Fumigation of DEE and Ch4 Enhance the Performance and Emission Characteristics of CI Engine by using WVO as Fuel

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Abstract: The ignition delay and engine exhaust emissions in a biogas- fueled diesel engine can be minimized by using a fuel having a high cetane number and high dissolved oxygen. In this respect, diethyl ether (DEE) is a promising choice. DEE can be produced from ethanol at an acceptable cost, and is renewable in nature. The high volatility of DEE prevents vapor lock in the fuel injection line. In a dual fuel mode, the DEE can be injected into the air intake port or can be fumigated with the biogas supply line to increase the efficiency and reduce the emission level. Research reports are available on the use of DEE in diesel engines operated in a dual fuel mode with biodiesel or diesel as pilot fuels, and biogas as primary fuel. The DEE injection in the dual fuel engine gave a shorter ignition delay and shorter combustion duration. The peak cylinder pressure and the heat release rate increased with the DEE injectio. The part load performance and the brake thermal efficiency (BTE) of the engine were increased. The HC, CO, and smoke emission were decreased, but the NO emission was reportedly higher.

Keywords: - Biogas,Waste, Duel Fuel, BTE, DEE

I. INTRODUCTION

The steam turbine in a steam power plant is another example of an external combustion engine. The steam engine may be called an intermittent external combustion engine and the steam turbine a continuous external combustion engine. A closed cycle gas turbine plant is also an example of an external combustion engine. Here, normally the air is a working substance which completes a the rmodynamic cycle. It receives heat from products of combustion of fuel in a heat exchanger and rejects heat from another heat exchanger to the surroundings. Here also the products of combustion do not enter into the turbine. Stirling engine is also an external combustion engine, In internal combustion engines, either the combustion of the fuel takes place inside the engine cylinder or the products of combustion enter into the cylinder as a working fluid. In reciprocating engines having cylinder and piston, the combustion of the fuel takes place inside the cylinder and such engines may be called intermittent internal combustion engines. In an open cycle gas turbine plant, the products of combustion of fuel enter into the gas turbine and work is obtained in the form of rotation of the turbine shaft. Such a turbine is an example of a continuous internal combustion engine.

The intermittent internal combustion engines are most popular because of their use as the prime mover in motor vehicles, and usually these engines are reciprocating engines, the reciprocating engine mechanism consists of a piston which moves in a cylinder and forms a movable gas. Tight seal. By means of a connecting rod and a crankshaft arrangement, the reciprocating motion of the piston is converted to rotary motion at the crankshaft. The steam turbine plant is the most popular continuous external combustion engine used for large electric power generation. The essential components are boiler, steam turbine, condenser, and feed pump.

The main advantages of an internal combustion engine over an external combustion engine

(a) Greater mechanical simplicity

(b) Higher power output per unit weight because of the absence of auxiliary units like boiler) condenser, and feed pump

(c) Lower initial cost

(d) Higher brake thermal efficiency as only a small fraction of heat energy of the fuel is dissipated to the cooling system.

The advantages of internal combustion engine accrue from the fact that they work at an average temperature which is much below the maximum temperature of the working fluid in the cycle.

The disadvantages of the internal combustion engine over the external combustion engine are:

(a) The IC engines cannot use solid fuels which are cheaper. Only liquid or gaseous fuels of given specifications can be efficiently used. These fuels arc relatively more expensive.

(b) The IC engines are not self-starting whereas the EC engines have a high-starting torque.

(c) The intermittent IC engines have reciprocating parts and hence they are susceptible to the problems of vibration.

II. EXPERIMENTAL SETUP

2.1 The Injection Pressure Variation

To acquire high degree of fuel atomization needs high injection pressure in the fuel injection system. For the purpose of sufficient evaporation of fuel in very short time. From that the fuel particles achieve sufficient spray penetration in order to exploit the fuel air charge in the cylinder. The fuel injection system should have measured the amount fuel desired, depending upon engine load and speed, and inject the fuel at desired rate in correct time. The appropriate shape and size of fuel particle obtained based on the particular combustion chamber. Generally, a supply pump withdraws the fuel from fuel tank and carries it's via a filter to the fuel injector. In present investigation the injection pressure varied from 180 to 210 bars. Normally the injection pressure is 180 bars for high speed diesel engines. In this the injection pressure is varying by tightening or loosening the screw provided top of the injector as shown in fig. 1 For measurement of fuel injection pressure on fuel injector system by using fuel injector pressure tester as shown in figure.

Figure 1: Fuel Injector

2.2 Load Measurement

The engine was coupled to an electrical dynamometer for loading. The dynamometer used to load the engine comprised a shunt wound DC generator and a load bank. In electrical dynamometer, the shaft rotation drives some form of electrical generator. The strength of the electromagnetic field coupling the rotating and stationary parts of the dynamometer can be adjusted in order to increase or decrease the resistance offered to the engine rotation. Dynamometer load was varied from 0 amps to 16 amps in steps of 4 amps.

2.3 Air and Fuel Flow Measurement

An "U" tube manometer was used to measure the airflow rate. One end of the manometer was left free to ambient and the other end was connected to surge tank. The fuel flow rate was measured on volume basis using a burette and a stop watch.

2.4 Exhaust Gas Temperature Measurement

The temperature of exhaust gas was measured with Chromel Alumel (K-Type) thermocouples. A digital indicator with an automatic room temperature compensation facility was used and it was calibrated periodically.

2.5 Cylinder Pressure Measurement

Kistler piezoelectric transducer with a range of 0 to 250 bar was used. In cylinder pressure was measured with a water-cooled piezoelectric transducer. The pressure transducer mounted on the engine is shown in Figure 3.4. The details of the pressure transducer and charge amplifier are given respectively.

• The piezoelectric transducer was mounted on the cylinder head and connected to a charge amplifier for measuring the pressure inside the engine cylinder.

• The transducer was cooled by circulating the cooling water through the inlet opening. The central opening was used for sending signal to the amplifier.

Figure 2: Photographic view of pressure transducer fitted in the cylinder head

2.6 Digital Data Acquisition System

A 12-bit ADC was used for the conversion of data into digital form. The data acquisition system could set the sampling speed and the total number of samples is taken continuously. Signals were stored in a file. One channel was fed with signals from the pressure transducer while the voltage was fed to the other channel from the TDC position encoder. Specifications of the ADC are given in Appendix 5. The ADC converter had external and internal triggering facility with sixteen single ended channels. Data from 100 consecutive cycles were stored continuously at each operating condition. Recorded signals were processed to obtain the peak pressure.

2.7 Measurement of Emissions

NOX, HC, CO emissions were measured by QROTECH, QRO- 401 exhaust gas analyser. The specifications are given in Appendix 6. Smoke intensity was measured using a Bosch smoke meter whose specifications are given in Appendix 7. The probe that was connected to the exhaust gas analyser was placed inside the exhaust pipe. HC and NOX were measured in ppm and CO in % by volume. The smoke intensity was measured in Bosch Smoke Number (BSN).

2.8 Test Conditions

The entire experimental work was carried out in the laboratory at room temperature (32oC) and atmospheric pressure (1.01325 bar). Before measuring all the engine exhaust emissions and cylinder gas pressure, the instruments such as gas analyzer and pressure sensor were calibrated and verified with the accuracy levels.

III. RESULT AND DISCUSSION

Brake specific fuel consumption kg/kw-hr: Specific fuel consumption (SFC) for a engine is defined as the amount of fuel that the engine needs to burn per hour to produce 1kW of Brake Power. Specific fuel consumption is the ratio of fuel consumption (kg/hr) to power output of the engine. on the graph shows that the BSFC increasing with the load but after 20 kg of weight, here BSFC decreases with load in between 5 to 20 kg of engine load. in the graph shows that the 30% biogas with DEE fumigation gave better fuel economy than other blending.

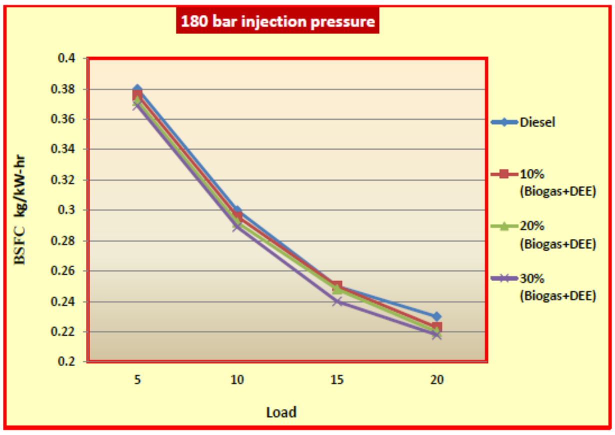

Figure 3: Variation of fuel consumption with load

Brake thermal efficiency: It is the efficiency of the actual engine, defined as the actual work (Indicated or Brake), and divided by the heat released by the combustion of fuel. The variations of brake thermal efficiency with load for both cases are shown in figure 4.2 Brake thermal efficiency is defined as the ratio of brake power to product of fuel consumption and calorific value. Brake thermal efficiency for all the fuels increases as the load increase. In the graph shows that the 30% biogas with DEE fumigation gave better brake thermal efficiency then other blending because biogas have more oxygen and DEE have more oxygen which give better combustion rate which increase brake thermal efficiency of the engine.

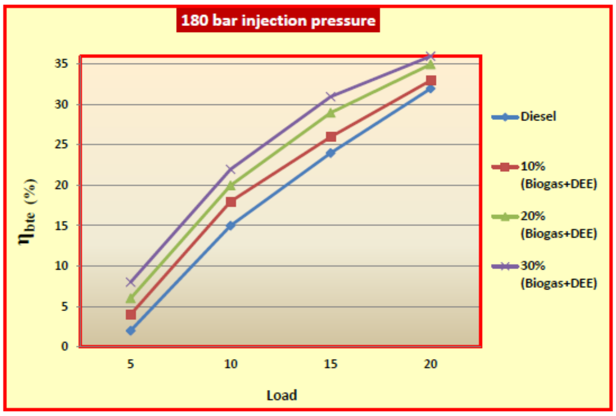

Figure 4: Variation of brake thermal efficiency with load

Smoke Opacity: In the graph shows that at 200 bar injection pressure the smoke opacity decrease with increases in percentage of biogas, here 30% of biogas gives minimum smoke opacity than other concentration of biogas. In the graph 4.25 shows that at 220 bar injection pressure the smoke emission decrease with increases in percentage of biogas, here 30% of biogas gives minimum smoke opacity than other concentration of biogas.

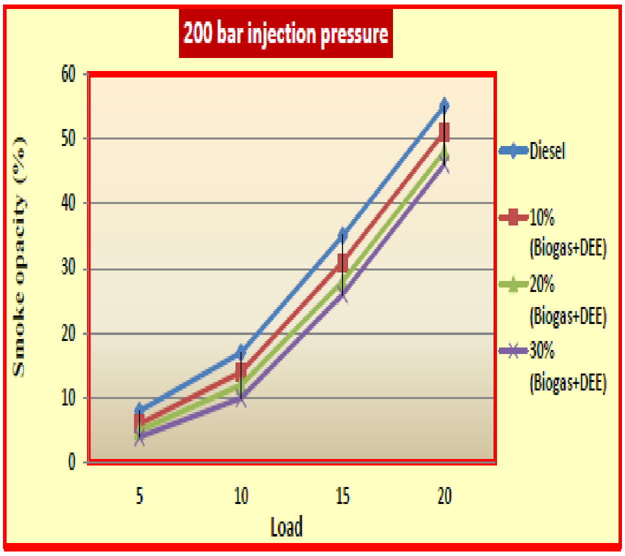

Figure 5: Variation of exhaust gas temperature with load at 200 bar injection pressure

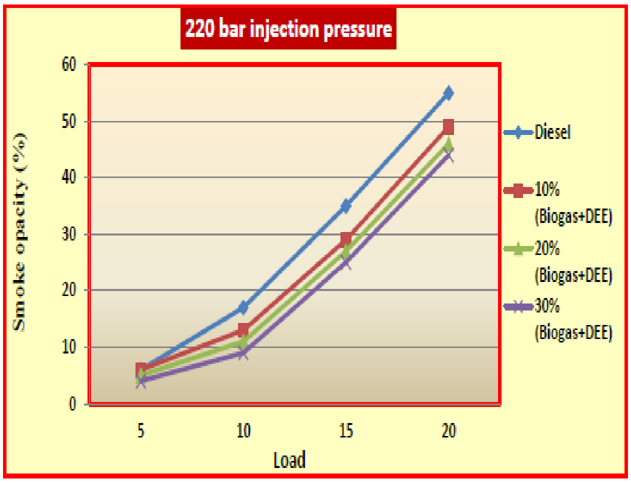

Figure 6: Variation of exhaust gas temperature with load at 220 bar injection pressure

IV. CONCLUSION

Brake specific fuel consumption at 220 bar injection pressure found less about (0.21 kg/kw.hr) other than 180 bar and 200 bar. Brake thermal efficiency at 220 bar injection pressure found higher (about 32%) other than 180 bar and 200 bar. Volumetric efficiency at 220 bar injection pressure found higher (about 86%) other than 180 bar and 200 bar. Mechanical efficiency at 220 bar injection pressure found higher (about 85%) other than

180 bar and 200 bar. Carbon monoxide emission found lesser (about 0.083%) in 220 bar injection pressure. Nitrogen oxide emission found lesser (about 590 ppm) in 220 bar injection pressure. Smoke Opacity found lesser (about 44 %) in 220 bar injection pressure.

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