not favourable for NO_x production.

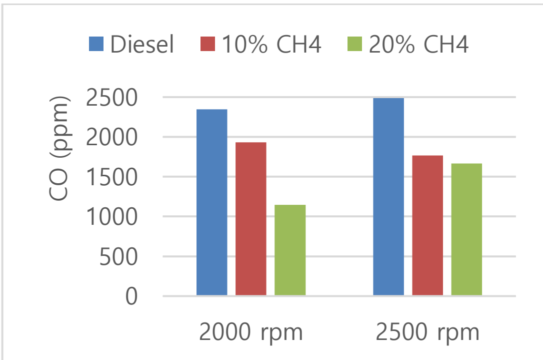

Figure 8: CO emissions for all cases.

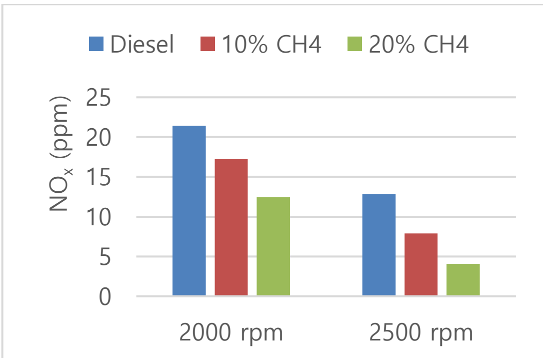

Figure 9: NO_x emissions for all cases.

5 Conclusions

The effect of natural gas operating in a dual-fuel configuration in a Diesel engine has been investigated via a simulation study. For the limited range of settings, it was found that increasing the fraction of energy content supplied by natural gas retards the main combustion phase without impacting on the initial combustion phase. At the higher engine speed, 20% CH_4 by mass was close to misfire; the thermal efficiency was slightly improved for the other cases with methane. Increasing the amount of natural gas increasingly reduced the emissions of CO and NO_x. Because the ECU would be able to dynamically control the amount of natural gas injected, based on the current results, it would appear that the best control strategy would be to reduce the amount of natural gas with increasing engine speed. Further research will seek to quantify such a control strategy and also investigate the impact of Diesel injection timing.

6 Acknowledgment

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References

[1] Commonwealth of Australia, Quarterly Update of Australia's National Greenhouse Gas Inventory: June 2016, 2016.

[2] Commonwealth of Australia, Tracking to 2020: An interim update of Australia's greenhouse gas emissions projections, December 2015.

[3] K. Poorghasemi, R.K. Saray, E. Ansari, B.K. Irdmousa, M. Shabakhti and J.D. Naber, Applied Energy 199 (2017) 430–446.

[4] R.G. Papagiannakis, S.R. Krishnan, D.C. Rakopoulos, K.K. Srinivasan and C.D. Rakopoulos, Fuel 202 (2017) 675–687.

[5] A. Yousefi, M. Birouk and H. Guo, Fuel 203 (2017) 642–657.

[6] H. Guo, W.S. Nell, W. Chippior, H. Li and J.D. Taylor, J. Eng. Gas Turb. Power (2010) 1–10.