ENGINE PERFORMANCE AND EMISSION CHARACTERISTICS OF SINGLE CYLINDER DIESEL ENGINE WORKING ON VARIOUS BLENDS OF PONGAMIA OIL

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ABSTRACT

Biodiesel is an alternative to conventional diesel fuel made from renewable resources. The oil from seeds (e.g Jatropa and Pongamia) can be converted to a fuel commonly referred as "biodiesel" no engine modifications are required to use biodiesel in place of petroleum- based diesel. Biodiesel can be mixed with petroleum – based diesel in any proportion. Biodiesel is reduces the Exhaust emissions. The climate change is presently an important element of energy use and development. The use of biodiesel resulted in lower emissions of unburned hydrocarbons, Carbon monoxide and particulate matter to measure the emissions such as carbon monoxide, particulate matter, and nitrous oxide exhaust gas analyzer is used.

As comparing of exhaust emissions diesel fuel procedure more emissions and biodiesel produce less emissions.

1.0 INTRODUCTION

Biodiesel is defined as mono-alkyl esters of long chain fatty acids derived from vegetable oils or animal fats which conform to ASTM D6751 (American Society for Testing & Materials). It is the name of a clean burning alternative fuel, produced from domestic, renewable resources and animal fats. Today's diesel engines require a clean –burning, stable fuel that performs well under a variety of operation conditions. It is the only alternative fuel that can be used directly in any existing, unmodified diesel engine. Because it has similar properties to petroleum diesel fuel, biodiesel can be blended in any ratio with petroleum diesel fuel. Specifications for use in diesel engines. Biodiesel refers to the pure fuel before blending with diesel fuel. Biodiesel blends are denoted as "BXX" with "XX" representing the percentage of biodiesel contained in the blend (ie: B20 is 20% biodiesel, 80% petroleum diesel). It is simple to use, biodegradable, nontoxic, and essentially free of sulfur and aromatics. It is made though a chemical process called transesterification where by the glycerin is separated from the fat or vegetable oil. Fuel-grade biodiesel must be produced to strict industry specifications in order to insure proper performance. It is better for the environment because it is made from, renewable resources and has lower emissions compared to petroleum diesel. It is less toxic than table salt and biodegrades as fast as sugar. It can be made in India from renewable resources such as Jatropha and Pongamia. Its use decreases our dependence on foreign oil and contributes to our own economy.

Dr. Rudolf diesel actually invented the diesel engine to run on a myriad of fuels including coal dust suspended in water, heavy mineral oil and you guessed it, vegetable oil. Dr. Diesel's first engine experiments were catastrophic failures. But by the time he showed his engine at the World Exhibition in Paris in 1900, his engine was running on 100% peanut oil.

Biodiesel Production

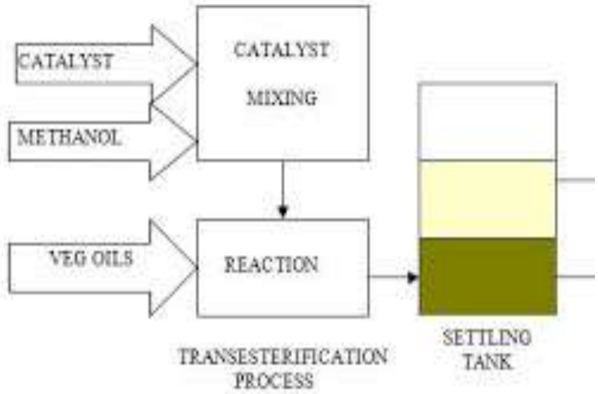
The production of Biodiesel, or alkyl esters, is well known. There are three basic routes to ester production from oils and fats.

1. Base catalyzed transesterification of the oil with alcohol.
2. Direct acid catalyzed etherification of the oil with methanol.
3. Conversion of the oil to fatty acids, and then to alkyl esters with acid catalysis.

The majority of the alkyl esters produced today are done with the base catalyzed reaction

Because it is the most economic:

- Low temperature (150F) and pressure (20psi) processing.
- High conversion (98%) with minimal side reactions and reaction time.
- Direct conversion to methyl ester with no intermediate steps.



VEG OIL: Jatropha, pongamia.
ALCOHOLS: Methanol, Ethanol.
CATALYST: Sodium hydroxide, Potassium hydroxide.

2.0 LITERATURE SURVY

Straight vegetable oils (SVO) can be used directly as a fossil diesel substitute; however, using this fuel can lead to some fairly serious engine problems. Due to its relatively high viscosity SVO leads to poor atomisation of the fuel, incomplete combustion, coking of the fuel injectors, ring carbonisation, and accumulation of fuel in the lubricating oil. The best method for solving these problems is the transesterification of the oil to produce biodiesel. Biodiesel is an alternative fuel similar to conventional or 'fossil' diesel. Biodiesel can be produced from straight vegetable oil, animal oil/fats, tallow and waste cooking oil. The process used to convert these oils to Biodiesel is called transesterification. The largest possible source of suitable oil comes from oil crops such as rapeseed, palm or soybean. In the UK rapeseed represents the greatest potential for biodiesel production. Most biodiesel produced at present is produced from waste vegetable oil sourced from restaurants, chip shops, industrial food producers, etc. Though oil straight from the agricultural industry represents the greatest potential source, it is not being produced commercially simply because the raw oil is too expensive. After the cost of converting it to biodiesel has been added on it is simply too expensive to compete with fossil diesel. Waste vegetable oil can often be sourced for free or sourced already treated for a small price. The result is Biodiesel produced from waste vegetable oil can compete with fossil diesel.

A. Murugesan and D. Subramaniam et.al.[1] was reported that to performance, emissions, and combustioncharacteristics of methyl/ethyl esters of pongamia, ethyl esters of neem and diesel blends in a diesel engine were experimentally investigated. For this study, methyl esters of pongamia, ethyl esters of pongamia, and ethyl esters of neem were added to diesel by volume of 20% (B20), 40% (B40), 60% (B60) and 80% (B80), as well as pure blend (B100). Fuels were tested in single cylinder, water-cooled, direct injection kirloskar diesel engine loaded by eddy current dynamometer. The effect of blends on engine performance, exhaust emissions, and combustion were examined at different loads. It was clear, up to 40% of methyl/ethyl ester did not affect the brake thermal efficiency.

Senthil R* and Silambarasan R et.al.[2] was reported that the performance and emission characteristics of a diesel engine fueled with annona-ethanol blend as a fuel. A single cylinder water-cooled four stroke diesel engine was usedin this experiment. The ethanol is blended with Annona Methyl Ester (AME) in the proportions of

60-40, 55-45, 50-50, 45-55. The performance and emission characteristics of annona-ethanol blends are evaluated by operating the engine at different load conditions. The performance parameters such as Brake Specific Fuel Consumption (BSFC), Brake Thermal Efficiency (BTE) and Exhaust Gas Temperature (EGT) were evaluated. Further, the exhaust emissions such as oxides of nitrogen (NOx) unburned hydrocarbon (HC), carbon monoxide (CO) and smoke were measure. It is found that annona-ethanol blend (A-E-50-50) showed slight increase in brake thermal efficiency with the reduction of exhaust gas temperature. Further, it is found that the slight reduction in NOx emission and smoke emission. It is also found that reduction in HC and CO emission was achieved. Hence, it is concluded that A-E 50-50 can be used as alternate fuel for DI diesel engine without any major modification.

Marousekj et.al.[3] was reported that to increase the efficiency of rapeseed oil recovery by pressure shockwaves and to assess the changes related to energetically utilization of the seedcake obtained. Mass balances and several design parameters (along with their manifestations on the seedcake) were analyzed to allow further optimization of the technology. It was found that the use of pressure shockwaves, in combination with the mechanical expeller, may increase oil yields up to the theoretical 100% maximum, or alternatively reduce expeller energy requirements while maintaining the same oil yield. Decreased amounts of oil in the seedcake correlate with reduced amounts of volatile matter, which means lower quantities of hazardous fumes generated during direct combustion. In addition, higher levels of seedcake disintegration accelerated the biogas production

Higa O, Kondo Y, Ueno et.al.[4] was reported that Kinetic data regarding the intensity of maceration and subsequent pretreatment with pressure shockwaves (50 MPa to 60 MPa) are described in detail and evaluated statistically. Mass balances as well as the study on liquid environment are reported, allowing further process optimization according to financial aspects. It was verified on a laboratory scale by Soxhlet apparatus that oil extraction over 94% may be reached. Achieving such a high level of disintegration opens wide options for application of hydrolysis in order to break apart the remaining lignocellulose cell walls and access the last oil remaining in the vacuoles.

3.0 DIESEL CYCLE AND RECIPROCATING INTERNAL COMBUSTION ENGINE

The diesel internal combustion engine differs from the gasoline powered Otto cycle by using highly compressed hot air to ignite the fuel rather than using a spark plug (compression ignition rather than spark ignition).

In the true diesel engine, only air is initially introduced into the combustion chamber. The air is then compressed with a compression ratio typically between 15:1 and 23:1. This high compression causes the temperature of the air to rise. At about the top of the compression stroke, fuel is injected directly into the compressed air in the combustion chamber. This may be into a (typically toroidal) void in the top of the piston or a pre-chamber depending upon the design of the engine. The fuel injector ensures that the fuel is broken down into small droplets, and that the fuel is distributed evenly.

The heat of the compressed air vaporizes fuel from the surface of the droplets. The vapour is then ignited by the heat from the compressed air in the combustion chamber, the droplets continue to vaporise from their surfaces and burn, getting smaller, until all the fuel in the droplets has been burnt. Combustion occurs at a substantially constant pressure during the initial part of the power stroke. The start of vaporisation causes a delay before ignition and the characteristic diesel knocking sound as the vapour reaches ignition temperature and causes an abrupt increase in pressure above the piston (not shown on the P-V indicator diagram). When combustion is complete the combustion gases expand as the piston descends further; the high pressure in the cylinder drives the piston downward, supplying power to the crankshaft.

As well as the high level of compression allowing combustion to take place without a separate ignition system, a high compression ratio greatly increases the engine's efficiency. Increasing the compression ratio in a spark-ignition engine where fuel and air are mixed before entry to the cylinder is limited by the need to prevent damaging pre-ignition. Since only air is compressed in a diesel engine, and fuel is not introduced into the cylinder until shortly before top dead centre (TDC), premature detonation is not a problem and compression ratios are much higher.

The p–V diagam is a simplified and idealised representation of the events involved in a Diesel engine cycle, arranged to illustrate the similarity with a Carnot cycle. Starting at 1, the piston is at bottom dead centre and both valves are closed at the start of the compression stroke; the cylinder contains air at atmospheric pressure. Between 1 and 2 the air is compressed adiabatically—that is without heat transfer to or from the environment—by the rising piston. (This is only approximately true since there will be some heat exchange with the cylinder walls.) During this compression, the volume is reduced, the pressure and temperature both rise. At or slightly before 2 (TDC) fuel is injected and burns in the compressed hot air. Chemical energy is released and this constitutes an injection of thermal energy (heat) into the compressed gas. Combustion and heating occur between 2 and 3. In this interval the pressure remains constant since the piston descends, and the volume increases; the temperature rises as a consequence of the energy of combustion. At 3 fuel injection and combustion are complete, and the cylinder contains gas at a higher temperature than at 2. Between 3 and 4 this hot gas expands, again approximately adiabatically. Work is done on the system to which the engine is connected. During this expansion phase the volume of the gas rises, and its temperature and pressure both fall. At 4 the exhaust valve opens, and the pressure falls abruptly to atmospheric (approximately). This is unresisted expansion and no useful work is done by it. Ideally the adiabatic expansion should continue, extending the line 3–4 to the right until the pressure falls to that of the surrounding air, but the loss of efficiency caused by this unresisted expansion is justified by the practical difficulties involved in recovering it (the engine would have to be much larger). After the opening of the exhaust valve, the exhaust stroke follows, but this (and the following induction stroke) are not shown on the diagram.

Advantages and disadvantages of Compression Ignition and spark Igition Engine.

a. Fuel economy

The man S80ME-C7 low speed diesel engines use 155 grams (5.5 oz) of fuel per kWh for an overall energy conversion efficiency of 54.4%, which is the highest conversion of fuel into power by any single-cycle internal or external combustion engine (The efficiency of a combined cycle gas turbine system can exceed 60%.[95]) Diesel engines are more efficient than gasoline (petrol) engines of the same power rating, resulting in lower fuel consumption. A common margin is 40% more miles per gallon for an efficient turbodiesel. For example, the current model Škoda Octavia, using Volkswagen Group engines, has a combined Euro rating of 6.2 L/100 km (46 mpg_{-imp}; 38 mpg_{-US}) for the 102 bhp (76 kW) petrol engine and 4.4 L/100 km (64 mpg_{-imp}; 53 mpg_{-US}) for the 105 bhp (78 kW) diesel engine.

While a higher compression ratio is helpful in raising efficiency, diesel engines are much more efficient than gasoline (petrol) engines when at low power and at engine idle. Unlike the petrol engine, diesels lack a butterfly valve (throttle) in the inlet system, which closes at idle. This creates parasitic loss and destruction of availability of the incoming air, reducing the efficiency of petrol engines at idle. In many applications, such as marine, agriculture, and railways, diesels are left idling and unattended for many hours, sometimes even days. These advantages are especially attractive in locomotives (see dieselisation).

Heat balance test on 4-stroke, single cylinder - Diesel engine test rig

Specifications

Make : Kirloskar model AVI
Bore: 80mm
Stroke: 110 mm
Rated Speed : 1500 rpm
BHP: 5Hp
No of cylinder: 1
Orifice diameter: 20mm
Type of Ignition: Compression Ignition
Method of Starting: Crank starting
Method cooling: Water cooling
Lubrication: Mechanical Lunbrication
Lubrication OIL: SAE – 40
Method of loading: Rope Break drum Dynamometer with Spring Balance

Starting the Engine:

1. Engage de-compression lever before cranking.
2. Crank the engine and disengage the de-compression lever.
3. Adjust the governor to attain the rated speed.

Procedure:

1. Open the three way cock so that fuel flows to the engine directly from the tank.
2. Open the cooling water valves and ensure water flows through the engine.
3. Start the engine and allow running on no load condition for few minutes.
4. Load engine by adding weights upon the hanger.
5. Allow the cooling water in the brake drum and adjust it to avoid spilling.
6. Allow the engine to run at this load for few minutes.
7. Adjust the cooling water regulators such that the temperature raise of Cooling water for engine jacket is around 50C and for calorimeter around 250C.
8. Note the following readings

a) Engine Speed
b) Weight on the hanger
c) Spring balance
d) Manometer
e) Time for 10cc of fuel consumption
f) Volume of Cooling water (Calorimeter) collected for 1 min.
g) Volume of Cooling water (Engine)collected for 1 min
h) Inlet and outlet temperatures of engine cooling water
i) Inlet and outlet temperatures of calorimeter cooling water
j) Inlet and outlet temperatures of exhaust gases
k) Ambient temperate

9. Repeat the above procedure for different loads.
10. Stop the engine after removing load on the engine

Observations

Full Load , W=*2 NR 9.81B.P 60 1000*Π ×× ×□…. kg

S.No Particulars ¼Load ½Load ¾Load Full Load

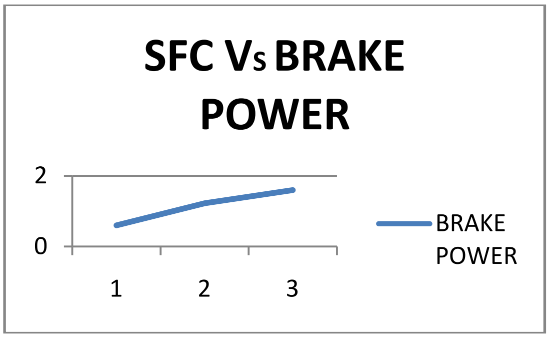
1. Rated Speed, N …..rpm
2. Difference of Water Manometer Reading, hw…….m
3. Time for 10cc of Fuel Consumption, t…..sec
4. Volume of Cooling water (Calorimeter) collectedfor 1 min. ,Vwc……… m3/min
5. Calorimeter Cooling Water Flow Rate, mwc…Kg/min
6. Volume of Cooling water (Engine)collected for 1min,Vwe……… m3/min
7. Engine Cooling Water Flow Rate, mwe ….. Kg/min
8. Engine Cooling water inlet temperature, T1 …0C

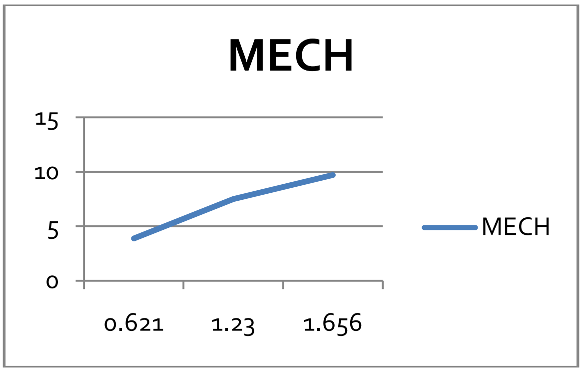
9. Engine Cooling water Outlet temperature, T2….0C
10. Calorimeter Cooling water inlet temperature, T3……0C
11. Calorimeter Cooling water outlet temperatureT4……0C
12. Calorimeter Exhaust gas inlet temperature, T5……0C
13 .Calorimeter Exhaust gas Calorimeter outlettemperature, T6 …..0C
14. Room temperature, T7 …. 0C

Calculations

1 Engine output (Brake Power) [B.P] =60 10002 NT□ □ □ …… KW
Where,
N = Rated speed ….. Rpm,
W0 = Weight of hanger = 1.0 kg
W1 = Weight on hanger ………. kg
W2 = spring balance reading ……….kg
Re = Effective brake drum radius = (R+r) … m
Where R is Brake drum radius
r is Rope radius
W = Net Load = [(W1-W2)+ W0] □ □ 9.81 ……… N
T = (W * Re)…………N-m
2 Indicated Power
Time for 10cc of fuel consumption, t = ……. Sec,
Mass of fuel consumption per min, mf = 60100010 . □ □ □*SpGravity of dieselt*..kg/ min.
Total Fuel Consumption, TFC = … mf □ □ 60….kg / hr.
Specific fuel consumption, SFC =*B PT F C*.. . ……..Kg / Kw-hr
Heat Input, HI =*60 60TFC CV*□ □ …….KW
Where CV is calorific value of given fuel = 45350 KJ / kg
Actual Air intake:
Manometer reading h1 = …………..cm of water
Manometer reading h2 = …………..cm of water
Difference in water level, hw =1001 2 *h* □ □*h*……………….m of water
Equivalent air column, ha =*Density of air/Density of waterhw* □ □ =1.16/1000 □ □*w h* ….m. of air
Orifice diameter, d = 0.03 m
Area of orifice, a =3.14□ □ (0.03^2) /4……..m2
Theoretical Volume of air intake, Va =60 □ □ Cd □ □ a □*a*□2*gh* m3 / min.
Where Cd = 0.62
Mass of air intake, ma = *a* □ □ □ Va……kg / min
Density of air *a* □ = 1.16 Kg/m3
Total mass of Exhaust Gas, mg = ma + mf kg/min
The specific heat of exhaust gas is determined by equating
Heat lost by exhaust gas = Heat carried by cooling water
Heat lost by exhaust gas ,Heg = mg □ □ Cpg □ □ (T5T6)
Heat gained by calorimeter cooling water, Hwc= mwc □ Cpw □ □ (T4T3)
Specific heat of gas, Cpg = ……. kJ/kg.k
Heat input, HI = T.F.C □ □ CV…… KJ/min
Heat Equivalent of B.P , HB.P = B.P □ □ 60 ………..KJ/min
Heat lost by exhaust gas, Heg = mg □ □ Cpg □ □ (T5T6) …..KJ/min
Heat Carried by engine Cooling water,Hwe= mw □ □ Cpw □ □ (T2 T1)… KJ/min
Heat unaccounted loss ,Hu = HI – (Heg + Hwe + HBP)

4.0 RESULTS AND PERFORMANCE CURVES:





Performance of IC Engine by using Bio- Diesel (B20)

Sl.no	Input Power (kw)	Output Power (kw)	Specificfuel Consumption (kg/kw-hr)	η_{bth} (%)	η_{mech} (%)	η_{vol} (%)	Air fuel ratio
1	4.6	0	0	0	0	80.0	0
2	4.2	1.1	0.33	27.5	41.1	81.7	0.03
3	5.6	1.8	0.28	32.1	51.4	80.2	0.05
4	8.2	2.3	0.32	27.9	57.5	91.6	0.06
5	11.3	4.7	0.21	43.2	63.8	80.9	0.13

Temperatures

S. no	T_1	T_2	T_3	T_4	T_5	T_6
1	31	31	25	29	67	25
2	31	32	7	25	143	12
3	31	32	20	21	183	6
4	29	28	24	23	52	20
5	26	22	21	22	190	15

Flow Meter Readings

Sl.no	Engine water flow rate - lPM_1	Calorimeter water flow rate - lPM_2
1	6	3

Performance of IC Engine by using Biodiesel (B15)

Sl.no	Input Power	Output Power	Sfc	Bth Efficiency	Mech Efficiency	Vol Efficiency
1	4.2	0	0	0	0	93.7
2	4.5	1.15	0.355	25.57	74.25	93.9
3	6.3	2.31	0.247	36.7	85	97.0
4	9.1	4.62	0.178	51.0	92	95.3

Temperatures

Sl.no	T_1	T_2	T_3	T_4	T_5	T_6
1	28	27	27	52	26	13
2	28	24	26	109	13	23
3	27	23	25	128	65	29
4	27	28	13	167	19	34

Performance of IC Engine by using Biodiesel (B25)

Sl.no	Input Power	Output Power	Sfc	Bth efficiency	Mech Efficiency	Vol Efficiency
1	3.90	0	0	0	0	93.7
2	4.68	1.15	0.367	49.61	39	93.9
3	6.30	2.311	0.247	36.75	56	93.9
4	9.1	4.622	0.178	51	71	93

Temperatures

Sl.no	T_1	T_2	T_3	T_4	T_5	T_6
1	29	29	30	84	56	66
2	27	28	25	108	16	42
3	28	29	24	120	24	54
4	29	24	26	160	52	56

Performance of IC Engine by using Diesel

Sl.no	Input power	Output power	Sfc	Bth efficiency	Mech efficiency	Vol efficiency
1	4.79	0	0	81.2	0	0
2	4.98	0.59	0.677	76.4	11.91	17.4
3	5.67	1.19	0.33	76.4	24.4	29.8
4	6.25	1.79	0.28	76.4	28.7	38.9
5	6.83	2.38	0.23	73.13	35.0	45.9
	10.25	4.77	0.17	75.3	47.4	63.0

Mass of Fuel Consumed (m_f):

$M_{f=}$ X_{CC}XSP.GR.OF FUEL/1000Xt

Where sp.gr of fuel(diesel)=0.827

X_{cc} is the volume of fuel consumed =10ml, t=time taken in sec.

Heat Input.HI

HI=m_fXcv KW

Where cv-calorific value of diesel=44631.96kj/kg

Out Put Power (or) Brake Power

Engine output BP=$2\prod NT/60000$ KW

Where N IS speed in RPM

T=FXrx9.81 N-M(r=0.15).

Specific Fuel Consumption

$SFC=m_f X3600/BP$ KJ/kg-hr.

Brake Thermal Efficiency η_{bth}:

$\eta_{bth}=3600x100/SFCXCV$

Mechanical Efficiency η_{mech}:

$\eta_{mech}=BP/IPX100$

where IP=BP+FP(from willian's line graph)

Calculation of Head of Air H_a:

$H_a=h_w\rho_{water}/\rho_{air}$

Volumetric Efficiency η_{vol}:

$\eta_{vol}=Q_a/Q_{th}X100$

where $Q_a=C_d a\sqrt{2gH_a}$, $C_d=0.62$ $a=\prod\backslash 4x(0.02)^2$, $Q_{th}=\prod\backslash 4Xd^2LN/60X2$.

D diameter of the engine=0.08m,L length of the stroke=0.01m AND N speed of the engine.

Heat Balance Sheet on Minute Basis:

Heat Supplied KJ / min % Heat distributed KJ/ min % Heat supplied by the fuel100 1.Heat in B.P

2. Heat carried by engine - Cooling water
3. Heat Carried by exhaust gases
4. Unaccounted losses (radiation, friction, etc.,100)

CONCLUSIONS:

Pongamia oil seems to have a potential to use as alternative fuel in diesel engines. Blending with diesel decreases the viscosity considerably. The following results are made from the experiment is -

1. The brake thermal efficiency of the engine with Pongamia oil -diesel blend was marginally better than with neat diesel fuel.
2. Brake specific energy consumption is lower for Pongamia oil -diesel blends than diesel at all loading.
3. The exhaust gas temperature is found to increase with concentration of Pongamia oil in the fuel blend due to coarse fuel spray formation and delayed combustion.
4. The mechanical efficiency achieved with Pongamia oil is higher than diesel at lower loading conditions. At higher loads, the mechanical efficiency of certain blends is almost equal to that of diesel.
5. The emission characteristics are higher than pure diesel but the Pongamia oil has relatively better performance with respect to other blends.

Pongamia oil can be accepted as a suitable fuel for use in standard diesel engines and further studies can be done with certain additives to improve the emission characteristics.

1. Performance and Emissions of c.i. Engine using Blends of Biodiesel and Diesel at Different Injection Pressures are experimentally investigated. The results of study may be summarized as follows:
2. Pongamia based biodiesels can be directly used in diesel engines without any modifications.
3. The performance is slightly reduced while brake specific fuel consumption is increased when using biodiesels.
4. The brake thermal efficiency of B10, B15 and B20 are better than B100 but still inferior to that of diesel and Compared with conventional diesel, exhaust emissions of CO and HC are reduced while NOx emissions are increased with biodiesel and its blends with diesel. The availability of abundant resources and environmental friendly emissions are recognized as strength of biodiesels leading them to potential candidates.

REFERENCES

[1]. .A. Murugesan1, D. Subramaniam; Analysis on performance, emission and combustion characteristics of diesel engine fueled with methyl–ethyl esters, Journal of Renewable and Sustainable Energy ,

November 2012 Accepted: November 2012

[2]. Senthil R* and Silambarasan, RDepartment of Mechanical Engineering, University College of Engineering Villupuram, Tamilnadu, India,Senthil and Silambarasan J Fundam Renewable Energy Appl 2015, 5:3

[3]. MaroušekJ, Itoh S, Higa O, Kondo Y, Ueno M, Pressure Shockwaves to Enhance Oil Extraction from JatrophaCurcas L.BiotechnolBiotechnological Equip 27.

[4]. MarousekJ, Use of continuous pressure shockwaves apparatus in rapeseed oil processing. (2013) Clean Technol EnvironPolicy15: 721-725.

[5]. Lin BF, Huang JH, Huang DY, Experimental study of the effects of vegetable oil methyl ester on DI diesel (2009). Engine performance characteristics and pollutant emissions. Fuel 88: 1779-1785.

[6]. Kannan D, NabiMN, HustadJE (2009)Influence of Ethanol Blend Addition on Compression .Ignition Engine Performance and Emissions Operated with Diesel and Jatropha Methyl Ester.Norwegian UnivSciTechnolSAE: 1808.

[7] Graboski MS,McCormick RL. Combustion of fat and vegetable oil derived fuels in diesel engines. Prog Energ Combust 1998;24:125–64.

[8] Lapuerta M, Armas O, Rodriguez-Fernandez J. Effect of biodiesel fuels on diesel engine emissions. Prog Energ Combust 2008;34:198–223.

[9] Basha SA, Raja Gopal K, Jebaraj S. A review on biodiesel production, combustion, emissions and performance. Renew Sust Energ Rev 2009;13:1628–34.