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Energy and Exergy Analysis of CI Engine Using Dual Biodiesel Blends with Diesel

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ABSTRACT

Many researchers were carried out the experiments with single biodiesel blends with diesel. But in this analysis, dual biodiesel blends is used to improve the performance of the engine with better energy. Neem oil and Pongamia pinnata oil is selected based on their availability and blended with diesel to form DPN (Diesel:Pongamia pinnata:Neem) biodiesel and their physical and chemical properties are compared with diesel. In this paper, energy and exergy analysis are applied to the experimental data of a direct injection, single cylinder, 4 stroke diesel engine using different kinds of double biodiesel blends with diesel. At peak load, the energy and exergy efficiencies of DPN 1 (90:5:5) are 25.6% and 46.1% respectively, which shows better result than other blends. Among the six blends, DPN 1 is having lower exhaust temperature of 305^{o}C at peak load. DPN 1 blend gives lower BSFC, exhaust temperature, HC, CO, CO_2, NOx and smoke opacity than other blends. The results showed that the DPN 1 (90:5:5) is having good energy and exergy values with diesel compared to other combination.

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INTRODUCTION

Due to the economic growth, the energy demand of the world is increasing every year. In the energy conversion process in diesel engines, energy efficiency is a key factor. Number of researchers studied on thermodynamics of efficiency, irreversibility and Exergy in compression-ignition engines. The thermal efficiency and the energy loss in the system are normally calculated from the 1st law of thermodynamics, while the maximum work output or Exergy is characterized by the 2nd law of thermodynamics. The Exergy of a system from the 2nd law is defined as the maximum useful mechanical work which can be produced as to thermal, mechanical and chemical equilibriums with its environment through reversible processes. Unlike energy, Exergy is not a conserved quantity and it can be destroyed during the process due to irreversibility. The Exergy loss was mainly occurred by the irreversibility of the combustion process, comparing with the Exergy losses from the heat transfer through the combustion chamber and the cooling water system (Sangsawang *et al*., 2012).

Ekrem (2010), who investigated the performance, emission and combustion of a diesel engine using neat rapeseed oil and its blends of 5%, 20%, 70%, and diesel fuel separately. The results indicated that the use of biodiesel produces lower smoke opacity and higher brake specific fuel consumption (BSFC) compared to diesel fuel. Rakapoulos *et al*., (2006) conducted the experiments on performance and emission analysis of diesel engine using various biodiesel samples like cottonseed oil, rapeseed oil, sunflower oil etc. Nitrogen Oxides (NOx) emissions were slightly reduced with the use of cottonseed and sunflower bio-diesel blends with respect to that of the neat diesel fuel. Kumar and Kumar (2012) performed the experimental investigation on C.I.engine with biodiesel blends of cotton seed methyl esters and neem oil methyl esters. It was observed that the smoke and emissions for the blends of cottonseed and neem are less, as compared to pure diesel. Properties of the 20% blend of cottonseed are nearer to the diesel fuel and it has the nearer value of performance with pure diesel and has low emission of carbon particulates. Arun Balasubramanian *et al*., (2013) investigated the possibilities of mixture of two biodiesel blends with diesel. Combinations of cottonseed and mustard oil with diesel (CMD) and pongamia pinnata and mustard oil with diesel (PMD) were taken for experimental analysis. For the maximum load, the value of specific fuel consumption and thermal efficiency of CPD-1 blend (10:10:80) was close to the diesel values. Azoumah *et al*., (2009) suggests the use of Exergy analysis combined with gas emissions analysis to optimize the performance of a compression ignition (CI) engine using biofuels such as cottonseed and palm oil in pure form or blended with diesel for different engine loads. Debnath *et al*., (2013)

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performed the experiments with palm oil methyl ester at 1500 rpm. The exergy analysis has shown that around 30% of the fuel exergy input that can be converted to useful means of energy or exergy. Sorathia and Yadav (2012) showed that the diesel-biogas dual fuel mode produced lower energy conversion efficiency and energetic end exergetic performance parameters of the engine computed and compared with each other. Pandiyarajan *et al*., (2011) experimented the second law analysis of diesel engine with diesel fuel. Energy analysis indicates that 15.2% of the total energy from the fuel input was recovered and only 1.6% of the chemical availability of the fuel is saved. Ismet sezer, (2011) investigated the thermodynamic and performance analysis of diesel engine with diethyl and dimethyl ether and declared that the brake power declines about 32.1% and 19.4% at 4200 rpm while brake specific fuel consumption increases about 47.1% and 24.7% at 2200 rpm respectively. The specific fuel consumption is also higher about 43.5% for dimethyl ether and 23.6% for diethyl ether than diesel fuel. Costa *et al*., (2012) performed the energetic and exergetic analysis of dual-fuel engine with the diesel engine. The energy efficiency ranged from 15.7% to 37.9% in diesel and 10 to 55.1% in dual mode, to power changing from 10 to 150 kW. The exergetic efficiency ranged from 14.6 to 35.4% in diesel pure mode and from 9.6 to 52.4% in dual mode.

Nomenclature:

A_d	Destructed availability (kW)
A_e	Exhaust gas availability (kW)
A_{in}	Input availability (kW)
A_s	Shaft availability (kW)
A_w	Cooling water availability (kW)
C	Carbon
CO	Carbon monoxide
CO_2	Carbon dioxide
C_{pw}	Specific heat capacity of water (kJ/kg K)
C_{pe}	Specific heat capacity of exhaust gas (kJ/kg K)
H	Hydrogen
HC	Hydro Carbon
LHV	Lower heating value (kJ/kg)
m_a	Mass flow rate of air (kg/s)
m_e	Mass flow rate of exhaust gas (kg/s)
m_f	Mass flow rate of fuel (kg/s)
m_w	Mass flow rate of cooling water (kg/s)
N	Speed of the engine (rpm)
NOx	Nitrogen Oxides
O	Oxygen
P_{amb}	Ambient pressure (kPa)
P_e	Exhaust gas pressure (kPa)
Q_e	Exhaust gas energy (kW)
Q_{in}	Input (fuel) energy (kW)
Q_s	Shaft energy (Brake power) (kW)
Q_u	Unaccounted energy (kW)
Q_w	Cooling water energy (kW)
R_e	Specific gas constant (kJ/kgK)
R_u	Universal gas constant (J/mol K)
S	Sulphur
T	Torque (N-m)
T_{wi}	Cooling water inlet temperature (^oC)
T_{wo}	Cooling water outlet temperature (^oC)
T_{amb}	Ambient temperature, (^oC)
T_{ei}	Exhaust gas inlet temperature, (^oC)
η_{ex}	Exergy efficiency
DPN 1	90 % Diesel+ 5% Pongamia pinnata biodiesel+ 5% Neem biodiesel
DPN 2	80 % Diesel+ 10% Pongamia pinnata biodiesel+ 10% Neem biodiesel
DPN 3	70 % Diesel+ 15% Pongamia pinnata biodiesel+ 15% Neem biodiesel
DPN 4	60 % Diesel+ 20% Pongamia pinnata biodiesel+ 20% Neem biodiesel
DPN 5	50% Diesel + 25% Pongamia pinnata biodiesel + 25% Neem biodiesel
DPN 6	0% Diesel + 50% Pongamia pinnata biodiesel + 50% Neem biodiesel

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Objective:

Most of the researchers were experimented the diesel engine with single biodiesel blended with diesel for optimum results. In this paper, the effects of two dissimilar biodiesel (pongamia pinnata and neem biodiesel) are combined in various blending ratio's which are analyzed based on first law and second law of thermodynamics. The theoretical investigation on the effective distribution of energy at various components of IC engine has been done by coupling the first and second laws of thermodynamics together. This depicts thermodynamic energy distribution of engine. However, in order to establish the combination on neem oil and cottonseed oil as an alternative to the diesel fuel, it is necessary to uncover the effect of engine operating parameters on thermo-mechanical energy-exergy distribution. Initially the combination of neem oil biodiesel and cottonseed oil biodiesel along with diesel were taken for the experimental analysis. Experiments were conducted using a single cylinder, 4 stroke, and direct injection diesel engine with different loads rated at 1500 rpm. The engine characteristics of six blends of biodiesel were compared, and then the exergetic analysis of the engine was done. The results obtained from the tests were then analyzed to explore the energy and exergy potential of fuel input, shaft work, cooling water and exhaust gas potential and exergy destruction. Also the effect of variation of engine parameters like brake specific fuel consumption, brake thermal efficiency, brake power, exergy efficiency and exhaust temperature are analyzed and discussed.

Experimental Setup and Procedure:

The experiments were conducted in a single cylinder, four stroke, direct injection Kirloskar engine fuelled with DPN biodiesel. Initially the biodiesel can be prepared by means of transesterification process with NaOH as a catalyst. The properties like kinematic viscosity, calorific value, flash point temperature for the test fuels (DPN1, DPN 2, DPN 3, DPN 4, DPN 5 and DPN 6) and diesel are analyzed by ASTM procedure. The properties are listed in the Table 1. It was observed that the property values lies within the range of biodiesel standards. The equipments like Ostwald viscometer, Pensky Martins closed cup apparatus, Digital bomb calorimeter are used to find out the properties of the fuel. From the Table 1, it was observed that DPN 1 is having desirable properties corresponding to diesel fuel than the other blends.

Table 1: Fuel properties of biodiesel and its blends.

Sl.No.	Test fuels	Density (kg/m^3)	Kinematic Viscosity (cSt)	Calorific Value (MJ/kg)	Flash Point Temperature (^{o}C)
	ASTM Method	D1298	D420	D 0445	D 0093
1	Diesel	830	4.0	46.6	58
2	DPN 1	832	4.1	46.1	69
3	DPN 2	834	4.3	45.4	80
4	DPN 3	837	4.4	45.0	91
5	DPN 4	840	4.6	44.5	102
6	DPN 5	844	4.7	44.1	113
7	DPN 6	856	5.3	41.5	168

The performance of test fuels were analyzed in a Kirloskar make single cylinder, four-stroke, direct injection diesel engine coupled with loading device. Experiments were conducted with varying loads while engine torque was kept constant. Fuel flow rates were measured with calibrated burette. The exhaust gas temperatures were measured using thermocouple. Parameters like brake specific fuel consumption, brake thermal efficiency and brake specific energy consumption were analyzed for different load conditions. The emissions from the engine were measured after the engine reached the steady working condition. The AVL make smoke meter is utilized to find the smoke opacity of exhaust gas. The Crypton make exhaust analyzer is used to measure the carbon monoxide (CO), carbon dioxide (CO_2), hydro carbon emission (HC) and nitrogen oxides (NOx). The experimental setup is shown in Fig. 1 and the Table 2 shows the test engine specifications.

Table 2: Test Engine Specifications.

Items	Specifications
Model	AV1
Made	Kirloskar
Type	Single cylinder, Four stroke
Bore x Stroke	87.5 x 110 mm
Rated Output	5.2 kW / 1500 rpm
Compression ratio	17.5 : 1
Injection Pressure	13.5 MPa
Type of Cooling	Water Cooling

Error Analysis:

The percentage error of various instruments that are utilized for this experiment is calculated. Percentage error is the ratio of minimum scale to the minimum value measured. The Table 3 listed the percentage of error for various instruments.

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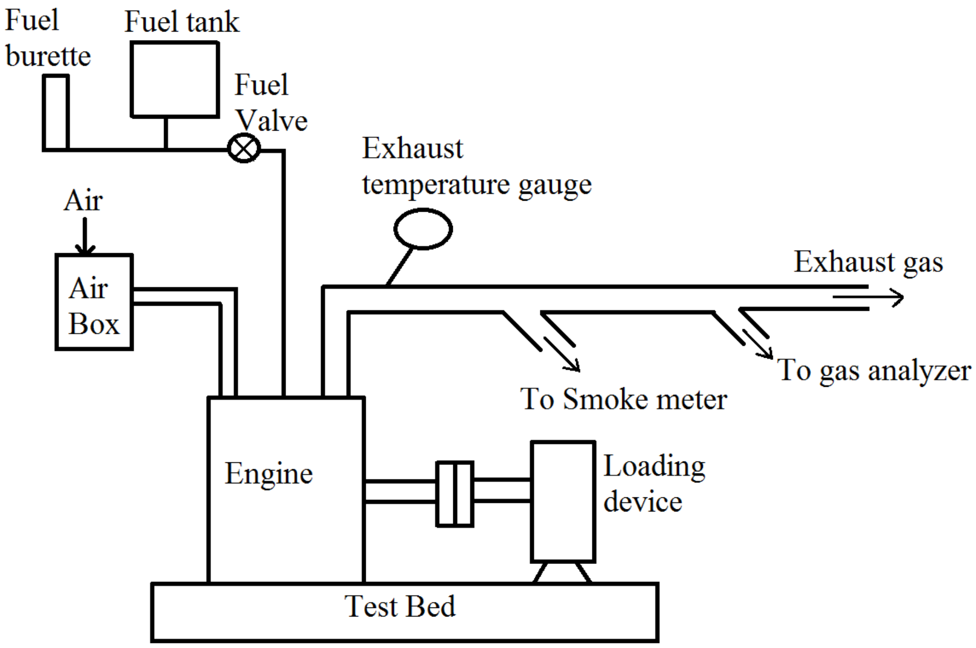

Fig. 1: Experimental setup.

Table 3: Error analysis.

Sl. No	Instruments	Error, %
1	Stop watch	1.25
2	Burette for fuel measurement	1.0
3	Digital bomb calorimeter	1.0
4	Thermometer	1.0
5	Weight balance	1.0
6	Manometer	1.0
7	Smoke meter	0.6
8	Exhaust Temperature Gauge	5.0

Energy And Exergy Analysis:

Energy Analysis:

In a CI engine, the fuel energy supplied per unit time (Q_{in}) is transferred in its different processes, shaft power (Q_s), energy in cooling water per unit time (Q_w), energy in exhaust gas per unit time (Q_e) and unaccounted energy losses per unit time (Q_u) in the form of friction, radiation, heat transfer to the surrounding, operating auxiliary equipments, etc. These different forms of energies are calculated according to the following analytical expressions. Fuel energy is calculated by inserting the value of mass flow rate and lower heating value in Eq.(1).

Fuel energy supplied per unit time, $Q_{in} = m_f \times LHV$, kW (1)

Torque and speed of the engine is substituted in Eq.(2) and calculated the shaft power of the engine.

Shaft power = Brake power of the engine, $Q_s = \frac{2\pi NT}{60}$, kW (2)

Mass of the water, specific heat value and inlet and outlet temperatures are substituted in Eq.(3) and the energy in cooling water is determined. Similarly energy in exhaust is determined using Eq.(4). The unaccounted energy loss is determined by Eq.(5).

Energy in cooling water per unit time, $Q_w = m_w \times C_p \times (T_{wo} - T_{wi})$, kW (3)

where, C_{pw} = Specific heat capacity of water.

Energy in exhaust gas per unit time, $Q_e = m_e \times C_p \times (T_{ei} - T_{amb})$, kW (4)

where, C_{pe} = Specific heat capacity of exhaust gas.

Unaccounted energy losses per unit time $Q_u = Q_{in} - (Q_s+Q_w+Q_e)$, kW (5)

Exergy analysis:

The availability can be described as the ability to perform useful amount of work by the supplied energy. In a CI engine the availability of the fuel (A_{in}) supplied is converted into different types of exergy, i.e., shaft availability (A_s), cooling water availability (A_w), exhaust gas availability (A_e) and destructed availability (A_d) in the form of friction, radiation, heat transfer to the surrounding, operating auxiliary equipments, etc. These forms of energies are calculated according to the following analytical expressions.

Inserting the values for the mass flow rate, lower heating value, and mass fractions equation (6) result in the input availability of the fuel.

$$A_{in} = m_f \times LHV \times \left\{ 1.0401 + 0.178\,(\frac{H}{C}) + 0.0432\,(\frac{O}{C}) + 0.2169\,(\frac{H}{C})[1 - 2.0628\,(\frac{H}{C})] \right\} \quad (6)$$

Shaft availability, A_s is a measure of brake power of the engine.

$$A_s = \text{Shaft availability of the engine} = Q_s = \frac{2\pi NT}{60}\text{, kW} \quad (7)$$

Cooling water availability (A_w) is calculated by inserting the values of cooling water energy, specific heat value and measured temperature values in Eq. (8).

$$A_w = Q_w \quad [m_{we} \times C_{pw} \times T_{amb} \times \ln\frac{T_{wo}}{T_{wi}}]\text{, kW} \quad (8)$$

Similarly exhaust gas availability (A_e) is found out by means of Eq. (9).

$$A_e = Q_e + [(m_f + m_a) \times T_{amb} \times \{C_{pe} \ln(\frac{T_{amb}}{T_{ei}}) \quad R_e \times \ln(\frac{P_{amb}}{P_{ei}})\}]\text{, kW} \quad (9)$$

Where $R_e = R_u$/Molecular weight

Destructed availability (A_d) is calculated by substituting the values of A_{in} , A_s, A_w and A_e in Eq. (10).

$$A_d = A_{in} - (A_s + A_w + A_e)\text{, kW} \quad (10)$$

Exergy efficiency (η_{ex}) is found out by means of,

$$\eta_{ex} = 1 - (\frac{A_d}{A_{in}})\text{, \%} \quad (11)$$

RESULTS AND DISCUSSION

Energy Analysis:

The fuel energy, Q_{in} enters the engine with each fuel and losing energy values (heat transfer by cooling water and exhaust losses). Fuel energy is directly proportional with the heating value of the fuel. The engine performance of the DPN biodiesel blends was evaluated by means of brake specific fuel consumption, brake thermal efficiency and engine exhaust temperature at different load conditions of the engine. Various comparisons are made between the performance parameters of the six blends and the results are discussed elaborately.

The effect of load on brake specific fuel consumption and thermal efficiency of DPN blends are shown in Fig. 2. Brake specific fuel consumption (BSFC) is nothing but the ratio of mass of fuel consumption and effective power whereas brake thermal efficiency indicates the engine's ability to convert chemical energy to mechanical energy and it is the ratio of brake power to the fuel energy input. In general BSFC was found to increase with increasing proportion of biodiesel blend and decreases sharply with increase in load. However, brake thermal efficiency of diesel engine obtained for different proportion of DPN blends are increased with the increasing in load.

Specific fuel consumption of DPN blends is having quite closer values with diesel fuel. Of all the six blends of DPN biodiesel, DPN 1 has lower BSFC than other dual biodiesel blends. At full load condition the value of BSFC for DPN 1 is 0.350 kg/kWh while DPN 6 is having the value of 0.38 kg/kWh. The loss of heating value of DPN biodiesel may be compensated with higher fuel consumption. Since the specific fuel exergy is associated to the calorific value, the fuel exergy values for the DPN blends are maintain a resemblance to the fuel energy values.

At maximum load, thermal efficiency of DPN 1 is 25.6% whereas DPN 2 is 25%. It is due to higher calorific energy of DPN 1 biodiesel compared to DPN 2. The thermal efficiency values decreases with increase in biodiesel content in the blends at all possible load conditions.

The effect of brake power on the exhaust gas temperature and nitrogen oxides for different fuels is shown in Fig.3. In general the exhaust gas temperature and NOx are increased with increase in engine loading for all the fuels proportion tested. This increase in exhaust gas temperature with load is obvious from the simple fact that more amount of fuel was required in the engine to generate that extra power needed to take up the additional loading. The exhaust gas temperature was found to increase with the increasing proportion of biodiesel. The higher average gas temperature, the presence of fuel oxygen and residence time at higher load conditions with the blend combustion caused higher NOx emissions.

DPN 1 has lower value of exhaust temperature than other dual biodiesel, due to increased amount of biodiesel content in it. The biodiesel content in the blends plays a major factor for the increase in temperature of exhaust gas at all possible load conditions. At maximum brake power, amount of NOx produced for DPN 1 is

324 ppm whereas DPN 2 produced 330 ppm. DPN 6 produced higher NOx because energy in exhaust gas is higher than other blends.

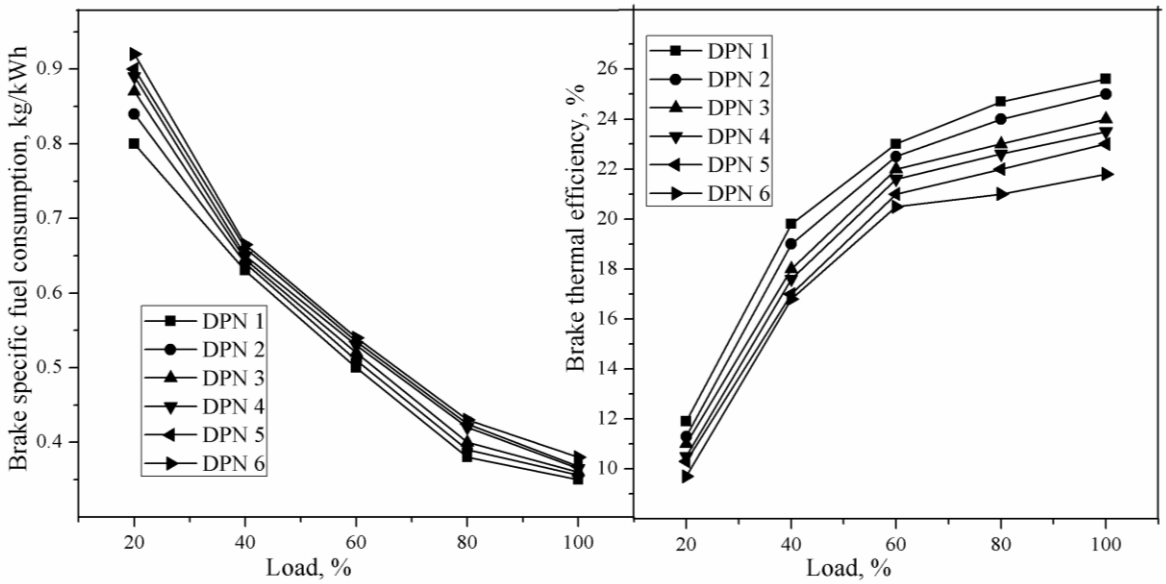

Fig. 2: Effect of load on SFC and brake thermal efficiency.

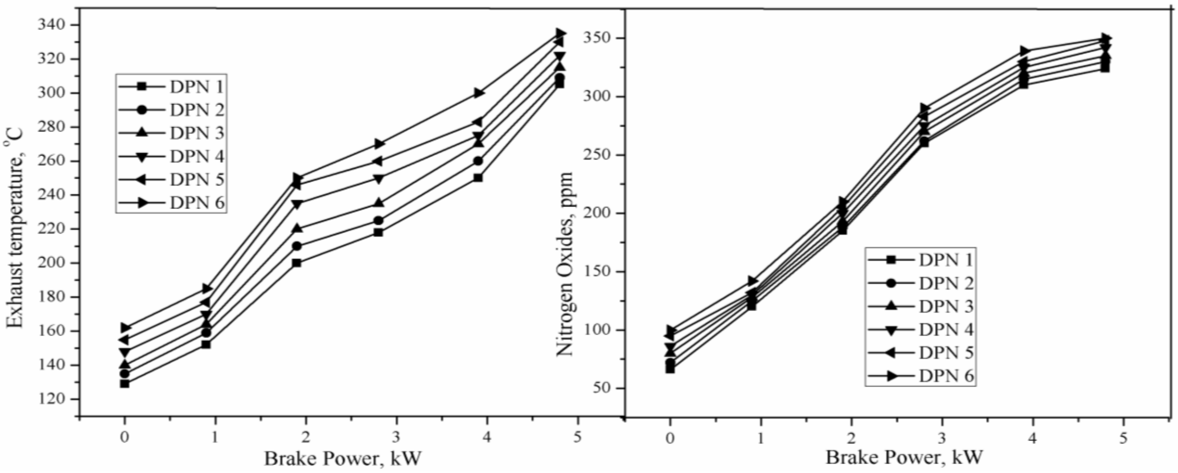

Fig. 3: Effect of brake power on exhaust gas temperature and NOx.

The variations of engine load on Carbon monoxide and carbon dioxide are shown in Fig.4. With increasing in engine load, the emissions of CO and CO_2 are increased. DPN 1 blend gives lower CO and CO_2 than other blends. This is due to the oxygen contents in the biodiesel which makes easy burning at higher temperature in the cylinder. DPN 6 blend gives higher CO and CO_2 than other blends. This is due to the high viscosity; the air-fuel mixing process is affected by the difficulty in atomization and vaporization of dual biodiesels. Moreover, DPN 6 has lower input fuel energy and higher exhaust gas energy. Higher the engine load, richer fuel-air mixture is burned and thus more CO is produced.

The variations of engine load on hydro carbon and smoke opacity emissions are shown in Fig.5. DPN 1 blend gives lower HC and smoke opacity than other blends. The DPN blends generally exhibit lower HC emission at lower engine loads and higher HC emission at higher engine loads. This is because of relatively less oxygen available for the reaction when more fuel is injected into the engine cylinder at high engine load. DPN 6 gives higher HC and smoke due to the higher density and viscosity. The high viscosity of pure biodiesel deteriorates the fuel atomization, and increases exhaust smoke.

Exergy Analysis:

The diesel engine is experimented with different kinds of DPN blends at no load, partial and full load conditions to evaluate the exergy. The exergy analysis of the engine is calculated by the formulas of 2^{nd} law of thermodynamics. Exergy is not conserved, and the irreversible processes in the engine, such as combustion, heat transfer, mixing, friction, etc., destroy a significant fraction of the fuel exergy. Due to the rate of exergy destroyed in the engine increases with increasing the combustion temperature, this exergy destruction can be reduced by preheating the application of energy and exergy analyses to a CI Engine combustion air and by reducing the amount of excess air. However, the higher temperatures may result in higher heat transfer which will remove the available energy. In addition, if not utilized, the higher availability will be expelled with the exhaust gases.

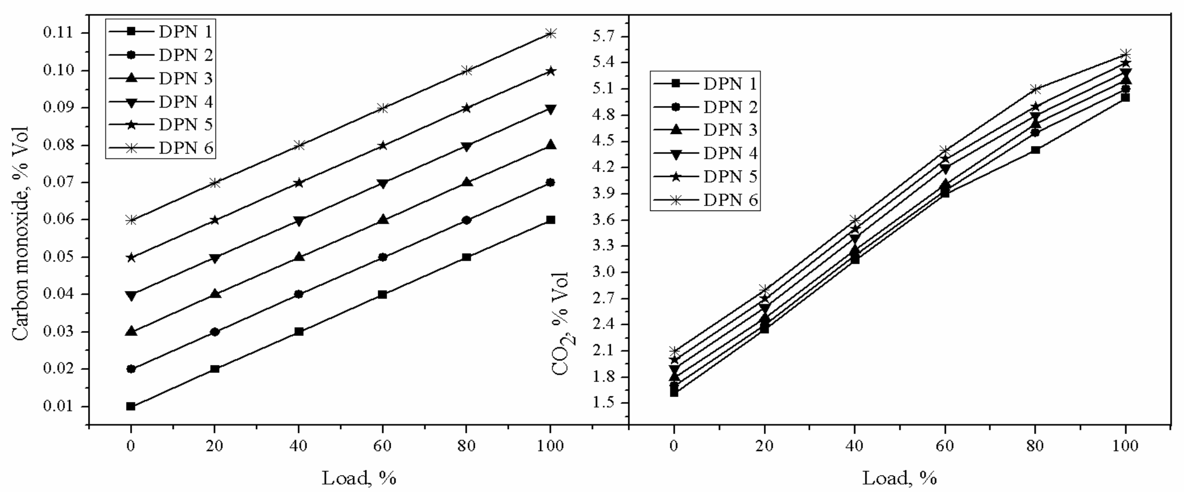

Fig. 4: Variations of engine load on CO and CO_2.

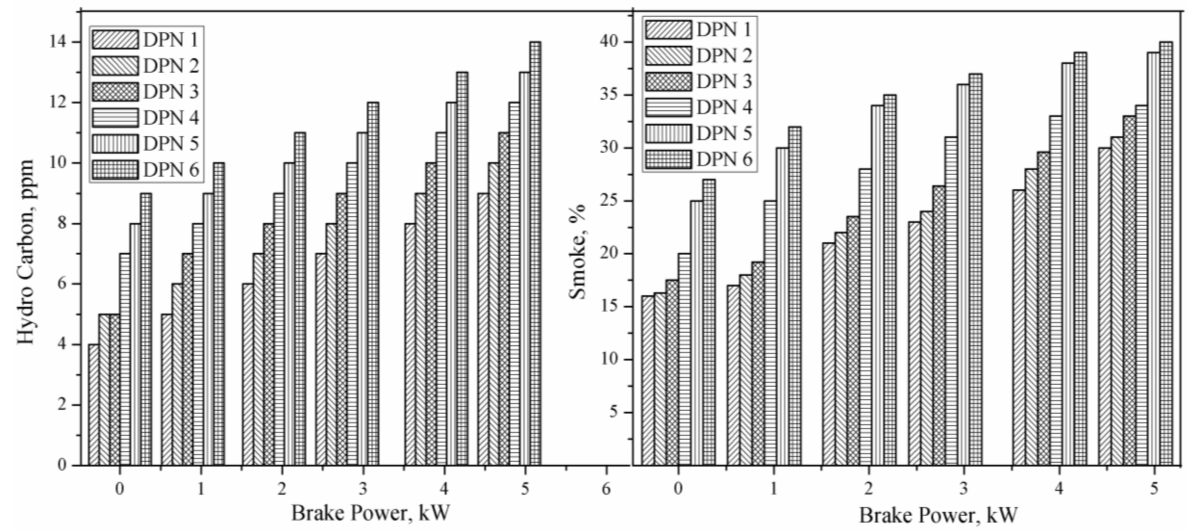

Fig. 5: Variations of engine load on HC and smoke.

The fuel energy (Q_{in}) values of DPN blends and the conversion of this fuel energy into useful shaft power (Q_s) output, energy losses in the engine by means of cooling water (Q_w) and exhaust gases (Q_e) are as shown in Fig. 6.

The various exergies like availability of the fuel (A_{in}), shaft availability (A_s), cooling water availability (A_w), exhaust gas availability (A_e) and destructed availability (A_d) of the different kinds of DPN blends is also shown in Fig. 6. For the same operating condition, all the exergies for DPN 1 is higher than DPN 6. Because, the net calorific value of DPN 1 is greater than that of the other DPN blends. The most important factor of the system inefficiency is the destruction of exergy (A_d) by irreversible processes. This indicates that improving the performance of the engine is of more importance than the recovery of low grade energy loss. However, from the exergy method, it is found that the actual exergy losses are insignificant compared to the irreversibility losses in the engine. By identifying and quantifying the exergy destruction or irreversibilities for specific engine parameters and parameters the engine performance can be improved. In this study, DPN1 is better than other DPN blends based on the exergy analysis.

The effects of load on thermal and exergy efficiencies are shown in Fig.7. The exergetic efficiency takes into account not only the first but also the second law of thermodynamics; it provides a better measure of the performance for a thermal system. DPN 6 gives higher exergy than DPN 1, because exergy losses are proportional to the heat flow rate. The higher the temperature of the engine surface from where the heat is rejected, the higher the exergy loss accompanying it is. Both the exergy loss accompanying heat loss and the rate of exergy destroyed by the combustion could be decreased by insulating the combustion chamber walls of the engine.

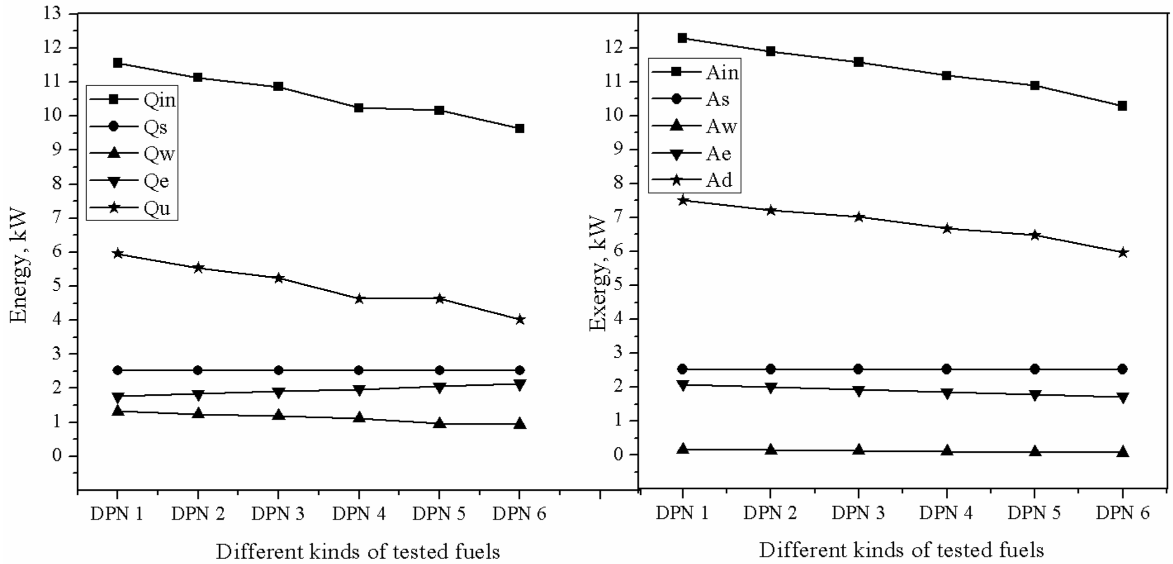

Fig. 6: Effect of brake power on input (fuel) and exergy of the DPN blends.

The first-law efficiency (thermal efficiency) is higher for DPN 1 because of same brake power and great fuel energy input (Q_{in}). It is seen that the fuel exergy inputs of DPN 1 blend is higher than the other DPN blends. At the same time, exergetic efficiencies of the engine follow distinct trends with the brake thermal efficiencies.

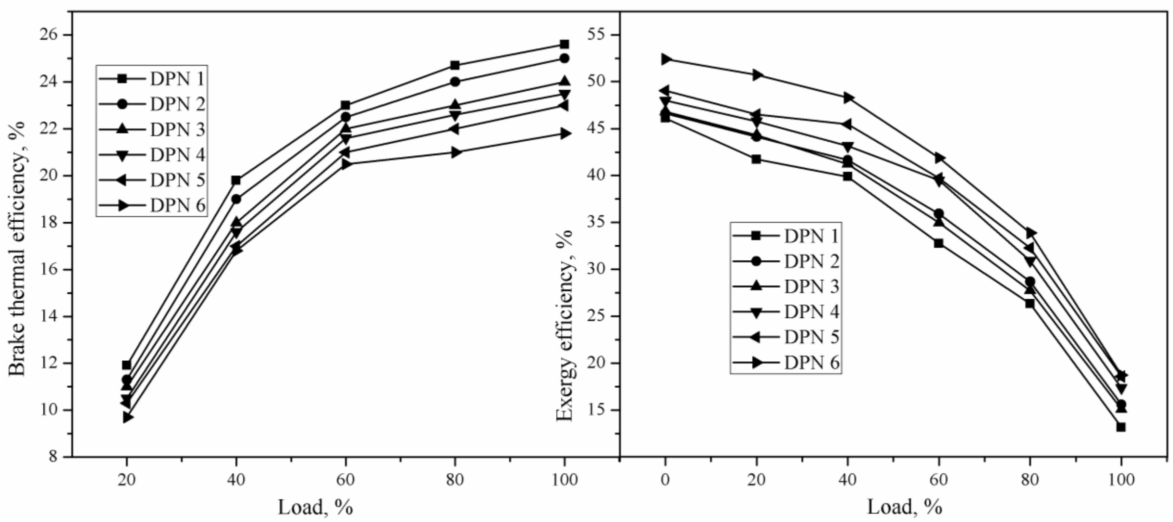

Fig. 7: Effect of load on brake thermal and exergy efficiencies.

Conclusion:

The energetic and exergy analysis of a single cylinder four stroke diesel engine using the dual biodiesel fuels of DPN 1, DPN 2, DPN 3, DPN 4, DPN 5 and DPN 6 have been performed. In this study, test engine was operated at constant speed and varying load conditions, without modifications to engine or injection system. The energy and exergy values are computed using experimental data and compared for all the six DPN blends. All energy and exergy performance parameters are increased with increasing engine load.

The thermal efficiency of engine fuelled by DPN 1 was slightly higher than other DPN blends due to the higher combustion efficiency and the lower heat losses. However, the exergy efficiency of engine fuelled by DPN 6 was slightly higher than the value of DPN 1. This is due to exergy losses are directly proportional to the heat flow rate.

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For the same operating condition, all the exergies (A_{in}, A_s, A_w, A_e and A_d) for DPN 1 is higher than other blends. Because, the net calorific value of DPN 1 is greater than that of the other DPN blends. DPN 1 blend gives lower BSFC, exhaust temperature, HC, CO, CO_2, NOx and smoke opacity than other blends. DPN 1 blend gives higher thermal efficiency than other blends. From the experimental test of this study, it can be confirmed that DPN 1 can be used for diesel engine for agricultural.

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