An Experimental Comparison Of Combustion, Performance And Emission In A Single Cylinder Thermal Barrier Coated Diesel Engine Using Diesel And Biodiesel

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***Abstract-* The use of methyl esters of vegetable oil known as biodiesel are increasingly popular because of their low impact on environment, green alternate fuel and most interestingly it's use in engines does not require major modification in the engine hardware. Use of biodiesel as sole fuel in conventional direct injection diesel engine results in combustion problems, hence it is proposed to use the biodiesel in low heat rejection (LHR) diesel engines with its significance characteristics of higher operating temperature, maximum heat release, higher brake thermal efficiency (BTE) and ability to handle the lower calorific value (CV) fuel. In this work biodiesel from Jatropha oil called as Jatropha oil methyl ester (JOME) was used as sole fuel in conventional diesel engine and LHR direct injection (DI) diesel engine. The low heat rejection engine was developed with uniform ceramic coating of combustion chamber (includes piston crown, cylinder head, valves and cylinder liner) by partially stabilized zirconia (PSZ) of 0.5 mm thickness. The experimental investigation was carried out in a single cylinder water-cooled LHR direct injection diesel engine. In this investigation, the combustion, performance and emission analysis were carried out in a diesel and biodiesel fueled conventional and LHR engine under identical operating conditions. The test result of biodiesel fueled LHR engine was quite identical to that of the conventional diesel engine. The brake thermal efficiency (BTE) of LHR engine with biodiesel is decreased marginally than LHR engine operated with diesel. Carbon monoxide (CO) and Hydrocarbon (HC) emission levels are decreased but in contrast the Oxide of Nitrogen (NOX) emission level was increased due to the higher peak temperature. The results of this comparative experimental investigation reveals that, some of the drawbacks of biodiesel could be made as advantageous factors while using it as a fuel in the LHR diesel engine. In the final analysis, it was found that, the results are quite satisfactory.**

I. Introduction

Diesel engines are the dominating one primarily in the field of transportation and secondarily in agricultural machinery due to its superior fuel economy and higher fuel efficiency. The world survey explicit that the diesel fuel consumption is several times higher than that of gasoline fuel. These fuels are fossil in nature, leads to the depletion of fuel and increasing cost. It has been found that the chemically treated vegetable oil often called as biodiesel is a promising fuel, because of their properties are similar to that of diesel fuel (DF) and it is a renewable and can be easily produced. Compared to the conventional DI diesel engine the basic concept of LHR engine is to suppress the heat rejection to the coolant so that the useful power output can be increased, which in turn results in improved thermal efficiency. However previous studies are revealing that the thermal efficiency variation of LHR engine not only depends on the heat recovery system, but also depends on the engine configuration, operating condition and physical properties of the insulation material (1–3).

The drawback of an LHR engine has to be considered seriously and effort has to be taken to reduce the increased heat loss with the exhaust and increased level of NOx emission. The potential techniques available for the reduction of NOx from diesel engines are exhaust gas recirculation (EGR), water injection, slower burn rate, reduced intake air temperature and particularly retarding the injection timing (4–6). It is strongly proven that the increasing thickness of ceramic coatings arrest the heat loss from the engine cylinder, in contrast decreases the power and torque. The optimized coating thickness can be identified through the simulation techniques (7). One of the viable significance of LHR engine is utilizing the low calorific value fuel such as biodiesel. Studies have revealed that, the use of biodiesel under identical condition as that for the diesel fuel results in slightly lower performance and emission levels due to the mismatching of the fuel properties mainly low calorific value and higher viscosity. The problems associated with the higher viscosity of biodiesel in a compression ignition (CI) engines are pumping loss, gum formation, injector nozzle coking, ring sticking and incompatibility with lubricating oil (8-12). The above identified problems with the use of biodiesel in conventional diesel engine can be reduced in LHR engines except for the injection problem.

The present investigation involves the comparison of combustion, performance and emission levels of diesel and biodiesel (Jatropha based) in conventional and LHR DI diesel engines.

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II. FUEL PREPARATIONS AND CHARACTERIZATION

The vegetable oil was transesterified-using methanol in the presence of NaOH as a catalyst. The parameter involved in the above processing includes the catalyst amount, molar ratio of alcohol to oil, reaction temperature and reaction time (13-15). The parameters for the biodiesel production are optimized such as Catalyst amount, Molar ratio (Alcohol to Oil), Reaction temperature, and Reaction time.

The raw biodiesel obtained was brought down pH to a value of 7. This pure biodiesel was measured on weight basis and the important physical and chemical properties were determined as per the BIS standards (14).It is evident that, the dilution or blending of vegetable oil with other fuels like alcohol or diesel would bring the viscosity close to the specification range for a diesel engine (16-17). The important physical and chemical properties of the biodiesel thus prepared and given in table 1.

Table 1 Properties of the diesel and biodiesel fuel

Characteristics	Diesel Fuel	B100
Density @ 15°C(kg/m^3)	837	880
Viscosity @ 40°C(cSt)	3.2	4.6
Flash point (°C)	65	170
Cetane number	47	50
Calorific Value (MJ/kg)	42	39.5

III. DEVELOPMENT OF TEST ENGINE

The engine combustion chamber was coated with partially stabilized zirconia (PSZ) of 0.5 mm thickness, which includes the piston crown, cylinder head, valves, and outside of the cylinder liner. The equal amount of material has been removed from the various parts of the combustion chamber and PSZ was coated uniformly. After PSZ coating, the engine was allowed to run about 10 hours, then test were conducted on it.

IV. EXPERIMENTAL PROCEDURE

The experimental setup and the specification of the test engine are shown in Fig.1 and table 3 respectively.

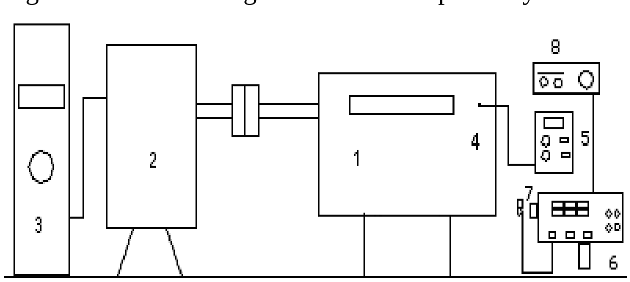

Fig.1 Experimental setup

i. Test engine
ii. Dynamometer
iii. Dynamometer controller
iv. Piezo electric pressure transducer
v. Charge amplifiers
vi. Data acquisition system
vii. Magnetic pickup
viii. Computer

The engine was coupled with an eddy current dynamometer for performance and emission testing. A piezoelectric transducer was mounted through an adopter in the cylinder head to measure the in-cylinder pressure. Signal from the pressure transducer was fed to charge amplifier. A magnetic shaft encoder was used to measure the TDC and crank angle position. The signals from the charge amplifier and shaft encoder were given to the appropriate channels of a data acquisition system.

The analyzer used to measure the engine exhaust emission was calibrated before each test. Using the appropriate calibration curve, the measurement error for each analyzer was reduced as per the recommendation by the exhaust analyzer manual. The emission values obtained in the form of ppm and percentage was expressed in term of specific mass basis (g/kWh). The NOx measurements were corrected for humidity following the procedure recommended by the Society of Automotive Engineers (SAE, 1993). Exhaust gas temperature was measured using an iron-constantan thermocouple and mercury thermometer was used to measure the cooling water temperature. Diesel and biodiesel was used in the conventional diesel engine and the PSZ coated LHR engine. Cylinder pressure data was recorded and the other desired datas were processed.

The experiments were carried out in a single cylinder, naturally aspirated, constant speed, water-cooled direct injection diesel engine with the following specifications

Table 2 Specification of test engine

No of stroke	Four stroke
No of cylinder	One
Bore, mm	87.5
Stroke, mm	110
Compression ratio	17. 5: 1
Rated power output	4.4 kW @ 1500 rpm
Injection Pressure, bar	200
Injection timing	24° BTDC

The test procedure was adopted from Beareu of Indian Standards BIS –10000(year 1985).

V. COMBUSTION AND HEAT RELEASE ANALYSIS

The combustion parameters can be studied through the analysis of heat release rate obtained from the pressure crank angle diagram. It is assumed that the mixture is homogeneous and uniform pressure and temperature at each instant of time during the combustion process. The heat release rate can be calculated from the first law of thermodynamics.

$$\frac{dU}{dt} = \frac{dQ}{dt} + \sum m_i\, h_i - p\frac{dV}{dt} \qquad (1)$$

Where

$\frac{dQ}{dt}$ - Heat transfer rate

$p\frac{dV}{dt}$ - Work done by the system

m_i - Mass of flow in to the system

h_i - Enthalpy of flow in to the system

p - Pressure at any crank angle

V - Volume at any crank angle

U - Internal energy at any crank angle

By neglecting the crevice volume and its effect, the equation (1) is reduced to

$$\frac{dU}{dt} = \frac{dQ}{dt} = m_f h_f - p\frac{dV}{dt} \qquad (2)$$

Where

m_f . Fuel flow rate

h_f - Enthalpy of the fuel

This equation (2) can be further reduced to

$$\frac{dQn}{dt} = \frac{dQch}{dt} - \frac{dQht}{dt} = p\frac{dV}{dt} + mCv\frac{dT}{dt} \qquad (3)$$

Where

$\frac{dQn}{dt}$ - Net heat release rate

$\frac{dQch}{dt}$ - Gross heat release rate

$\frac{dQht}{dt}$ - Heat transfer rate to the wall

From the ideal gas relation PV=mRT this equation (3) is further modified in to

$$\frac{dQn}{dt} = \frac{n}{n-1}p\frac{dV}{dt} + \frac{1}{n-1}V\frac{dP}{dt} \qquad (4)$$

The pressure at any angle obtained form the pressure crank angle diagram makes it possible to find out the heat release at any crank angle.

VI. RESULTS AND DISCUSSIONS

A. *Cylinder Pressure*

In a CI engine the cylinder pressure is depends on the fuel-burning rate during the premixed burning phase, which in turn leads better combustion and heat release. Figure 2 shows the typical variation of cylinder pressure with respect to crank angle. The cylinder pressure in the case of biodiesel fueled LHR engine is about 7.5 % lesser than the diesel fueled LHR engine and higher by about 2.75 % and 8.33% than conventional engine fueled with diesel and biodiesel. This reduction in the incylinder pressure may be due to lower calorific value and slower combustion rates associated with biodiesel fueled LHR engine. However the cylinder pressure is relatively higher than the diesel engine fueled with diesel and biodiesel.

It is noted that the maximum pressure obtained for LHR engine fueled with biodiesel was closer with TDC around 2 degree crank angle than LHR engine fueled with diesel The fuel-burning rate in the early stage of combustion is higher in the case of biodiesel than the diesel fuel, which bring the peak pressure more closer to TDC.

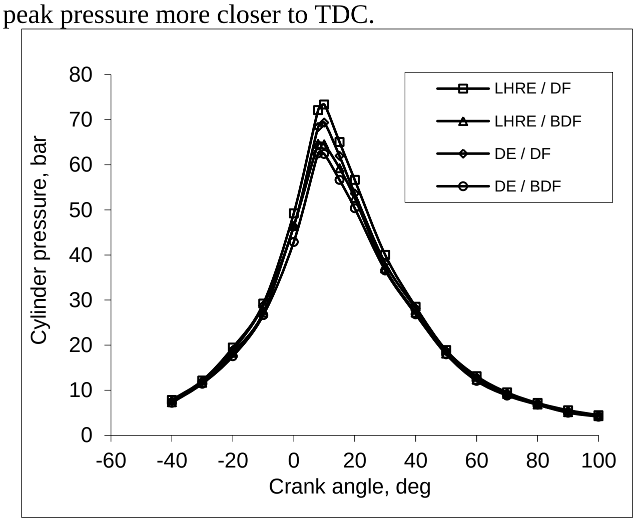

Fig. 2 Variation of cylinder pressure with respect to crank angle at full load

B. *Heat Release Rate*

Figure 3 shows the variation of heat release rate with respect to crank angle. It is evident from the graph that, diesel and biodiesel fuel experiences the rapid premixed combustion followed by diffusion combustion. The premixed fuel burns rapidly and releases the maximum amount heat followed by the controlled heat release. The heat release rate during the premixed combustion is responsible for the cylinder peak pressure.

The maximum heat release of LHR engine with biodiesel is lower about 8.1% than LHR engine fueled with diesel and higher about 2.4% and 7.02% respectively than conventional engine fueled with diesel and biodiesel. It was found that, premixed combustion in the case of biodiesel fuel starts earlier than the diesel fuel and it may be due to excess oxygen available along with higher operating temperature in the fuel and the consequent reduction in delay period than that of diesel fuel. It may be expected that high surrounding temperature and oxygen availability of fuel itself (bio diesel) reduce the delay period. However higher molecular weight lower calorific value and slightly higher value of viscosity bring down the peak heat release during the premixed combustion period. The heat release is well advanced due to the shorter delay period and early burning of the biodiesel. It is found that, the heat release rate of biodiesel, normally accumulated during the delay period follows the similar trends like diesel fuel.

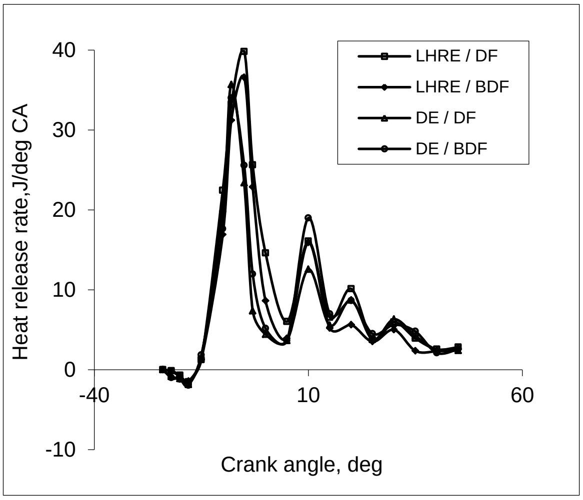

Fig. 3 Variation of heat release rate with respect to crank angle at full load

C. *Cumulative Heat Release Rate*

Figure 4 shows the variation of cumulative heat release with respect to crank angle. In general, the availability of oxygen in the biodiesel fuel itself enhances the combustion and thus increases the net heat release. In this investigation at full load, the net heat release for LHR engine fueled with biodiesel is lower by about 8.34% than LHR engine fueled with diesel and higher by about 2.35% and 7.04% respectively than LHR engine with biodiesel and conventional diesel engine fueled with diesel and biodiesel.

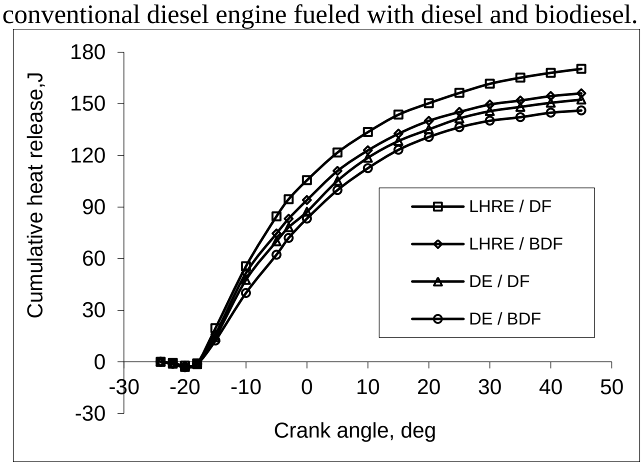

Fig.4 Variation of cumulative heat release with respect to crank angle at full load

D. *Brake Thermal Efficiency*

Figure 5 shows the variation of brake thermal efficiency with engine power output. The maximum efficiency obtained in the case of LHR engine fueled with biodiesel at full load was lower by about 2.92% than LHR engine fueled with diesel and higher by about 1.77% and 5.6% respectively than conventional diesel engine fueled with diesel and biodiesel. In overall, it is evident that, the thermal efficiency obtained in the case of LHR engine fueled with biodiesel is substantially good enough within the power output range of the test engine.

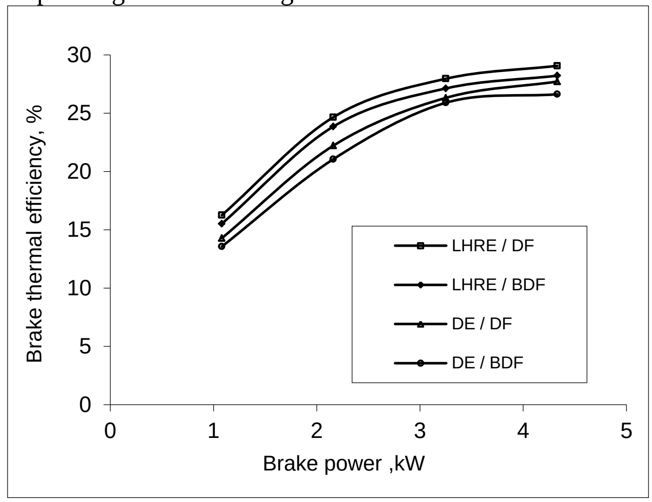

Fig. 5 Variation of brake thermal efficiency with engine power output

E. *Specific Fuel Consumption*

The variations of brake specific fuel consumption (SFC) with engine power output for different fuels are presented in figure 6. At maximum load the specific fuel consumption of LHR engine fueled with biodiesel is higher by about 6.27% than LHR engine fueled with diesel and lower by about 3.77% and 11.14% respectively than conventional engine fueled with diesel and biodiesel.

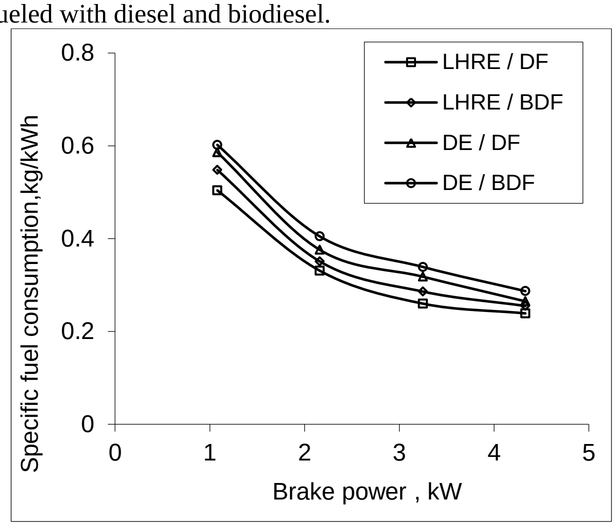

Fig. 6 Variation of Specific fuel consumption with engine power output

This higher fuel consumption was due to the combined effect of lower calorific value and high density of biodiesel. The test engine consumed additional biodiesel fuel in order to retain the same power output.

F. *Specific Energy Consumption*

Figure 7 shows the variation between specific energy consumption (SEC) and engine power output. The heat input required to produce unit quantity of power is proportionately varying with SFC. Higher the energy required at low load and decreases by increasing the load. It is found that the specific energy consumption of LHR engine with biodiesel is higher by about 1.27% than the LHR engine with diesel fuel and lower by about 3.78% and 11.15% respectively for conventional diesel engine with diesel and biodiesel

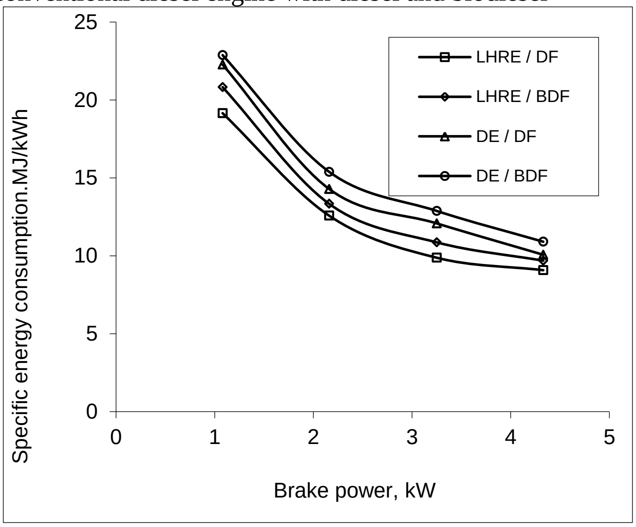

Fig.7 Variation of specific energy consumption with engine power output

G. *Exhaust Gas Temperature*

Figure 8 shows the variation of exhaust gas temperature with engine power output. At full load, the exhaust gas temperature of LHR engine fueled with biodiesel gives lower value by about 2.52% than LHR engine fueled with diesel and higher by about 2.83% and 6.13% respectively than conventional engine with diesel and biodiesel. The higher operating temperature of LHR engine is responsible for the higher exhaust temperature. The exhaust gas temperature of biodiesel varying proportionately with engine power output as in the case of diesel fuel. It may be due to the heat release rate by the biodiesel during the expansion is comparatively lower than diesel.

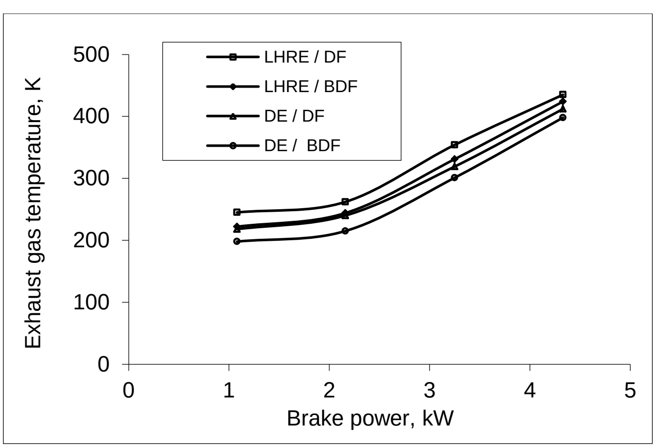

Fig.8 Variation of exhaust gas temperature with engine power output

H. *Carbon Monoxide*

The variation of carbon monoxide (CO) with engine power output is presented in figure 9. The fuels are producing higher amount of carbon monoxide emission at low power outputs and giving lower values at higher power conditions. Carbon monoxide emission decreases with increasing power output. At full load, CO emission for LHR engine with biodiesel fuel is lower by about 10.73%, 26.82% and 31.89% respectively than LHR engine with diesel, conventional engine fueled with biodiesel and diesel. With increasing biodiesel percentage, CO emission level decreases. Biodiesel itself has about 11% oxygen content in it and it may helps for the complete combustion. Hence, CO emission level decreases with increasing biodiesel percentage in the fuel.

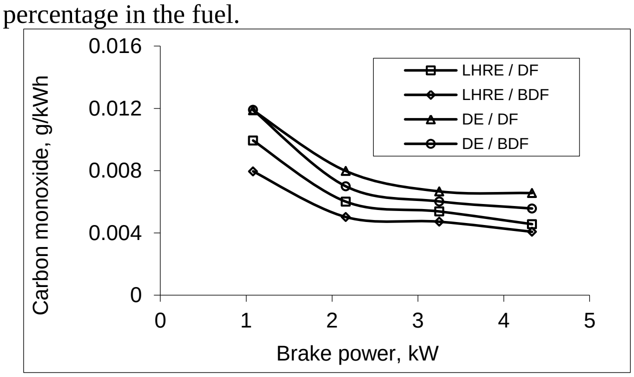

Fig. 9 Variation of carbon monoxide with engine power output

I. *Unburned Hydrocarbon*

The variation of hydrocarbon (HC) with respect to engine power output for different fuels are shown in figure 10.The high operating temperature in LHR engine makes the combustion nearly complete than the limited operating temperature condition as in the case of diesel engine. At full load hydrocarbon emission levels are decreases for LHR engine fueled with biodiesel than LHR engine fueled with diesel and diesel engine fueled with diesel and biodiesel such as 9.8%, 17.21% and 21.31% respectively. The air fuel mixture, which was accumulated in the crevice volume, was reduced due to the high temperature and availability of oxygen, which in turn leads to reduction in unburned hydrocarbon emissions.

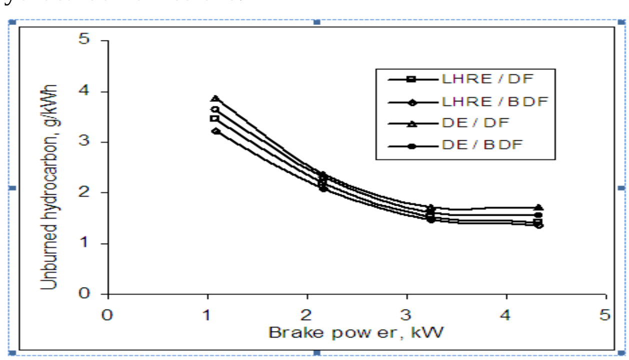

Fig. 10 Variation of hydrocarbon with engine power output

J. Oxides Of Nitrogen

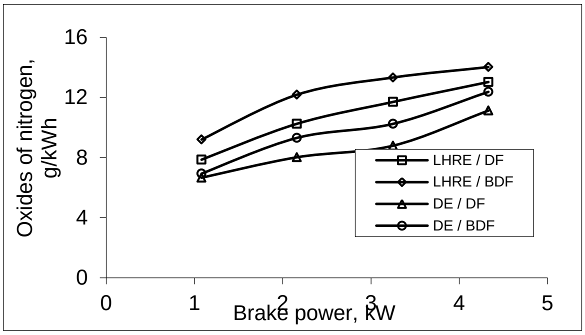

Fig. 11 Variation of oxides of nitrogen with engine power output

Figure 11 shows the variation of oxides of nitrogen with engine power output. The main reason for the formation of oxides of nitrogen in an IC engines are high temperature and availability of oxygen. At maximum load, NOx emission for LHR engine with biodiesel fuel is higher about 7.09%, 13.35 and 20.37% respectively than LHR engine fueled with diesel and conventional diesel engine with biodiesel and diesel. In LHR engine, the operating conditions are in favor of NO species and such as the availability of oxygen in the fuel itself other than the oxygen available in the air and high temperature due to insulation coating, which enhance the NO species formation.

K. Particulate Matter

Fig.12 shows the variation of particulate matter with engine power output. The particulate matter of LHR engine with biodiesel fuel is higher about 3.1% than LHR engine with diesel and lower by about 3% and 11.1% respectively than conventional diesel engine fueled with diesel and biodiesel. It is clearly found that, the particulate matter for biodiesel was higher irrespective of the engine used compared with the diesel fuel. This is due to the incomplete combustion of biodiesel fuel.

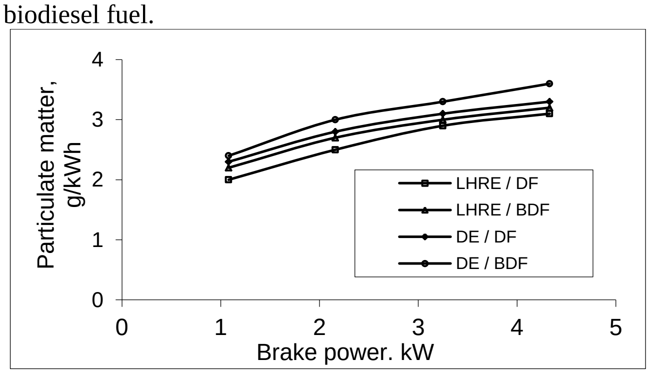

Fig. 12 Variation of particulate matter with sengine power output

VII. CONCLUSION

The biodiesel produced from Jatropha oil by transesterification process reduces the viscosity of the oil in order to match the suitability of diesel fuel. The diesel engine is modified in to LHR engine by means of partially stabilized zirconia (PSZ) coating. The various combustion parameters such as cylinder pressure, rate of heat release, cumulative heat releases were analyzed and the following conclusions were arrived it.

i. At full load condition, the cylinder pressure in the case of biodiesel fueled LHR engine was lower than that of the diesel fueled LHR engine. Even though this reduction under identical condition is substantial. The absolute value of this cylinder peak pressure is well within operating limits of the test engine.
ii. The final analysis of the heat release shows that, the value of net heat release in the case of biodiesel fueled LHR engine is substantially good enough for the effective work done of the test engine.

The performance characteristics such as brake thermal efficiency, specific fuel consumption and specific energy consumption and various emission characteristics were compared and summarized as follows.

i. The maximum efficiency obtained in the case of LHR engine fueled with biodiesel was lower than the LHR engine operated with diesel fuel. However the efficiency of the LHR engine with biodiesel fuel is well within the expected limits.
ii. The exhaust gas temperature of LHR engine fueled with biodiesel was lower than LHR engine fueled with diesel throughout the operating condition. The low exhaust gas temperature indicates the heat release rate during the late combustion was comparatively lower than diesel fuel.
iii. The specific fuel consumption of LHR engine with biodiesel was higher than LHR engine fueled with diesel. The higher consumption of fuel due to low calorific value and high viscosity. Even though it could be expected to the offset by the cost of biodiesel.
iv. The specific energy consumption of LHR engine with biodiesel was higher than LHR engine fueled with diesel fuel.
v. It was found that, CO and HC emissions for LHR engine with biodiesel was considerably lower than LHR engine fueled with diesel. This reduction of emissions due to excess oxygen availability along with higher operating temperature.
vi. NO emission for LHR engine with biodiesel fuel was higher than LHR engine fueled with diesel. The operating conditions of LHR engine were favorable to NO formation. However this increase in emission level was within the acceptable limits.
vii. The particulate matter of LHR engine with biodiesel fuel is higher than LHR engine fueled with diesel due to incomplete combustion.

The above comparative study clearly reveals the possibility of using the biodiesel in LHR direct injection diesel engine. The combustion, performance and emission characteristics show the suitability of biodiesel in LHR engine.

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Nomenclature

LHRE	-	Low heat rejection engine
BDF	-	Biodiesel fuel
DE	-	Diesel engine
DF	-	Diesel fuel
DI	-	Direct injection
PSZ	-	Partially stabilized zirconia
BTE	-	Brake thermal efficiency, %
CV	-	Calorific value, MJ/kg
CO	-	Carbon monoxide, g/kWh
HC	-	Hydrocarbon, g/kWh
NO	-	Nitric oxide, g/kWh
NO_x	-	Oxides of nitrogen, g/kWh
EGR	-	Exhaust gas recirculation
LHR	-	Low heat rejection
SEC	-	Specific energy consumption, kJ/kWh
SFC	-	Specific fuel consumption, kg/kWh
TDC	-	Top dead center
BTDC	-	Before top dead center
BIS	-	Bureau of Indian standards
U	-	Internal energy, kJ/kg K
Q	-	Heat transfer rate, kJ/s
m	-	Mass of fuel. KJ/s
h	-	Enthalpy, kJ/kg
p	-	Pressure, bar
V	-	Volume, m^3
t	-	Time, s
T	-	Temperature, K
C_v	-	Specific heat at constant volume, kJ/kgK
λ	-	Equivalence ratio