© Applied Science Innovations Pvt. Ltd., India Carbon – Sci. Tech. 8/2(2016)21-28

Carbon – Science and Technology

ISSN 0974 – 0546

http://www.applied-science-innovations.com

RESEARCH ARTICLE Received:10/03/2016, Accepted:15/04/2016

Investigation on the performance and emission parameters of dual fuel diesel engine with mixture combination of hydrogen and producer gas as secondary fuel

A. E. Dhole(1, a, *), R. B. Yarasu(1), D.B.Lata(2)

1) Department of Mechanical Engineering, Government College of Engineering, Amravati, Maharashtra, India.
2) Department of Mechanical Engineering, Birla Institute of Technology, Mesra, Ranchi 835215, India.

Abstract: This study presents experimental investigation in to the effects of using mixture of producer gas and hydrogen in five different proportions as a secondary fuel with diesel as pilot fuel at wide range of load conditions in dual fuel operation of a 4 cylinder turbocharged and intercooled 62.5 kW gen-set diesel engine at constant speed of 1500 RPM. Secondary fuel Substitution is in different percentage of diesel at each load. To generate producer gas, the rice husk was used as source in the downdraft gasifier. The performance and emission characteristics of the dual fuel engine are compared with that of diesel engine at different load conditions. It was found that of all the combinations tested, mixture combination of PG:H_2=(60:40)% is the most suited one at which the brake thermal efficiency is in good comparison to that of diesel operation. Decreased NO_x emissions and increased CO emissions were observed for dual fuel mode for all the fuel combinations compared to diesel fuel operation.

Keywords- Alternative fuels, Dual fuel engine, Hydrogen, Producer gas

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1 Introduction: The use of alternative fuel can reduced the dependence on petroleum based fuel which is a march towards energy safety. Hydrogen is thought to be the most viable alternative fuel for vehicle because of its clean, high efficiency and reproducibility characteristics. It requires smaller ignition energy, has a wider fire range and faster burning speed in comparison with gasoline and diesel. Amongst the other gaseous fuels, producer gas derived from biomass gasification is a healthier option as an environment friendly fuel. This fuel gas, in addition to being CO_2 neutral, generates lesser quantity of undesirable emissions [1]. As India is an agricultural country, it has huge variety of biomass feed stock available in enormous quantity. Since these are obtainable locally, biomass gasifier based power generation may be an appropriate option for decentralized power generation in many parts of the country. In the current context of the petroleum fuel, this recognizes for better utilization of these resources by thermo chemically converting in to producer gas [2]. Important properties of hydrogen and producer gas with its measured compositions are given in Table (1) [3].

Several researchers have carried out works on either hydrogen or producer gas only as a secondary fuel in diesel engines. Dhole et al. [3] compared the effect of hydrogen and producer gas as secondary fuels on performance and emissions of a dual fuel diesel engine. The drop in average value of thermal efficiency from 32.35% on diesel to 28.7% dual fuel mode using pigeon pea stalks as biomass fuel was observed by Das et al. [4]. However, thermal efficiency with wood chips and corn cabs in dual fuel mode was comparable to diesel. Singh et al. [5] investigated that by using diesel and refined rice bran oil in different proportion and producer gas from a wood gasifier in dual fuel mode and mixed fuel mode at different loads, the brake thermal efficiency decreases as compared to pure diesel. Lata et al. [6]

observed significant and eco-friendly performance of an engine using mixture of hydrogen and LPG while experimenting with hydrogen, LPG, and a mixture of LPG plus hydrogen in various proportions in different combinations as secondary fuels and diesel as a pilot fuel.

Saravanan et al. **[7]** found improvement in brake thermal efficiency with H_2 as an enrichment medium and diesel as an ignition source. Gomes Antunes et al. **[8]** achieved higher fuel efficiency in hydrogen-fueled engine by approximately 43% as compared to 28% in the conventional diesel engine due to direct injection of hydrogen. Banapurmath et al. **[9, 10]** used producer gas as secondary fuel with diesel, honge oil, rice bran, neem oil in dual fuel mode which resulted in lower brake thermal efficiency than single fuel operation. It was proved by Ramadhas et al. **[11, 12]** that the existing diesel engine is capable of running successfully in dual-fuel mode of operation using coir-pith and wood chips derived producer gas.

Table (1): Important properties of Hydrogen and Producer gas with its composition

Sr. No.	Properties	Diesel	Producer gas (PG)	Hydrogen (H_2)
1.	Lower Heating Value (kJ/kg)	42,800	6000	1,20,000
2.	Minimum Ignition energy (mJ)	--------	--------	0.26
3.	Flame speed (cm/s)	2.0-8.0	20-30	265-325
4.	Flammability limit (% vol in air)	0.6-7.5	7.0-21.6	4-75
5.	Flammability Limit (Equi. Ratio)	0.6-2.0	--------	0.1-7.1
6.	Diffusion Coefficient (cm^2/s)	--------	--------	0.61
7	Type of gasifier	Downdraft, batch feeding		
8	Feeding	Manual		
9	Fuel consumption	7kg/h (approximately) of rice husk		
10	Hopper capacity	100 kg (approximately) of rice husk		
11	Gas cooling medium	Water		
12	Generated producer gas compositions.	CO = 23.0 ± 4%, H_2 = 21.6± 3%, CO_2 = 10.2 ±4%, N_2 = 43.1 ± 3%, CH_4 = 2.1 ±3% .		

Roy et al. **[13]** analyzed the performance and exhaust emissions of a supercharged producer gas-diesel dual engine for the hydrogen content in producer gas. At the lower end of the optimum fuel-air equivalence ratio ø=0.42, the engine power with the high hydrogen content producer gas was 12% higher whereas, at the upper end of the optimum ø=0.79 it was 2% higher than that of the low hydrogen content producer gas. Sridhar et al. **[14]** revealed that the low energy density producer gas can be used to operate commercially available natural gas engines by employing suitably designed carburetor. Although it causes a loss of power to an extent of 20-30%, but it paves path for the option of adapting commercially available gas engines for large scale power generation application. This loss in power is recompense to a much larger proportion since these generate fewer amounts of NO_x and nearly zero SO_x towards green house gas emissions.

Hassan et al. **[15]** investigated that in dual fuel producer gas-diesel operation, supercharging is an effective way for improvement of combustion characteristics with reduction of unburned gas emission. Sahoo et al. **[16]** concluded in his review that dual fuel concept is a promising technique for controlling both NO_x and soot emissions even on existing diesel engine. But, emissions like UBHC, CO are higher for part load gas diesel engine operations. Further, it was observed that with increased engine speed or advanced injection timings, or with increased amount of pilot fuel, the thermal efficiency of dual-fuel engines improves. Uma et al. **[17]** investigated that both at diesel and dual fuel mode, the engine performance decreased with increase in emissions at part load conditions. Dual fuel operation increases the CO and reduces NO_x than diesel engines at all operated load condition.

In all the referred work, research on the use of producer gas produced from rice husk in combination with hydrogen in to four cylinder heavy-duty turbocharged diesel engines has not been reported. This paper presents effects of using mixture of producer gas and hydrogen in four different proportions as a secondary fuel with diesel as pilot fuel at wide range of load conditions in dual fuel operation of a 4 cylinder turbocharged and intercooled 62.5 kW gen-set diesel engines.

2. Experimentation:

The experimental setup used in the present paper is same as in reference **[3]** and has been illustrated here in brief for the sake of clarity. A diesel engine test setup was developed to carry out the experimentation on dual fuel engines. Table (2) shows specification of an engine (2). The diesel engine was modified by attaching hydrogen gas cylinder with the intake manifold to work on dual fuel mode through flame traps, mass flow meters. It was followed by a one-way non-return valve and common flame arrestor by keeping turbocharger and its bypass active. Down draft gasifier was used for generation of producer gas using rice husk. This is directly connected to the inlet manifold through the valve for the controlling of its supply. The engine was coupled to a 62.5 kW D.C. generator. The load on the engine was varied by introducing five water pumps and twelve 3 kW industrial water heaters in a set of four each. The engine was run at a constant speed of 1500 RPM. Figure (1) shows schematic layout of the diesel engine test setup used during the experiments.

The rice husk was fed to the gasifier through its top opening. Air entered in the combustion zone and producer gas generated leaves near the bottom of the gasifier at the temperature of about 500^oC. The hot producer gas was allowed to cool by passing through the water cooler where its temperature was reduced to $40\text{-}50^oC$. The cooled gas with moisture was then passed through the filter to remove tar and other particles. Gas passed through pebble bed and then through bubble cap filter column. Later it allowed passing through cotton yarn column for absorbing the moisture and security filter for fine filtering. A valve was provided at the outlet of filter pipe to control the gas flow. To measure the flow rate of producer gas, an orifice connected to the surge tank was used. The producer gas and air were mixed in the intake pipe and the mixture entered into the engine. The increase in air flow rate decreases the gas flow to the intake, as the ratio of air and gas flow rate is almost remains constant.

A piezoelectric pressure transducer (pressure range 0-250 bars) and a charge amplifier was used to measured the cylinder pressure. This pressure data were transferred to data acquisition system for further analysis. A Kistler make crank angle encoder with an accuracy of 1^o was used for angle measurement. The pressure data were obtained for an average of 100 cycles after 15 minutes of engine operation on stabilized conditions. The mass flow rate of hydrogen and producer gas was measured by mass flow meters in liters per minute. The experiments were performed for five times to ensure repeatability (2).

The experiments were conducted on the diesel engine setup under the following Cases.

(i) Case I : Engine runs on neat diesel only.
(ii) Case II : Engine runs on diesel as pilot fuel and a mixture of producer gas and hydrogen as secondary fuel.

The experimental chart in the form of test matrix is shown in Table (3).

Table (2): Engine specification

No.	Parameter	Engine Specification	No.	Parameter	Engine Specification
1	Make / Model	Ashok Leyland ALU WO4CT	9	Compression ratio	17.5:1
2	General Details	4-Stroke, CI, DI, Constant Speed, water cooled, turbo charger, Gen-Set engine	10	Injection Pressure	260 (bar)
3	No. of Cylinder	4	11	Injection Timing	16^0 BTDC
4	Bore	104 (mm)	12	Rated Power kW	62.5 at 1500 rpm
5	Stroke	113 (mm)	13	Inlet Pressure	1.06 (bar)
6	Rated Speed	1500 (rpm)	14	Inlet Temperature	313 (K)
7	Swept volume	3839.67 (cc)	15	Nozzle Diameter	0.285 (mm)
8	Clearance vol.	84.90 (cc)	16	Number of hole	5

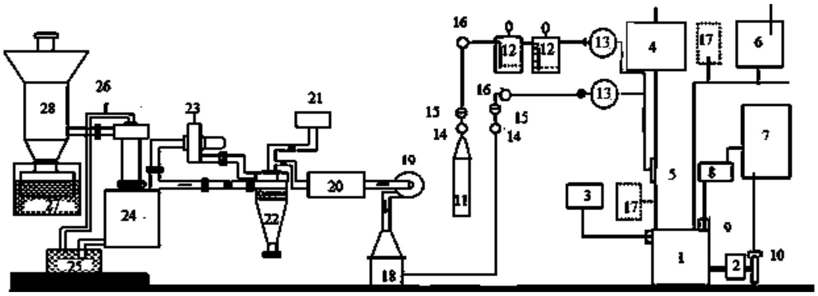

Figure (1): Schematic of the experimental set up.

1-Engine, 2-Gen-set,3-Diesel tank and measurement system, 4-Air tank and measurement system, 5-Gas mixture,6-Gas analyzer, 7-PC based data acquisition system, 8-Charge amplifier, 9-Cylinder pressure sensor, 10-Crank angle encoder,11-Hydrogen gas cylinder, 12-Hydrogen gas flame trap, 13-Gas flow meter, 14-Gas cylinder control valve, 15-Pressure regulator, 16-Solenoid switch valve, 17-Temperature and Pressure measurement locations, 18-Fabric filter, 19-Blower, 20-Organic filter, 21-Burner, 22-Cyclone, 23-Starting blower, 24-Cooling tower, 25-Water pipe, 26-Water tank, 27-Water seal, 28-Gasifier.

Table (3): Experimental test matrix

Case No.	Primary Fuel	Secondary Fuel	Load (%)	Secondary Fuel Substitution as % of diesel at each Load %	Mixture Composition (% of PG + % of H_2) in each % Mixture (M)
I	Diesel	--------------	13, 40,60,80	--------------	--------------
II	Diesel	M(PG + H_2) % M=Mixture	13, 40,60,80	M-30, M-40, M-50, M-60, M-70	PG-90% + H_2-10% PG-80% + H_2-20% PG-70% + H_2-30% PG-60% + H_2-40%

3. Results and discussion:

3.1 Analysis of brake thermal efficiency (η_{bth}): In this case, results are examined for 30% and above of mixture ($PG+H_2$) since, it was revealed that η_{bth} drops at lower gaseous fuel substitution and load conditions. Figure (2) shows variation of brake thermal efficiency with four different mixture combinations of gaseous fuel substitution at 13%, 40% and 80% loads respectively. It is observed that the η_{bth} is less than pure diesel operation (Case I) at all load conditions. Reason for this may be the pilot diesel fuel is low in quantity and hence, fewer ignition centers are formed. Also, lower η_{bth} may be due to rise in ignition delay of diesel with the presence of producer gas in dual fuel mode, lower burning rate of producer gas itself [11] and by the slow progress of the combustion. Furthermore, reduce amount of fresh air entering the combustion chamber, in-complete combustion and lower calorific value of producer gas are the major factors for the reduction in brake thermal efficiency [12].

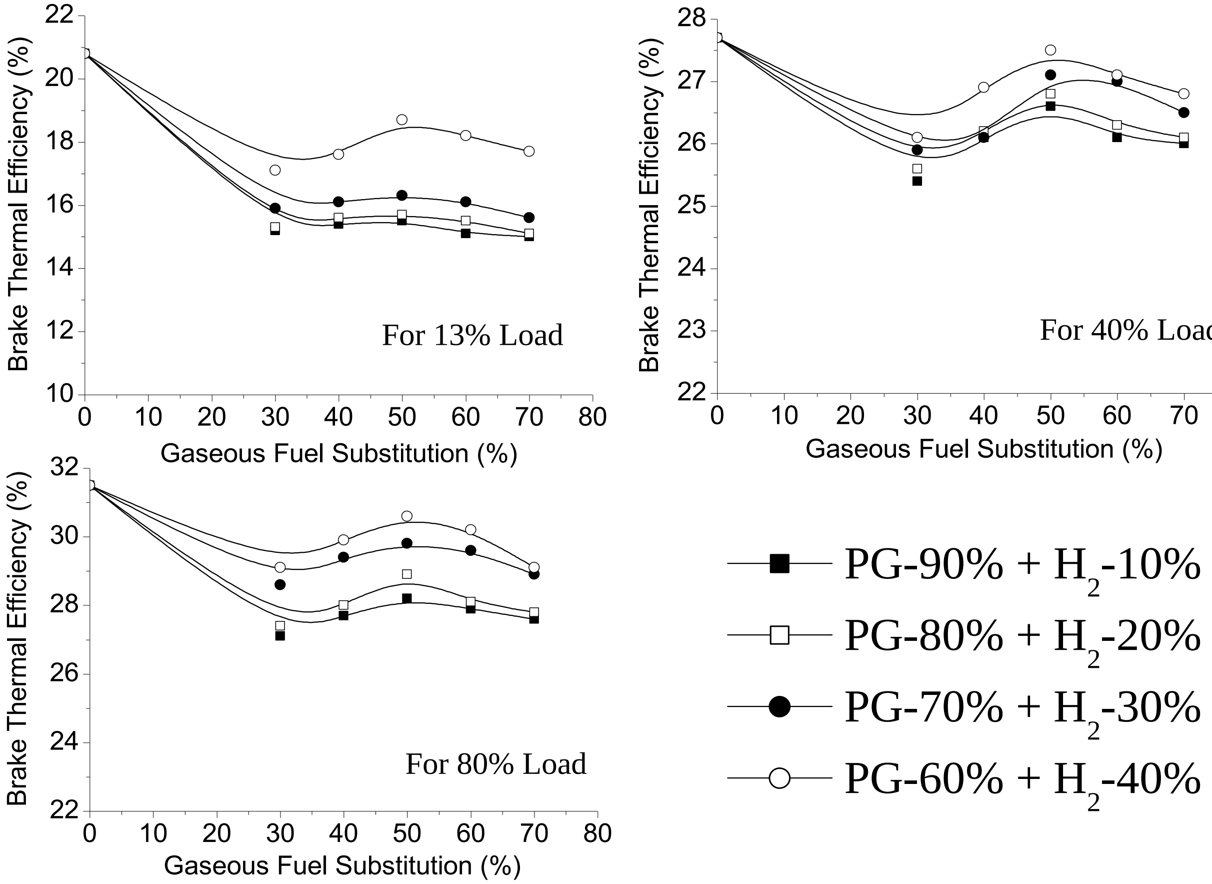

Figure (2): Brake thermal efficiency vs. mixed gaseous fuel substitution at different loads.

It is further observed that as the percentage of hydrogen in the mixture increases, the efficiency get increases up to 40% of hydrogen substitution due to the fact that the laminar burning velocity of producer gas is 0.5 m/sec as compared to 2.65 m/sec of hydrogen. The presence of hydrogen enhances the burning velocity of mixture and thus efficiency increases [18]. Also, producer gas flame tends to become unstable, while, hydrogen-air flames likely to be stable. Therefore, by increase in hydrogen fraction leads to stabilization of flame [19]. However, further addition of hydrogen beyond 60:40 reduces the efficiency viz 50:50. It might be due to increase in hydrogen fraction, flame destabilization takes place because of reduction in Markstein length (Markstein length measures the effect of curvature on a flame; larger the Markstein length, greater the effect of curvature on burning velocity) [6]. Further, other hydrocarbons do not have enough oxygen to burn since hydrogen used the majority part of the oxygen available for combustion.

3.2 Carbon monoxide (CO): In common diesel engine, the carbon oxidation reaction is nearly completed due to the presence of more excess air. Figure (3) exhibits variation of CO with four different mixture combinations of gaseous fuel substitution at 13%, 40% and 80% loads conditions. It is observed that dual fuel operation produces more CO at all load conditions than pure diesel operation. At 13% and 80% load conditions, of all the mixture combinations tested, mixture combination PG:H_2=60:40 at 50 % of its substitution gives maximum 78.9% and 41.9% rise in CO emission respectively, as compared to pure diesel operation. At low load conditions, gaseous fuel-air mixture near the pilot is burned due to less turbulence. Thus some partial oxidation product like carbon monoxide may come out in the exhaust. At higher concentration of gaseous fuel, the concentration of the partial oxidation product could increase [7, 20]. This is considered to be the reason for the rise in CO emissions. Maximum rise in CO emission at 80% load is due to higher mean gas temperature and combustion rate. Higher emissions of CO in dual fuel mode could be recognized to lower heating value of producer gas, lower adiabatic flame temperature and lower mean effective pressure [9].

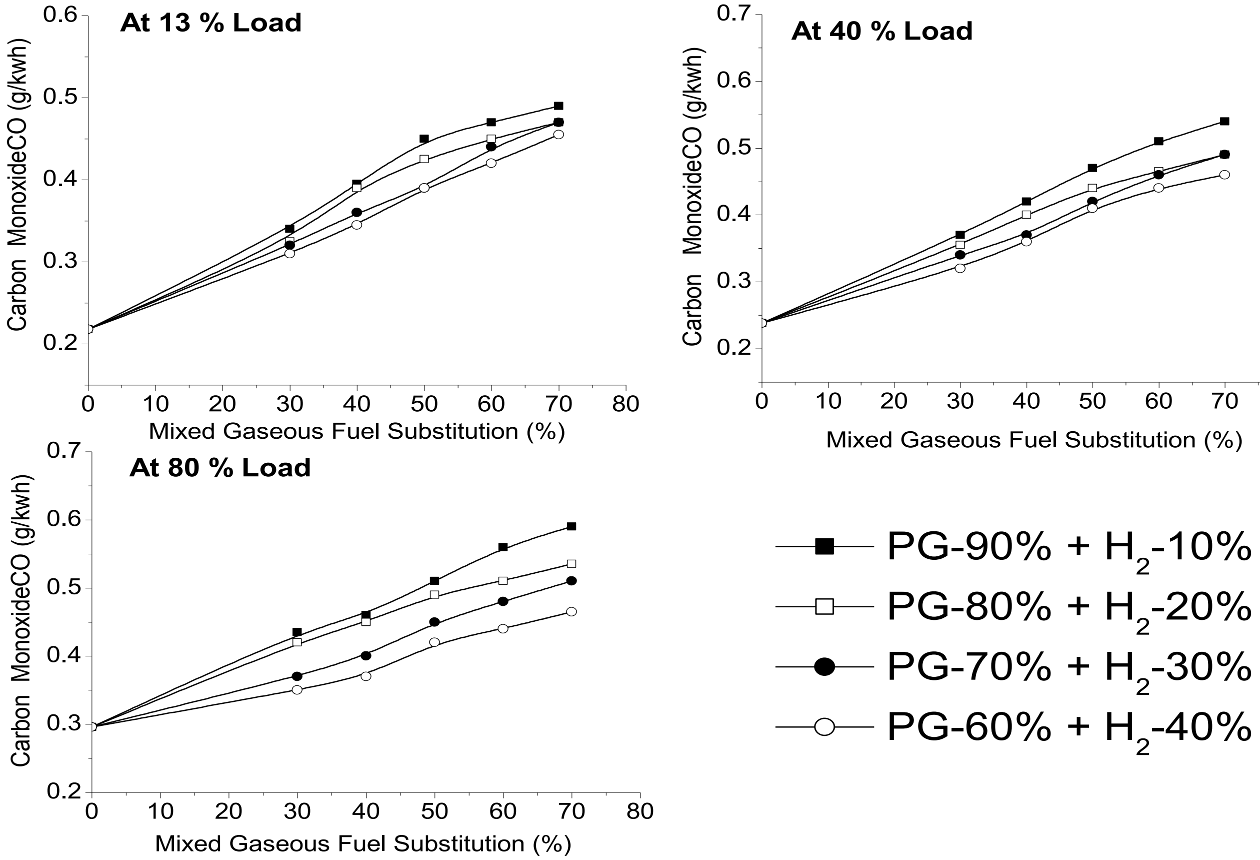

Figure (3): Carbon monoxides vs. mixed gaseous fuel substitution at different loads.

Further, hydrogen shows different behavior in dual fuel engine due to presence of liquid hydrocarbon. The moment ignition starts, the spontaneous combustion occurs due to the presence of more percentage of hydrogen. As a result, diesel fuel is more subjected to higher combustion temperature in an atmosphere of lack of oxygen. The higher concentration of CO emission in dual fuel mode gives an indication of incomplete combustion. The mixture of gaseous-air flow to the engine reduces the amount of oxygen required for complete combustion [9]. In general, since the hydrogen does not contain any carbon particle it reduces CO and the percentage of CO present in the exhaust is due to the burning of lubricating oil and incomplete combustion of diesel fuel and producer gas. Further, CO emission is increased at all load conditions due to delayed ignition period.

3.3 Oxides of nitrogen (NO_x): In the dual fuel engine development of NO_x mostly depends on diesel pilot spray region. It increases with the increase in the size and amount of pilot diesel fuel. Also, the NO_x emission rises with the increase in cylinder temperature, oxygen concentration and combustion duration [6].

Figure (4) exhibits variation of NO_x with four different mixture combinations of gaseous fuel substitution at 13%, 40% and 80% loads conditions. It is observed that dual fuel operation produces less NO_x at all load conditions than pure diesel operation. At 13% and 80% load conditions, of all the mixture combinations tested, the mixture combination PG:H_2=60:40 at 50 % of its substitution gives maximum 45.0% and 55.37% drop in NO_x emission respectively, as compared to pure diesel operation. This may be because of increase in hydrogen substitution simultaneously increases the mole fraction of H_2O i.e. the moisture increases which finally brought down the peak temperature. Hence NO_x decreases with the increase in hydrogen substitution [2]. Also, reduction in NO_x may be due to the lower adiabatic flame temperature of producer gas and absence of organic nitrogen in producer gas [9, 11]. Furthermore, drop in high temperature region around the diesel flame due to more uniform temperature distribution obtained with the gaseous fuel-air mixture [15, 16].

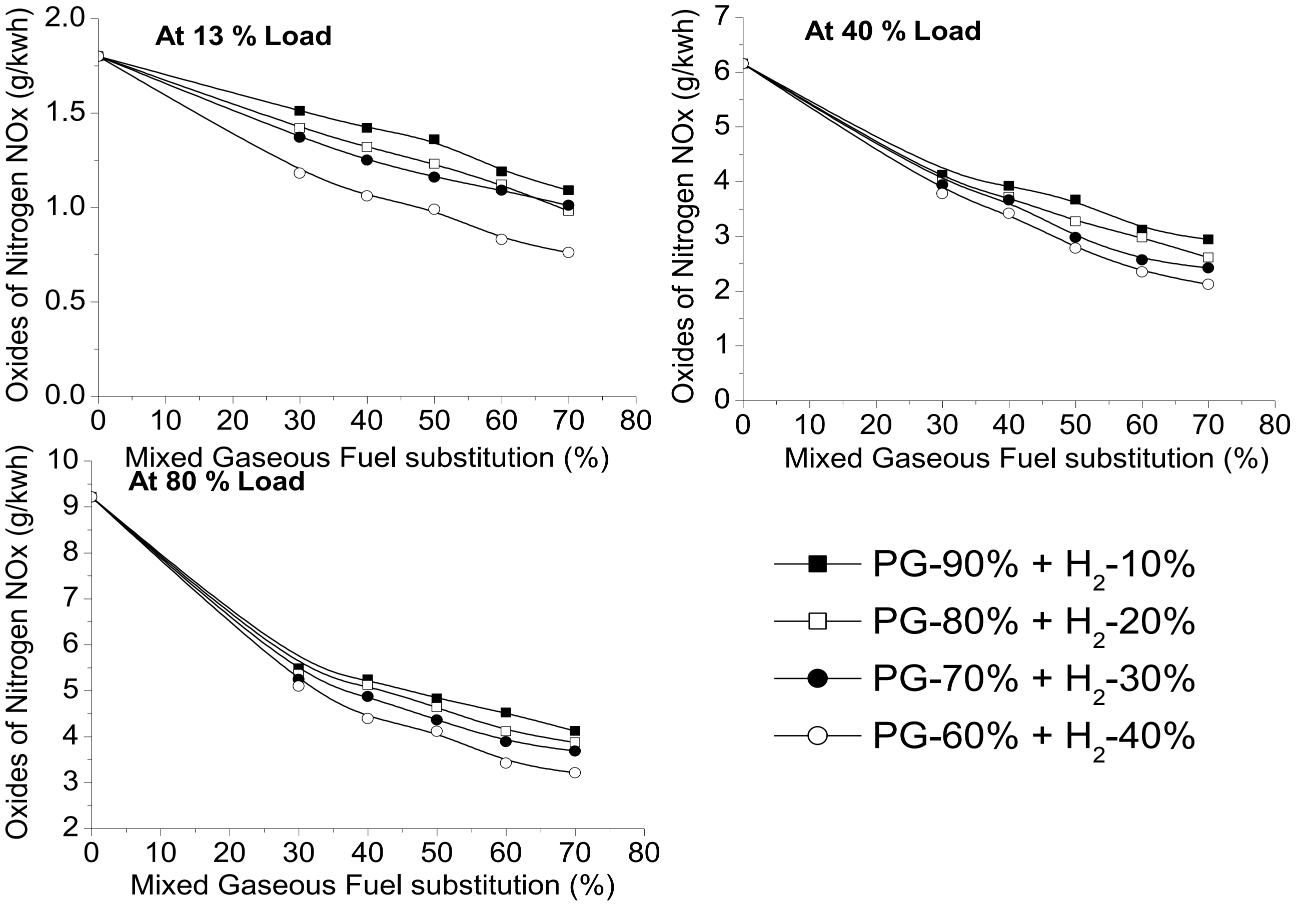

Figure (4): Oxides of nitrogen vs. mixed gaseous fuel substitution at different loads.

4 Conclusions: On the basis of the results and discussions presented above, the following conclusions could be drawn.

1. The performance study of CI engine operated on diesel, hydrogen and producer gas in dual fuel mode as CI engine fuels indicates no major modification required in an existing diesel engine.
2. Mixture combination PG:H_2 = 60:40 is the most suited combination of all the combinations tested.

3. A mixture combination PG:H_2 = 60:40 as secondary fuel exhibits better brake thermal efficiency in comparison with other proportions of the same mixture. However, this is less than pure diesel.
4. Further increase of hydrogen proportion in the mixture combination deviate the performance of an engine.
5. Rise in CO emission for mixture combination PG:H_2 = 60:40 of all mixture combinations tested in duel fuel mode is less. However, this is much more than pure diesel.
6. NO_x emissions for all type of the fuel combinations in duel fuel mode are found to be lower than single diesel fuel engines. Moreover, for mixture combination PG:H_2 = 60:40 it is least.
7. Replacement of mixture combination PG:H_2 = 60:40 at 50% of diesel replacement is giving better results from all performance and emission parameters point of view.

Also, the analysis shows that the mixture combination PG:H_2 = 60:40 is always considerable beyond 40% load condition. Although, it reduces the brake thermal efficiency slightly, however, drops down the formation of NO_x as compared to pure diesel. Besides, its use avail wide scope for the unexploited biomass for the generation of producer gas which ultimately reduce burden over the use of fossil fuel.

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