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BIOGAS –DIESEL DUAL FUEL ENGINE EXHAUST GAS EMISSIONS

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ABSTRACT

The work is aimed at developing combustion model along with writing simulation code of biogas- diesel duel fuel engine to study the effect of emissions. An exhaust gas simulation code is written for determining the mole fraction of different constituents of exhaust gases .The proportion of CO_2 in the biogas-diesel mixture affecting the mole fraction of exhaust species when the bio-gas is burnt along with diesel also be simulated and results are presented. This paper examine the effect of variations of carbon dioxide in bio-gas on the emissions of the engine, mole fractions of CO_2, H_2O,N_2, O_2, CO, H_2,H,O,OH & NO.

Key words: Bio-gas, diesel, mole fraction, exhaust emission.

1. INTRODUCTION

Now-a-days the whole world is facing energy crisis. Available sources of liquid fuel will be depleted after few years. Commonly used gaseous fuels of I.C. engines are, LPG, CNG. But they have their own limitations. LPG is explosive, CNG is expensive. Methane separated from biogas is equivalent to CNG but economical than CNG. In this situation biogas can serve as best alternative fuel [1]. Bio-gas is one of the promising renewable energy sources. Which is compound gas consisting mainly of methane (CH_4) and carbon dioxide (CO_2), biogas also contain traces of nitrogen, hydrogen, oxygen and hydrogen sulphide. When diesel engine runs on biogas the combustion is poor as compare to diesel fuel, the major reason of poor combustion in the presence of carbon dioxide in the biogas [2].

Exhaust gas emission in an internal combustion engine can be controlled by different methods, i.e., by modifying the engine design, treating the exhaust gas and by fuel modifications. Robert [3] conducted studies for the change in engine designs like changing inlet and out let valve opening time. Hunter [4] studied the exhaust gas emissions using secondary air along with fuel injection to achieve better mixing. Mc Connell [5] tried to improve exhaust emission using fuel properties and antismoke additives. Exhaust gas treatment methods using catalytic converters to oxidize and reduce exhaust emission have treated at Volvo lambda [6].sharma[7] used after burners with a spark plug in the exhaust muffler for the combustion of unburnt hydrocarbons and exhaust gas recirculation methods to reduce NOx emissions. Calton [8] has also used fuel additives like cynaruric acid. Hansen [9] investigated the combustion of ethanol and blends of ethanol with diesel fuel. It was observed that the effects of adding ethanol to diesel fuel were increased ignition delay, increased rates of premixed combustion, increased thermal efficiency and reduced exhaust smoke. Czerwinski [10] used a rapeseed oil, ethanol and diesel fuel blend and compared the heat release curves with diesel fuel. It was observed that the addition of ethanol caused longer ignition lag at all operating conditions. During the literature survey studies very little information is available on complete exhaust analysis of diesel engine using diesel and biogas fuels. Thus, there is an urgent need to develop cheap and simple methods of reducing exhaust gas emissions levels from the compression ignition engines. Therefore, in this study authors have used biogas, low cost (compared to the pure diesel), &renewable alternative fuel along with diesel for reducing exhaust gas emissions of diesel engine.

The main objectives of the study are:

(i) To analyze the biogas-diesel duel fuelled engine exhaust pollutants.

(ii) To see the effect of using biogas on the emission levels from biogas-diesel duel fuelled engine exhaust.

(iii) To determine mole fractions of constituents of exhaust emission. The ratio of number of moles of a substance in a mixture of solution to the total number of moles is the mole fraction.

Selection Criteria of the Biogas as combination of Diesel:

Bio-gas used in the current study has been selected on the basis of some chemical and thermodynamic properties as described in table.1.

Property	Diesel	Bio-gas
Calorific value, kJ/kg	44 500	35000-40000
Self ignition temperature, □C	725	700
Boiling point range, □C	260-320	300
Ignition delay period, s	0.002	-
Flame propagation rate, cm/s	10.5	40
Flame temperature, □C	1400	900
Specific gravity at 32□C	0.83	0.8
Sulphur content by weight, %	0.8	0.5

Lower self ignition temperature, boiling point and high flame propagation rate of bio-gas have been found more suitable alternative fuel for use in diesel engine along with diesel.

Bio-gas, with its remarkable combustion properties in the conventional internal combustion engine, appears to be the best transportation fuel of the future if it can be produced economically. Importantly, it can be used in existing internal combustion engines, yielding unprecedented efficiencies and low levels of exhaust pollution [11].

Most research in dual fuel engine has concentrated on defining the extent of dual fueling and its effect on emissions and performance [12, 13]. Natural gas in combination with diesel was tried and found to be very effective in NOx reduction but engine operation can suffer from high hydrocarbons (HC) emissions and poor performance, especially at high loads [14,15]. The auto ignition of methane was studied experimentally to obtain ignition delay data as a function of engine cylinder pressure and temperature by Sandia National Laboratory [16,17]. Karim, et al. [18,19] And Gunea [20], et al. concluded that at low out puts, much of the primary gaseous fuel remain unburned leading to high hydrocarbon (HC) and CO emissions which is mainly due to very lean operation and a weak ignition source. At high loads a large amount of gaseous fuel admission results in uncontrolled reaction rates near the pilot spray causing rough engine operation. Experimental investigation on a LPG – diesel dual fuel engine by Poonia, et al., [21, 22] revealed that at low loads, the brake thermal efficiency is always lower than diesel values but is better at high loads. Also, at low outputs increasing the pilot quantity and intake temperature improves the thermal efficiency. The HC and CO emissions were found to increase in the dual fuel mode. Daisho, et al., [23] showed that emission characteristics and brake thermal efficiency of a dual engine using natural gas can be improved by reciprocating exhaust gas, increased brake thermal efficiency with increase intake temperature is also reported.

This paper involves the simulation program for determining the mole fraction of each of the exhaust species when the biogas is burnt along with diesel and the results are presented. The proportion of biogas in the biogas-diesel blend affecting the mole fraction of the exhaust species is also simulated. In running diesel engine it is necessary to feed 15 to 20 percent diesel along with gas and in the situation the consumption of gas is about 0.42 to 0.50 M^3 per horse power per hour.

Problem formation: For any of the fuel combinations, it becomes imperative to optimize the combination depending upon not only the performance but also depending upon exhaust emissions. For this if the optimal performance and reduced emissions if simulated, can practically be achieved effectively. Here in this paper an effort is made to simulate the exhaust emissions in dual fuel mode. The optimal combination of fuels depending upon the exhaust can be effectively found out and practically applied. This program code is even valid with slight variation to the other combination of fuels. Thermodynamics is able to predict the equilibrium state that results from burning a fuel-air mixture given only the initial conditions. Combustion is a chemical reaction between a fuel and oxygen, which is accompanied by the production of a considerable amount of heat. The composition of the exhaust gas produced is a function of temperature as well as equivalence ratio (ratio of actual fuel air ratio to theoretical fuel air ratio). A lean mixture has $\varphi < 1$. A rich mixture has $\varphi > 1$. The mixture is said to be stoichiometric if $\varphi = 1$. Many components are present in the exhaust gas because of dissociation of some species. The heat of combustion of a fuel is defined as the heat transferred out of a system per unit mass or mole of fuel when the initial and final states are at the same temperature and pressure [24]. Based on the combustion stoichiometric theory, a computer program had been developed for blended fuels to calculate the mole fractions of the exhaust gases . Thermodynamic data for elements, combustion products and many pollutants are available in a compilation published by the National Bureau of Standards, called the JANAF (Joint Army-Navy-Air Force) tables (1971). For single component fuels the data presented by Stull, Westrum and Sinke, (1969) is in the same format as that of JANAF tables. A compilation by Rossini (1953) is useful for hydrocarbon fuels at temperatures as high as 1500K

Inputs to the Program: The fuel is to be specified in terms of the C, H, O and N atoms in the fuel. For the blend of two fuels considered i.e., Diesel and Hydrogen, the percentage with which they blend in the mixture also has to be specified. The other parameters that need to be specified are equivalence ratio, pressure and temperature. For the calculation of equilibrium constant, the data for constants is considered from JANAF tables. The molar-air fuel ratio is calculated from the number of Carbon, Hydrogen, Nitrogen and Oxygen atoms present in the fuel.

The mole fractions for all the remaining species is obtained in terms of y_3, y_4 and y_6 i.e, the mole fractions of N_2 , O_2 and H_2 respectively as

$$\left.\begin{array}{l} y_7=C_1*(y_6)^{0.5}; \\ y_8=C_2*(y_4)^{0.5}; \\ y_9=C_3*(y_4)^{0.5}*(y_6)^{0.5}; \\ y_{10}=C_4*(y_4)^{0.5}*(y_3)^{0.5}; \end{array}\right\} \quad (8\ a)$$

where

$$\left.\begin{array}{l} C_1 = K_1 / P^{1/2} ; \\ C_2 = K_2 / P^{1/2} ; \\ C_3 = K_3 ; \\ C_4 = K_4 ; \end{array}\right\} \quad (8\ b)$$

Where log K_p value are obtained from JANAF tables

RESULTS & CONCLUSIONS

Simulated results:

In this paper the exhaust emission results are presented to validate the prediction.

The figure1. Show the change in mole fraction of CO_2, for various percentages of bio-gas substitutions at different constant equivalence ratios .It can be noted that for all equivalence ratios except for 1.2, the mole fraction of CO_2 increases continuously for 20% to 40% bio-gas substitution.

As the temperature increases, the mole fraction of CO_2 decreases since the dissociation increases with temperature. The figure 2 show the change in mole fraction of H2O, for various percentage of bio-gas substitutions at different constant equivalence ratios .It can be noted that for all equivalence ratio, the mole fraction of H_2O increases continuously for 20% to 40% bio-gas substitution. As the percentage bio-gas increases, the mole fraction of H2O also increases; highest value of H2O is seen at equivalence ratio of 1.2.

Figure 3, shows the change in mole fraction of NO for various percentages of bio-gas substitutions for different constant equivalence ratios for temperature of 1500 K .With the increase in percentage of bio-gas, the NO decreases slightly. As the value of equivalence ratio is increasing, the mole fraction of NO is decreasing for temperature of 1500 K. NO present in the exhaust is very less as the concentration of N atoms is very less. The lack of dissociation of N_2 molecules is the result of the strong triple covalent bond. Equilibrium concentrations of NO are rather flat and peak in the lean region falling rapidly in the rich region. In most combustion systems, NO levels are well below the equilibrium concentrations because of relatively slow formation reaction as the temperature increases, the mole fraction of NO increases due to increase in dissociation of N_2.

As shown in figure 4, figure 5 the decreasing behavior of O_2, OH respectively was observed, and the trend was the same as that was observed in case of NO.

Figure6. Shows the results for CO against the percentage bio-gas substitution. It can be noted that as the bio-gas percentage is increasing the mole fraction of CO decreases. CO is the minor species in the lean combustion and as the mixture gets richer the mole fraction of CO increases due to incomplete combustion of carbon. Further, as the temperature increases CO_2 dissociates to form CO and hence mole fraction of CO increases as temperature increases.

The decreasing behavior of N_2 was observed in figure 7.as % of CO2 increases in bio-gas.

The decreasing behavior of H_2, H& O with increasing CO2 of bio-gas can be observed from figures 8,9 &10 respectively. Experimental values are shown from fig.11 to fig.15

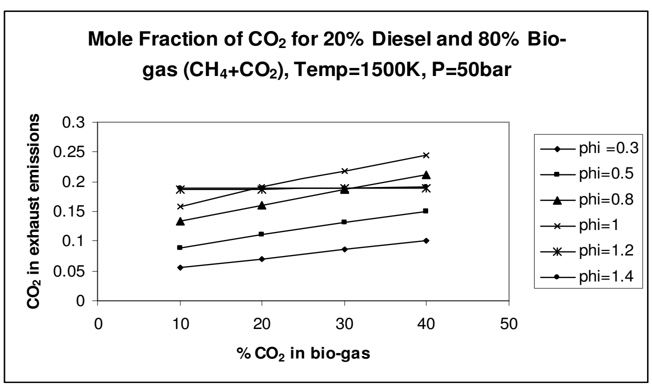

Fig.1 Effect of CO2 in bio-gas fuel substitution on CO2 in exhaust emissions

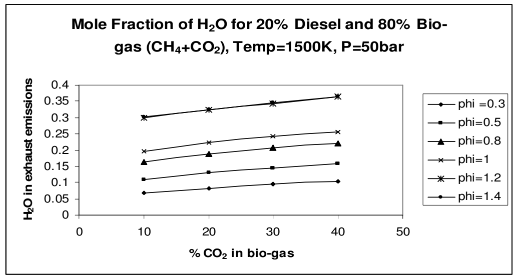

Fig.2 Effect of CO2 in bio-gas fuel substitution on H2O in exhaust emissions

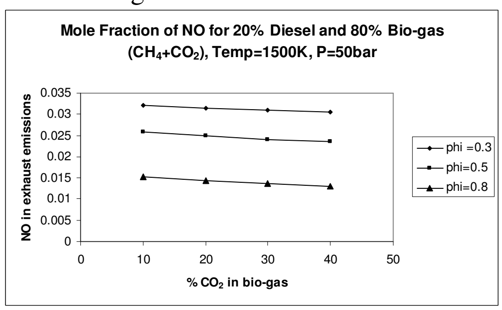

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Fig.3 Effect of CO2 in bio-gas fuel substitution on NO in exhaust emissions

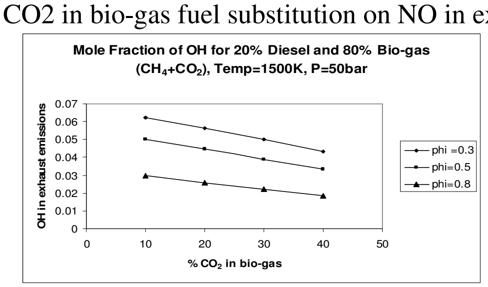
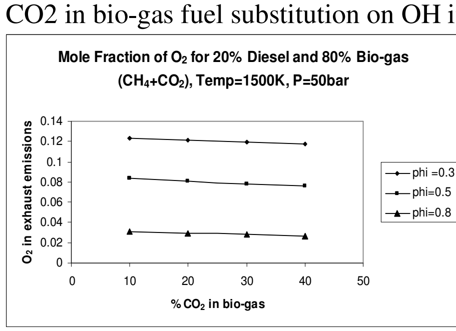


Fig.4 Effect of CO2 in bio-gas fuel substitution on OH in exhaust emissions



CONCLUSIONS

1. Compression ignition engines of the dual-fuel type, fuelled with various gaseous fuel resources, produce less exhaust emissions than conventional diesel engines without any substantial increase in operating and capital cost. Smoke and particulate
Emissions from dual-fuel engines generally tend to be much lower than with the corresponding diesel operation.
2. For all equivalence ratios except for 1.2 the mole fraction of CO_2 increases continuously from 10% & 40% bio-gas substitution.
3. It is seen that the NO emission decreases as the amount of biogas Introduced increases owing to the presence of CO_2. NOx i.e. NO+ NO_2 emissions are very much dependent on combustion chamber temperature.EGR is one of the most effective techniques currently available for reducing NOx emission in internal combustion diesel engines. NOx emissions are mainly affected by two factors: (i) the presence of oxygen in the charge; and (ii) the reaction temperature
4. Carbon monoxide occurs only in engine exhaust. It is the product of incomplete combustion due to insufficient amount of air in air- fuel mixture. As the bio-gas percentage increases the mole fraction of CO increases, and for more than 1 equivalence ratio, the increase is sharp.
5. The mole fraction of H_2O increases with bio-gas substitution; this decreased the peak combustion temperatures.

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