ENERGY ANALYSIS OF A DI DIESEL ENGINE FUELED WITH PUNNAI OIL METHYL ESTER AND ITS DIESEL BLENDS

C.Bibin[1], Vinoth Kumar.C[2], Manoj Kumar.N[3], Naveen.A[4]
1- Assistant professor, 2,- U.G Students
Department of Mechanical Engineering, RMK College of Engineering and Technology, Chennai

ABSTRACT
The major objective of the present study was to analyze the energy characteristics of 10% to 40% (by volume) blending of punnai oil methyl ester (PME) with diesel as fuels in the diesel engine at various loads. From steady state experimental results, an energy analysis was carried to determine the Energy loss in engine cooling, Energy loss with exhaust gas and unaccounted Losses. Engine IP was more for the blends up to B30 compared to diesel fuel operation, but it significantly reduced in case of the blend B40 indicating more energy losses with this fuel blend. The energy analysis also indicates higher energy loss for the blend B40.
Keywords: Biodiesel, Diesel engine, Fuel Energy, Punnai oil Methyl Ester.

1. INTRODUCTION

Rapid depletion of petroleum reserves and the increasing concern for the environment has forced the scientific community to develop eco friendly alternative fuels, especially, to the diesel oil for its partial or full replacement. Vegetable oil obtained from non edible sources is considered promising alternate fuel for diesel engine compared to their edible counterpart due to the food vs. fuel controversy. Biodiesel derived from non edible plant species such as Jatropha curcas (Ratanjot),Pongamia pinnata(karanja), Azadirachta indica (Neem), Madhuca indica (Mahua), Hevea brasiliensis (Rubber seeds) are receiving significant attention as possible renewable alternate fuels in India. Biodiesel is a clean burning, renewable alternate fuel that has minimum sulfur and aromatic content. Sulfur is a pollutant directly and no significant air pollution reduction strategy can work without getting sulfur out of fuels. Reduction in sulfur content is the most notable restriction in the recent years, which has direct economic consequence on the investment made by the oil companies and finally on the fuel price. Biodiesel has definitely the advantages from this point of view. However it has higher viscosity and lower heating value. High viscosity of fuel leads to poor atomization of the fuel spray and incomplete combustion resulting in gradual coking of the injector tips, oil ring sticking and thickening and gelling of the engine lubricant oil. Biodiesel is less suitable for low temperature application as its cloud and pour points are higher than diesel. At low temperature, fuel forms wax crystals, which can clog fuel lines and filters in a vehicle’s fuel system. Probably for these reasons, the 5-20% (by volume) blending with standard diesel has been considered as suitable at present for using in existing diesel engines without any modifications. But the most suitable biodiesel blending that gives optimum performance in an unmodified diesel engine depends upon the type of biodiesel and the engine configurations.

Many researchers have experimentally evaluated the energy analysis of conventional diesel engines fuelled with bio-diesel and its blends. An energy balance study on an engine helps in understanding how the energy is lost which in turn help in finding means to reduce the same to improve the performance of the engine in terms of efficiency and power output. This is the main reason behind most energy studies performed on engines. Besides energy balances studies help characterize the effect of changes in engine operating and design parameters on the overall engine system. Once it is identified by the energy balance, one can attempt to either maximize or oppose its use due to the benefits and drawbacks. If there were significant losses through certain parts of the system, the energy balance study would show this and help identify the cause. Researchers [4,5] have made energy analysis of diesel engine from theoretical calculations of heat transfer and exhaust energy. Research works based on experimental observations are very few [6,7] particularly with biodiesel and its blends as engine fuels.

In the present investigation, biodiesel was prepared from Punnai oil using a two step acid base catalyzed trans-esterification process in a laboratory scale. The properties of the various blends were determined. An experimental study was conducted to evaluate and compare the different blending in a standard, fully instrumented, four stroke, DI, Kirloskar 'TAF1' Diesel engine. The series of tests were conducted using the fuels at various loads. The various losses are measured and analyzed.

2. PROPERTIES OF DIESEL AND VARIOUS BLENDS OF BIODIESEL

Biodiesel obtained from punnai oil was mixed with diesel and **t**he properties of diesel and various blends were evaluated.These are summarized in Table 1 below.

Table 1. Properties of diesel and various blends of biodiesel

Property	Diesel	B10	B20	B30	B40
Density at 15°C (g/cc)	0.846	0.850	0.855	0.859	0.866
Kinematic viscosity at 40°C (cSt)	2.34	2.64	2.84	3.07	3.28
Higher heating value (kJ/kg)	45553.0	45489.9	45418.1	45348.9	45247.4
Cetane Index	46.60	46.34	46.50	46.34	45.39
Flash point (°C)	46.0	47.0	49.0	53.0	55.0
Pour point (°C)	3.0	-3.0	0.0	3.0	6.0
Sulpher content (ppm)	489.0	440.0	390.0	302.0	274.0

3. EXPERIMENTAL SET-UP

The experimental set up is a single-cylinder, four-stroke, naturally aspirated, DI diesel engine with specifications given in Table 2. Necessary instruments for combustion pressure, fuel pressure and crank-angle measurements are provided in the set up. One piezo sensor mounted in the engine cylinder head senses the in cylinder pressure and the pressure signals from this are fed to a charge amplifier. A high precision crank angle encoder is used to give signals for top dead centre (TDC) and the crank angle. The signals from the charge amplifier and the crank angle encoder are supplied to a data acquisition system which is interfaced to a computer through engine indicator for obtaining pressure crank angle (p-θ) diagram. There are provisions made in set up also for interfacing airflow, fuel flow, temperatures and load measurement. A differential pressure transducer is used to measure air flow rate. The engine is coupled with an eddy current dynamometer for controlling the engine torque through computer. Thermocouples are used to measure different temperatures, such as exhaust temperature, coolant temperature, and inlet air temperature. Two rotameters are provided for engine cooling water and calorimeter water flow measurement. A Labview based engine performance analysis software package is provided for on line performance evaluation. A gas analyzer was used to measure the concentration of gaseous emissions. The gas analyzer calibrates automatically every time they are started and display the quantity of exhaust gases.

Table 2. Engine Specifications.

Make and Model	Kirloskar TAF 1
Number of cylinder	1
Stroke	Four stroke
Type of cooling	Air cooling
Ignition	Compression ignition
Bore	87.5 mm
Stroke	110mm
Compression Ratio	17.5:1
Speed (Constant)	1500 rpm
Rated Power	4.4 kW
Fuel injection timing	23^0 bTDC
Type of dynamometer	Eddy current dynamometer
Injection pressure	200 bar

3.1 Experimental procedure

The engine was first made to run at a fixed compression ratio by supplying the diesel fuel to the engine and then the various blends of diesel and biodiesel one by one. The load was varied from no load to over load in five steps in case of all the fuels investigated. For each fuel three test run were performed under identical conditions to check for the repeatability of all results. The repeatability of the results was found to be within an acceptable limit. At each load, the engine speed was measured by the crank angle encoder; cylinder pressure was measured by the piezo electric sensor mounted in the cylinder head. The fuel flow and the air flow were measured by the flow transducers. The signals that were obtained from various sensors were fed to the engine indicator for storing the data and interfacing with computer. The stored data were analyzed by using the analysis software package.

4. ENERGY ANALYSIS

Experiments with engines very often involve an energy balance on the engine. In a reciprocating internal combustion engine the chemical energy of the fuel is the input energy and not all the input energy is converted into useful work. A major part of the heat energy produced as a result of combustion of fuel is used for cooling the engine, a significant part of the fuel energy is lost with the exhaust gases and there are unaccounted energy losses which may be due to radiation from high temperature in-cylinder combustion gases and the surface of the combustion chamber. The unaccounted loss may also include the heat energy consumed by friction. Radiation is considered to be an important mode of heat transfer in diesel engines which is mainly caused by presence of soot particles in the combustion gases. Due to these losses the thermal efficiency of the engine reduces. Now from first law of thermodynamics, the total heat energy produced from combustion of fuel must be equal to the sum of the useful power produced and the various losses.

Following are the calculation involved.

Fuel Energy Input (kW) = $\dot{m}_f$ × Calorific Value

Where $\dot{m}_f$ is the fuel flow rate of the engine at a given load in kg/sec.

BP (kW) = $\frac{2\pi NT}{60000}$; where N is the engine rpm, T is the engine torque measured by the dynamometer.

Frictional power (kW) = IP-BP

Heat lost to coolant (kW) = $\dot{m}_{w.engine} C_{pw}(T_2 - T_1)$

Where $\dot{m}_{w,engine}$ is the mass flow rate of engine cooling water. The volume flow rate of cooling water is measured by adjusting the flow rate of rotameter (engine) and this is multiplied with the density of water to obtain $\dot{m}_{w,engine}$ T_1 and T_2 are the engine water inlet and outlet temperatures respectively. C_{pw} is the specific heat of water evaluated at the mean of the temperatures T_1 and T_2. The density of water is also evaluated halfway between T_1 and T_2. The computerized engine supplier provided a method for calculation of the exhaust heat loss as given below.

Exhaust loss (kW) $= (\dot{m}_a + \dot{m}_f) C_{pex}(T_5 - T_{amb})$;

T_{amb} is the ambient temperature. It was seen that for calculating the heat lost with exhaust gases, it requires the specific heat of the exhaust gas. It can be calculated using the following equation which is derived from an energy balance in the exhaust calorimeter and for that, it requires that data logging should be done after the engine reaches the steady state.

$$C_{pex} = \frac{\dot{m}_{w,cal} C_{pw}(T_4 - T_3)}{(\dot{m}_a + \dot{m}_f)(T_5 - T_6)}$$

In these equations, $\dot{m}_{w,cal}$ is the water flow rate through the Calorimeter which is measured by adjusting the flow rate of the rotameter (Calorimeter). T_3 is the Calorimeter water inlet temperature, T_4 is the Calorimeter water outlet temperature, T_5 is the exhaust gas temperature at Calorimeter inlet temperature, T_6 is the exhaust gas temperature from calorimeter outlet. Yuksel and Ceviz [8] determined the losses through the exhaust using a heat balance in the exhaust gas calorimeter. However in our case, after conducting the experiments, it was seen that there was variation in the values of C_{pex} calculated at a given load for a particular fuel and also the exhaust losses calculated from the average specific heat values were found to be high. Besides, there was uncertainty in the results found because the heat loss in the calorimeter was totally ignored in the calculation of C_{pex}.

Therefore, the exhaust energy was calculated from the enthalpies of products [5]. The remainder of the energy that was not measured which can be found from energy conservation is termed as unaccounted miscellaneous losses. Based on the experimental emission results obtained at various loads and considering equilibrium of these species in the combustion products, the molar based reaction equation can be expressed as follows

$$C_xH_yO_z + \lambda(O_2 + 3.76N_2) \rightarrow$$
$$aCO + bCO_2 + cH_2O + dO_2 + eC_xH_yO_z + fNO + gN_2$$

The values of these coefficients in the equation per mole of fuel were calculated at various loads and the exhaust energy in kW was calculated using the following equation. The molecular formula of diesel is approximated as $C_{12}H_{23}$ [9] and the chemical formula of pure biodiesel is derived from the fatty acid composition of punnai oil as $C_{17.75}H_{33.43}O_{1.98}$ [10] and from these the chemical formula for the blending have been obtained.

$E_{ex} = \dot{n}_f \Delta\bar{h}$, where $\Delta\bar{h}$ is the term for enthalpy change of the products at the EGT and the reference temperature expressed as $\Delta h = h(T) - h(T_{ref})$ and $\dot{n}_f$ is the molar fuel flow rate.

4.1 Fuel energy

The fuel energy input to the engine for the five different tested fuels is shown in Fig. 1. The PME and its diesel blends provide slightly higher energy to the engine than diesel fuel in order to produce the same brake power output at different loads. This was due to higher fuel consumption with respect to the blends, although the lower heating value was less for them. Since the fuel energy input was more for the PME and its diesel blends, therefore, the brake power output being the same for all the fuels at a particular load, it resulted in lower brake thermal efficiency in case of the blends.

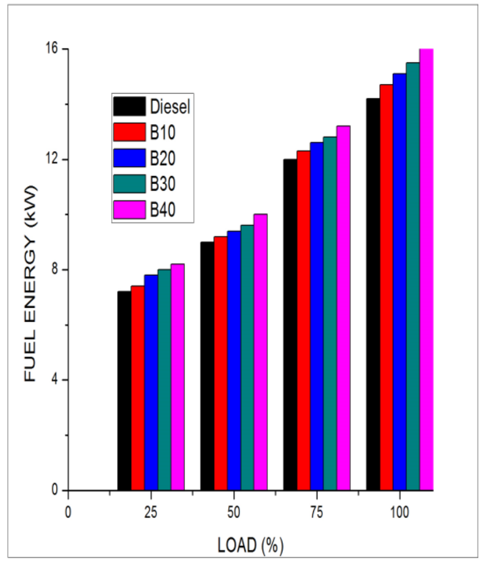

Fig. 1: Fuel energy input to the engine at various loads for the tested fuel

4.2 Energy loss in engine cooling

Fig. 2 shows the loss of energy in cooling the engine by cold water which was circulated through the engine jacket. The heat carried away by a coolant medium generally consists of heat transferred to the combustion chamber walls from the gases in the cylinder, heat transferred to the exhaust valve in the exhaust process, and a substantial fraction of the friction work[11]. It was seen that the energy lost to the coolant increases with increasing load for all the fuels. This was because; at higher load due to burning of relatively more amount of fuel, the in-cylinder temperature increases and therefore the temperature of engine cooling water leaving the engine jacket (T_2) also becomes high. Since the flow rate of engine cooling water is maintained at the same level at all engine loads, therefore with increase in the value of T_2, the heat lost to the coolant increases. Heat losses for engine cooling were in general found to be more for the PME blends at all the loads. Early premixed combustion and subsequent heat release in the late combustion phase in case of the biodiesel blends may have resulted in slightly higher overall cylinder temperatures than diesel due to which more heat was absorbed by the engine cooling water. Since the energy absorbed by the coolant also accounts for the energy transferred to the lubricating oil from friction of moving engine components, therefore the higher frictional losses in case of the blends may have also added to the increase in temperature of coolant leaving the engine. However, in case of the blend B40 a significant reduction in cooling loss was observed at 75% load and full engine load. This may be due to reduction in the cylinder temperature resulting from incomplete combustion of this particular fuel blend due to its higher viscosity, poor atomization and lack of available oxygen for burning a rich fuel mixture. At full engine load, there was 2.893%, 4.23%, 9.37% increase in energy loss to the coolant with B10, B20 and B30 respectively as against a reduction of 4.89% in case of B40 compared to cooling loss with respect to diesel. This may be noted that cooling losses may be minimized by increasing the coolant temperature that can be achieved by lowering coolant flow rate which in turn decreases the mass flow rate of fuel consumption.

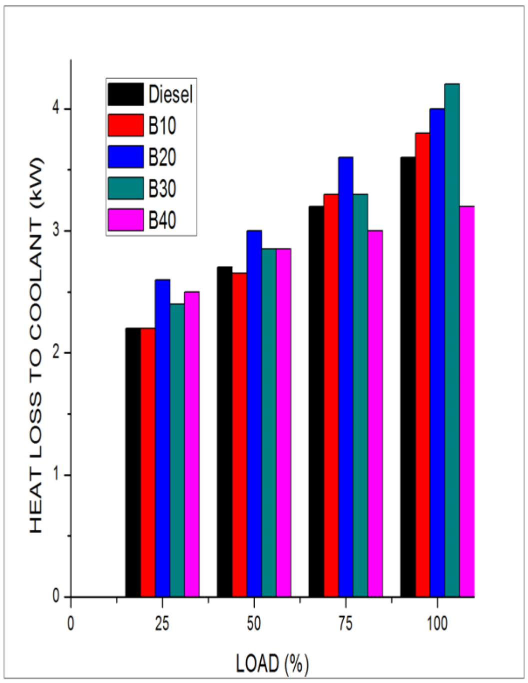

Fig. 2: Energy loss in engine cooling at various loads for the tested fuels

4.3 Energy loss with exhaust gas

A major part of the input energy not available for useful work is the energy carried away by the exhaust gases. The energy loss accompanying the exhaust gas is shown in Fig. 3. It was observed that there was increase in the exhaust loss with increasing load in almost all the fuel operations. With the increase in load there is an increase in the amount of fuel injected to the combustion chamber while the amount of air inducted being more or less constant at various loads. The heat release increases with load due to combustion of higher amount of fuel and as a result the temperature of the products of combustion increases. Higher the temperature of combustion gases more is the EGT and hence the loss of energy accompanying exhaust gas is also more at higher engine load. It was seen that the EGT was more for the PME blends therefore the energy loss accompanying exhaust gas was also found to be slightly higher for the blends. At 75% load the exhaust gas loss was almost found to be same for diesel, B10 and B20. At full engine load operation, the energy loss accompanying exhaust gas was slightly more with B10 and B20 fuel operations as compared to diesel fuel operation. However, with B30 and B40 fuel operations at full engine load, the exhaust loss decreased slightly. The calculated losses were found to be 3.013, 3.073, 3.045, 2.991, 2.941 kW respectively for diesel, B10, B20, B20 and B40 respectively which were almost comparable even though the EGTs for the blends were more. Reduction in exhaust loss with B40 was mainly due to lower values of EGT and the molar ratios of the O_2 and N_2 in the combustion products as compared to the other blends.

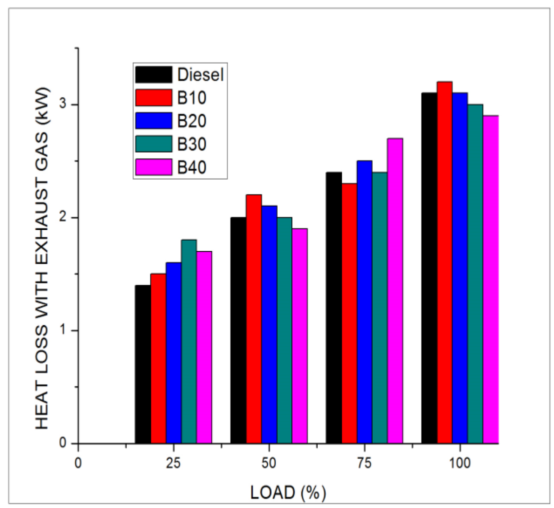

Fig. 3: Energy loss with exhaust gas at various loads for the tested fuels

4.4 Unaccounted Losses

The unaccounted losses may include the heat rejected to the oil (if separately), convection and radiation heat transfer from the engine's external surface. These may also include the losses resulting from incomplete fuel combustion. These losses at various engine loads for all the tested fuels are presented in Fig. 4. It was observed that these losses were more in case of the PME blends at all the loads except for the blend B10 at 25% and 50% load and B20 at 25%

load. What was more interesting and important to observe that these losses were significantly high for the blend B40 at all engine loads. Therefore higher unaccounted losses in case of B40 at various loads and more particularly at full engine load implies that a significant portion of these losses with B40 at various loads were either mainly due to higher radiation or losses due to the combustion inefficiency. These losses mainly account for the losses due to inefficient combustion. Higher combustion losses as well as lower engine IP in respect of B40 clearly indicates the problem relating to the combustion of this particular PME blend. These losses for diesel, B10, B20, B30 and B40 at full engine load were 1.33, 1.43, 1.67, 1.56 and 3.9 kW respectively and account for 9.74%, 10.15%, 11.6%, 10.72% and 24.17% of the fuel energy input. It was also seen that these losses increased with load for all the fuels which may be due to incomplete combustion of relatively rich fuel air mixture at higher load.

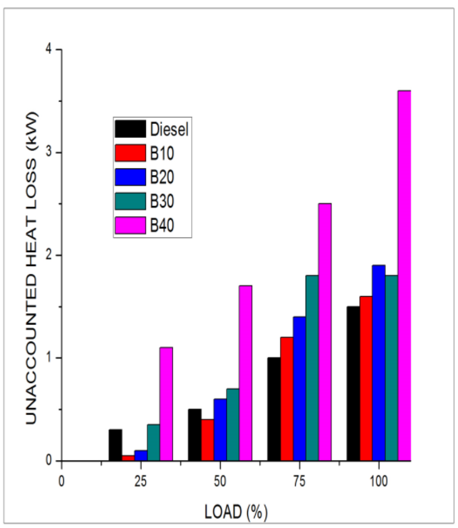

Fig. 4: Unaccounted energy loss at various loads for the tested fuels

5. CONCLUSION

A comparison of physical and fuel properties of different PME blends with pure diesel fuel indicates that the blends are quite similar in nature to diesel fuel with viscosities being slightly higher and calorific values being lower for the blends. Fuel energy input was more for the PME blends at various loads despite the lower heating value of the blends. Engine cooling losses were slightly more for the blends at various loads, however for B40 it was comparatively less at 75% and full load. This could be due to reduction in cylinder temperature resulting from incomplete combustion of this particular fuel blend. The energy loss accompanying exhaust gas was also found to be slightly higher for the PME blends at lower engine loads. However, at 75% load the exhaust gas losses were almost found to be same for diesel, B10 and B20 and slightly being higher for B30 and B40. At full engine load operation, these losses were slightly more with B10 and B20 and slightly lower with B30 and B40 as compared to diesel fuel operation. The most important observation was the unaccounted heat losses with respect to the fuel blend B40 which was significantly high at all engine loads. The viscosity was more for the blends and due to higher viscosity and particularly for this blend; the fuel did not atomize properly leading to poor combustion, which ultimately resulted in higher unaccounted losses. Use of fuel injection pump employing higher injection pressure could be a solution to this problem relating to use of higher level of biodiesel blending. Based on the study it can be recommended that PME blending up to 30% can be used in an unmodified Kirloskar diesel engine without any significant loss in performance.

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