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Soot Emission Analysis in Combustion of Bio-gas Diesel Dual Fuel Engine

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Abstract

Soot emission in bio-gas diesel dual fuel engine has been analyzed by numerical simulation with 2-stape soot formation model of Magnussen. The result shows that soot formation mainly occurred in diffusion combustion phase of diesel pilot jet. Soot peak value is proportional to the first peak value of ROHR, and it is found at around the same crank angle position with the second peak of ROHR. At a given engine speed and diesel content in the fuel, the highest soot peak value is obtained with slightly rich mixture whereas soot concentration in exhaust gas increases monotonically with increasing equivalence ratio. Increasing diesel content in the fuel increases both soot peak value and soot concentration in exhaust gas. At a given equivalence ratio and diesel content in the fuel, engine speed has a moderate effect on soot formation rate but a significant effect on soot combustion rate. Soot concentration in the exhaust gas practically vanished as equivalence ratio under 0.98 and 15% diesel content in the fuel. This is the ideal operation regime of biogas diesel dual fuel engine in view of soot emission control.

Keywords

Biogas; Renewable energy; Biogas-diesel dual fuel engine; Soot emission; Magnussen model

1. Introduction

Depletion in fossil fuel resources and a growing awareness of the serious effects of global warming have forced numerous governments all over the world to formulate and enforce application of renewable energy to reduce greenhouse gas emission. Bio-fuels that are environment friendly present attractive behaviors in combustion and emissions (Yoon & Lee, 2011; Nabi & Hustad, 2012). Among bio-fuels, bio-gas is considered as an abundant resource of renewable energy in developing countries. Application of this fuel on diesel engines for transportation, irrigation and non-grid power generation purposes is preferred due to its high thermal efficiency. This is especially important not only in the developing countries where meeting the growing demand of fossil fuels is a major economic challenge, but also in the developed countries where greenhouse gas emission reduction is an important commitment according to the COP21 convention.

Biogas, mainly consisting of CH_4 and CO_2 has almost the same properties as natural gas. However, as a renewable fuel, the use of bio-gas presents two profits: it provides an alternative source of energy for fossil fuel that saves and protects the environment from the harmful greenhouse gases, carbonic and methane that would be emitted into the atmosphere.

Due to low energy density and cetane number, bio-gas cannot be used as a single fuel in compression ignition (CI) engine, but it can be used in the dual fuel mode. The engine, which uses conventional diesel fuel and bio-gas fuel, is referred to as bio-gas diesel dual fuel engine. It is basically a modified diesel engine in which bio-gas fuel, called the primary fuel, is the main source of energy input to the engine. The primary bio-gas fuel is compressed with air, but does not auto ignite due to its high self-ignition temperature. A small amount of diesel, usually called the pilot, is injected as in a normal diesel engine near the end of compression of the primary fuel-air mixture. This pilot diesel fuel, auto ignites first and acts as a deliberate source of ignition for the combustion of the gaseous fuel-air mixture. The pilot diesel fuel contributes only a small fraction of the engine power output. Load of the engine is controlled mainly by bio-gas fuel (Ga, Nam, Tien & Tu, 2014; Ashok, Ashok & Kumar, 2015). Thus, the combustion process and emission in a dual fuel engine is complex as it combines the features of spark ignition engines and compresses ignition engines (Wagemakers & Leermakers, 2012).

Soot is primarily identified as major constituent of the emission emerging from diesel combustion. The presence of diesel pilot injection in biogas dual fuel engine should be thus considered in view of pollution emission. Several technologies have so far been developed to reduce soot emission from diesel combustion combustion (Zhu, Cheung & Huang, 2010; Cheng, Cheung, Chan, Lee, Yao & Tsang, 2008). The combination use of gaseous fuels and conventional diesel fuel to reduce soot emission has been reported by numerous researchers.

Tomita et al. carried out research on a hydrogen diesel dual fuel engine (Tomita, Kawahara, Piao, Fujita & Hamamoto, 2001) and they reported a reduction in soot concentration to near zero levels at all diesel injection timings and at all equivalence ratios of hydrogen. Tsolakis, Hernandez, Megaritis and Crampton (2005) obtained similar results of reduction in the total mass of soot emissions in combustion of hydrogen diesel dual fuel. Besides, a considerable number of studies (Saravanan, Nagarajan, Sanjay, Dhanasekaran & Kalaiselvan, 2008) and (Saravanan & Nagarajan, 2008) reported a reduction in soot emissions when CH_4 was used as a primary fuel in a CH_4 diesel dual fuel engine. It is generally believed that the more carbon a fuel molecule contains, the more likely the production of soot is by the fuel during combustion (Tree & Svensson, 2007). Moreover, the chemical structures, such as aromatics, C=C and cyclic molecules, are regarded to have increased soot producing tendency (Smith, 1981). In dual fuel mode, some amount of diesel is replaced by the gaseous fuels, which are higher ratio of hydrogen to carbon; thus, soot emission is consequently reduced.

Thus, in view of soot emission reduction, it is particularly a positive merit in favor of dual fuel engines converted from existing diesel engine. Although there are numerous research works on experimental and theoretical investigations concerning the dual fuel operating mod, a comparatively lower number of specific research works on soot emission of biogas diesel dual fuel engines are found. The past investigations concentrated mainly on the performance and fuel consumption characteristics for biogas diesel dual fuel engines. Mustafi and Raine (2008), studied emissions from a dual fuel engine fuelled with natural gas and biogas; however, dual fuel combustion characteristics and their effects on emissions were not presented there. An effort using biogas biodiesel in dual fuel application are found in references (Yoon & Lee, 2011; Luijten & Kerkhof, 2011) where engine performance, combustion and emission characteristics are investigated.

Although the results of these above primary works show that bio-gas diesel dual fuel operation seem to be regarded as an appropriate way for simultaneously controlling emissions from diesel engines and saving petroleum based diesel fuel, it is necessary to investigate into more detail the reduction characteristics of exhaust emissions with special emphasis on soot formation in order to establish bio-gas as a real alternative fuel for diesel engines.

The emission characteristics of dual fuel engine are affected by both the pilot diesel fuel as ignition source and the primary premixed bio-gas fuel. The experimental investigations of combustion and emission of such kind of engine is very complicated and expensive. Thus 3-D modeling takes into account the interactions between different phenomena including turbulent flow, spray, combustion, soot formation and naturally the geometry of combustion chamber presents more advantage. It allows a precise investigation of the problem as it provides all required properties at any point within the combustion chamber and at any time. In other words, numerical modeling makes it possible to explore combustion characteristics and soot emission that may be difficult and/or expensive to achieve with experiments.

Fundamental aspects concerning CFD one dimension simulation of combustion process and soot formation in diffusion flames have been carried out by (Vignon & Ga, 1995). They found out that soot formation given by Magnussen model is fitted well to experimental data. An integral model of soot formation based on Magnussen model has been established by Bui et al. to investigate soot formation in combustion of an experimental diesel engine (Ga, Mai, Brun & Vignon, 1999). Good coherence between model and experiments has been observed. The model was then applied to successfully calculate soot formation in combustion of furnace (Ga & Lu, 2002). Multidimensional CFD simulation of combustion in an IDI diesel engine has been studied by (Zellat, Rolland & Poplow, 1990) using KIVA code. Strauss, Schweimer & Ritscher, 1995 had studied the combustion and pollutant formation processes in diesel engine using SPEED CFD code. Sera, Bakar & Leong (2003) formulated dual fuel engine simulation model and studied the combustion process of a natural gas diesel dual fuel engine, and good levels of agreement were obtained between measured and predicted results. Singh, Kong, Reitz, Krishnan & Midkiff (2004) studied the combustion and emissions of a natural gas diesel dual fuel engine and showed that dual fuel engine combustion results in significant reduction in soot emissions. In brief, CFD simulation is an efficient method to explore combustion and soot emission of compression ignition engine (Barzegar, Shafee & Khalilarya, 2013).

The paper aims to analyze the effect of different operation parameters of biogas diesel dual fuel engine on soot emission. The issue of reduction of soot emission is explored by controlling diesel/biogas ratio, fuel -air equivalence ratio and swirl in combustion chamber of a biogas diesel dual fuel engine, which is converted from a Vikyno EV2600NB single cylinder diesel engine. The research is carried out by numerical simulation with help of FLUENT CFD code and soot emission model of Magnussen.

2. Numerical Simulation

In the present study, the Computational Fluid Dynamics (CFD) code FLUENT was used to model a complex combustion phenomenon and soot formation in biogas diesel dual fuel engine. The numerical modeling was taking into account the effect of turbulence via k-ε model, droplet breakup of diesel jet via wave model. Soot formation is modeled by Magnussen mechanism (Magnussen & Hjertager, 1977). The mechanism can be depicted by two participating processes, the formation and oxidation of soot. In this study, the two-step soot model was applied. With respect to the model, in addition to solving the transport equation for soot mass fraction, the model also requires the solution of another transport equation for radical nuclei concentration. The net rate of soot generation is the balance of soot formation and soot combustion. Soot formation is proportional to radical nuclei concentration whereas soot combustion. Soot formation is proportional to fuel concentration, oxygen concentration and ε/k (ε: turbulence energy dissipation; and k: turbulence kinetic energy).

The net rate of nuclei generation is given by the balance of the nuclei formation rate and the nuclei combustion rate. The nuclei formation rate is proportional to fuel concentration and $e^{-E/RT}$. The nuclei combustion rate is proportional to soot combustion rate.

Numerical simulation is performed on a single cylinder, Vikyno EV2600NB DI diesel engine having an omega bowl shaped piston. The engine specifications are given in Table 1.

Calculations are theoretically carried out on a closed system from intake valve closure at 0°CA to exhaust valve opening at 360°CA with different operation conditions. The diesel fuel is injected via a single hole injector at top of combustion chamber. Initial conditions for pressure and temperature in the combustion chamber are 0.95bar and 350K, respectively. Bio-gas contains 70%CH_4 and 30%CO_2. The amount of diesel pilot injection or diesel content in the fuel namely D15, D20 and D25 corresponds to 15%, 20% and 25% energy contribution of diesel in total energy of fuel mixture, respectively.

Table 1. Engine specifications

Cylinder volume (cm^3)	1181
Bore (mm)	118
Stroke (mm)	108
Number of cylinders	1
Compression ratio	16.5
Rated output (HP)/rate speed (rpm)	20/2200
Maximum output (HP)/maximum speed (rpm)	25/2400

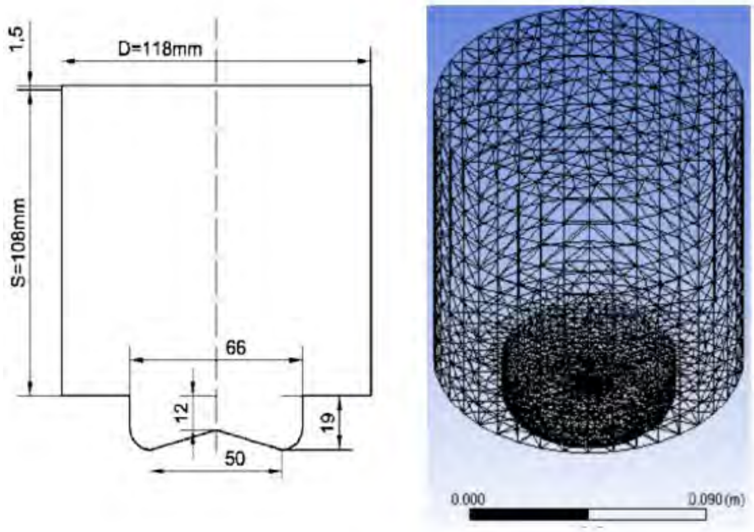

Figure 1. Calculating space and meshing

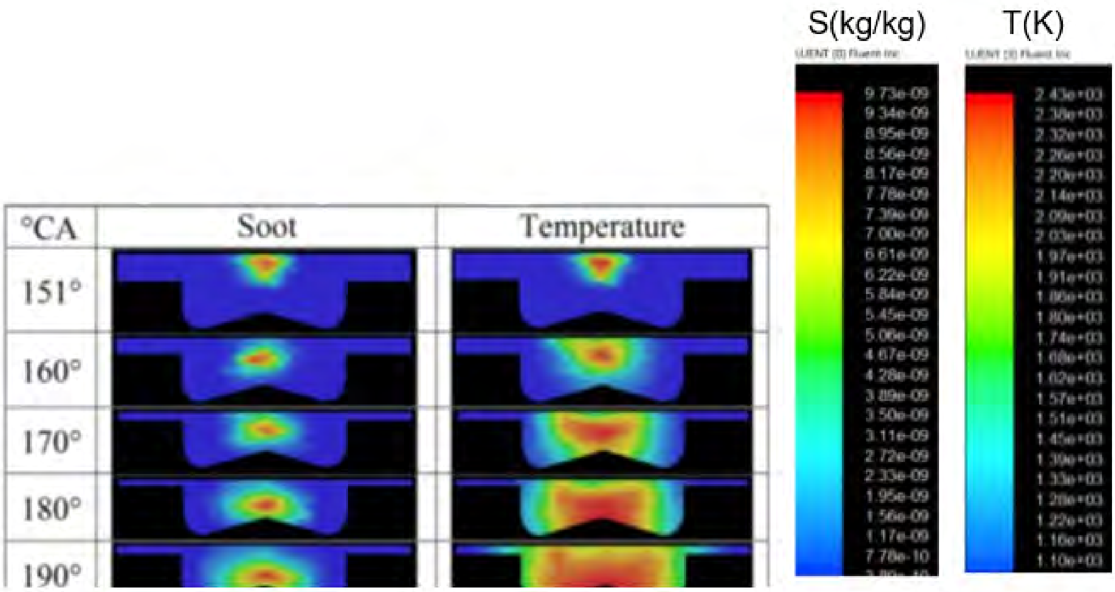

Figure 2. Soot and temperature distribution in combustion chamber (D15, φ=1.1, n=2400 rpm)

Fig. 1 presents geometry of combustion chamber and calculating space. Dynamic meshing of in-cylinder space of the engine is generated automatically by ANSYS code. There are 9279 nodes in the space with high density in combustion chamber.

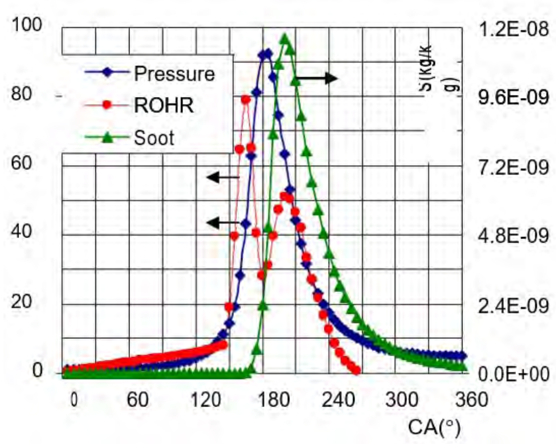

Figure 3. Rate of heat release, pressure and soot concentration vscrank angle of biogas diesel dual fuel engine combustion (D15, φ=1.2, n=2400rpm)

The biogas-air mixture is suggested to be homogenous. The combustion process is modeled via partially premixed model. Calculation procedure of effects of fuel composition and operation condition on engine performance have been presented in detail in (Ga, Hai, Tu & Hung, 2015). The distributions of soot concentration and temperature are illustrated in Fig. 2 for a typical operation condition. The result shows that soot concentration is accumulated in high temperature region where diesel concentration is available.

3. Result and Discussion

3.1. Analysis of Soot Emission in Dual Fuel Combustion

As it has been mentioned above, soot emission is the balance of soot formation and soot combustion that depend on temperature, fuel concentrations, oxygen concentration and turbulence intensity in the combustion chamber. These effects can be analyzed via the rate of heat released (ROHR) in dual fuel engine.

Fig. 3 presents ROHR, in-cylinder pressure (p) and soot concentration (S) as functions of crank angle (CA). ROHR curve has two peaks in which the first peak is higher than the second. Peak of pressure is located between these two ROHR peaks. Soot concentration is continuously increasing with increase in pressure and reaches a peak at around the same crank angle position with the second peak of ROHR. After this peak, soot concentration is decreased gradually due to oxidation.

In fact, ROHR curve is mainly due to fuel consumption via three burning phases of the dual fuel combustion process: (1) rapid premixed combustion of the pilot diesel fuel, (2) diffusion combustion of pilot diesel fuel and premixed combustion of the primary biogas and (3) premixed combustion of biogas and the left over pilot diesel fuel. During the first phase of combustion, a small part of pilot diesel fuel and a little quantity of the biogas entrained by the spray is consumed. ROHR in this phase is mainly due to the diesel fuel burned. Soot formation mechanism in dual fuel mode in this phase is thus not quiet different from diesel fuelling mode. The second phase is due to the burning of a maximum part of the biogas and a part of the rest of the pilot diesel fuel. Biogas burns mainly during this phase of the combustion. Soot formation in this phase increases due to diffusion combustion of diesel droplets and soot agglomeration at high combustion temperature. Finally, the third phase is due to the combustion of the rest of the two fuels that are not burned in the last phase. In this phase soot formation decreases while soot combustion increases; hence the resultant soot generation in expansion stroke decreases gradually. Soot concentration at the end of expansion stroke, namely soot in exhaust gas, is the main concern of the research.

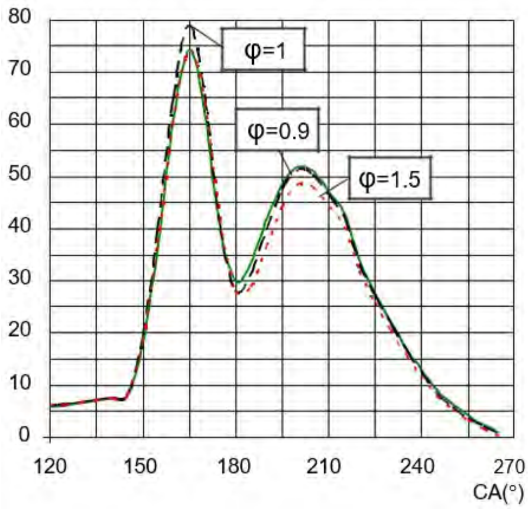

Figure 4. RORH vs crank angle with different equivalenceratios (D15, n=2400rpm)

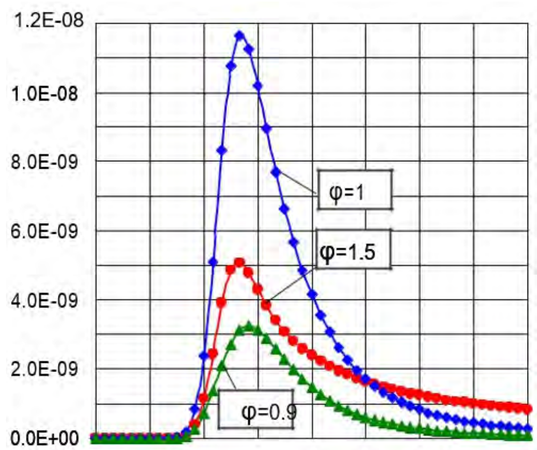

Figure 5. Sootconcentration vs crank angle with different equivalence ratios (D15, n=2400rpm)

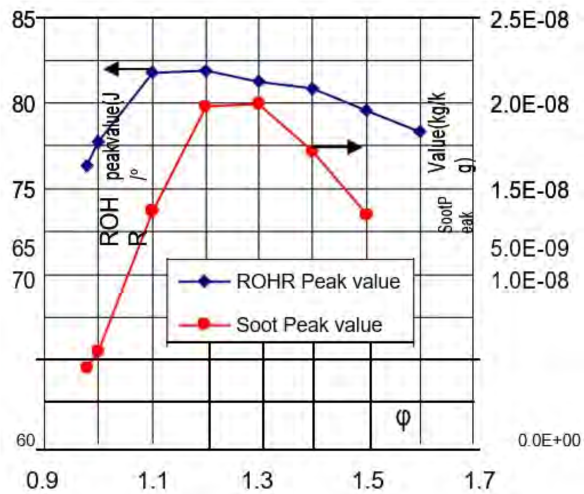

Figure 6. ROHR peakvalue and soot peak value vs equivalence ratio (D20,n=2400rpm)

Fig. 4 shows ROHR traces of combustion for three different equivalence ratios of 0.9, 1.0 and 1.5 at constant engine speed of 2400rpm and 15% diesel content in the fuel. At a given diesel content in the fuel and at constant engine speed, it can be observed from the figure that the peaks of ROHR decrease in the cases of equivalence ratios of 1.5 and 0.9 relative to equivalence ratio of 1.0. As equivalence ratios of the fuel-air mixture are higher or lower than stoichiometric, it is suggested that this may result in the slower flame propagation speeds than the maximum speeds achievable during stoichiometric combustion. When the fuel-air mixture is relatively close to stoichiometric, the peak of soot concentration is much higher as shown in Fig. 5. The peak soot values are $3.25.10^{-9}$, $1.16.10^{-9}$, $5.06.10^{-9}$ kg/kg for φ=0.9, 1.0 and 1.5, respectively. Rich or lean mixture results in a decrease in soot concentration due to low ROHR. In any case of operation, soot concentration is proportional to the first peak of ROHR. With lean mixture, both fuel concentration and combustion temperatures are low, resulting in a low soot concentration. With rich mixture, fuel concentration is high but soot concentration is not increased due to low combustion temperature. In general, at a given operation regime, the maximum soot concentration is obtained in mixture slightly higher than stoichiometric as shown in Fig. 6.

Fig. 7 presents a general comparison to verify the interactions of soot emissions with diesel content in the fuel. It confirmed the above observation that soot formed at early stages of combustion due to diffusion combustion of diesel fuel. At the same equivalence ratio, high diesel content in the fuel satisfied soot formation increases. At the same equivalence ratio of φ=1, soot peak value is lower by about 30%, and soot in exhaust gas is lower by about 60% as diesel content in the fuel decreased from 25% to 15%. Significant increase of soot emission in exhaust gas as diesel content increases can be explained by the fact that diesel is the highly carbonated fuel providing largely radical nuclei particularly the rich fuel zones inside the combustion chamber. Moreover, diesel fuel contains aromatic compounds favored soot formation.

It can be speculated that the formation of the majority of soot was caused by the quantity of diesel fuel injected into the combustion chamber, which was minimized in the case of dual fuel operations. Soot particles form primarily from the carbon in the diesel fuel. In biogas, the carbon/hydrogen ratio is lower when compared to diesel, so soot formation is less in low content of diesel in fuel as a result. Besides, soot oxidation processes, being a function of temperature, in dual fuel combustion are more important than those in diesel engines, due to higher in-cylinder temperatures that result from increased heat release rates. Therefore, reduction in soot emissions can be attributed to (1) a direct consequence of flame temperature reduction in early phase of combustion and the lowered concentration of diesel, (2) increased oxidation of soot nuclei/precursors in the soot forming region by increasing turbulence intensity and the enhanced concentrations of OH in combustion region.

3.2. Effects of fuel concentration

Fig. 8 shows the reductions in X_f.exp(-E/RT) implying a beneficial effect of equivalence reduction (X_f: fuel concentration; E: activation energy; R: universal gas constant; T: temperature). At a constant engine speed 2100rpm, a reduction in equivalence ratio from φ=1.3 to φ=1.0 causes up to 25% reduction in X_f.exp(-E/RT). Otherwise, it can be seen on Fig. 9 that the reduction of equivalence ratio results in a net increase in $X_{O2}.\varepsilon/k$ (where X_{O2}: concentration of oxygen). A reduction in equivalence ratio from φ=1.3 to φ=1.0 causes up to 17% increasing in $X_{O2}.\varepsilon/k$. However, this effect is less at the lean mixture due to reduction of turbulence intensity resulting from reduction of flame speed. At the same condition, the increase of $X_{O2}.\varepsilon/k$ only 6% as φ decreases from 1.3 to 0.9. Therefore, as previously discussed, it could be suggested that a decrease in equivalence ratio promotes soot oxidation and reduces initial soot formation at a given engine speed. These effects are more important as mixture richer than stoechiometric.

Fig. 10a and Fig. 10b show the effect of equivalence ratio on the total soot mass concentration at a given diesel content in the fuel. The peak of soot concentration (S_{max}) is increased as equivalence ratio increased from 0.98 to around 1.2 and 1.3. As mixture is richer than this value, the soot concentration peak is decreased very fast but contrarily soot concentration in the exhaust gas ($S_{exhaust}$) increased. This is due to the low temperature of incomplete combustion, which is not favorable for soot oxidation.

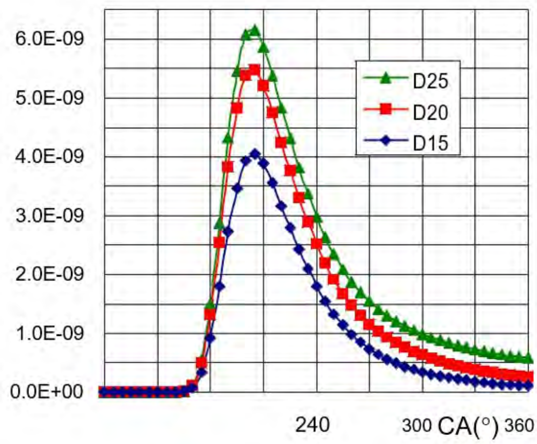

Figure 7. Soot concentration vs crank anglewith different diesel content in thefuel (n=2400rpm, φ=1)

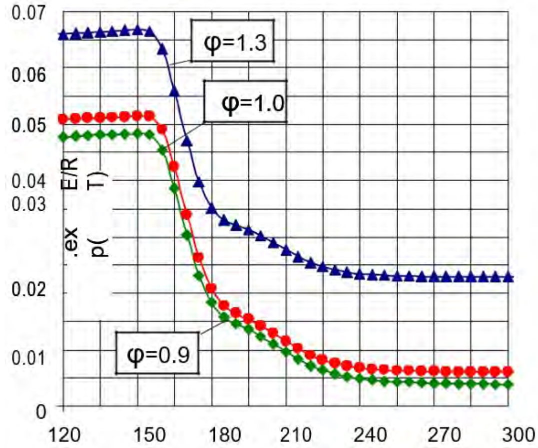

Figure 8. Variation of Xf.exp(-E/RT) in function of crank angle
with different equivalence ratio (D20, n=2100rpm)

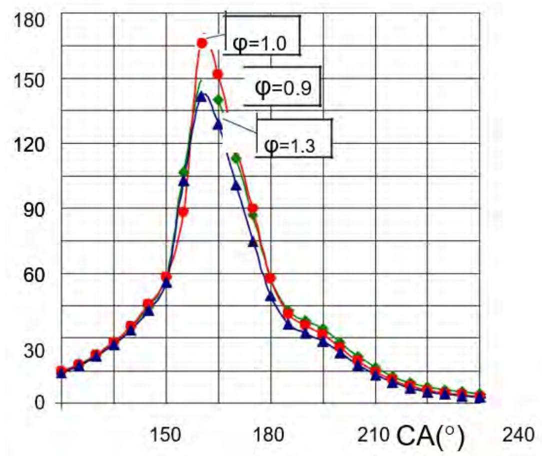

Figure 9. Variation of XO2ε/k in function of crank angle with
different equivalence ratio (D20, n=2100rpm)

Fig. 11 shows maximum soot concentration and soot concentration in exhaust gas plotted for various equivalence ratios and diesel contents in the fuel. The soot concentration in exhaust gas appears to be monotonically increasing with increasing equivalence ratio of the mixture, whereas maximum soot concentration presents a peak at φ=1.2, 1.25 and 1.27 for 15%, 20% and 25% diesel content in the fuel, respectively. One could notice that the peak of soot concentration in function of equivalence ratio tends to have a richer mixture as diesel content increased. The soot is then oxidized in the leaner regions of the mixture so that most of the soot is burnt before exhaust to the atmosphere. Soot concentration in the exhaust gas is practically vanished as equivalence ratio is under 0.98 and 15% diesel content in the fuel. This is the ideal operation regime of biogas diesel dual fuel engine in view of soot emission control.

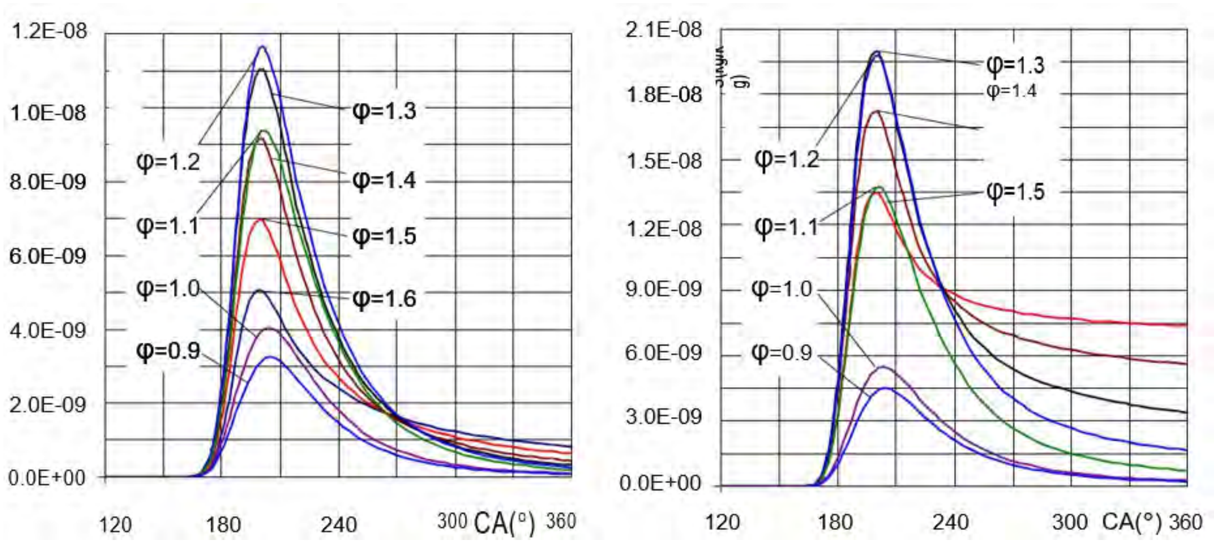

Figure 10. a: Effects of equivalence ratio on soot concentration (D15, n=2400rpm)
b: Effects of equivalence ratio on soot concentration (D20, n=2400rpm)

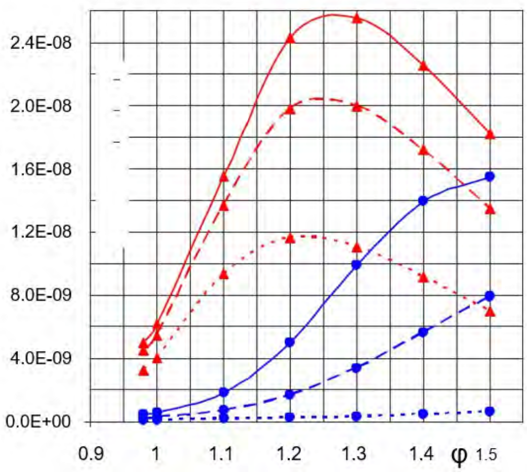

Figure 11. S_{max} (æ), $S_{exhaust}$ (—) vsequivalence ratio with different diesel content in the fuel (: D25, –: D20, ···: D15,n=2400rpm)

3.3. Effect of swirl

Fig. 12 and Fig. 13 show that rate of increasing of X_f.exp(- E/RT) is expected to be smaller than that of X_{O2} .ε/k as increasing engine speed. One possible explanation for this is that at higher engine speed the mixture displaced due to the swirl severely affects the ε/k despite the increase in in-cylinder temperatures arising from improvement of combustion effects to exp(-E/RT). It can be seen that as engine speed increases, ε/k is increased particularly during the second phase of combustion and hence enhanced soot combustion. This suggests that at a given equivalence ratio and diesel content in the fuel, increasing of engine speed results a reduction in soot emission.

The variation in soot emissions with crank angle at different engine speeds is portrayed by Fig. 14. It can be observed from the figure that soot emissions decrease with the increase of engine speed at a given equivalence ratio. With equivalence ratio φ=1 and 20% diesel content in the fuel, peak values of soot concentration are $2.1.10^{-8}$, $1.3.10^{-8}$ and $1.0.10^{-8}$ for engine speed of 1800rpm, 2100rpm and 2400rpm respectively, i.e. a decreasing of 50% peak value of soot concentration as engine speed increases from 1800rpm to 2400rpm. Soot concentration in exhaust gas is reduced by about 60% as the same condition. Increasing the engine speed improves soot combustion due to increasing ε/k, and this is the reason that might have an impact on soot emission.

In general, the above positive results are obtained in this simulation research in favor of biogas diesel dual fuel engine on soot emissions; it can be concluded that it is a promising technology for controlling soot emissions in existing conventional compression ignition engines with minor engine hardware modifications, thus saving fossil fuel and saving the human life from the hazardous effects of exhaust gas pollutants from the conventional diesel engines.

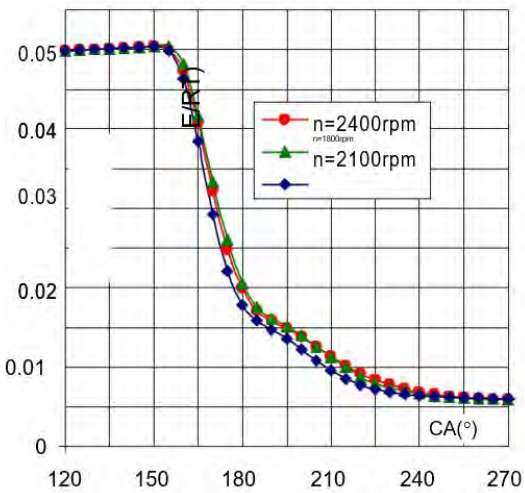

Figure 12. Variation of X_f.exp(-E/RT) in function of crank angle with different engine speeds (D20, φ=1)

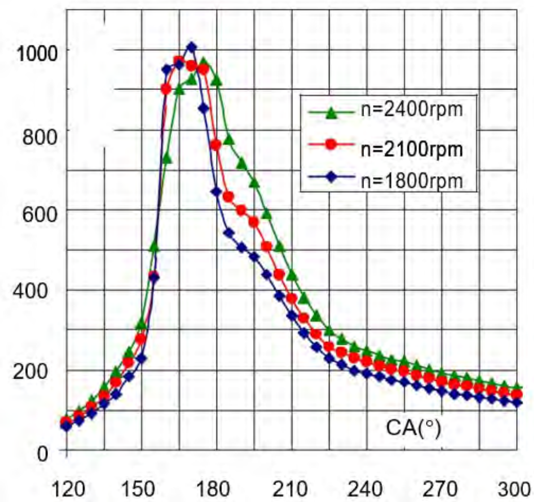

Figure 13. Variation of $X_{O2}\varepsilon$/k in function ofcrank angle with different engine speeds (D20, =1)

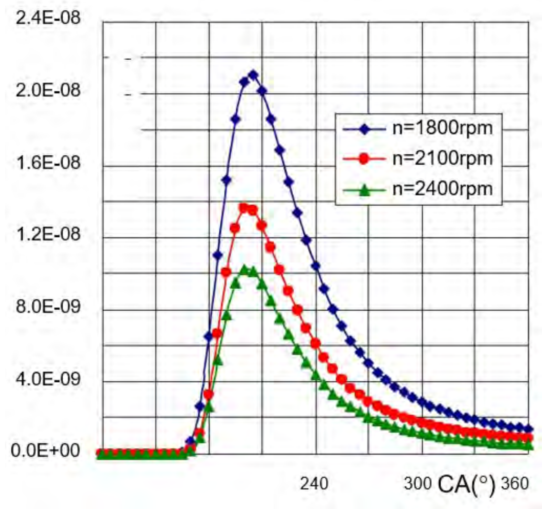

Figure 14. Soot concentration vs crank angle with different engine speeds (D20, φ=1)

4. Conclusions

The soot emission characteristics of the biogas diesel dual fuel engine have been analyzed by numerical simulation. The following conclusions are drawn:

1. Soot formation mainly occurred in the phase of diffusion combustion of diesel pilot jet in biogas diesel dual fuel engine. In any case, soot peak value is proportional to the first peak of ROHR, and it is found at around the same crank angle position of the second peak of ROHR curve.
2. In biogas diesel dual fuel engine, soot formation rate is not quiet different form diesel engine, but soot combustion rate is much higher. Soot emission can be reduced by shortage pilot injection time and increasing biogas quantity in combustion for a higher value of the second peak of ROHR.
3. Increasing diesel content in the fuel increases both soot peak value and soot concentration in exhaust gas. At the same equivalence ratio of φ=1, soot peak value is lower by about 30% and soot in exhaust gas is lower by about 60% as diesel content in the fuel decreased from 25% to 15%.
4. At a given engine speed and diesel content in the fuel, the highest soot peak value is obtained with slightly rich mixture. In the case of biogas diesel dual fuel engine, the highest soot peak value is obtained with φ in the range of 1.2 and 1.3 as diesel content in the fuel varies from 15% to 25%. However, soot concentration in the exhaust gas increases monotonically with equivalence ratio.
5. At a given equivalence ratio and diesel content, engine speed has a moderate effect on soot formation but a considerate effect on soot combustion. Soot concentration in exhaust gas reduces by about 60% and soot peak value reduces by about 50% as engine speed increases from 1800rpm to 2400rpm at a given equivalence ratio and diesel content in the fuel.
6. Soot concentration in the exhaust gas is practically vanished as φ=0.98 and 15% diesel content in the fuel. This is the ideal operation regime of biogas diesel dual fuel engine in view of soot emission control.

References

1. Ashok, B., Ashok, S. D., & Kumar, C. R. (2015). LPG diesel dual fuel engine – A critical review. *Alexandria Engineering Journal,54*(2), 105-126.

2. Barzegar, R., Shafee, S., & Khalilarya, S. (2013). Computational fluid dynamics simulation of the combustion process, emission formation and the flow field in an in-direct injection diesel engine. *Thermal Science,17*(1), 11-23.

3. Bui Van Ga, Bui Thi Minh Tu/ Environmental Science and Sustainable Development, ESSD

4. Cheng, C., Cheung, C., Chan, T., Lee, S., Yao, C., & Tsang, K. (2008). Comparison of emissions of a direct injection diesel engine operating on biodiesel with emulsified and fumigated methanol. *Fuel,87*(10-11), 1870-1879.

5. Ga, B. V., & Lu, L. V. (2002). Integral Model for Soot Formation Calculation of Turbulent Diffusion Flames in Industrial Furnaces. In *6th European Conference on Industrial Furnaces and Boilers*. Portugal: Estoril-Lisbon.

6. Ga, B. V., Hai, N. V., Tu, B. T., & Hung, B. V. (2015). Utilization of Poor Biogas as Fuel for Hybrid Biogas-Diesel Dual Fuel Stationary Engine. *International Journal of Renewable Energy Research,5*(4), 1007-1015.

7. Ga, B. V., Nam, T. V., Tien, L. M., & Tu, B. T. (2014). Combustion Analysis of Biogas Premixed Charge Diesel Dual Fuelled Engine. . *International Journal of Engineering Research & Technology (IJERT,3*(11), 188-194.

8. Ga, B.V., Mai, P. X., Brun, M., Vignon, J.M. (1999). Modele integral de flammes turbulentes de diffusion pour le calcul de la combustion Diesel. *Entropie,216,* 52-59.

9. Luijten, C., & Kerkhof, E. (2011). Jatropha oil and biogas in a dual fuel CI engine for rural electrification. *Energy Conversion and Management,52*(2), 1426-1438.

10. Magnussen, B., & Hjertager, B. (1977). On mathematical modeling of turbulent combustion with special emphasis on soot formation and combustion. *Symposium (International) on Combustion,16*(1), 719-729.

11. Mustafi, N. N., & Raine, R. R. (2008). A Study of the Emissions of a Dual Fuel Engine Operating with Alternative Gaseous Fuels. *SAE Technical Paper Series*.

12. Nabi, M. N., & Hustad, J. E. (2012). Influence of oxygenates on fine particle and regulated emissions from a diesel engine. *Fuel,93*, 181-188.

13. Saravanan, N., & Nagarajan, G. (2008). An experimental investigation of hydrogen-enriched air induction in a diesel engine system. *International Journal of Hydrogen Energy,33*(6), 1769-1775.

14. Saravanan, N., Nagarajan, G., Sanjay, G., Dhanasekaran, C., & Kalaiselvan, K. (2008). Combustion analysis on a DI diesel engine with hydrogen in dual fuel mode. *Fuel,87*(17-18), 3591-3599.

15. Sera, M. A., Bakar, R. A., & Leong, S. K. (2003). CNG Engine Performance Improvement Strategy Through Advanced Intake System. *SAE Technical Paper Series*.

16. Singh, S., Kong, S., Reitz, R. D., Krishnan, S. R., & Midkiff, K. C. (2004). Modeling and Experiments of Dual-Fuel Engine Combustion and Emissions. *SAE Technical Paper Series*.

17. Smith, O. I. (1981). Fundamentals of soot formation in flames with application to diesel engine particulate emissions. *Progress in Energy and Combustion Science,7*(4), 275-291.

18. Strauss, T. S., Schweimer, G. W., & Ritscher, U. (1995). Combustion in a Swirl Chamber Diesel Engine Simulation by Computation of Fluid Dynamics. *SAE Technical Paper Series*.

19. Tomita, E., Kawahara, N., Piao, Z., Fujita, S., & Hamamoto, Y. (2001). Hydrogen Combustion and Exhaust Emissions Ignited with Diesel Oil in a Dual Fuel Engine. *SAE Technical Paper Series*.

20. Tree, D. R., & Svensson, K. I. (2007). Soot processes in compression ignition engines. *Progress in Energy and Combustion Science,33*(3), 272-309.

21. Tsolakis, A., Hernandez, J. J., Megaritis, A., & Crampton, M. (2005). Dual Fuel Diesel Engine Operation Using H2. Effect on Particulate Emissions. *Energy & Fuels,19*(2), 418-425.

22. Vignon, J.M. & Ga, B.V. (1995). Calcul de flammes de diffusion verticales par un modele integral. In *Joint Meeting the French and German Section of the Combustion Institute.* France

23. Wagemakers, A., & Leermakers, C. (2012). Review on the Effects of Dual-Fuel Operation, Using Diesel and Gaseous Fuels, on Emissions and Performance. *SAE Technical Paper Series*.

24. Yoon, S. H., & Lee, C. S. (2011). Experimental investigation on the combustion and exhaust emission characteristics of biogas–biodiesel dual-fuel combustion in a CI engine. *Fuel Processing Technology,92*(5), 992-1000.

25. Yoon, S. H., & Lee, C. S. (2011). Experimental investigation on the combustion and exhaust emission characteristics of biogas–biodiesel dual-fuel combustion in a CI engine. *Fuel Processing Technology,92*(5), 992-1000.

26. Zellat, M., Rolland, T., & Poplow, F. (1990). Three Dimensional Modeling of Combustion and Soot Formation in an Indirect Injection Diesel Engine. *SAE Technical Paper Series*.

27. Zhu, L., Cheung, C., Zhang, W., & Huang, Z. (2010). Emissions characteristics of a diesel engine operating on biodiesel and biodiesel blended with ethanol and methanol. *Science of The Total Environment,408*(4), 914-921.