Combustion Analysis of Biogas Premixed Charge Diesel Dual Fuelled Engine

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***Abstract*—Biogas engines are mostly converted from commercial compression or spark ignition engines. Some available researches have evaluated the use of biogas in relatively high power engines. However the similar publications on small engines is still limited. This paper suggests a general way to predict the performance of a biogas premixed charge diesel dual fuelled engine converted from a small single cylinder, stationary diesel engine which are very popular in rural areas of developing countries. Combustion of biogas-air mixture ignited by diesel injection in the combustion chamber of the engine is analyzed by CFD software FLUENT. Effects of biogas composition and operation parameters on engine performance are considered. The simulation is applied initially on a typical biogas diesel dual fuel engine converted from Vikyno EV2600NB diesel engine but this general procedure can be then applied on other kind of diesel engines. The prediction given by the simulation are very helpful for conversion technology of various kinds of diesel engines to biogas diesel dual fuel engine.**

Keywords*—biogas; dual fuel engine; FLUENT; combustion simulation

Nomenclatures:

P_e : Brake effective power
P_i : Indicated power
p_i : Cylinder pressure
W_i : Indicated cycle work
n : Engine speed
T : Mean mixture temperature
φ : Crank position (CA degree)
φ_s : Advance ignition timing (CA degree before TDC)
ϕ : Fuel-Air equivalence ratio
MxCy : Biogas containing x% CH_4 and y% CO_2 (volume fraction)
CA : Crank angle
TDC : Top Dead Center

I. INTRODUCTION

The world faces twin problems of energy crisis and environmental degradation. Substitution of fossil fuels by renewable energy is a promising way to resolve these problems. Biogas is an attractive alternative energy source because in view of energy crisis, it can act as a alternative fuel using for internal combustion engine and otherwise, it is renewable in nature, thereby not net contributors to the green house gases.

The energy utilization of biogas is maximized when it is converted into electricity, which is easy to use and transfer via a biogas generator and a small gas engine at a farm, which makes the process eco-friendly and energy efficient [1]. Biogas are very stable against knocking and can therefore be used in engines of higher compression ratios than petrol engines and thus, gains higher brake thermal efficiency and power [2]. However experimental results of Huang et al [3] indicate that the presence of carbon dioxide in biogas lowers the cylinder pressure, hence engine power and thermal efficiency are reduced.

Many engines converted from commercial compression or spark ignition engines can be fuelled with biogas. In comparison with biogas spark ignition engine, biogas-diesel dual fuel engine presents numerous advantages in practice. The experimental results of operating a diesel engine on dual fuel revealed better engine performance in terms of brake thermal efficiency and lower emissions [4]. Diesel engine can be easily converted to fumigated dual-fuel engine. Otherwise, this method has the advantage that in case of a shortfall in biogas supply during an important operation, the engine switches over smoothly without interruption to conventional diesel operation. This is very beneficial for the small scale biogas production in rural area where a standby engine is usually needed.

Dual fuel engines introduce a premixed air-gaseous fuel mixture, which is ignited at the final stage of the compression stroke by a diesel injection (pilot fuel). This diesel quantity plays also the role of lubrification of the nozzle. Quality of injected pilot fuel is another factor that has an effect on dual fuel engine performance, since composition and cetane number of liquid fuels affect ignition delay period and premixed combustion duration. Injection diesel provides an additional way to control output power of the engine fuelled with poor biogas [5] but it reduces the economic property of biogas engine [6].

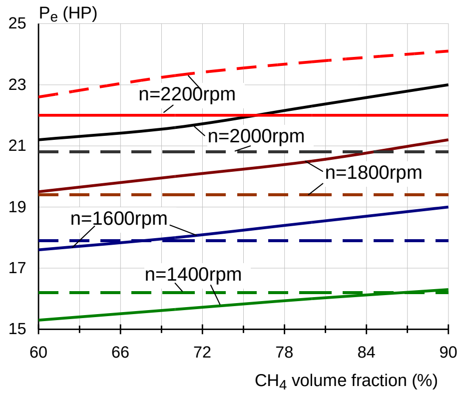


Fig. 1. Comparison of maximum brake power given by diesel engine (---) and that given by biogas-diesel dual fuel engine (—) fuelled with biogas containing various CH_4 compositions at different regimes

Diesel engine has low global air-fuel equivalence ratio, so when it is converted to biogas-diesel dual fuel engine, we can profit excess air to supply more biogas in order to improve engine output power. Tippayawong et al [7] observed that short-term performance between conventional diesel and dual-fuel operations was comparable whereas Mitzlaff et al [8] pointed out that the power output was found to be slightly higher in dual fuel mode than for diesel fuel operation. Comparison of Vikyno EV2600NB engine power operating on diesel mode and biogas diesel dual fuel mode with various biogas compositions and engine regimes is shown in Fig. 1. At engine speed of n=1400rpm and ϕ=1, power of dual fuel mode is higher than that of diesel mode as CH_4 composition in biogas exceeds 85%. Similarly, at n=1600rpm, power of dual fuel mode is higher than that of diesel mode as CH_4 composition in biogas exceeds 73%. With engine speed greater than 1800 rpm, dual fuel engine power exceeds diesel power with CH_4 composition in biogas is very low, about 50%-60%. Higher engine regime is, lower CH_4 composition in biogas at which power of dual fuel engine exceeds power of diesel engine is. Therefore in many cases, poor biogas without CO_2 removal can be used on dual fuel engines while maintaining the rated power of original diesel engines.

Some investigations have evaluated the use of biogas in concrete dual fuel engine [9], [10], [11], but the available way to improve performance of engine in general at different regimes and different biogas compositions is still limited. Otherwise most of the present application of gaseous fuel engines has been focused on medium or relatively high power, multi-cylinder engines with specially designed combustion systems and exhaust gas treatment systems, but the experiences on small biogas engines were rarely available. However, a significant number of small or low-power single-cylinder engines are currently being operated world-wide. The present study was designed to observe the performance of an a small engine fuelled with biogas under different operating conditions. Because the available diesel engine in the market is very various, thus experimental research on each engine are complex and expensive. Simulation study is appropriate solution to resolve this difficulty.

In previous works [12], [13] we have studied biogas spark ignition engine converted from diesel engines. The present study was focused on biogas diesel dual fuel engine converted from a Vikyno EV2600NB diesel engine. The object was to determine a set of operating conditions to match and accommodate the variable fuel composition, optimizing performance with minimal engine modifications, and suitable for use with biogas supplies in developing countries.

II. Simulation Procedure

Simulation of combustion in combustion chamber of dual fuel engine converted from Vikyno EV2600NB diesel engine is carried out with help of the software ANSYS FLUENT. The detail of meshing and boundary conditions input are presented elsewhere [14].

TABLE I. ENGINE SPECIFICATIONS

Type	EV2600NB
Bore x Stroke (mm)	118 x 108
Displacement (cm^3)	1181
Rated power (HP)/Rated speed (rpm)	20/2200
Maximum torque (kgm)/Engine speed (rpm)	8.92/1400
Compression Ratio	16.5
Combustion chamber	Omega

The base engine for this research was a Vikyno EV2600NB, single cylinder, four-stroke, direct injection (DI), stationary diesel engine. The major engine specifications are given in Table 1. The engine was modified to run on dual fuel mode and its original fuel injection system was maintained for the dual fuel operation.

Fig. 2a and Fig. 2b shows typical results given by the simulation. They illustrate the variation of CH_4 concentration and mean mixture temperature in combustion chamber of dual fuel engine fuelled with biogas containing 60% CH_4 at stoichiometric fuel-air equivalence ratio. As the ignition of dual fuel engine is conducted by pilot diesel jet, the combustion is started at jet tip in omega combustion chamber, instead at its top position as SI engine. The flame propagation is very fast because the mixture is well prepared and ignited by high energy of diesel jet torch.

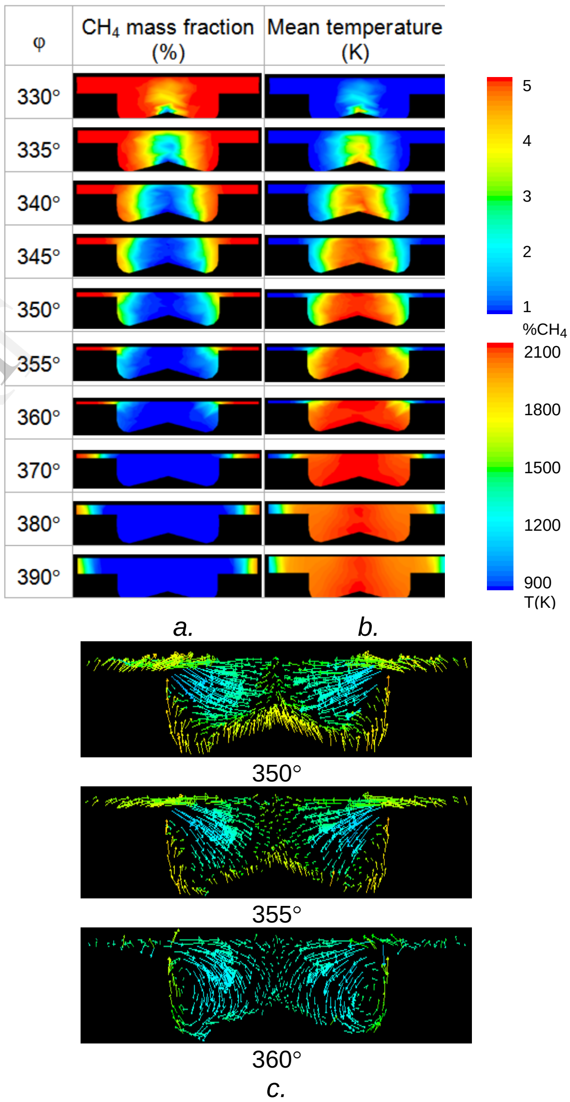


Fig. 2. Variation of CH_4 concetration (a), mean mixture temperature (b) and mixture velocity (c) in combustion chamber of dual fuel engine (Biogas M80C20; n=1400rpm; φ_s=30°; ϕ=1)

Fig. 2c illustrates the fluid velocity field in combustion chamber as the engine fuelled with biogas containing 60% CH_4. It can be seen that the swirl increases significantly as the piston approaches to the TDC.

III. RESULT AND DISCUSSIONS

The effects of ignition timing, fuel-air equivalence ratio, engine speed and biogas composition to the performance of biogas diesel dual fuel engine will be presented in the following section. The effect of compression ratio has been presented else where [15].

A. Effects of advance ignition timing

Fig. 3 shows cylinder pressure diagram of biogas diesel dual fuel engine related to ignition timing. It can be seen that peak pressure increases significantly with increase in advance ignition timing. The maximum cylinder pressure moves toward TDC as increase of ignition timing. This leads to an increase of both compression and expansion works. Gain or loss of indicated cycle work depends on balance of these two works. It can be seen from Fig. 4 that optimum ignition timing is 30° before TDC as biogas diesel dual fuel engine operates at 2000 rpm and fuelled with biogas containing 70% CH_4.

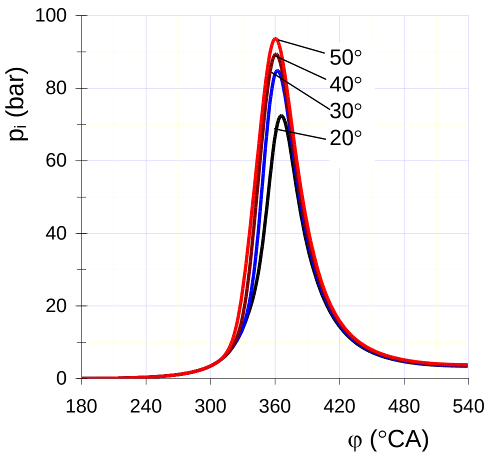


Fig. 3. Cylinder pressure diagrams with various advance ignition timing angle φ_s: 20°, 30°, 40°, 50° (Biogas M70C30;n=2000rpm)

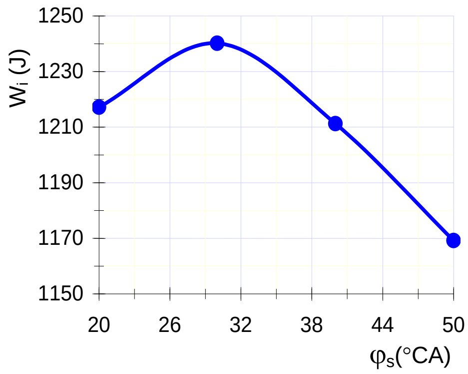


Fig. 4. Effect of advance ignition timing angle on indicated cycle work (Biogas M70C30, n=2200rpm)

B. Effects of fuel-air equivalence ratio

Induction of gaseous fuel, called primary fuel, reduces the consumption of diesel (substitution level). However, when a dual fuel engine operates at part load (low fuel-air equivalence ratio) with high substitution levels, the thermal efficiency is lower than in diesel engines. Negative effects of part load and high substitution levels on dual fuel engine performance are a result of the ignition delay increase and poor flame propagation of the air-gaseous fuel mixture, which in these conditions is closer to the lower flammability limit [16]. Fig. 5a illustrates the variation of CH_4 concentration in combustion chamber of dual fuel engine with crank angle at various fuel air equivalence ratio. Ignition timing was maintained constant. The results show that increase of fuel-air equivalence ratio leads to an increase of CH_4 concentration at the end of combustion process. This causes an increase of mean mixture temperature in expansion stroke. The peak of mean mixture temperature moves toward the left with increase of fuel-air equivalence ratio (Fig. 5b).

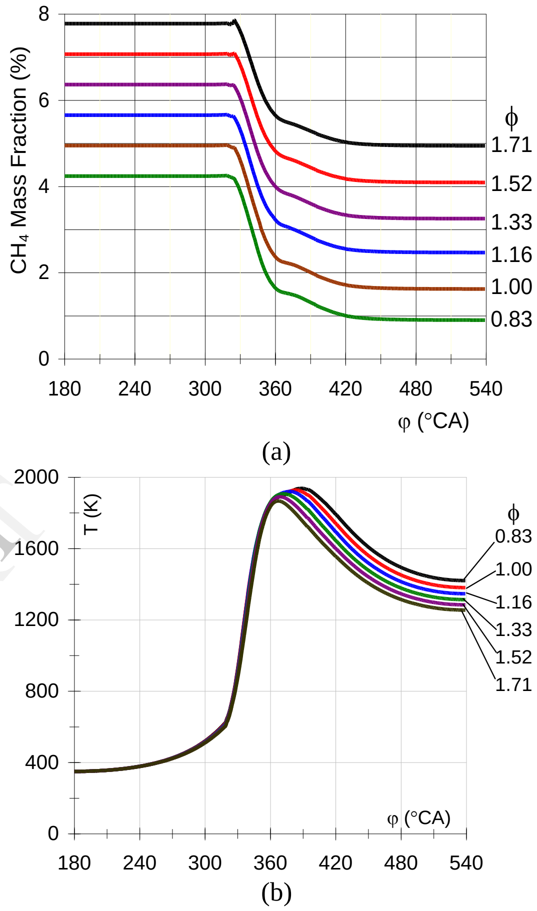


Fig. 5. Variation of CH_4 concentration (a) and mean mixture temperature (b) with crank angle at different equivalence ratio (Biogas M60C40, n=2000rpm, φ_s= 40°)

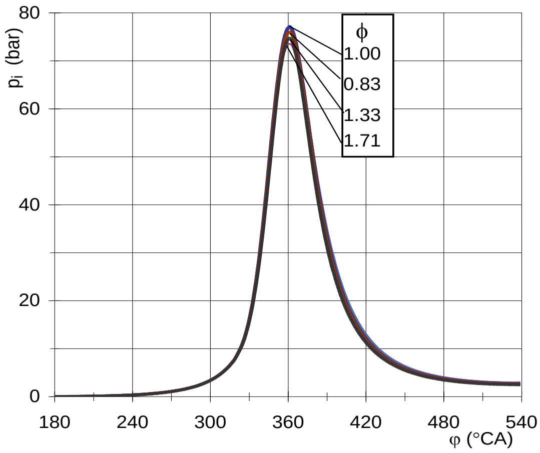


Fig. 6. Effect of equivalence ratio on cylinder pressure diagram (Biogas M60C40, n=2000rpm, φ_s= 40°)

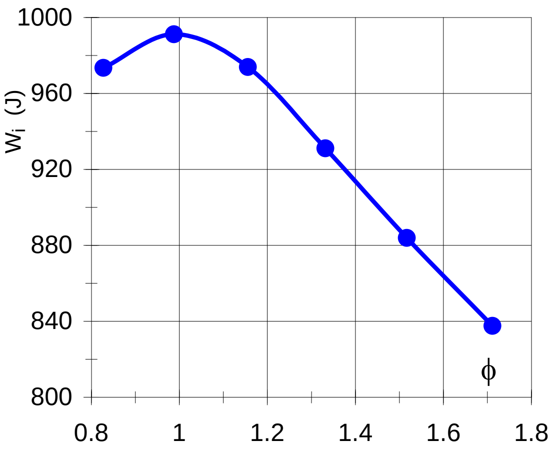


Fig. 7. Variation of indicated cycle work with equivalence ratio (Biogas M60C40, n=2000rpm, φ_s= 40°)

The cylinder pressure diagrams of dual fuel engine fuelled with biogas containing 60% CH_4 are shown in Fig. 6. The variation of indicated cycle work with equivalence ratio is shown on Fig. 7. It is found that equivalence ratio has a great effect on indicated cycle work. The peak of indicated cycle work is obtained at ϕ around stoichiometric value. Mixtures richer or leaner than this point will cause incomplete combustion or slow the burning rate and hence lead to a drop in indicated cycle work.

C. Effects of engine speed

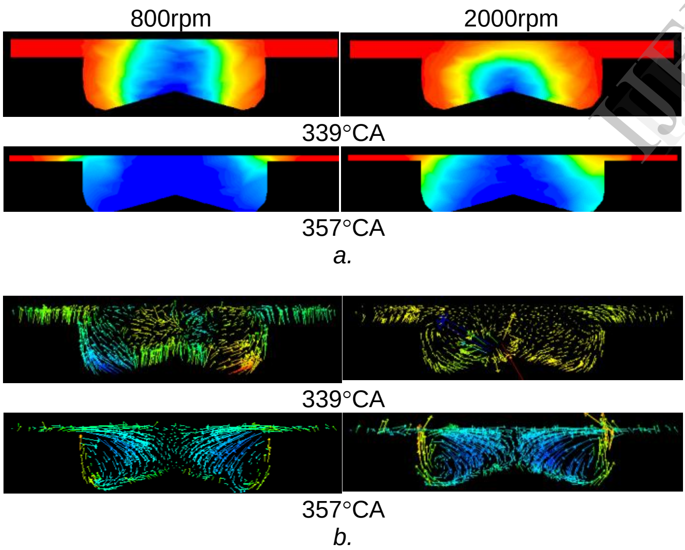


Fig. 8. Comparison of burnt zone (a) and turbulence movement of fluid (b) in combustion chamber at low (800rpm) and high (2000rpm) engine speed

The presence of carbon dioxide in the biogas reduces the burning velocity hence engine speed has significant effect on engine performance. In fact this can be explain via 2 reasons: (1) the higher engine speed, the lower time is for each degree of CA and (2) the higher engine speed, the greater turbulence intensity of flow is for fluid movement in combustion chamber. The first reason leads to a drop of fuel consumption for each CA degree but contrarily the second reason leads to an increase of burning velocity. The engine performance depends on the balance of these two reasons. Fig. 8a illustrates the volume of burnt mixture at 339°CA and 357°CA as engine fuelled with biogas containing 70% CH_4 and running at speed of 800rpm and 2000rpm. Advance ignition angle is 30°CA. Turbulence movement of fluid in combustion chamber can be predicted as shown in Fig. 8b with two above engine speeds. Turbulence intensity increases significantly near TDC. At low engine speed, time for each CA degree is greater than that at high engine speed. This mean that the quantity of fuel burnt corresponding to each CA degree at low engine speed is greater than that at high engine speed. Hence it can be seen that in early phase of combustion, at the same crank position, the lower the engine speed, the larger is the volume of burnt mixture. However near TDC, as combustion velocity is more effected by turbulence movement, the difference of burnt mixture volumes with low and high engine speed is not evident.

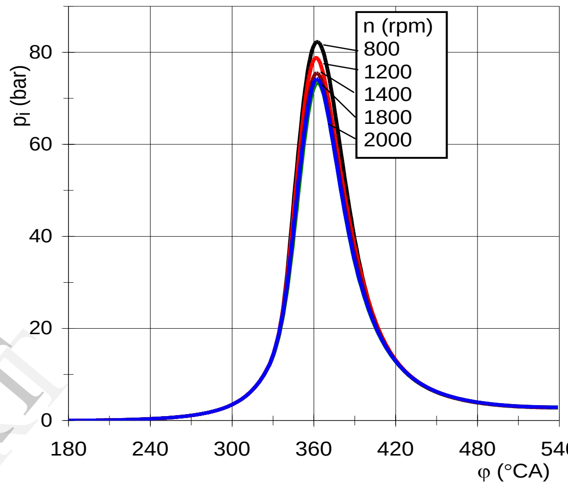
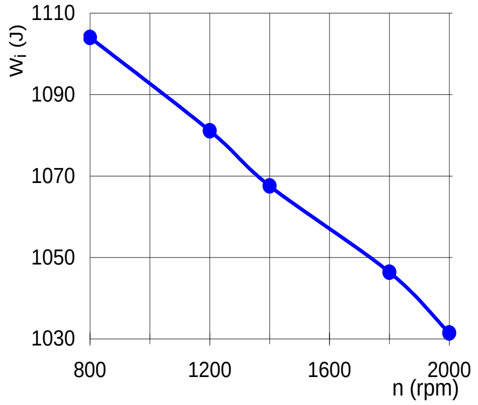


Fig. 9. Effect of engine speed on cylinder pressure diagram (Biogas M60C40;φ_s= 30°; ϕ=1)

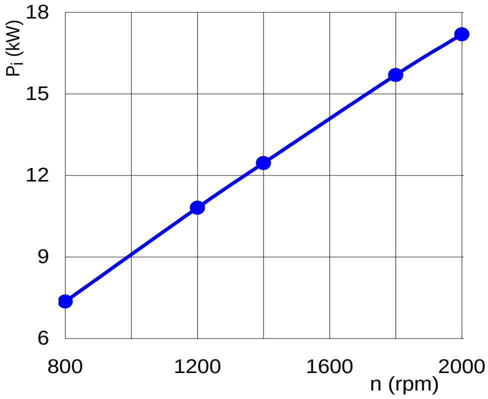
Fig. 9 illustrates the effects of engine speed on cylinder pressure diagrams with biogas containing CH_4 of 60%, stoichiometric mixture and advance ignition angle of φ_s=30° before TDC. We find that as the engine speed goes up, the peak of cylinder pressure decreases due to the reduction in the heat release for each crank angle degree.

Fig. 10a shows the variation of indicated cycle work with engine speed of dual fuelled engine with biogas containing 60% CH_4, equivalence ratio ϕ=1 and injection timing φ_s=30° before TDC. The results show that for engine speed goes up from 800rpm to 2000rpm, the indicated cycle work of the engine decreases about 7%.

Indicated power of engine is proportional to product of indicated engine cycle work and engine speed. As indicated cycle work decreases with an increase of engine speed thus the variation of indicated power with engine speed (namely indicated characteristic curve) is not linear. Fig. 10b illustrates the characteristic curve of dual fuel engine fuelled with biogas containing 60% CH_4, stoichiometric mixture and advance ignition angle 30° before TDC.



(a)



(b)

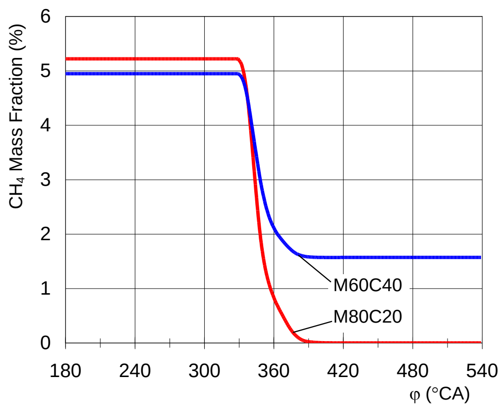
Fig. 10. Variation of indicated cycle work (a) and maximum indicated power with engine speed (Biogas M60C40; n=2000rpm;φ_s=30°)

D. *Effect of quality of biogas*

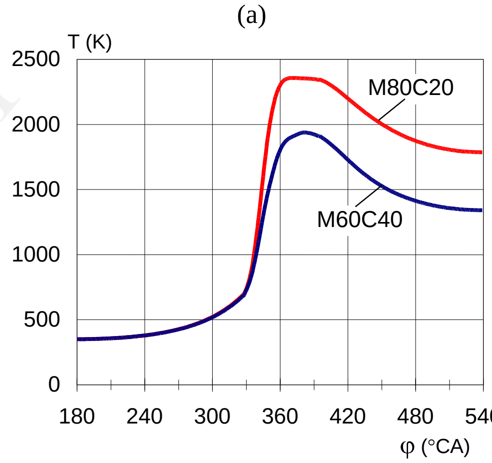
The presence of carbon dioxide in the biogas reduces the burning velocity which ultimately affects the performance of the engine [11]. This particularly becomes significant at low loads due to its influence in burning rate inhibition [17], [18], [19]. However, the operating conditions can be optimized [12], and a fast burning engine design could improve performances [20]. Therefore, new solutions are to be found at the level of the combustion process to increase efficiency.

In order to examine the effects of biogas quality on engine performance, simulations were conducted with CH_4 fractions from 60% up to 80% with a constant equivalence ratio ϕ=1. Fig. 11a shows the variation of CH_4 concentration in combustion chamber of the dual fuel engine running at speed of 1400rpm and fuelled with biogas containing 80% and 60% CH_4. It can be seen that with the increase in carbon dioxide in biogas, the flame propagation rate is reduced, leading to a longer combustion duration and a correspondingly lower rate of CH_4 consumption. This confirms the experimental results of Huang et al [3]. With stoichiometric mixture and biogas containing 80% CH_4, almost CH_4 is burnt at the end of combustion process. While at the same conditions but with biogas containing 60% CH_4, a significant quantity of fuel remains unburnt at the end of combustion process due to decrease of burning velocity.

Comparison of mean mixture temperature in combustion chamber of dual fuel engine fuelled with biogas containing 60% and 80% CH_4 running at speed of 1400rpm is shown on Fig. 11b. It is possible to see that at lower CH_4 composition, due to the heat release rates were lowered in comparison with higher CH_4 content in biogas, the maximum mean mixture temperature decreases. At the same equivalence ratio ϕ=1, maximum mean mixture temperature of biogas containing 80% CH_4 is 400K higher than that of 60% CH_4 at engine speed of 1400rpm. The differences of maximum pressure in these 2 cases are 17bar and 20bar corresponding to engine speed of 800rpm and 1400rpm respectively (Fig. 12a and Fig. 12b). Consequently the indicated cycle work falls as CH_4 content in biogas decreases from 80% to 60%.



(a)

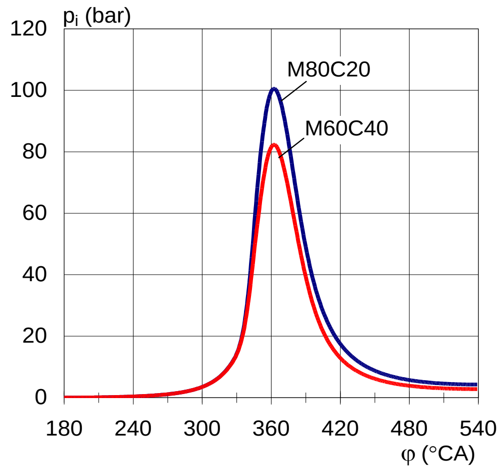


(b)

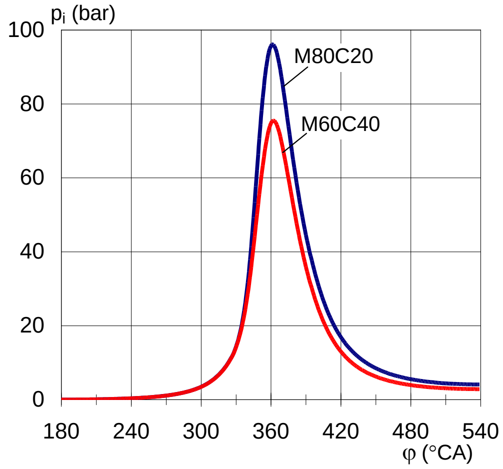
Fig. 11. Effect of CH_4 composition in biogas on fuel consumption rate (a) and on mean mixture temperature (b) (n=1400rpm;φ_s=30°; ϕ=1)

Variation of indicated cycle work with engine speed at φ_s=30°, equivalence ratio ϕ=1 and biogas containing 60%, 70% and 80% CH_4 is shown in Fig. 13. CH_4 composition in biogas has a significant effect on engine performance. As composition of CH_4 in biogas falls from 80% to 60%, indicated cycle work decreases 220J and 150J as engine speed of 800rpm and 2000rpm respectively. The differences of indicated engine power are as results (Fig. 14). As engine speed 2200rpm and CH_4 composition in biogas decreases from 80% to 70% and 60%, the indicated power of dual fuel engine decreases 1.3kW and 2.7kW respectively.

A comparative study between simulation results and experimental data was presented in [14]. The brake effective power can be obtained by multiplying indicated power (Fig. 14) with mechanical efficiency and then it is compared with brake power given by experiment at the same operation conditions. The results were very well consistence [14].



(a)



(b)

Fig. 12. Effect of CH_4 composition in biogas on cylinder pressure diagram at engine speed of n=800rpm (a) and n=1400rpm (b) (φ_s=30°; ϕ=1)

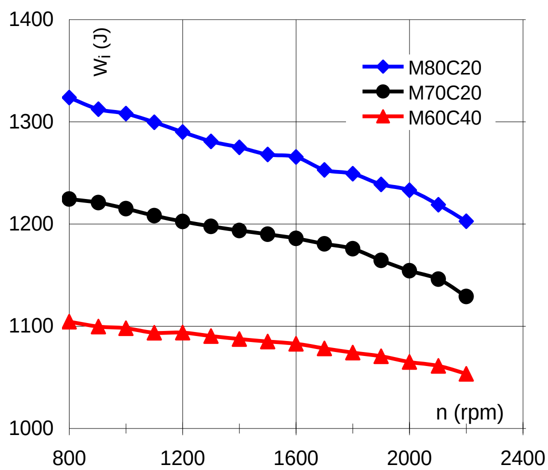


Fig. 13. Effect of CH_4 composition in biogas on variation of indicated cycle work with engine speed (φ_s=30°; ϕ=1)

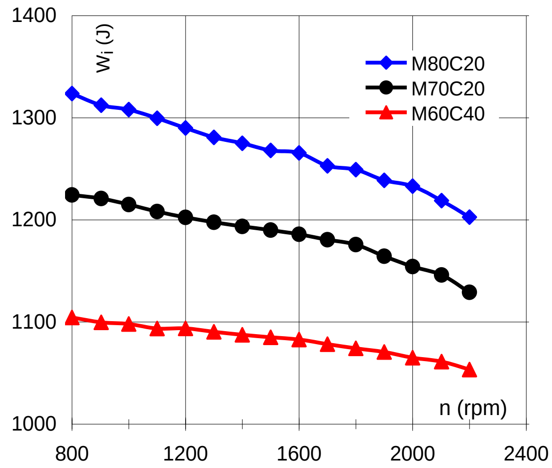


Fig. 14. Effect of CH_4 composition in biogas on variation of indicated cycle work with engine speed (φ_s=30°; ϕ=1)

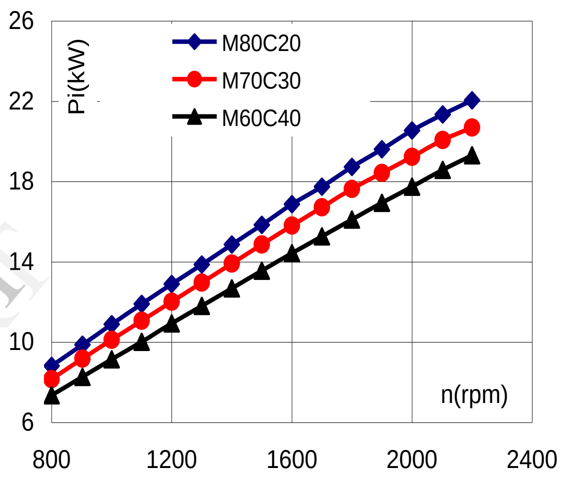


Fig. 15. Effect of CH_4 composition in biogas on variation of maximum indicated power with engine speed (φ_s=30°; ϕ=1)

IV. CONCLUSIONS

The following conclusions may be drawn from the results of the present study:

- Rated brake power of dual fuelled engine with poor biogas can be higher than that of original diesel engine. This can occur even though with poor biogas containing 50%-60% CH_4 as engine speed exceeds 1800rpm.
- Optimum ignition timing is 30°CA before TDC as the engine fuelled with biogas containing 70% CH_4 at engine speed of 2000rpm.
- Mean mixture temperature in expansion stroke increases with rich mixture but unburnt fuel in exhaust gas increases with lean mixture. Peak of indicated cycle work is recorded as fuel-air equivalence ratio is around stoichiometric value.
- With same fuel-air equivalence ratio ϕ=1, for CH_4 composition in biogas change from 80% to 60%, maximum mean mixture temperature decreases 400K

at engine speed of 1400rpm. This leads to a drop of maximum cylinder pressure of 17bars and 20bars corresponding to engine speed of 800rpm and 1400rpm respectively.

- With stoichiometric fuel-air equivalence ratio, as CH_4 composition in biogas change from 80% to 60%, indicated cycle work decreases 220J and 150J at engine speed of 800rpm and 2000rpm respectively. Indicated power is non linear relationship with engine speed.

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